INL/RPT-24-80435
Revision 0

Light Water Reactor Sustainability Program

Heat Augmentation from LWRs to Support High-Temperature Industrial Applications

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U.S. Department of Energy

Office of Nuclear Energy

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Prepared for the
U.S. Department of Energy
Office of Nuclear Energy
Under DOE Idaho Operations Office
Contract DE-AC07-05ID14517

SUMMARY

The United States and countries around the world are seeking to reduce dependence on fossil fuels to achieve climate goals and ensure national energy security. In the United States, industrial process heat based on fossil fuel sources accounts for approximately 30% of all greenhouse gas emissions. Replacing those heat sources with low-carbon “clean” heat is receiving considerable attention. This work focuses on the technical aspects of delivering heat from light-water reactors and upgrading that heat to high temperature (350–550°C) using mechanical heat pumps to support a wide array of industrial processes.

A generic layout of a nuclear power-based heat pump process is illustrated in Figure E1. In this work two heat pump cycles are evaluated for feasibility: the steam compression cycle and the reverse Brayton cycle, including simple and complex split compression designs with heat recuperation. Thermodynamic and practical process temperature and pressure limits are also considered.

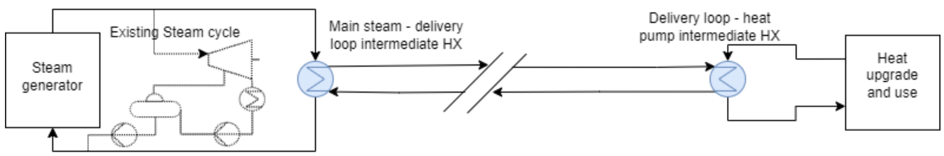

Figure E1. Generic layout of considered heat delivery and augmentation system.

Heat pumps performance is typically measured using a coefficient of performance (COP), which is defined as the amount of delivered heat divided by the required electrical input. In that calculation, the low temperature input heat is assumed to be available without any energy cost. However, heat from nuclear plants can be used to produce electricity; so, in this work, an adjusted coefficient of performance (COP_{ADJ}) is developed that accounts for loss in electric power production due to diverting nuclear heat to a heat pump. Assuming the nuclear plant thermal conversion efficiency is 35%, the theoretical maximum COP_{ADJ} is 2.86, which can only be achieved if zero electricity is consumed by the heat pump.

The performance of several heat pumps configurations is highlighted in Figure E2. The left panel shows different combinations of heat transfer fluid temperatures entering and exiting the industrial process for sensible heating. As the load inlet and exit temperatures increase, COP_{ADJ} decreases. While steam compression cycles provide the best thermodynamic performance, those cycles require pressures exceeding 200 MPa and compressor outlet temperatures over 1,000°C for heat delivery at the highest evaluated demand temperature of 550°C. For comparison, typical steam compression cycles are limited to approximately 6 MPa, and world class leading steam turbine technology in ultra-supercritical coal plants has only achieved approximately 35 MPa and 700°C. Those systems experienced severe challenges and have all been abandoned. Generally, high-pressure compressors are limited to approximately 80 MPa with a temperature limitation of approximately 400°C.

The right panel in Figure E2 shows expected COP_{ADJ} values for simple and advanced cycles providing heat to a load that operates at a uniform temperature (such as for a phase change process) and also shows performance curves that adhere to practical temperature and pressure limits in the compressor of 700°C and 30 MPa, respectively. The maximum COP_{ADJ} value that is achieved is approximately 2 for a relatively small temperature boost from 290°C to 350°C. As the demand temperature increases to 500°C, the COP_{ADJ} quickly decreases to approximately 1.4. These expected low COP_{ADJ} values greatly challenge the feasibility of using heat pumps to upgrade the heat from nuclear plants. For example, general guidance is that the COP of air-source heat pumps should be at least 2 for feasibility, and the COP of geothermal heat pumps should be at least 3.

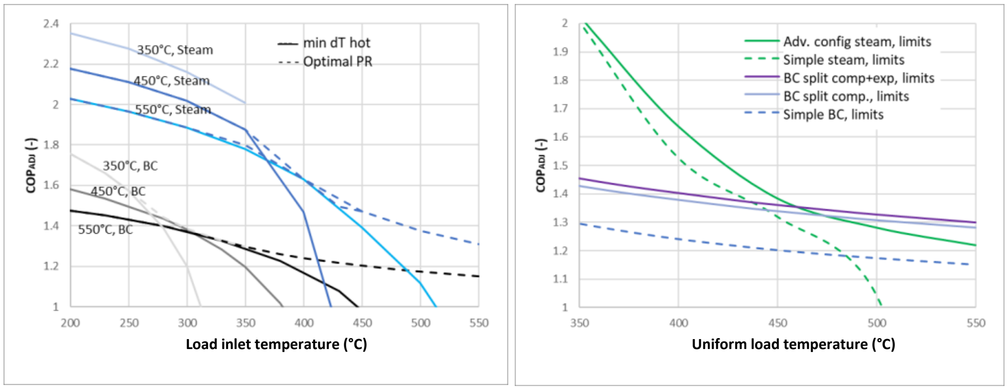

Figure E2. Comparison of COP_{ADJ} values for a simple Brayton cycle (BC) and steam compression system for sensible heat demand with different load exit temperatures as functions of load inlet temperature (left panel); and similar comparison for additional cycles, such as split compression and expansion cycles, with uniform load temperature considering technical limitations (right).

In addition to technical considerations, economic feasibility must also be evaluated. A provisional net present value analysis (NVA) has been completed that compares nuclear power-based heat pump operations to electric heating, and the key results are plotted in Figure E3. The economic feasibility of nuclear power-based heat pumps depends primarily upon three factors, which are (1) the average price for which nuclear electricity can be sold, the system-level COP_{ADJ} value, and capital expenditure CAPEX of the heat pump system. It must be noted that the system-level COP_{ADJ} includes assumptions regarding thermal losses from the system. Heat pump system maintenance cost and capacity factor (operating hours per year) are also important but are neglected in this provisional analysis. Figure E3 shows system-level COP_{ADJ} values that are needed to achieve economic parity with electric heating for a range of heat pump parameters. Assuming a system-level thermal loss of 10% and a CAPEX cost of $1,000/kWe, an advanced Brayton cycle heat pump with a COP_{ADJ} of 1.3 delivering heat to a uniform-temperature process (the most favorable heat load case) becomes competitive at electricity prices above approximately 60 $/MWhe. For a case of typical electricity prices in the range of $30/MWh, the system COP_{ADJ} must exceed 1.5 if 10% thermal losses are considered. Including heat pump maintenance cost and capacity factor could significantly increase the required system COP_{ADJ} to achieve economic parity with electric heating.

Figure E3 also shows system COP_{ADJ} values that are required to achieve economic parity with electric heating assuming that heat losses can be reduced to negligible levels. For a heat pump system with a CAPEX cost of $1,000/kWe and negligible thermal losses to achieve economic parity with electric heating for an electricity price of $30/MWh, the system COP_{ADJ} values can be as low as 1.38.

Based on the results in Figure E2, this provisional analysis indicates that nuclear-based heat pumps are not likely to achieve economic parity with electric heating for applications that require heat at temperatures hotter than 400 °C. It is also important to note that these simple analyses do not consider interruption risks to the industrial process due to dependency on complex equipment operating at high temperatures and pressures. Including interruption risks will tend to increase the favorability of electric heating methods because those technologies have well established reliability, modularity, and cost.

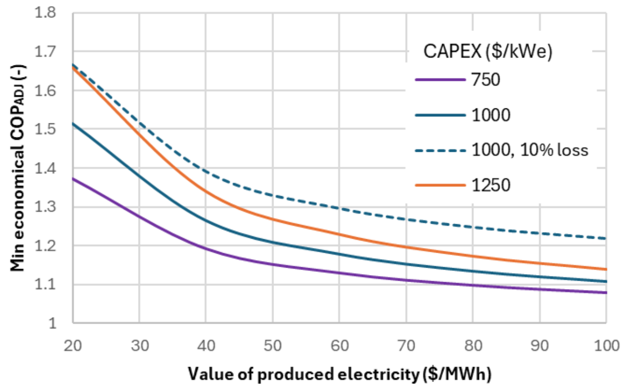

Figure E3. Minimum required COP_{ADJ} for heat pump systems to achieve economic parity with electric heating as a function of electricity price.

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FIGURES

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ACRONYMS

CHP	Combined heat and power
COP	Coefficient of performance
LWR	Light-water reactors
NPP	Nuclear power plants
NPV	Net present value
RD&D	Research, Development and Demonstration
SG	Steam generator

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HEAT AUGMENTATION FROM LWRS TO SUPPORT HIGH-TEMPERATURE INDUSTRIAL APPLICATIONS

1. INTRODUCTION

The United States and countries around the world are seeking to reduce dependence on fossil fuels to achieve climate goals and ensure national energy security. In the United States, industrial process heat based on fossil fuel sources accounts for approximately 30% of all greenhouse gas emissions [1]. Replacing those heat sources with low-carbon "clean" heat is receiving considerable attention. This work focuses on the technical aspects of delivering heat from light-water reactors and upgrading that heat to high temperature (350–550°C) using mechanical heat pumps to support a wide array of industrial processes.

Nuclear energy is an important component of decarbonization efforts [2] [3]. While the historical role of nuclear power plants (NPPs) in the United States has been providing baseload power, nuclear power promoters, vendors and operators are arching for nuclear power promoters, vendors and operators are searching for other business cases that could increase revenue and potentially assist in reducing carbon emission of energy-intense industrial processes in a combined heat and power (CHP) approaches [4] [5]. Indeed there is already notable experience using current light-water nuclear plants to supply heat for district heating [6] [7], desalination [8] [7], and a limited number of cases where process steam is delivered to other industries [6]. Recent projects demonstrate the viability of heat delivery over relatively long distances. For example, a 2023 utility district heating system piped hot water at 150°C a distance of 26 km from a 2 x VVER 1000 Temelín plant to a city with a population of 100,000 inhabitants, delivering 750 TJ annually at a 100 MW peak duty [9]. Nuclear-sourced district heating is also considered in Reference [10].

When considering industrial process heat delivery, LWRs are limited by the steam saturation temperature corresponding to the turbine inlet pressures, typically around 275°C (6 MPa). In some novel LWR systems, such as the NuScale VOYGR 77, the saturation temperature is as low as 240°C (3.3 MPa) [11]. Delivering process heat is less common than district heating; however, a large system was recently commissioned that supplies about 340 MWth of 1.8 MPa steam superheated to 248°C (about 50°C of superheat), assuming a condensate return temperature of approximately 100°C. The heat source is a VVER-1000 nuclear plant with two units that are located more than 23 km from the chemical production industrial complex [12] [13] [14].

Conceptually, the delivered heat can be upgraded to higher temperatures to extend the range of applications in which the low-carbon nuclear heat can be used. This report analyses options to upgrade heat from light-water reactors (LWRs) and compares those options to direct electric heating. Two heat pumping approaches are evaluated, including a steam (vapor) compression cycle using water as a working fluid and a reverse Brayton cycle operating in a cooling mode with air as the working fluid. Air was chosen as the working fluid for the reverse Brayton cycle because it has a similar performance to other leading gases and has the additional advantages of being inexpensive and relatively inert. The analyses focus on basic thermodynamic limits and early technical considerations.

High-temperature mechanical heat pumps with target temperatures of approximately 250°C are gaining attention [15] [16] for upgrading industrial waste heat because that temperature is suitable for most industrial heat loads [17] [18]. For these applications, special attention is given to the selection of the working fluids [19], and only few options show promise for temperatures augmentation of heat from nuclear reactors. Standard working fluids, such as fluorocarbons and various refrigerants, have temperature limitations below the needed working temperature or are highly toxic, such as decane and propylbenzene [20] [21] [22]. Potential working fluids with strong solvent properties cause additional complexity for equipment design, especially bearings.

One concept [23] suggests using biphenyl, which is a relatively exotic working fluid. It has a normal boiling point near 250°C and critical point (using Peng-Robinson equation of state) at 516°C and 4 bar. A supercritical heat pump cycle was proposed, delivering sensible heat at up to 600°C for H_2 thermochemical production. Stand-alone cascade vapor compression cycles with biphenyl as the working fluid is claimed to have coefficient of performance (COP) of 2.3–3.8 for cycles that operate between these temperature levels of Reference [24]. Similarly, siloxanes have been suggested for delivery temperatures in the range of 350–500°C, though with maximum temperature lifts not exceeding about 150°C [25]. These theoretical studies, however, neglect that these types of organic fluids typically undergo thermal decomposition at much lower temperatures [26] [27].

Chemical and sorption heat pumps were suggested as an alternative to mechanical heat pumps with the potential of high lifts and potentially high COP [28]. Development of these systems can take advantage of similar work taking place for thermochemical energy storage [29], and there have also been proposals to integrate this technology with nuclear heat [30]. However, these systems still suffer from issues associated with materials durability, such as corrosion, and generally have low technological readiness level. Physical experiments have been mostly limited to small scale (kW) laboratory systems that have not exhibited clear success [31].

In this report, simple layouts for the mechanical heat pumps are evaluated as well as more complex arrangements that include split compression, expansion and recuperation. Another important consideration is the heat transfer characteristics between the hot fluid exiting the heat pump and the industrial process. The system performance depends on whether that heat transfer occurs at a uniform temperature, such as for a phase change process or at variable temperatures due to sensible heating. This report considers both of those options as well and estimates the range of technically feasible solutions by considering the maximum likely system pressure and temperature limits at key points in the system. Last, an approximate range of economic feasibility is estimated by comparing high-level costs heat pump systems to those of electric heating. Heat pump solutions are not considered economically feasible if their total costs, including total capital costs and operating costs, are higher than those of electric heating.

1.1 Industrial High Temperature Heat Demand

Standard process steam systems are typically employed for process heat delivery at temperatures below approximately 300°C. In many cases, applications that require high-temperature process heat are designed and optimized for direct fossil fuel combustion. Potential issues associated with supplying high-temperature process heat include the following factors:

- Heat transfer in current processes is predominantly achieved through radiation.
- Heat transfer limitations may require that the temperature of the heat transfer fluid be at significantly higher temperatures than the maximum temperature of the process.
- The process may involve direct fuel conversion, where the fuel composition plays a crucial role in the process, such as in iron and steel industries.
- The byproducts from certain industries, such as pulp/paper and petroleum refineries, are used as fuel to provide heat. Many industrial processes are optimized for utilization of the waste byproducts, and supplying external heat destroys the value-added role these byproducts provide. There are additional modifications for reprocessing byproducts and waste that are no longer used as fuel.

As shown below, direct electrification or non-carbon fuel combustion may be more feasible than delivering and upgrading nuclear heat for applications hotter than 300°C. One study [32] found there is relatively low demand for external process heat in the range of 300–750°C for petroleum refinery, methanol, pulp and paper, and iron and steel industries. The amount of process heat in the range of 250–500°C is also relatively low, as indicated in the industrial heat demand distribution in Figure 1. Based on this information, it seems upgrading LWR heat for applications between 250–500°C represents a niche market, and upgrading heat for applications above 500°C is decidedly impractical.

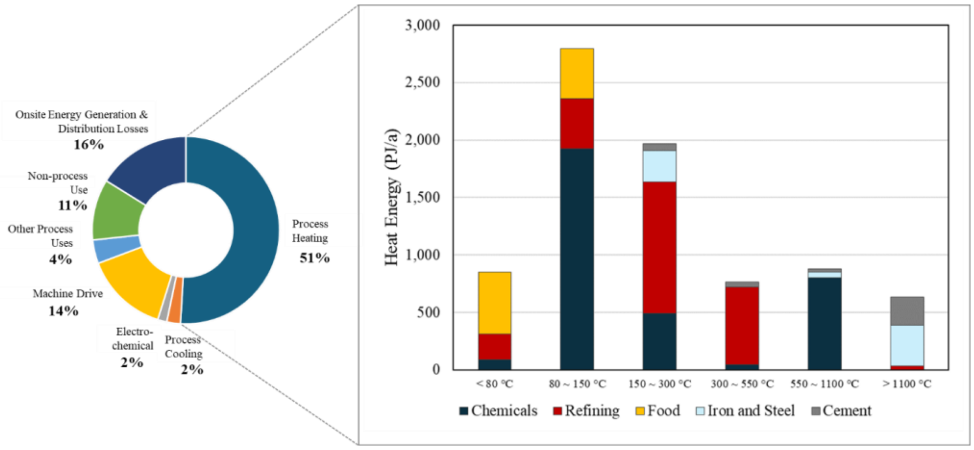

Figure 1. Breakdown of on-site energy use at U.S. manufacturing facilities, along with the distribution of process heat temperature ranges by industrial subsector [33].

2. METHODOLOGY

The analysis is based on standard thermodynamic mass and energy balance calculations using a heat transfer pinch point approach. A summary of the assumptions is provided in Section 2.6. The size of pinch point temperature difference is selected as technically attainable, yet optimistic. Similarly, there are selected values of compressors and turbines isentropic efficiencies. Optimistically, mechanical and electrical (motor) efficiencies are included in the isentropic efficiencies. Fluid properties are based on the Coolprop database [34].

2.1 Heat Source

The heat source for the heat pump is considered to be the main steam line in a standard 1,000 MWe class Westinghouse pressurized-water reactor plant. The main steam parameters are steam pressure p_{steam} = 6 MPa and quality x = 1. Nominal feedwater temperature at the steam generator inlet is considered to be T_{FW} = 225°C.

2.2 Performance Indicators

2.2.1 Coefficient of Performance

Standard efficiency metrics are well established for typical CHP systems in which steam is produced at high pressure and temperature. In such processes that require low-temperature heat, a portion of the steam is extracted from the turbine system at intermediate temperature and pressure after performing work in the high-pressure stage. Extracting heat from the turbine system slightly decreases the efficiency of electric power production but significantly increases overall energy utilization efficiency by decreasing the amount of heat that is rejected to the surroundings. For those CHP systems, the overall efficiency of the process is calculated based on the sum of the output electric and thermal powers. An issue with that approach is it lumps the electric and thermal power together in the efficiency evaluation even though the quality and potential usefulness of those different powers are not the same. The same approach can be used to evaluate the efficiency of a coupled turbine/heat pump system; however, its value is further diminished because it does not capture the increased quality of the heat exiting the heat pump compared to the heat entering the heat pump. In fact, it completely fails to recognize the benefit of the heat pump, and for that reason, the performance of heat pumps is typically reported as a coefficient of performance

(COP), which is defined as the heat delivered by the heat pump divided by the electric power consumed by the heat pump as shown in Equation (1):

$$COP = \frac{Q_{delivered}}{W_{in}} \tag{1}$$

Notably, the standard COP definition does not account for the heat entering the heat pump, which is assumed to come from the surroundings with energy penalty. In the case of a heat pump coupled to a power plant steam system, that standard definition, as illustrated in Figure 2, is incomplete because it does not account for the heat from the turbine system that is consumed by the heat pump.

Consequently, in this work, we develop an adjusted COP, denoted COP_{ADJ}, which is defined as the heat delivered by the heat pump divided by the sum of the electric power consumed by the heat pump and the heat entering the heat pump as shown in Equations (2) and (3):

$$COP_{ADJ} = \frac{Q_{delivered}}{W_{in}+W_{loss\ production}} \tag{2}$$

$$COP_{ADJ} = \frac{Q_{delivered}}{W_{in} + Q_{input} \times \eta_{NPP}} \tag{3}$$

Although still not perfect, this parameter does capture the benefit of the heat pump compared to a resistive heater, which has a COP of unity, while also accounting for the heat extracted from the turbine system.

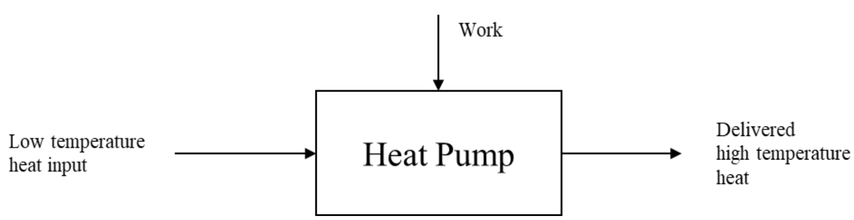

Figure 2. A generic heat pump block diagram.

In Equation (3), η_{NPP} is the power conversion efficiency of the nuclear plant and is assumed to be a constant 35% throughout the analysis. This approximation is not exact and depends upon the specific turbine control system. For example, controls such as group nozzle, partial arch, uniform throttling, or possibly even sliding pressure (which is not common) could be used to decrease the steam flow through the turbines, which generally lowers thermodynamic cycle efficiency. However, higher steam quality at lower pressure due to turbine control valve throttling may increase the turbine isentropic efficiency, partly mitigating the negative impact of lower steam flow rate. Such behavior has been observed in a modified model of a nuclear plant in which steam was extracted from the main steam line for a heat application [35].

A consequence of approximating the plant conversion efficiency as 35% is the value of COP_{ADJ} cannot exceed 2.86, even if the electric power consumption of the compression system (W_{in}) is zero. This is because every MWh of extracted heat corresponds to 350 kW of lost power production. As shown below, including the electric power consumption of the compressor system but still neglecting thermal losses, the maximum ideal efficiency is not expected to be much higher than 2, which greatly challenges the feasibility of using heat pumps to upgrade the heat from nuclear plants. For example, general guidance is that the COP of air-source heat pumps should be at least 2 for feasibility, and the COP of geothermal heat pumps should be at least 2.8 [36].

2.2.2 Compression Pressure and Pressure Ratio

Pressure and pressure ratio are important for physical design and assessment of technical feasibility of any compressor. Higher absolute pressures generally increase costs, even though the unit volume of fluid decreases. As shown later in the report, even though some concepts appear thermodynamically favorable, the associated operating pressures limit technical feasibility. It should be noted that for dynamic, non-volumetric compressors, the number of stages and compressor cost increases with PR. The maximum pressure ratio for compressors of commercial gas turbines is approximately 24 [36] [37].

2.2.3 Return Condensate Temperature to Steam Generator

The steam generator inlet has a nominal temperature requirement to ensure the reactor operates within the desired conditions. Hot fluid returning from the industrial process must not cause the temperature of the condensate entering the steam generator to exceed the threshold. One option for decreasing the feedwater temperature and increasing options for the heat pump system is to reduce the flow of extraction steam from the turbines and use the return condensate to support feedwater heating.

2.2.4 Mass Flow Rate Ratio

The ratio of the mass flow rates of the heat delivery fluid to the heat pump working fluid is another important consideration. As this ratio increases, the flow rate of the delivery fluid can become impractically high, resulting in large pipe diameters and pumps with very high costs. This phenomenon is a major factor explaining why steam has better performance transporting the latent heat than fixed gases.

2.3 Heat Demand Types

Three model cases of heat delivery, representing various types of heat users, are considered in this work:

- Uniform heat capacity demand
- Uniform temperature demand
- Uniform temperature demand followed by cooling.

The first case represents thermal demand where sensible heat is delivered to a material to raise its temperature. The second case corresponds to heat transfer occurring at a fixed temperature. Examples include phase change during a boiling process, heat to support endothermic chemical reactions, and curing of materials (which often involve a phase change and chemical reactions).

Any process involves both uniform heat capacity demand and uniform temperature demand, such as preheating a feedstock followed by curing in a furnace. A percentage of 20% sensible heat is estimated to be a good representation of this type of industrial processes so a reference mixed-mode heat delivery case is considered in this work that consists of 20% sensible heat delivery and 80% uniform temperature demand at 200°C. Note that many processes use heat recuperation or other means of waste heat recovery so heat requirements at lower temperatures are fulfilled using recovered heat. For this reason, 200°C is considered as the lowest temperature for heat demand. Figure 3 presents heat– temperature curves (Q-T diagrams) pointing out also at distinct cases of pinch point locations during the heat transfer for each of the considered cases.

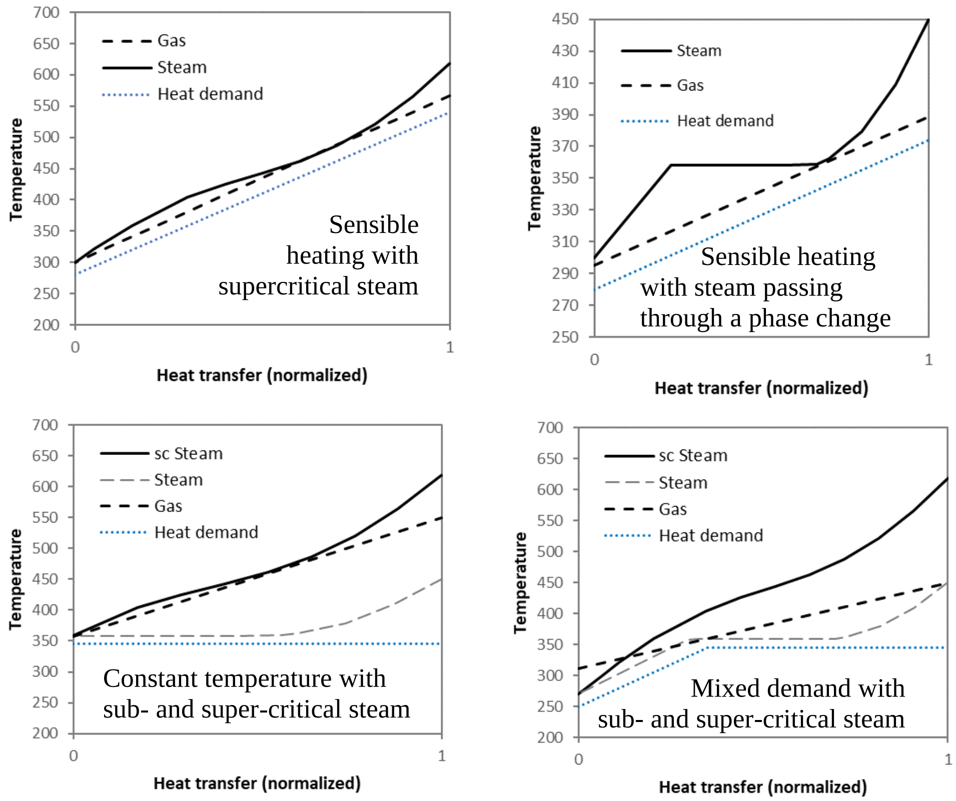

Figure 3. Examples of representative heat demand-temperature (Q-T) diagrams.

2.4 System Configurations

When considering the general layout of a heat upgrade and delivery system, several practical engineering aspects must be considered. First, the delivery steam should be separated from original NPP steam for multiple reasons, including safety, regulation, potential contamination, and chemical treatment differences between steam in the nuclear plant and steam delivered to any coupled industrial process. To reduce costs, it is preferable to transport the fluids at lowest possible temperatures and at moderate pressures, so the heat pump is located close to the industrial process, as shown in Figure 4. In this situation, a portion of the steam from the steam generator is diverted to an intermediate heat exchanger at the NPP site where it is condensed and preferably subcooled before being returned to the condenser or feedwater stream. In this analysis, the heat pump working fluid is assumed to be the same as the heat delivery fluid, which may or may not be steam.

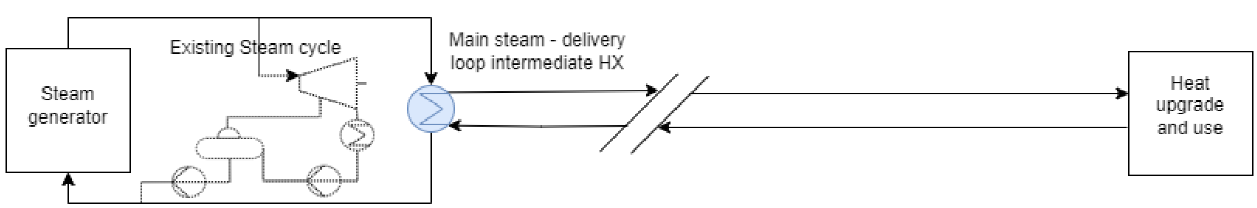

Figure 4. Simplified heat delivery and temperature upgrade (heat pump) system assuming a single working fluid.

In the case that the heat pump working fluid is not the same as the heat delivery fluid, an additional intermediate heat exchanger and heat transfer loop are required, as illustrated in Figure 5. The additional intermediate heat exchanger and heat transfer loop pose additional system challenges in terms of temperature drop, thermal losses, system complexity, reliability, and cost, as described in greater detail below.

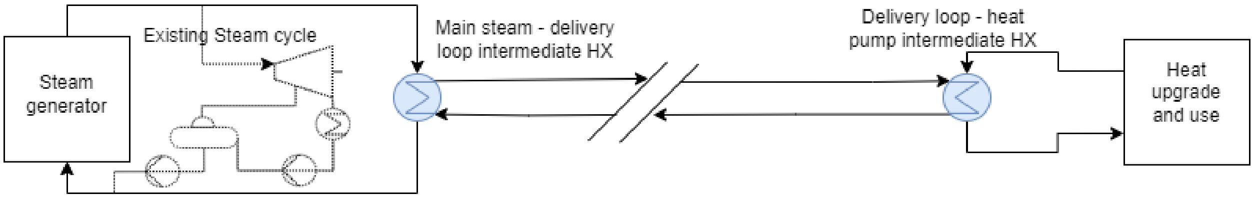

Figure 5. Heat augmentation and delivery loop with separate heat delivery and heat pump working fluids.

2.4.1 Steam Compression Systems

Standard heat pump fluids, such as fluorocarbons and various refrigerants, have temperature limitations below the needed working temperature for nuclear heat-based applications or are highly toxic, such as decane and propylbenzene [20] [21] [22]. Potential working fluids with strong solvent properties cause additional complexity for equipment design, especially bearings. Because of these challenges, steam offers an attractive option for a nuclear-couple heat pump. A simplified steam delivery and compression system is schematically shown in Figure 6. Liquid water is evaporated in a heat exchanger heated by steam from the steam generator, and the resultant steam is sent to the heat pump, which is located near the end use site. In the heat pump, steam is compressed, which also increases the fluid temperature. The compressed steam delivers high-temperature heat to the load. The working fluid exiting the load may still be at pressures and temperatures above the vapor dome, so an expander may be used to recover some of the pressure difference and at the same time increase the wetness after the expansion. Practical limits for steam quality after expansion determine the pressure at the expansion valve inlet. A separator provides a means to circulate residual steam back into the delivery steam line upstream of the compressor and prepare the condensate for the return trip to the nuclear site.

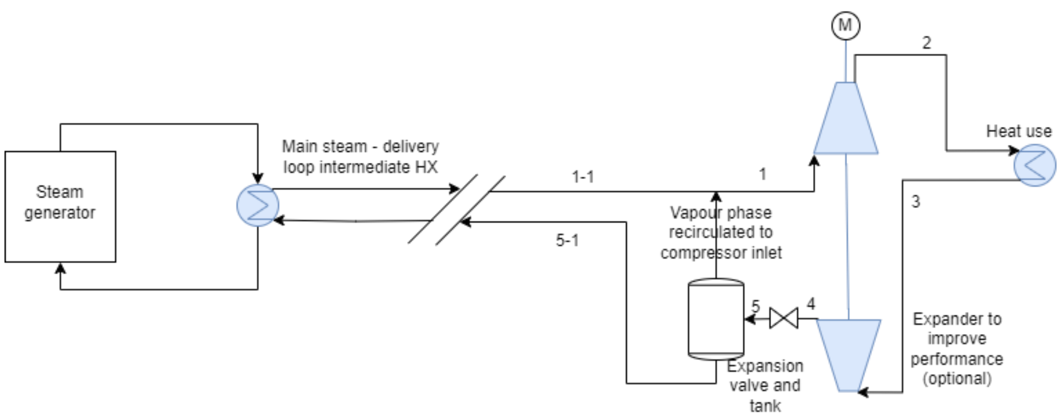

Figure 6. Steam compression cycle for thermal energy upgrade.

One issue is the temperature of the condensate set back to the steam generator is expected to be approximately 275°C; however, the nominal allowed inlet temperature to the steam generator is only 225°C. There are two leading options on how to manage this issue. The first option is to reject the heat from the condensate return line to the ambient or for a tertiary use. The second option is reducing the extraction of steam from the turbines and modifying the feedwater heaters, so that the condensate returning from the industrial process supports feedwater heating. Neither effect is analyzed in detail in this work but may be considered in future work.

2.4.2 Brayton Cycle

In addition to a steam compression cycle, we also evaluate a Brayton cycle, as shown in Figure 7. Unlike the steam compression option, Brayton gas cycles do not involve phase changes in the heat pump. An expander downstream of the heat load is a necessity—not just for recovering power but also for decreasing the fluid temperature.

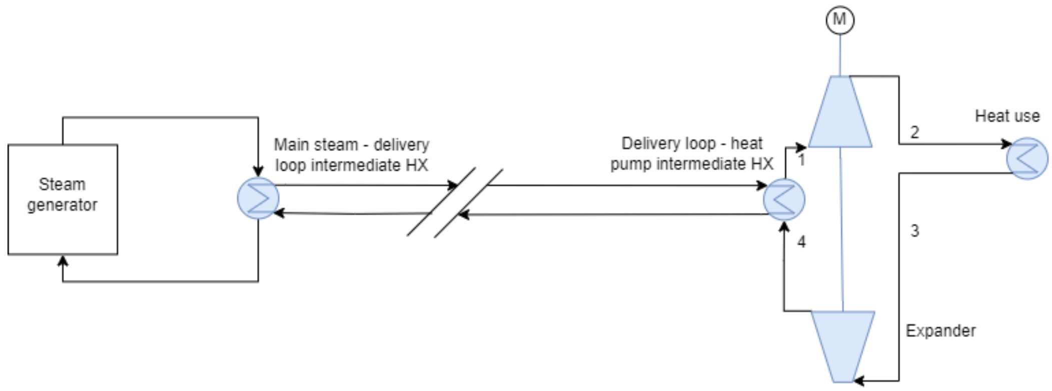

Figure 7. Simple Brayton gas cycle for thermal energy upgrade.

2.4.3 Advanced Cycle Configurations (Split Compression, Expansion, Recuperation)

In situations in which the temperature of the working fluid exiting the heat load is hotter than that of the fluid entering the compressor, a recuperator can be implemented, as shown in Figure 8. This feature is beneficial when the heat is delivered from a gaseous phase without phase change, such as in a Brayton cycle and partly in a supercritical steam compression cycle. In a standard vapor compression cycle, this approach is beneficial if the working fluid has a steep slope in the saturated vapor curve in T-s diagram. Beyond these cases, recuperation increases compressor power due to increased superheat at the inlet while the condensing pressure remains the same and only liquid subcooling before the expansion valve can be slightly increased.

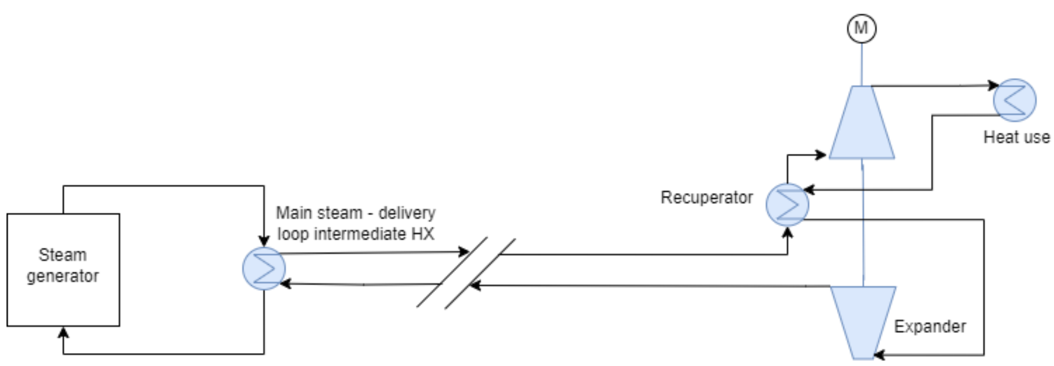

Figure 8. Recuperated Brayton heat pump cycle.

In cases that require delivery of ultra-high delivery temperature, the required pressures and temperatures for a single compressor can become unfeasible. A potential mitigation in such a situation is to split the compression into two stages with separate heat delivery at each stage, as seen in Figure 9. The pressures and temperature after individual compressors may be more practical.

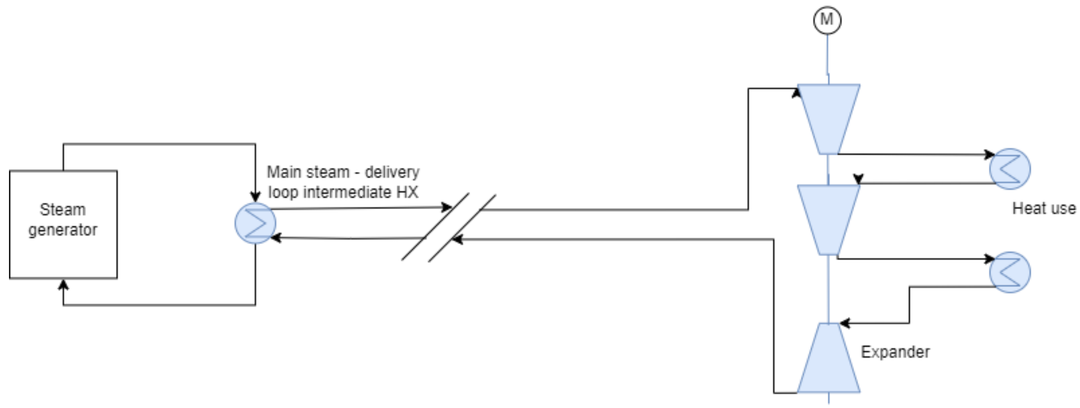

Figure 9. Split compression Brayton heat pump cycle.

Both approaches described above can be combined in a split compression recuperated system, as shown in Figure 10, which can also improve the thermodynamic performance. When using steam as the working fluid, a separator may also be needed, as shown above in Figure 6.

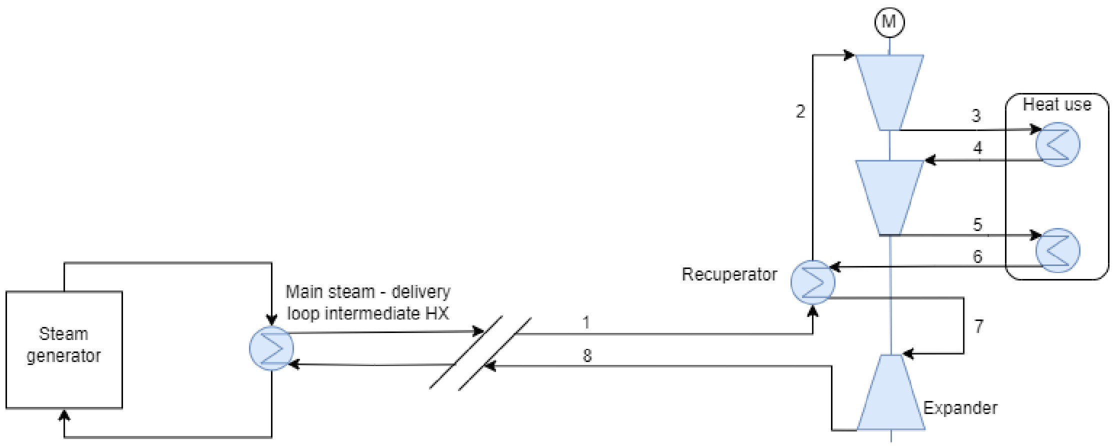

Figure 10. Split compression recuperated Brayton heat pump cycle.

The final advanced design for consideration is a split compression and expansion recuperated cycle and is known to have the best theoretical performance for power production. Because steam-based systems involve the expansion of two phases and part of the expansion cannot be realized in the turbines in most cases due to excessive liquid content, the final advanced design is only considered for Brayton cycles. The system layout, which includes multiple compression stages and expander stages with heat recuperation, is shown in Figure 11.

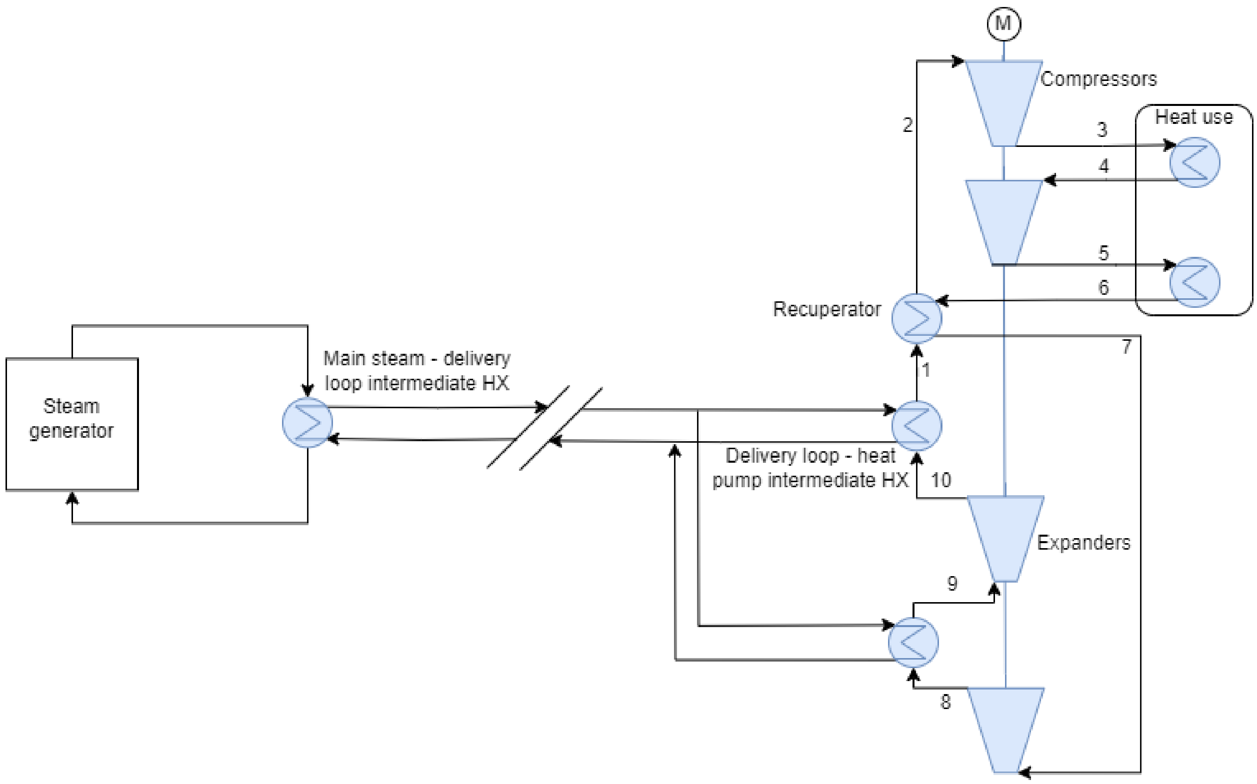

Figure 11. Split compression and split expansion recuperated Brayton heat pump cycle.

2.4.4 Electric Boosting

Electric boosting (i.e. not direct electric process heating, but electric heaters used for increasing heat transfer fluid temperature) as an upgrading option can only boost the fluid temperature, not pressure, which limits potential applications. Figure 12 illustrates the concept and notes this option only works when the return fluid temperature (and thus load exit temperature) is below the temperature of steam generator. The only exception could be to modify the turbine steam extraction system and feedwater heater system as described above in Section 2.4.1 so that heat from the fluid returning from load is used to support feedwater heating. For gas systems with electric topping heat, transferring the heat to the load with a small temperature difference requires very high mass flow rates that may require impractical heat delivery pipe diameters and pumps.

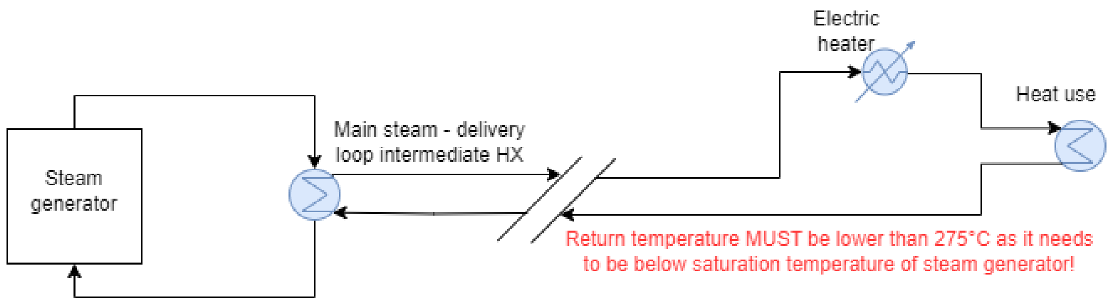

Figure 12. Illustration of a process with direct electric heating for heat upgrade.

2.5 Economic Viability Estimate

The economic viability estimate employs standard net present value (NPV) methods. The approach adopted in this work is to compare the heat pump and direct electric heating approaches. Therefore, only a cost differential between the two options is calculated, and the resulting potential application space is obtained from a parametric study. In this approach, the solution meets the following criteria for Equation (4).

$$NPV_{Electric\ Heater} \geq NPV_{Heat\ Pump} \tag{4}$$

In this simplified analysis, project duration is taken as 20 years plus a construction period of 1 year. Discount rate is 6%. Revenue for heat pump operations comes from avoided costs by consuming less electricity than would be used in a direct electric heating process. In this approach, the NPV of the electric heating reference case is thus always negative value.

Capital cost is estimated as \$100/kW for electric heating [38] [39] [40]. Estimating the capital cost of the heat pump system is much more difficult. Given the extent of components and system complexity (turbo-compressor and expander, heat exchangers, heat delivery system), it is estimated that the capital expenditure (CAPEX) would be comparable to similarly sized power conversion systems for power production in gas turbines and steam cycles. Based on several References [41] [42] [43], a range of \$750/kWe to \$1250/kWe is explored. Operation costs excluding electric power consumption are expected to be 1% of CAPEX. The assumed capacity factor is based on 8,000 annual operating hours.

The combination of sale price of excess electricity and heat pump system capital cost then provides through Equation (4) a value of COP_{ADJ} above which the heat pump system operation is economically favorable. Note the COP_{ADJ} from solving the Equation (4) should include additional heat and pressure losses excluded from this analysis. An illustrative example of their impact on the required heat pump COP will be shown in Section 3.

2.6 Summary of Assumptions.

The list below contains a summary of assumed parameters used in the analysis. Minimum temperature differences and compressor/expander efficiencies are chosen to represent the optimistic end of the technically feasible range. The heat source was specified as a standard LWR plant. Pressure losses for this initial primarily thermodynamic study are neglected, and their impact may be explored later for selected cases. A low pressure for Brayton cycle is chosen as 250 kPa as being reasonable without excessive overpressure. As all the fluids except for water are in the highly superheated regions, varying this pressure has a negligible impact on the thermodynamic performance as follows:

- ΔT_{min_gas} – 20 K – minimum temperature difference for heat transfer with gaseous fluid on one or both sides of the HX
- $\Delta T_{min\_boil/cond\text{-}liq}$ – 3 K – minimum temperature difference for heat transfer with boiling or condensation on one or both sides of the HX
- $\Delta T_{min\_liq\text{-}liq}$ – 5 K – minimum temperature difference for heat transfer with liquid fluid on both sides of the HX
- η_{exp} – 90% – expander (turbine) isentropic efficiency
- η_{comp} – 78% – compressor isentropic efficiency
- η_{NPP} – 35% – reference nuclear plant thermal efficiency
- p_{SG} – 6 MPa – steam generator pressure
- x_{SG_out} – 100% – steam quality at the steam generator outlet (saturated)

- $x_{exp\_out\_min}$ – 90% – minimum acceptable steam quality at expander outlet
- T_{SG_in} – 225°C – nominal steam generator inlet temperature
- Δp_{loss} – 0% – pressure losses.

The list below contains parameters that are varied in the parametric study and their ranges:

- Required load inlet temperature for sensible heating and uniform load temperature – 300–550°C
- Assumed load exit temperature for sensible heating – 200–500°C
- Brayton cycle fluids – Air, N_2, CO_2, He, Ar.

3. RESULTS

Results are presented for all process heating types (temperature profiles) with the simple cycle cases. Next, parameters for cases with uniform heating temperature are illustrated using the advanced configuration of split compression recuperated system.

3.1 Steam Compression (Simple) Cycle

A simple vapor compression cycle is demonstrated for sensible and uniform temperature heating. As the load exit temperature approaches the higher heating temperature, the process transforms to uniform heating temperature, which will show the upper limit for each case. The uniform heating temperature case is then verified against technical limits on maximum pressure and temperature and on advanced cycle configurations with recuperation, split compression, and expansion.

3.1.1 Sensible Heating

3.1.1.1 Load inlet temperature of 550°C

First, we present results for sensible heat delivery at the load entry temperature of 550°C. The thermodynamic parameters of the steam compression cycle options are illustrated as a temperature-entropy (T-s) diagram in Figure 13. The numbers in Figure 13 correspond to a return (load exit) temperature of 550°C with an optimized pressure ratio. The steam compression cycle has the minimal temperature difference typically in the middle of the heat transfer process. For 550°C case, the system always requires a supercritical state, and heat transfer is illustrated in Figure 14. Adjusting the system for reaching this minimal temperature difference yields the lowest pressures and temperatures in the system. An improved COP may be achievable at a higher pressure ratio and thus higher compressor outlet pressures and temperatures that would likely not be feasible. The anticipated feasible, optimized systems are represented by orange lines. Systems designed between these two limits would require technically complex and costly solutions because the steam would not always be in a supercritical state.

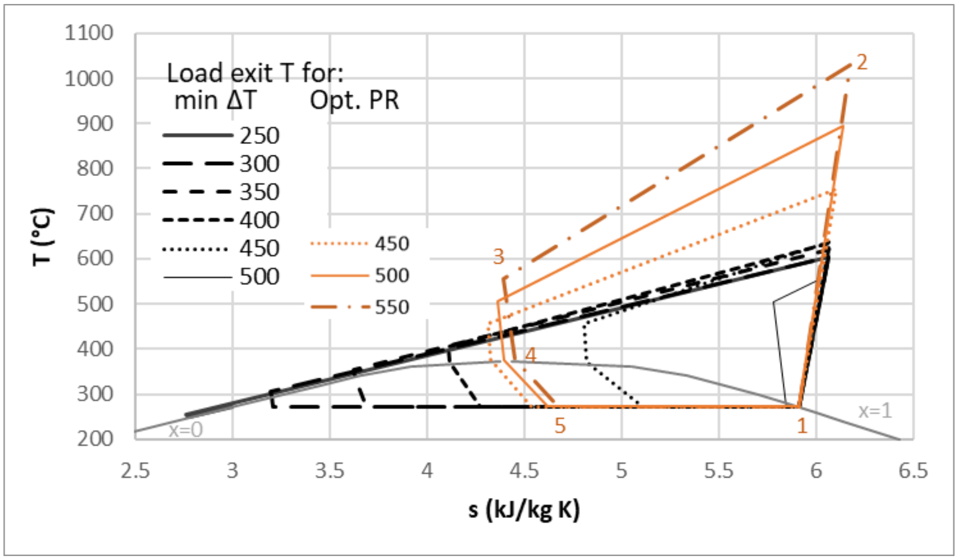

Figure 13. T-s diagrams of steam heat pumping at a load entry temperature of 550°C.

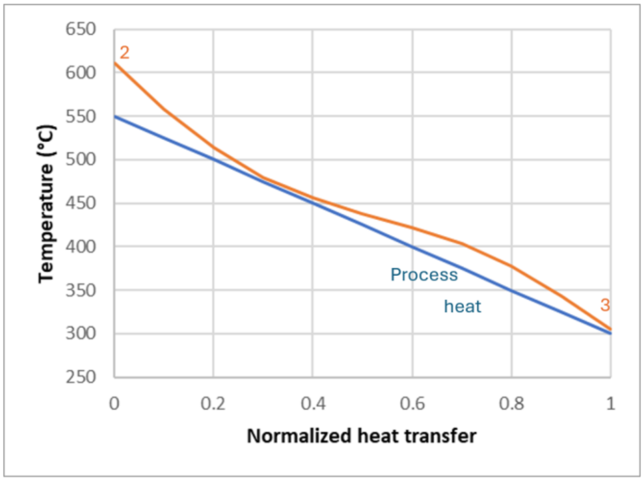

Figure 14. Q-T diagram of heat transfer from compressed steam to heat a process to 550°C (sensible heating).

Figure 15 shows COP and COP_{ADJ} for the steam compression systems as functions of load exit temperature of the heat load. At the highest temperatures, the COP curve for optimized pressure ratios diverges from the pinch-point-driven cases. Regardless of relatively high COP values, the COP_{ADJ} is limited in the entire range to below 2, which is generally considered to be impractical, especially considering that this simplified analysis neglects potentially significant thermal and electrical losses. Note that a COP below 2 means that compressor input comprises more than 50% of delivered duty. Another important observation is the COP values are highly dependent on the lower process heating (heat transfer fluid return) temperature.

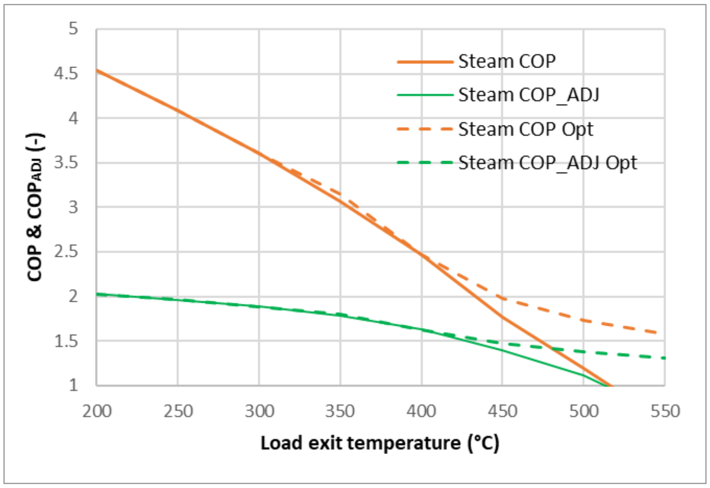

Figure 15. COP and COP_{ADJ} for the steam heat pumping with load entry temperature of 550°C, including both pinch-point-driven and pressure-ratio-driven cases.

Values of other parameters of interest are shown in Figure 16. The left panel shows temperature at the compressor outlet. Note that temperatures around 600°C are reasonably close to temperatures found in axial compressors, especially for natural gas fired turbines. A limitation to keep in mind is first stages of gas turbine without active cooling have temperature limits around 1100°C. Operation with steam involves additional challenges, such that active cooling may be unavoidable. In case of volumetric compressors, active cooling mechanisms do exist for systems operating at high temperatures (following the principle of engine jacket cooling).

The middle panel shows the pressure ratios and pressures at the compressor outlet. The pressure ratios are moderate, but the absolute pressures start at 37 MPa, and the pressure for optimized pressure ratio goes as high as 220 MPa, corresponding to 550°C delivery and uniform heating temperature (for phase-change, constant temperature chemical reactions and similar processes). At the lower end, a pressure of 37 MPa and temperatures of 600–700°C correspond to parameters of advanced ultra-supercritical thermal power plants [44], which have suffered many issues in development and application. For example, in Germany, these were the first plants phased out by the utilities [45]. Any decision to pursue development of a compressor system operating at these parameters should be carefully evaluated in terms of reliability and affordability.

The last parameter, volumetric flow rate parameter, indicating the volume to be transferred per unit mass of heat source steam (thus linked to volumetric flow rate per MWth) has very low values due to the latent heat of steam.

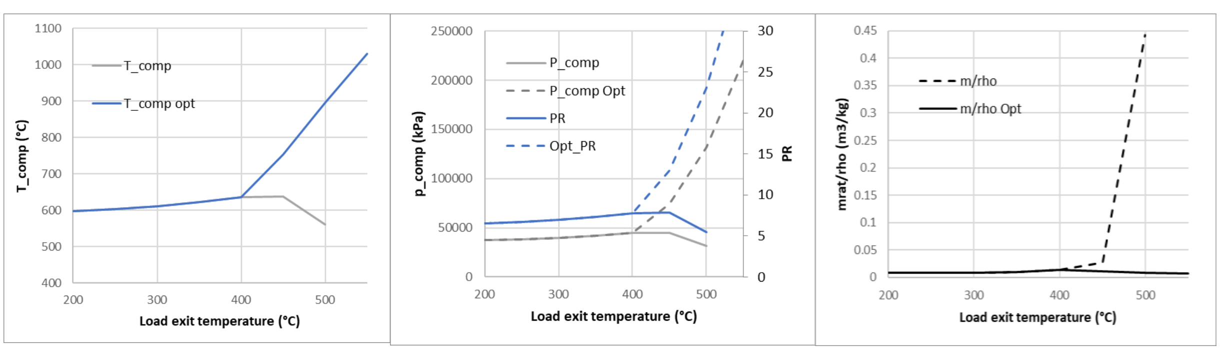

Figure 16. Parameters for technical assessment of the steam compression systems delivering 550°C sensible heating—temperature after compressor, pressure after compressor and compression ratio, and volumetric flow rate parameter.

The return temperature of the condensate after expansion and separation entering the intermediate heat exchanger is illustrated in Figure 17. The expansion step may produce subcooled liquid. Still over most of the range, the steam generator (SG) inlet temperature is not met, and additional features will be needed to address this issue.

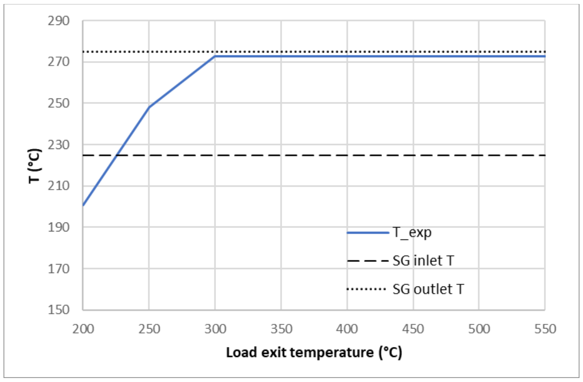

Figure 17. Return steam/condensate temperature after expansion for a load inlet temperature of 550°C.

3.1.1.2 Load entry temperature of 450°C

Similar results are obtained for a load entry temperature of 450°C. Figure 18 shows the T-s diagram and reveals the system is supercritical, as in the previous case, though with notably lower pressures and more realistic temperatures at the compressor outlet. The adjusted COP (Figure 19) is slightly higher for this case, compared to the case of a load entry temperature of 550°C, and the other parameters are also more favorable as well (Figure 20), except for the temperature of the return condensate after expansion. The temperature at that point is practically identical to that of the previous case and therefore is not shown in a separate figure.

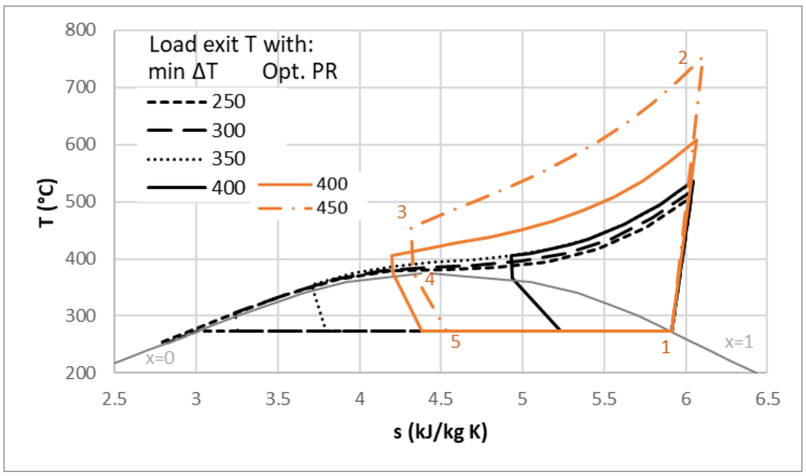

Figure 18. T-s diagrams of steam heat pumping at a load entry temperature of 450°C.

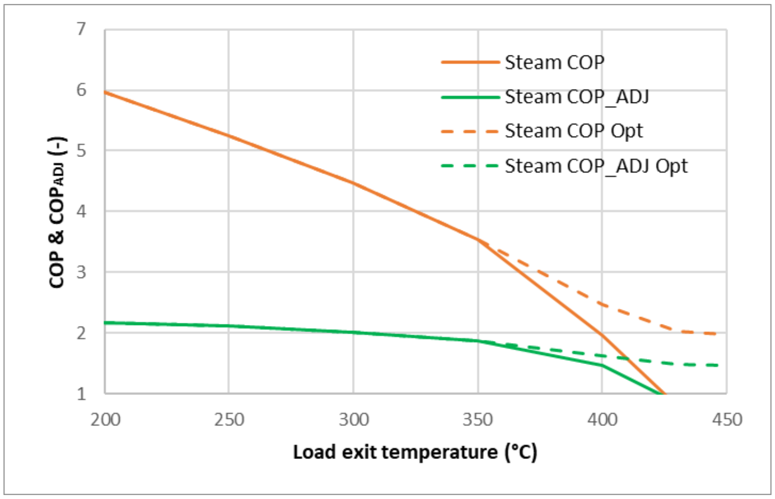

Figure 19. COP and COP_{ADJ} for the steam heat pumping with a load entry temperature of 450°C, including both pinch-point-driven and pressure-ratio-driven cases.

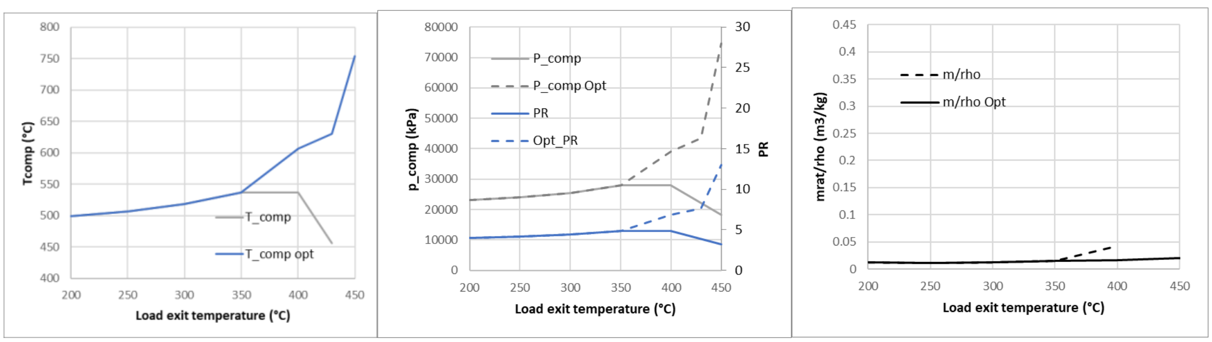

Figure 20. Parameters for technical assessment of the steam compression systems delivering 550°C sensible heating—temperature after compressor, pressure after compressor and compression ratio, and volumetric flow rate parameter.

3.1.1.3 Load entry temperature of 350°C

In this case, the temperature is sufficiently low that the steam is subcritical, as shown in Figure 21. The pinch point is at the point of saturated vapor, as seen in the Q-T diagram in Figure 22, which is also a reason why the system pressures differ for each case. Another important feature is the higher temperature difference (20 K) at the point of saturated vapor (about 25% of transferred heat) due to lower heat transfer coefficient from superheated steam before it starts condensing.

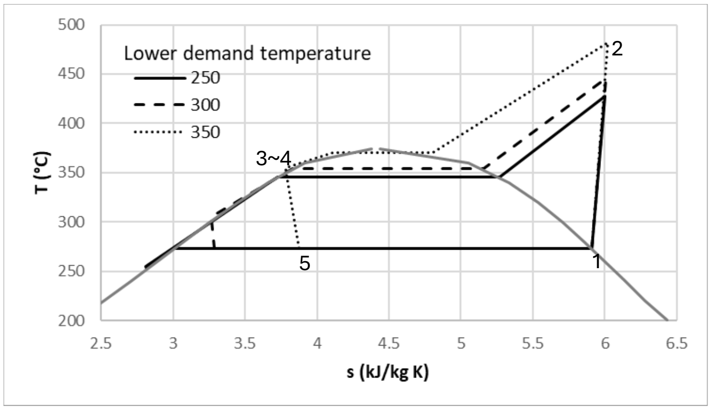

Figure 21. T-s diagrams of steam heat pumping with highest delivery temperature of 350°C.

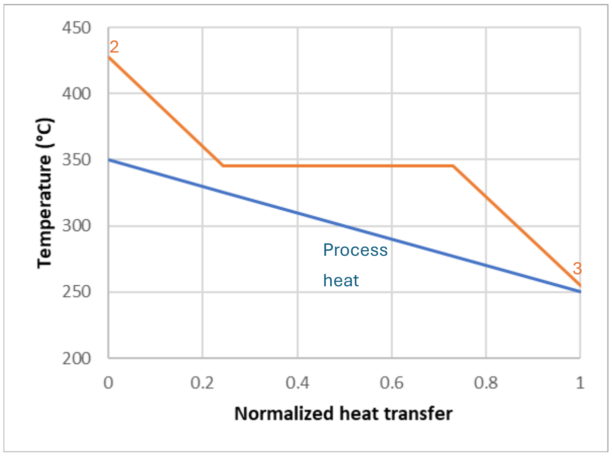

Figure 22. Q-T diagram of heat transfer from compressed steam to heat up a process to 350°C (sensible heating).

Because the temperature lift is low, the COP values are high, as shown in Figure 23. The adjusted COP values are substantially lower, as discussed above. The expected temperatures and pressures, which are technically attainable without difficulty, are presented in Figure 24.

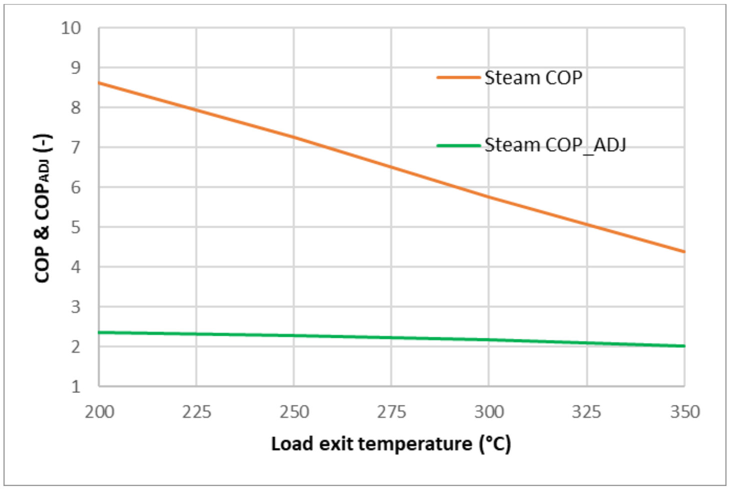

Figure 23. COP and COP_{ADJ} for steam heat pumping with load inlet temperature of 450°C, comparison between maintaining pinch point after compressor and results for optimized pressure ratio.

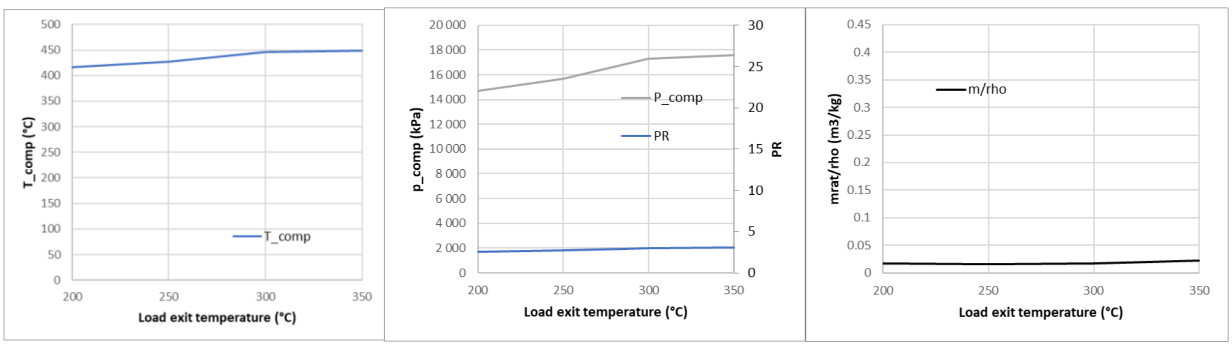

Figure 24. Parameters for technical assessment of the steam compression systems delivering 350°C sensible heating—temperature after compressor, pressure after compressor and compression ratio, and volumetric flow rate parameter.

3.1.1.4 Summary of steam results for sensible heating

Figure 25 summarizes the expected values of COP and COP_{ADJ} for the cases described above. All the ideal-case values of adjusted COP are below 2.5, even for load exit temperatures as low as 200 °C, indicating that the system economics could be challenging for these heat pump concepts.

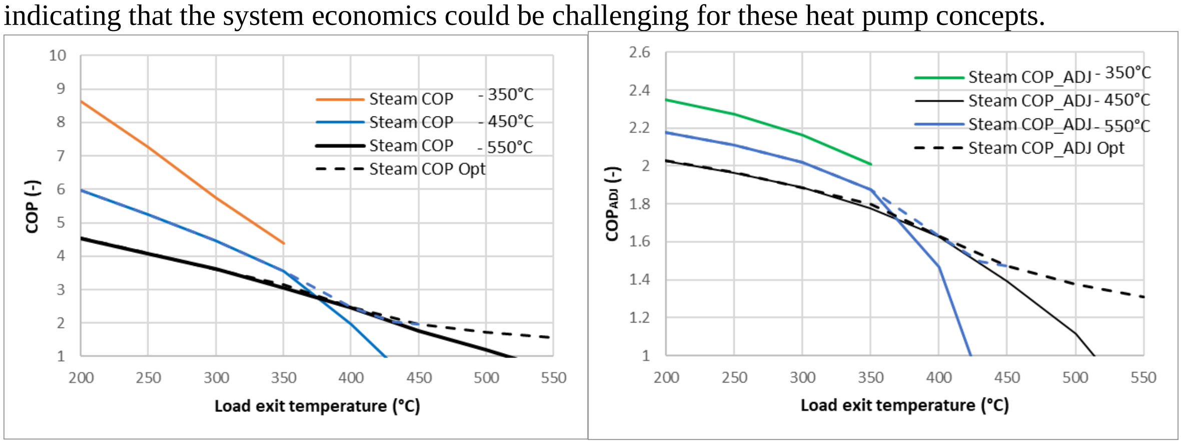

Figure 25. Summary of expected COP and COP_{ADJ} values for all steam compression system designs.

3.1.2 Uniform Temperature Heating

Uniform temperature heating offers a smaller range of options compared to sensible heating. The expected COP values are plotted with respect to temperature in Figure 26, which indicates the adjusted COP can be slightly higher for uniform temperature heating compared to sensible heating. The other key temperature and pressure parameters are shown in Figure 27. Supercritical pressures above approximately 30 MPa are very difficult to achieve safely and reliably in real systems, such that heating at temperatures hotter than about 370°C are not likely to be practical.

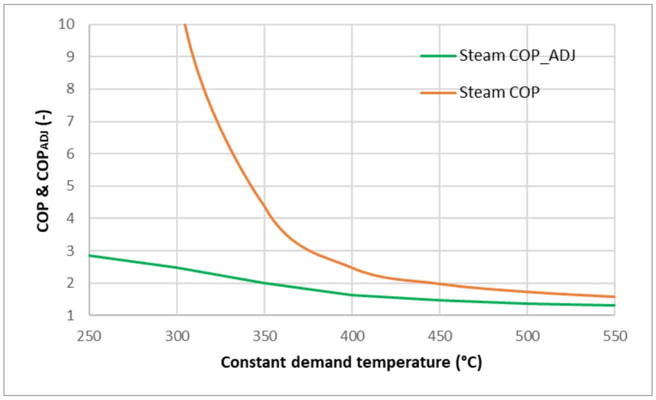

Figure 26. COP and COP_{ADJ} for steam compression system delivering heat to a load with uniform temperature heat demand.

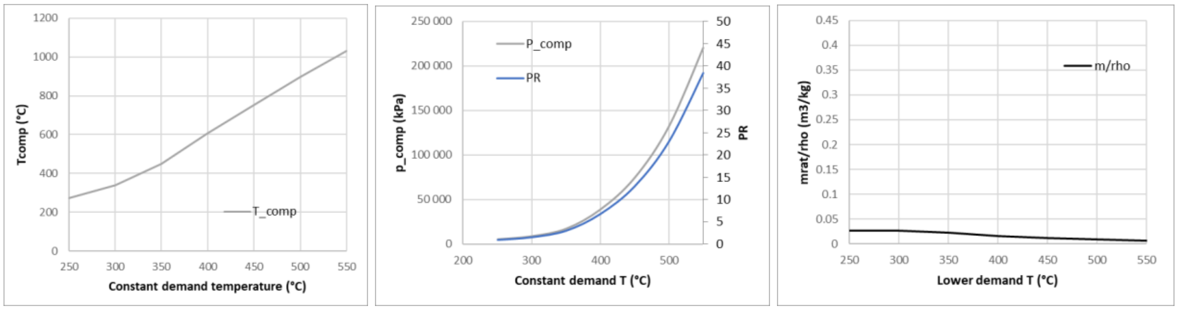

Figure 27. Technical assessment parameters of steam compression systems delivering to heat to loads with uniform temperature heat demand. Parameters include temperature after compressor, pressure after compressor and compression ratio, and volumetric flow rate parameter.

The results above are for optimized compression pressure ratios. To illustrate the performance at lower pressures, the impact of pressure ratio on COP for the case of 500°C heat delivery at uniform temperature is illustrated in Figure 28. Temperature and volumetric flow rate parameters are presented in Figure 29. Importantly, the pressures are greater than the potentially feasible maximum of approximately 30 MPa. Additional mitigation of the excessively high pressures can be achieved using split compression, as shown below.

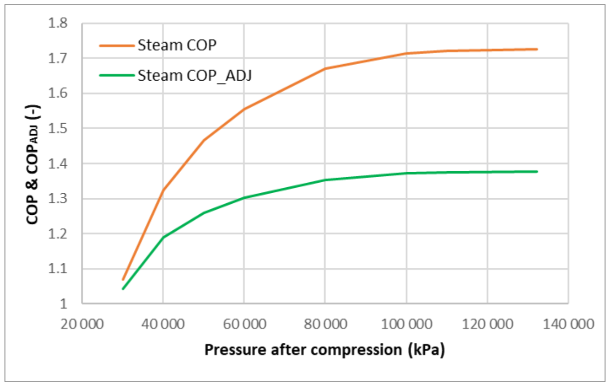

Figure 28. COP and COP_{ADJ} for steam compression system delivering heat to a uniform temperature load of 500°C at sub-optimal compression pressures.

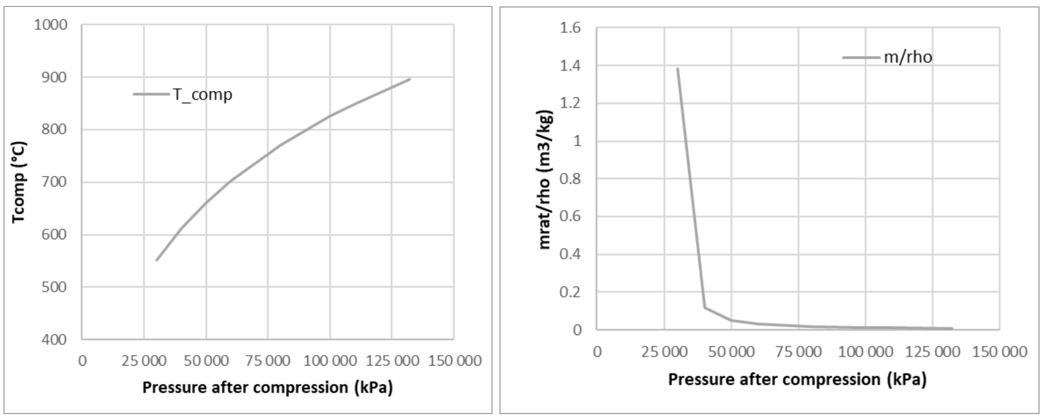

Figure 29. Parameters for technical assessment of the steam compression systems delivering to uniform temperature heating demand—temperature after compressor and volumetric flow rate parameter.

The expected COP values for different load heat exchanger temperatures are plotted in Figure 30, which also includes dashed lines that indicate the practical compressor pressure and temperature limitations of 30 MPa and 750°C. The practical upper limit of the adjusted COP is approximately 2 (1 unit of source energy to 1 unit of compressor power) at a load heat exchanger temperature slightly hotter than 400°C. Figure 31 shows key compressor temperature and pressure values, which further shows that practical temperature and pressure limitations have substantial impact.

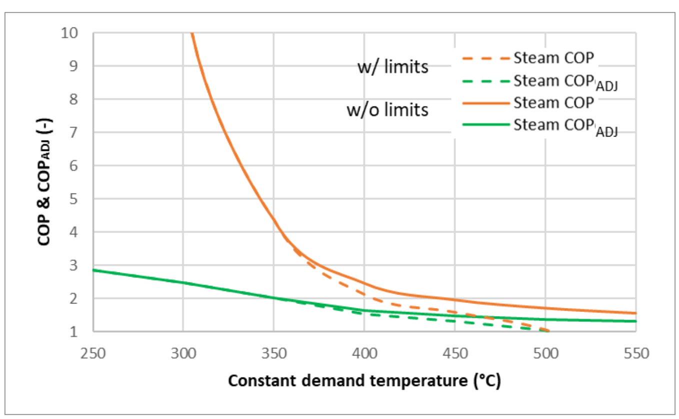

Figure 30. COP and COP_{ADJ} for steam compression systems delivering heat to a uniform temperature demand with and without consideration of practical maximum pressure and temperature limits.

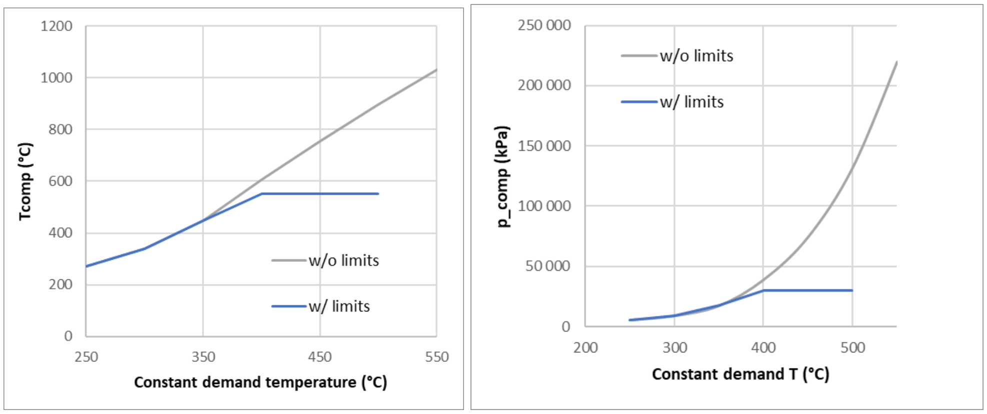

Figure 31. Parameters for technical assessment of the steam compression systems delivering to uniform temperature heating demand with and without maximum pressure and temperature limits.

3.1.3 Advanced Cycles: Uniform Temperature Heating

This section explores prospects on advanced cycle configurations using split compression and heat recuperation using a uniform heating temperature case. The associated T-s diagram ignoring practical pressure and temperature limitations are shown in Figure 32. Expected COP values for the advanced cycles are compared to those of the simple cycles in Figure 33 and show that the more sophisticated design yields marginal improvement in adjusted COP values with a maximum improvement of about 11% at the highest temperature. Expected values of the compressor temperature, pressure, and volumetric flow rate are shown in Figure 34. Again, the benefits of the more advanced cycle configuration are slight.

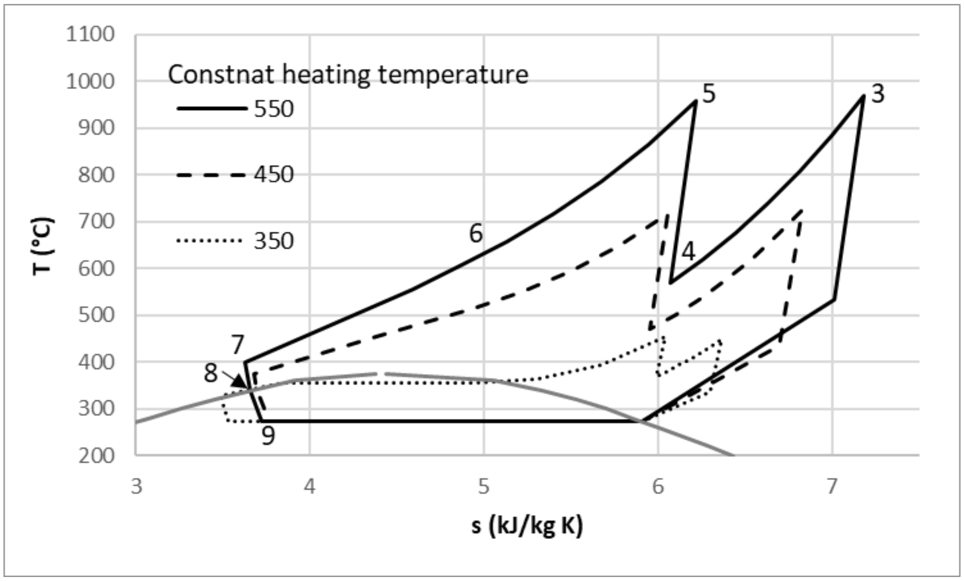

Figure 32. T-s diagrams of steam heat pumping using advanced configuration for uniform temperature profile of heat delivery.

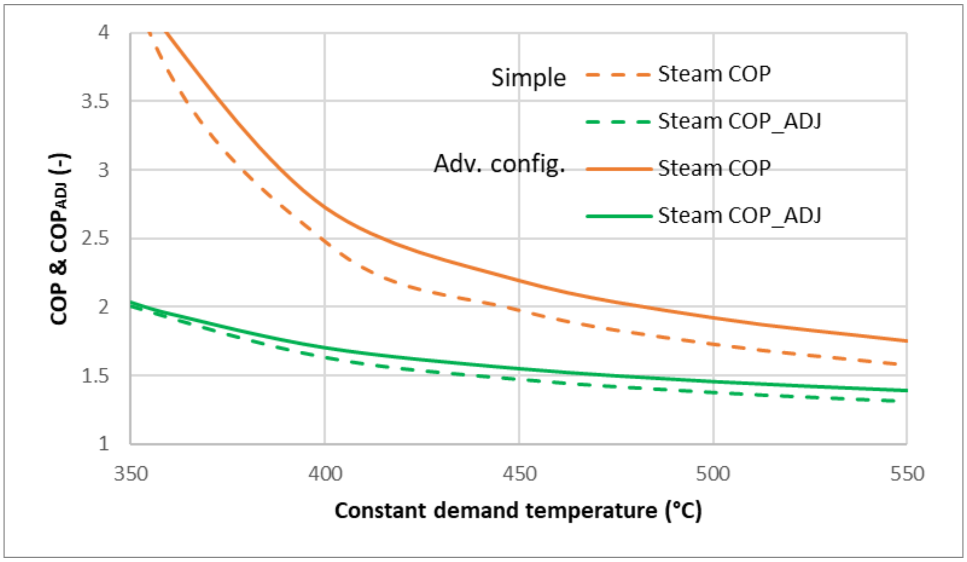

Figure 33. COP and COP_{ADJ} for steam compression system delivering heat to a uniform temperature demand without limits of maximal pressure and temperature—comparison of simple and advanced configuration.

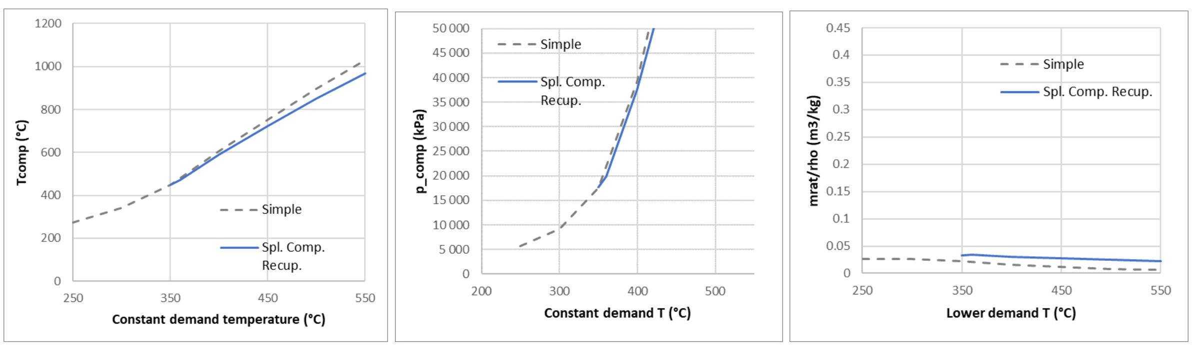

Figure 34. Parameters for technical assessment of the steam compression systems delivering to uniform temperature heating demand—comparison between simple and advanced configuration without any limits on parameters.

The impact of practical technical pressure and temperature limits of 30 MPa and temperature 750°C on the expected COP values are shown in Figure 35. The benefit of the advanced cycle compared to the simple cycle becomes clear as the demand temperature increases. The simple cycle cannot operate above approximately 500°C—while the advanced cycle can function up to the highest temperature of 500°C, although with unacceptably, poorly adjusted COP. The compressor outlet pressures are nearly identical for the simple and advanced configurations. Figure 36 shows the impact of the practical temperature and pressure limits on the other key parameters and shows that advanced design allows the volumetric flow rate to be substantially decreased for load exit temperatures greater than 450°C.

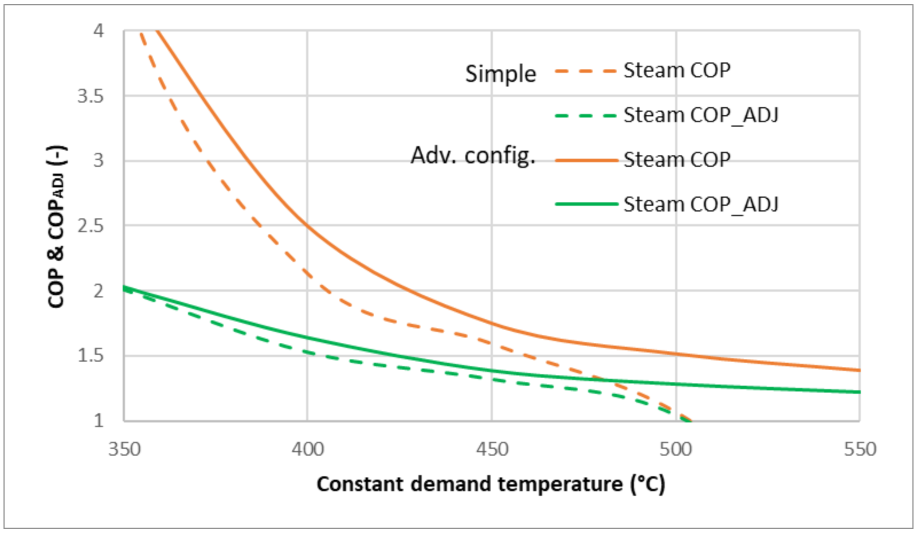

Figure 35. COP and COP_{ADJ} for steam compression system delivering heat to a uniform temperature demand with maximal pressure and temperature limits in place.

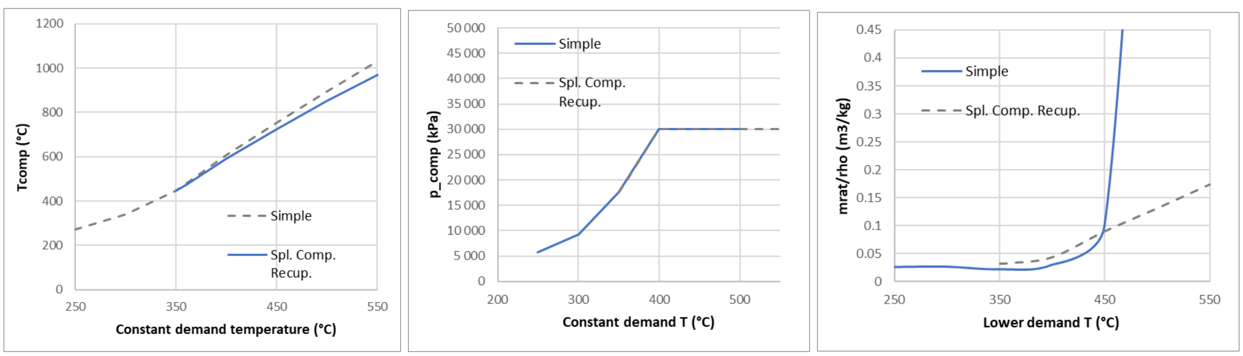

Figure 36. Parameters for technical assessment of the steam compression systems delivering to uniform temperature heating demand—comparison between simple and advanced configuration including practical compressor temperature and pressure limitations.

3.1.4 Mixed-mode Heating

The T-s diagram in Figure 37 illustrates the mixed-mode heating in which 20% of the heat is delivered as sensible heat between 200°C and a specified temperature. The heat delivery profiles are illustrated by Q-T diagrams in Figure 38 for two cases that include heat delivery from a supercritical fluid and heat delivery at subcritical steam pressure. The expected COP values, compared in Figure 39, show that mixed-mode heating offers the opportunity for slightly higher expected COP values than for uniform heating profile. For the lowest heating temperature of 250°C, there is no work needed and the adjusted COP reaches the maximal value of nearly 3. The technical parameters are very similar to those of the uniform temperature profile shown in Figure 31.

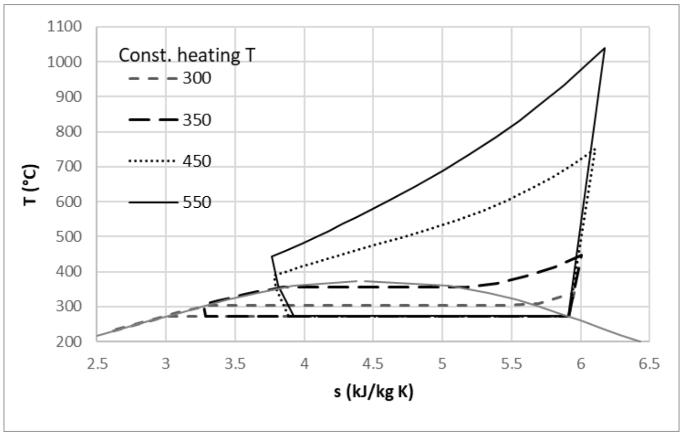

Figure 37. T-s diagrams of steam heat pumping for mixed-mode heat delivery.

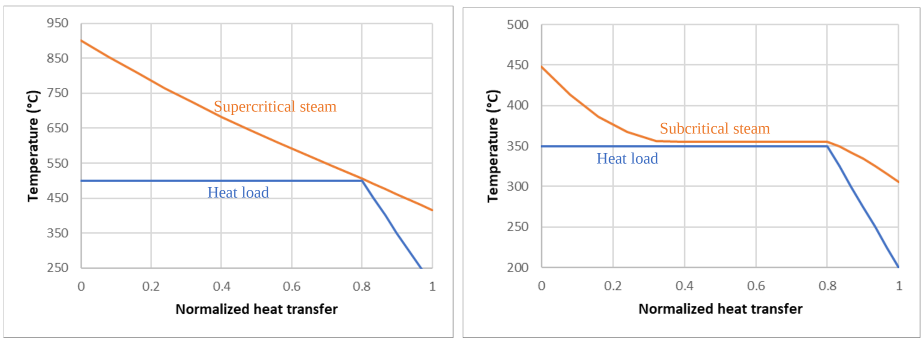

Figure 38. Q-T diagram of heat delivery for mixed-mode heat delivery with the uniform temperature portions of heat delivery at 500°C (heat delivery from supercritical fluid) and 350°C (from subcritical fluid).

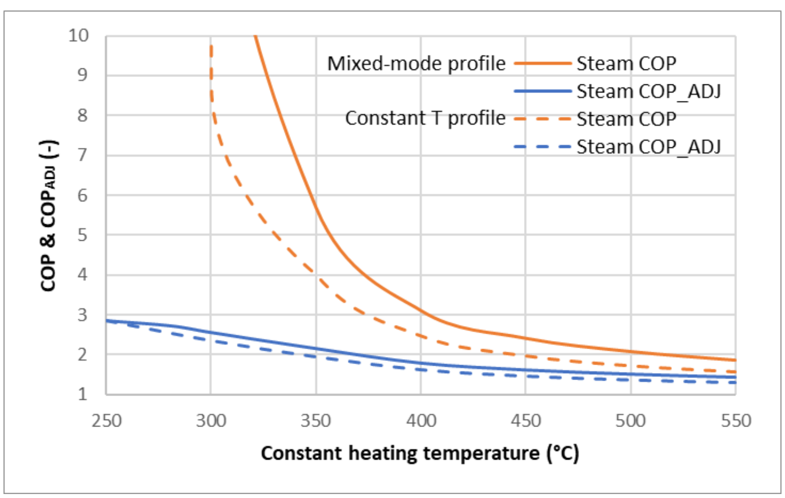

Figure 39. COP and COP_{ADJ} values for mixed-mode heat delivery using a steam compression cycle.

3.2 Brayton Cycle

3.2.1 Sensible Heating

3.2.1.1 Load entry temperature of 550°C

Similar to the summary on steam cycles, we first present results for Brayton cycle systems with sensible heat delivery at the highest demand temperature of 550°C. In Figure 40, the thermodynamic parameters of the Brayton cycle for these condition with air as the working fluid are illustrated in a T-s diagram. The Brayton cycle can be designed to have a minimal temperature difference at the hot end, represented by solid lines in Figure 40 to reduce the pressure ratio and temperature at the compressor outlet. For these cases, the hottest load exit temperature that is thermodynamically possible is approximately 430°C. As the load exit temperature for heat utilization becomes hotter, the thermodynamically optimal solution occurs at higher pressure ratios and thus higher compressor outlet pressures and temperatures. These Brayton cycles that are optimized by increasing the pressure ratio are represented by the dotted lines in Figure 40, including a case in which the load exit temperature is 480°C. The Brayton cycle can be designed to operate between these two extreme cases of minimal temperature difference at the hot end and optimized COP; however, those cases are not shown in Figure 40. Additionally, the two horizontal dashed lines show the steam generator inlet and outlet temperatures. Preferably, the gas exiting the expander should be no hotter than the steam generator inlet temperature plus a small amount to account for thermal losses and the required temperature difference across the intermediate heat exchangers.

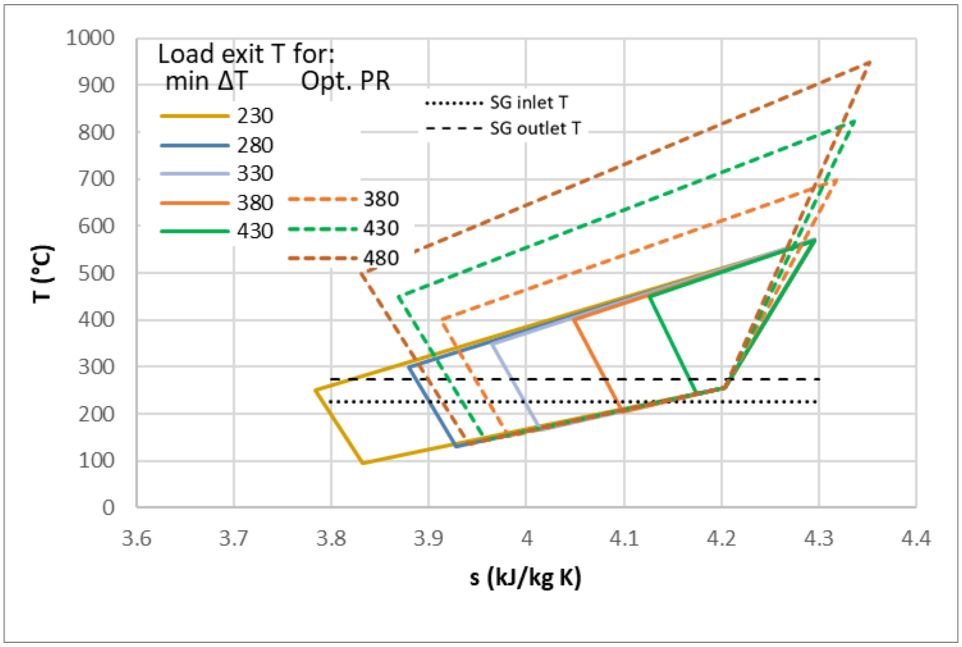

Figure 40. Diagrams of air BC for heat pumping with highest delivery temperature of 550°C.

Figure 41 plots expected COP and COP_{ADJ} for several gases as functions of load exit temperature and shows that the results for all explored gases exhibit a high degree of similarity. The fact that all the gases exhibit similar performance is expected because all of them are operating well within their superheated regions. Considering their close thermodynamic performance, the selection of a superior gas for nuclear heat pump applications will be based on other technical aspects, such as cost, density, and corrosion. Subsequent analyses below focus on air because it is abundant, inexpensive, and relatively unreactive.

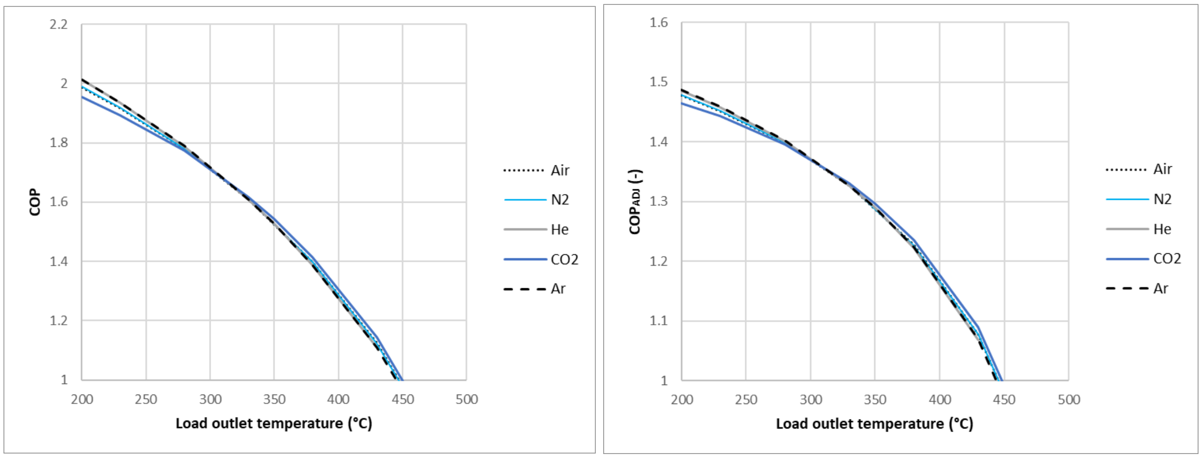

Figure 41. COP and COP_{ADJ} values as functions of load exit temperature for several gases with load inlet temperature of 550°C. For all cases, the pinch point is maintained after the compressor.

Figure 42 compares the expected COP values for cases with and without optimized pressure ratios. The expected COP values of pinch-point-limited solutions show a clear departure from those of the optimal pressure ratio cases for load exit temperature hotter than approximately 350°C. The maximum adjusted COP occurs at the coldest load exit temperature and is less than approximately 1.5.

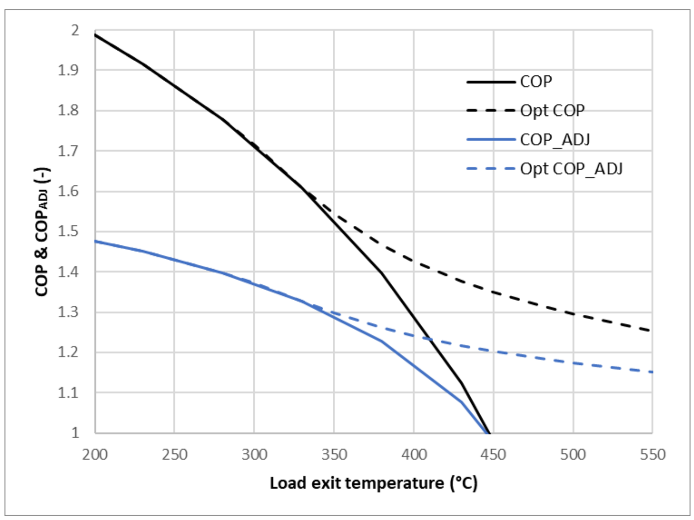

Figure 42. Expected COP and COP_{ADJ} values for a load inlet temperature of 550°C and air as the working fluid. Two cases are compared, including a case in which the pinch point is maintained after the compressor and a separate case with an optimized pressure ratio.

The calculated pressure ratios and volumetric flow rate parameters for a load inlet temperature of 550°C and for different gaseous working fluids are shown in Figure 43. The calculated values of the volumetric flow rate for cases with optimized pressure ratio have a maximum for all gases at approximately 330°C, while the volumetric flow rate monotonically increases with increasing load exit temperature for cases with a minimum temperature difference at the hot end.

For comparison, gas turbines commonly have pressure ratios of approximately 15-30, which give rise to high pressures up to approximately 30 bars and high temperatures up to approximately 500°C in compressors without intercooling. Achieving very high compression ratios for CO_2 might be also challenging because of its supercritical state properties.

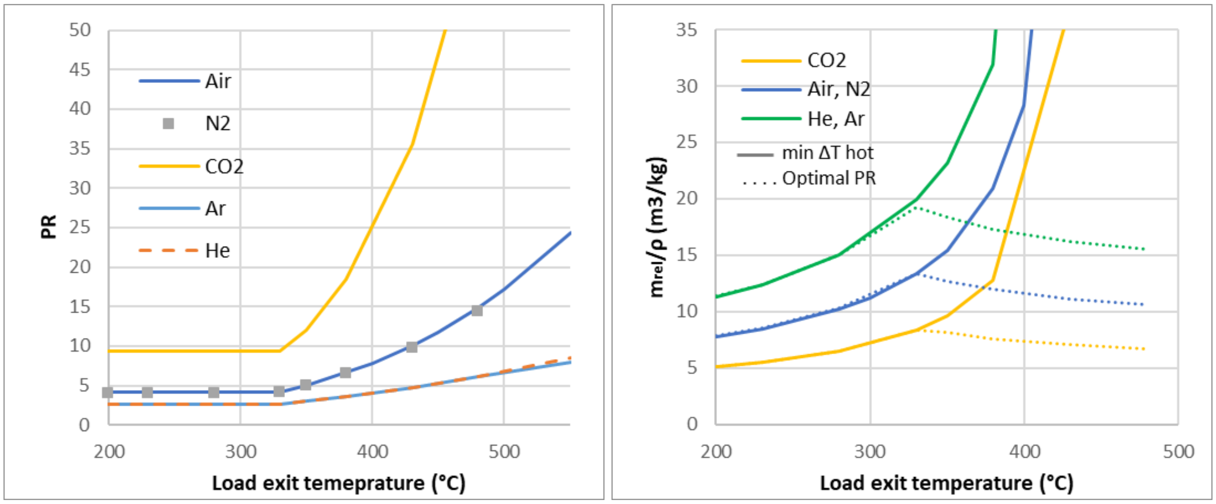

Figure 43. Thermodynamically optimal pressure ratios and volumetric flow rates for providing process heat at 550°C. The horizontal pressure ratio sections correspond to maintaining the pinch point after the compressor.

The air temperature downstream of the expander is shown in Figure 44 for cases in which the pinch point is maintained after the compressor. The air temperature downstream of the compressor increases as the load exit temperature becomes hotter than approximately 330°C because the pressure ratio increases. Compressor temperature hotter than 750°C are above typical feasibility limits.

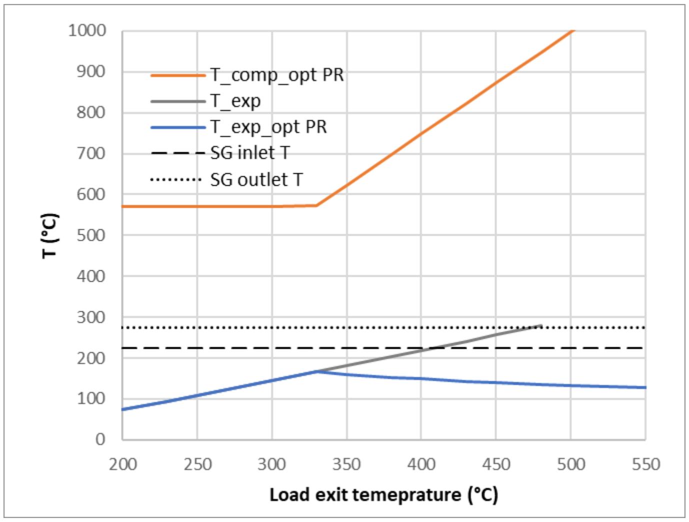

Figure 44. Air temperature after the expander for a case delivering heat at a load inlet temperature of 550°C.

3.2.1.2 Load inlet temperature of 450°C

Expected COP values for different fluids are nearly the same, so only COP values with air as the working fluid are presented. Figure 45 presents expected COP values for cases with the pinch point maintained after the compressor and with optimized pressure ratios. The expected COP values of pinch-point-limited solutions show a clear departure from those of the optimal pressure ratio cases for load inlet temperature hotter than approximately 300°C. The maximum adjusted COP occurs at the coldest load inlet temperature and is less than 1.6.

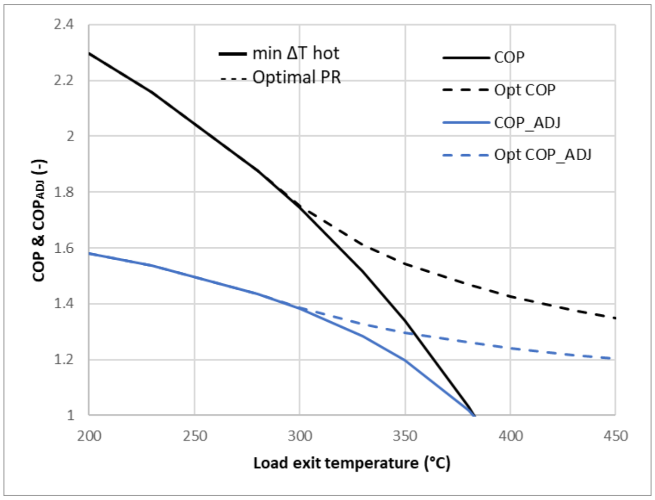

Figure 45. Expected COP and COP_{ADJ} for a load inlet temperature of 350°C with air as the working fluid. Two cases are compared, including a case where the pinch point is maintained after the compressor and a separate case with an optimized pressure ratio.

The calculated pressure ratios and volumetric flow rate parameters for a load inlet temperature of 450°C and for different gaseous working fluids are shown in Figure 46. The overall trends are the same as for the case with a load inlet temperature of 550°C presented in Figure 43. The calculated values of the volumetric flow rate for cases with optimized pressure ratio have a maximum for all gases at approximately 300°C, while the volumetric flow rate monotonically increases with increasing load inlet temperature for cases with a minimum temperature difference at the hot end.

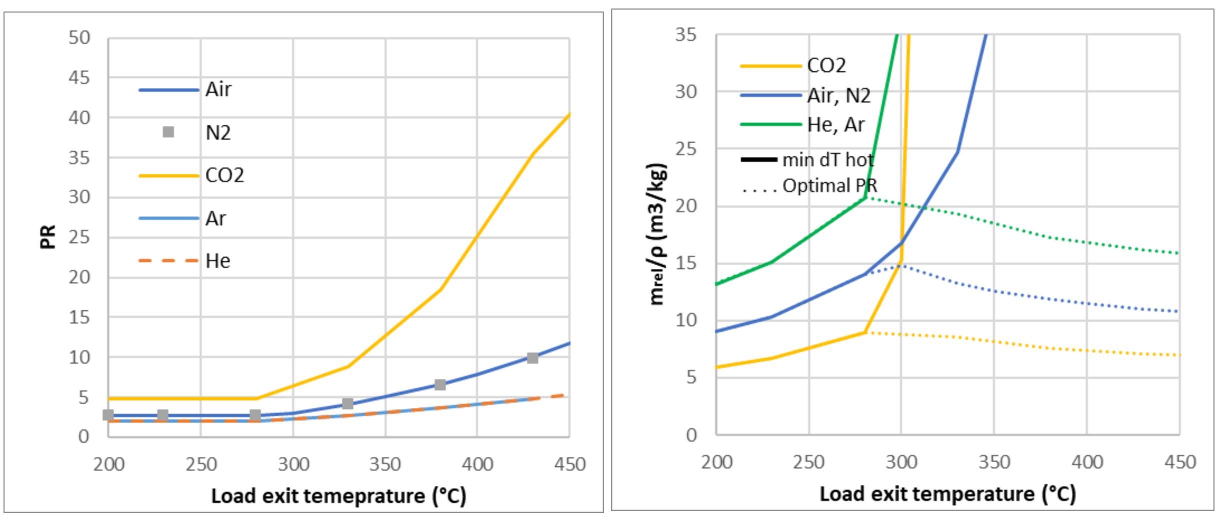

Figure 46. Thermodynamically optimal pressure ratio for providing process heat at 450°C (maintaining pinch point after compressor corresponds to the values of horizontal sections) (left) and a volumetric flow rate parameter (right), both based on the load inlet temperature.

3.2.1.3 Load entry temperature 350°C

Figure 47 presents expected COP values for cases with the pinch point maintained after the compressor and with optimized pressure ratios. The expected COP values of pinch-point-limited solutions show a clear departure from those of the optimal pressure ratio cases for load inlet temperature hotter than approximately 240°C. The maximum adjusted COP occurs at the coldest load inlet temperature and is less than approximately 1.8.

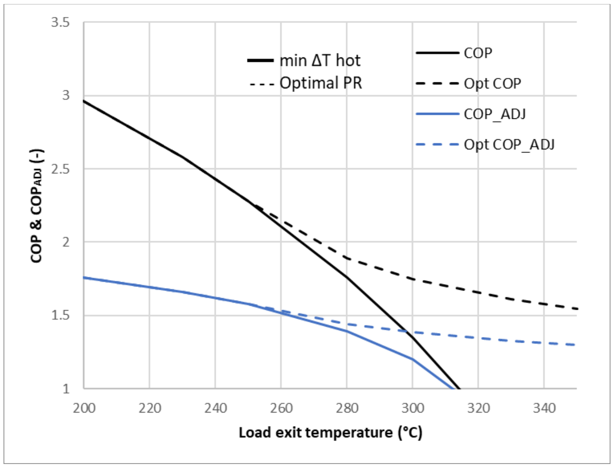

Figure 47. COP and COP_{ADJ} for the air with a load inlet temperature of 350°C, comparison between maintaining pinch point after compressor and results for optimized pressure ratio.

The calculated pressure ratios and volumetric flow rate parameters for a load demand temperature of 350°C and for different gaseous working fluids are shown in Figure 48. The overall trends are the same as for the cases with hotter load inlet temperatures shown above. The volumetric flow rate parameters for the optimized pressure ratio cases with CO_2 and air remain below approximately 11 m^3/kg and 25 m^3/kg, respectively, which are similar to those for the other cases with hotter load inlet temperatures shown above. It should be noted that these volumetric flow rate parameters are more than 20 times greater than those of the steam working fluid cases summarized in Section 3.1, such that far larger and more expensive pipes and pumps will be required. These extreme flow rates are required because heat transfer occurs between a gas temperature differential that is less than 50°C.

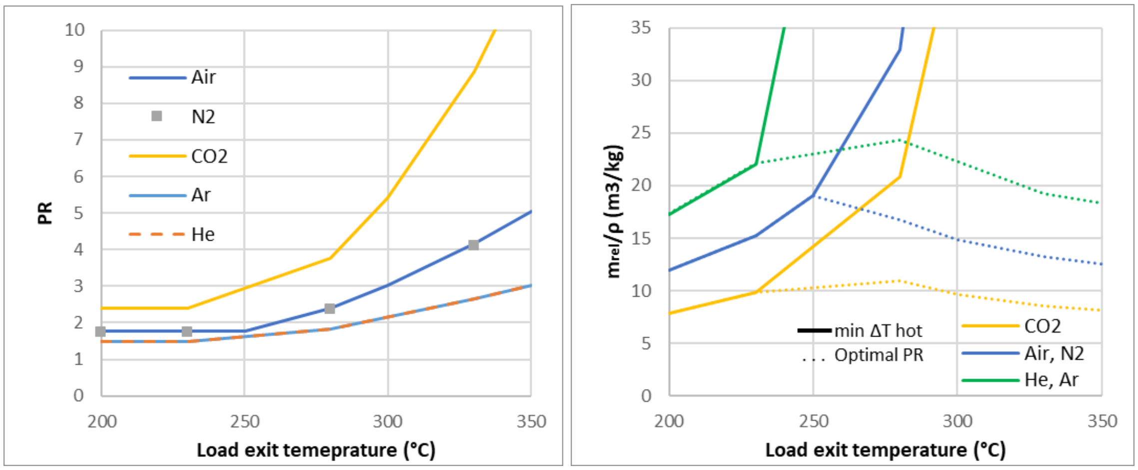

Figure 48. Thermodynamically optimal pressure ratios and volumetric flow rates for providing process heat at 450°C.

3.2.1.4 Summary

Figure 49 shows the expected COP values for cases with the three different inlet load temperatures considered above. The trends and key maximum values have already been discussed above. Plotting the data in this way does show that the deviation in the expected COP values between cases with optimal pressure ratios and cases in which the pinch point is maintained after the compressor occurs at lower temperatures as the load inlet temperature becomes hotter. Another important point is the adjusted COP values are all below 1.8, such that deploying these complex thermo-mechanical systems will need to be carefully assessed before a decision is made to proceed with construction.

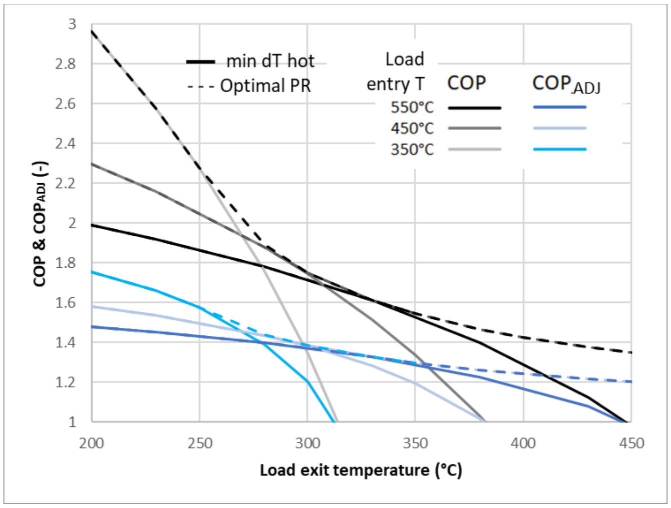

Figure 49. Combined curves of COP and COP$_{ADJ}$ for load inlet temperatures of 550°C, 450°C, and 350°C.

3.2.2 Uniform Temperature Heating

Figure 50 shows the expected COP values for cases with optimized pressure ratio to maximize the COP value and also includes sub-cases with and without practical pressure and temperature limitations of 750°C and 30 MPa at the compressor outlet. The temperature limit causes the COP and adjusted COP to underperform compared to the case without temperature limitations at a heating temperature of approximately 400°C. The adjusted COP with practical limitations ranges from approximately 1.6 at a heating temperature of 250°C (for which a heat pump is not necessary because heat at that temperature can be supplied by LWRs) down to just over 1 at 550°C. As discussed above, these low adjusted COP values indicate this process is likely not economically viable. Figure 51 shows the impact of the practical temperature and pressure limits on the other key parameters. The key impacts are that the compressor temperature and pressure ratio are held to below approximately 760°C and 8, respectively, and the volumetric flow rate parameter does not decrease below approximately 12m^3/kg.

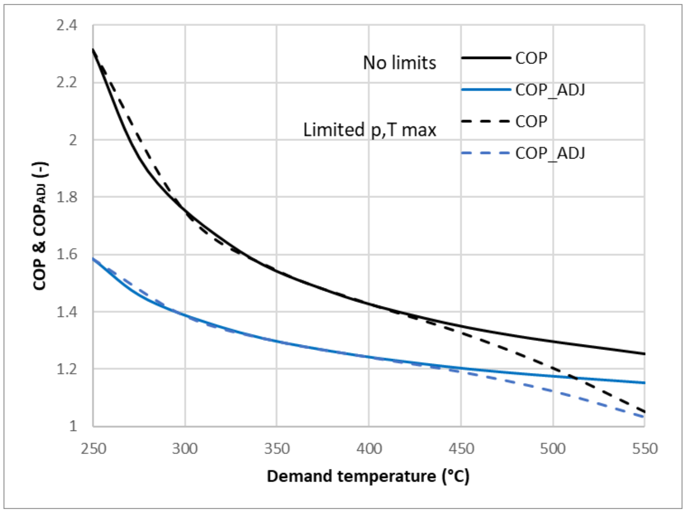

Figure 50. Expected COP and COP_{ADJ} for an air-based Brayton cycle system with uniform demand temperature, including with and without practical pressure and temperature limits.

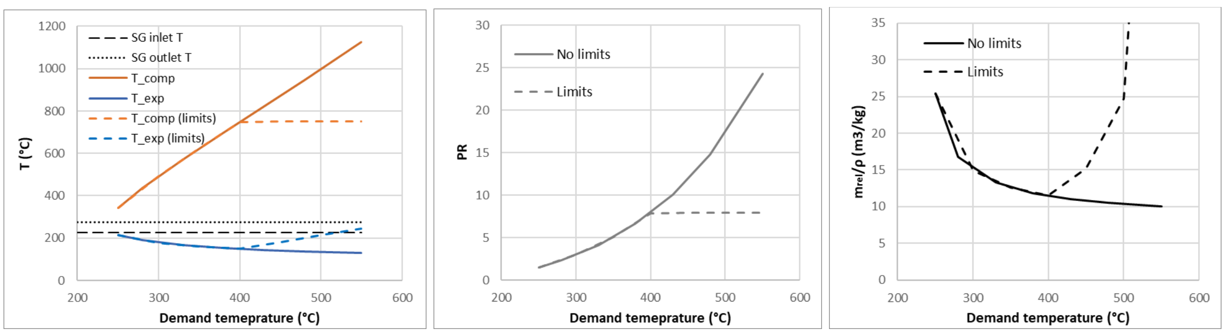

Figure 51. Parameters for technical assessment of the uniform heating temperature case in a Brayton cycle.

3.2.3 Advanced Cycles: Uniform Temperature Heating

A regenerated split compression Brayton cycle with a uniform heating temperature is compared to the corresponding simple case in Figure 52. The impact of practical temperature and pressure limits is also shown in Figure 52 and reveals that those limitations start to impact the performance at a heat delivery temperature of 450°C. At that condition, the adjusted COP of the simple cycle is approximately 1.2, while that of the advanced cycle is approximately 1.38. At a heat delivery temperature of 550°C, the adjusted COP of both the simple and advanced cycles decreases to just over 1 due to the practical temperature limitation. Figure 53 compares the other key parameters for the simple Brayton cycle with those of the regenerated split compression Brayton cycle. The thermodynamically optimal solutions of advanced cycles do not violate the imposed limit of 750°C over the entire explored range. In this case, the pressures depend on the arbitrary selected pressure levels of heat input and compressor inlet, which do not exceed 400 kPa. The case with split expansion has only marginal additional COP benefit as opposed to split compression single expansion cycle. It has slightly higher pressure ratio but also higher temperature after expansion, limiting the heat input to be only over low-temperature range and increasing mass flowrate. Due to the added complexity while only minimal benefit, this configuration is not pursued further.

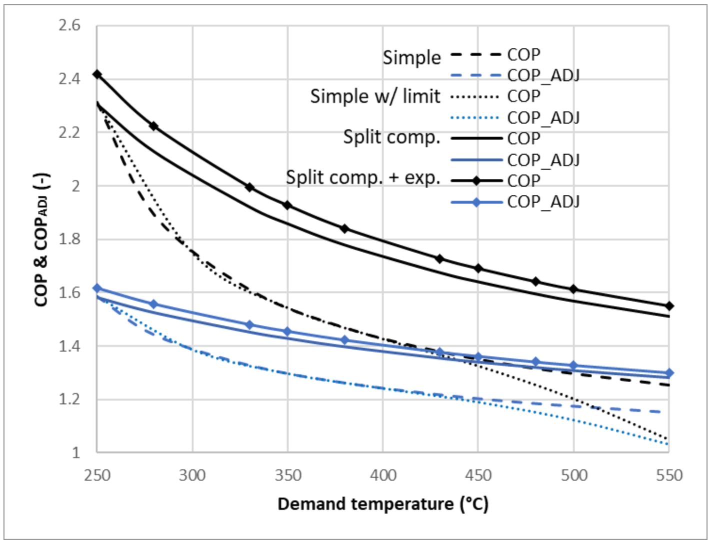

Figure 52. COP and COP_{ADJ} for Brayton system delivering heat to a uniform temperature demand without limits of maximal pressure and temperature—comparison of simple and advanced configuration.

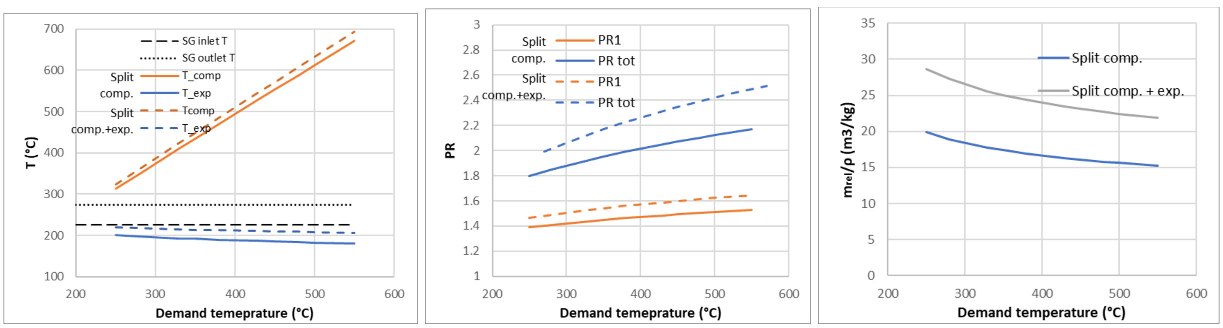

Figure 53. Parameters for technical assessment of the recuperated split compression (and eventually expansion) Brayton systems delivering to uniform temperature heating demand. Note that pressure and temperature limits were not reached.

3.2.4 Mixed-mode Heat Delivery in Brayton Cycles

Figure 54 compares the expected COP values for the simple Brayton cycle with heat delivery at uniform temperature to that of the same Brayton cycle operating with a mixed-mode heat delivery as introduced above. Similar to the cases with steam vapor compression, the Brayton cycle with mixed-mode heat delivery yields slightly higher adjusted COP values than the same cycle with heat delivery at uniform temperature; however, the highest adjusted COP value is still less than 1.8. Figure 55 shows the other key parameters for the simple Brayton cycle with mixed-mode heat delivery. The range of pressure ratio is slightly lower than for the constant temperature heating with simple cycle (Figure 51) but still very high when compared to advanced cycle configurations requiring less 2.6 for highest delivery temperatures (Figure 53).

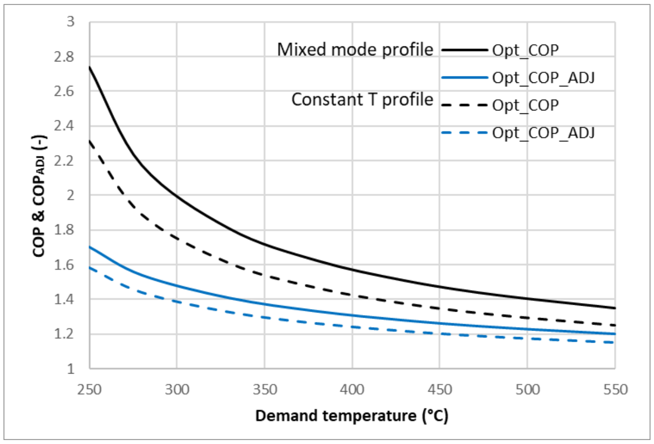

Figure 54. COP and COP_{ADJ} for air Brayton heat pump cycle based on the combination of uniform demand temperature and sensible heat up from 200°C.

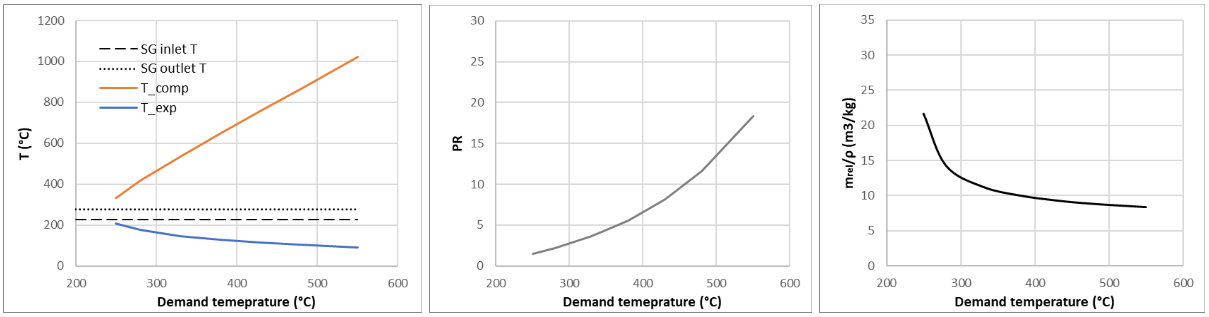

Figure 55. Parameters for technical assessment of the combination of uniform demand temperature and sensible heat up from 200°C.

3.3 Electric Boosting

For a convenient comparison, only a single point of load inlet temperature is considered here, which is 200°C (or lower). Return condensate in the nuclear plant enters the steam generator at a temperature of 225°C. In previous COP graphs, those would be points at their left edges. Uniform temperature heating cannot be provided at temperatures higher than the saturation temperature because of heat transfer/pinch point limitations. Sensible heating is considered only up to 450°C; heating at a higher temperature typically requires extreme and unpractical temperatures at the heater outlet. Already for demand at 300°C, the required steam temperature is nearly 850°C, which may cause material issues. COP and heater outlet temperatures are presented in Figure 56.

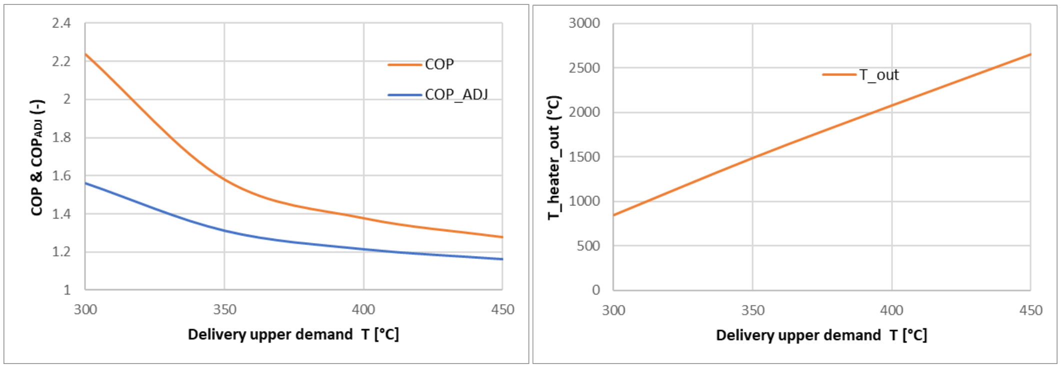

Figure 56. Electric heating COP and heater outlet temperature based on inlet load temperature for sensible heating (with load exit temperature of 200°C).

4. TECHNICAL FEASIBILITY AND SUPPLY CHAIN

For a preliminary assessment of technical feasibility, commercially available solutions for compression technology are briefly assessed in terms of the maximum practical pressure and pressure ratio cited above (maximum practical pressure and pressure ratio of 30 MPa and 16, respectively). An example of a technical solution in which natural gas combustion turbine technology is repurposed for a heat pump application is also provided.

4.1 Compressor Technology Overview

4.1.1 Steam Compressors

Steam compression is a mature, though relatively niche, technology applied within industries. Typical applications include industrial vapor recompression, such as steam recycling in pulp drying process [46].

There are several manufacturers of steam compressors. Spilling GmbH (Hamburg, Germany) uses piston technology with discharge pressures up to 6 MPa and delivers heat as hot as approximately 11 MWth per compressor unit. Other steam compressor manufacturers include Hanwha Power Systems (Seongsan-Gu, Korea) with a declared discharge pressure 1.9 MPa using centrifugal technology (a discharge pressure of 7.5 MPa is also mentioned) [47]. Atlas Copco (Greenville, South Carolina, United States) has a reference of a mechanical vapor recompression delivering 1.35 MPa steam [48] and the applied technology of integrally geared radial multi-stage compressors has a maximum of 20.5 MPa [49]. Radial steam compressors for a small temperature rise of the saturated steam in range of 6–24°C are also commercially available [50]. It is clear the required conditions are far from existing steam compressor technology, and a custom development based on a high-pressure compressor is needed.

Small piston high-pressure compressors can achieve 30 MPa [51] but work with fixed gases (typically air), and their mode of operation is more thermal compression than isentropic compression, which is less efficient. This operation tends to be intermittent, requiring periodic downtime for cooling. Looking at more robust and industrial-scale applications, centrifugal barrel configuration multi-stage compressors claim to have a limit around 70–100 MPa [52] [53]. For example, MAN Energy Solutions (Augsburg, Germany) supplies barrel compressors that can deliver up to 65 MPa [54]. With a maximal suction volumetric flow rate of 0.55 m^3/s, the heat delivery parameters in case of steam compression for uniform temperature delivery at 450°C correspond to delivered duty of 130 MWth. Even though steam is not specifically mentioned, specialty gases, including wet gases, are mentioned within requested options. Maximal temperatures are not specified and will likely result from material limitations. Generally, maximum pressures of radial compressors are considered to be approximately 70 MPa [53]. By direct consultation with several compressor manufacturers, temperatures above 400°C cause substantial issues, and a long-term dedicated development would need to be undertaken. While technically possible, the resulting cost may be multiple times the cost of compressors at lower temperatures.

4.1.2 Air and Generally Brayton Cycle Compressors

For Brayton cycles, the mass (and volumetric) flow rates are directly comparable to combustion gas turbine systems. For example, a split compression recuperated configuration delivering 100 MWth of heat at 500°C (uniform heating profile) has a gas mass flow rate of 900 kg/s, whereas a 450 MWe Siemens turbine SGT5-8000H has flue gas flow rate of 935 kg/s [37]. Unless a high-pressure and high-density gas (CO_2) is selected as a working fluid, axial turbo-compressor technology is needed to achieve the required flow rates. Compressor and expander technologies for heat pump heats at applicable nuclear conditions are feasible, although they are currently tailored to gas combustion turbines.

4.2 Heat Exchanger Technology Overview

Standard shell-and-tube heat exchangers are available for applications that use steam as the working fluid and that meet applicable process requirements for the intermediate heat exchanger. It is anticipated that other types of heat exchangers, such as compact heat exchangers and fin-and-tube heat exchangers for Brayton cycle systems, can be developed that meet the needed requirements for the intermediate heat exchanger. At the industrial site, the high pressures and temperatures may require specific, new technology. The shell-side pressure in shell-and-tube heat exchangers are limited by the shell-side thickness. On the tube side, solutions exist for pressures up to 100 MPa [55]. Printed circuit heat exchangers have similar upper pressure limits of 100 MPa and temperatures of 800°C or higher [56] [57]. Even though relevant technologies do exist, it should be remembered that they likely have very high capital costs.

4.3 Technical Overview of Explored Options

Table 1 provides an overview of parameters for heat pumping options for sensible heating applications with simple cycles and without limits on technical feasibility. Contrary to that, Table 2 provides same parameters for uniform temperature heating profiles, including practical limits for pressure and temperature. The tables are color-coded to indicate technical feasibility and compare COP values. Red indicates infeasible pressure or temperature and unacceptable COP, while green indicates possible conditions. It is clear steam compression cycles generally provide better performance than gas Brayton cycles; however, when technical limitations are considered, their relative performance is similar.

Table 1. Performance and technical parameters for heat pumping options delivering heat to sensible heat demand (simple configuration, without considering practical temperature and pressure limits).

	550°C		450°C		350°C	
Steam compression	min	max	min	max	min	max
Pressures (MPa)	37	220	23	75	15	17.5
Temperatures (°C)	600	1030	500	750	420	450
Volume flow	Ok	Ok	Ok	Ok	Ok	Ok
Return T	Ok (low T)	Complications	Ok (low T)	Complications	Ok (low T)	Complications
Readily available HX	PCHE	n/a	S&T/PCHE	PCHE	S&T	S&T/PCHE
Readily available comp	Near limits	n/a	Issues	limits	Possible	Potentially issues
COP_{ADJ}	1.3	2	1.5	2.2	1.95	2.35
Air Brayton						
Pressures (MPa)	1	6	0.7	3	0.5	1.3
Temperatures (°C)	570	1120	470	870	370	620
Volume flow	High	High	High	High	High	High
Return T	Ok	Ok	Ok	Ok	Ok	Ok
Readily available HX	Fin & tube	Fin & tube	Fin & tube	Fin & tube	Fin & tube	Fin & tube
Readily available comp	Yes	Temp beyond limits	Yes	Temp issues	Yes	Temp limits
COP_{ADJ}	1.0	1.5	1.20	1.6	1.3	1.8

Table 2. Performance and technical parameters for heat pumping options delivering heat to uniform heat demand profile by simple and advanced configuration, including practical temperature and pressure limits. Reaching 550°C using a steam compression cycle requires exceeding practical temperature and pressure limits.

	550°C		450°C		350°C	
Steam compression	Simple	Advanced	Simple	Advanced	Simple	Advanced
Pressures (MPa)	n/a	30	30	30	17.6	17.6
Temperatures (°C)	n/a	750	630	551	450	450
Volume flow	n/a	Ok	Ok	Ok	Ok	Ok
Return T	n/a	Complications	Complications	Complications	Complications	Complications
Readily available HX	n/a	PCHE	PCHE	PCHE	S&T/PCHE	S&T/PCHE
Readily available comp	n/a	limits	limits	limits	Potentially issues	Potentially issues
COPADJ	n/a	1.22	1.32	1.39	2.01	2.03
Air Brayton						
Pressures (MPa)	2.0	0.4	2.0	0.4	1.3	0.4
Temperatures (°C)	750	670	750	550	620	440
Volume flow	High	High	High	High	High	High
Return T	Ok	Ok	Ok	Ok	Ok	Ok
Readily available HX	Fin & tube	Fin & tube	Fin & tube	Fin & tube	Fin & tube	Fin & tube
Readily available comp	Temp limits	Temp limits	Temp limits	Yes	Temp limits	Yes
COPADJ	1.0	1.3	1.2	1.3	1.3	1.5

For Brayton air cycles, there are immediate technical advantages due to similarities with natural gas combustion turbines. Figure 57 illustrates a potential turbine conversion for a Brayton cycle heat pump application. Considering that the lowest pressure in the system would be higher than atmospheric pressure, the first several stages of the compressor would not be needed. The inlet temperature would correspond to the design air temperature at the given compressor stage, further aligning the need for only minimum modifications to the existing equipment. Compressed air at highest temperature would provide delivery heat to the industrial load. The temperatures before the expander would be notably lower than for current combustion gas turbines, so the expansion turbine stages would be redesigned. It is anticipated that only 3–5 aerodynamic stages would be needed and simpler design for expansion could be used. These modifications would have relatively low-cost impacts. Finally, air inlet and exhaust would be connected with the air passing through a heat input exchanger using heat from the NPP.

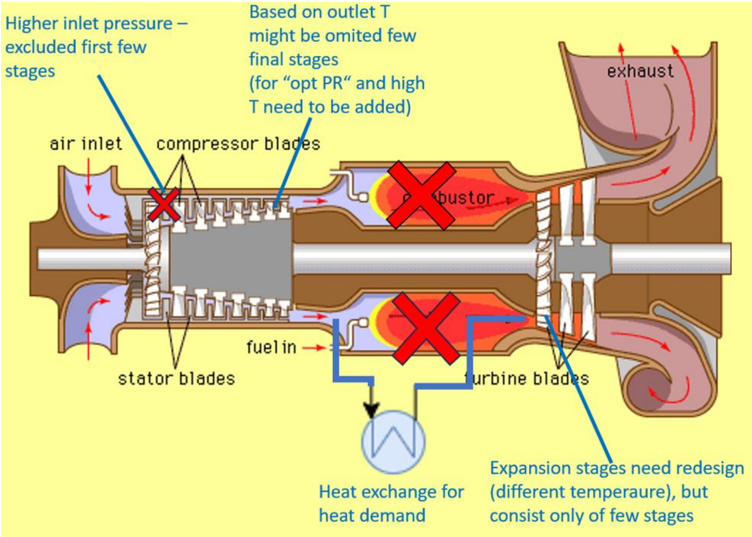

Figure 57. Simplified drawing of options to modify a gas turbine system for a Brayton cycle heat pump (modified from Reference [58]).

5. ECONOMIC RESULTS – FEASIBILITY COMPARISON WITH ELECTRIC HEATING

To estimate the economic feasibility, we calculate the minimal system COP that achieves similar NPV as direct electric heating. The results are plotted in Figure 58, which presents the system COP_{ADJ} values that are needed achieve economic parity with electric heating for different assumed heat-pump CAPEX values as a functions of electricity sales price. Three of the considered cases neglect system thermal losses, and one case assumes a heat-pump system CAPEX cost of $100/kWe where thermal losses are 10% of the heat that is delivered by the system. The value of 10% thermal losses is considered reasonable, and it is noted that some sources suggest thermal losses can be as high as 20% in steam systems with long lines [59]. Assuming a CAPEX cost of $1,000/kWe, the heat pump with uniform heating becomes competitive at electricity prices above about $60/MWh, based on a COP_{ADJ} of approximately 1.3 for an advanced Brayton cycle as listed in Table 2. For a case of low electricity prices in the range of $30/MWh, the system COP_{ADJ} must exceed 1.5 if 10% thermal losses are considered.

There is clearly a range of electricity prices for which nuclear heat-based heat pumps could be economical. Still, a careful consideration of the feasibility range and potential benefits needs to be performed before conclusions are drawn. Additionally, unless the heat pump is developed as modular system, providing same configuration and parameters as a solution across multiple customer types and locations, the customary approach, especially for compressor-expander assembly, might further increase costs and reduce reliability. Electrical heating has large advantage in modularity as well as maximal temperatures.

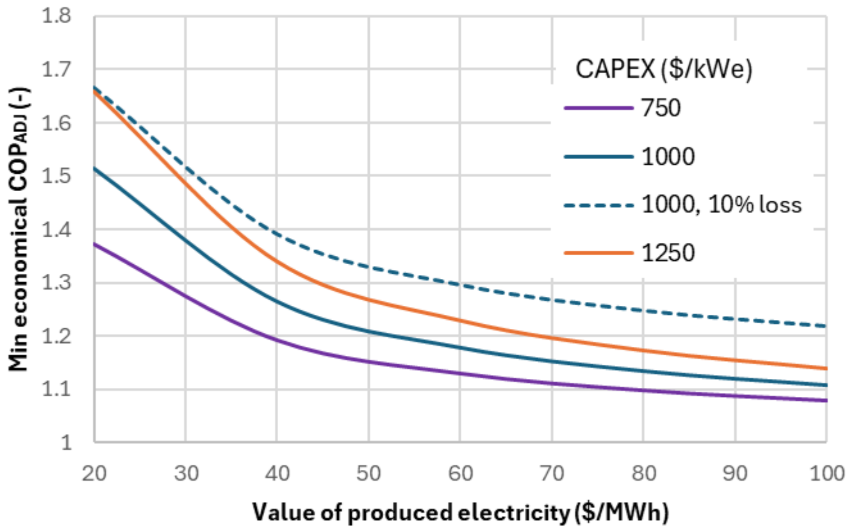

Figure 58. Minimum required COP_{ADJ} for a heat pump delivering heat at a uniform temperature of 550°C to achieve economic parity with direct electric heating as a function of electricity cost.

6. FUTURE WORK AND PROSPECTIVE CONCEPTS

Future work could focus on developing one or several illustrative case studies or sizing examples for heat pump systems, considering specific industrial heat demand scenarios. Within such a case, more detailed cost estimates may be developed, and system risk and reliability impact can be included. This would involve considering the costs of equipment, installation, and operation. This would help to understand and compare the benefits and drawbacks of the heat pumping approach.

Estimates of the levelized cost of heat including the nuclear operation could then provide further insights into the overall cost-effectiveness of heat pump systems compared to conventional natural gas and thermally decoupled electrified heating systems.

Further thermodynamic analysis could be extended to include multiple ranges of technical limits to all explored options (i.e., also for sensible and combined heating profiles). This would involve assessing the performance and efficiency of the system under different operating conditions to ensure optimal design and operation.

On the other hand, one further approach recommended for future work is to use the heat pumps for pumped thermal energy storage, allowing flexible operation of the nuclear plant. The principle is in upgrading heat, storing it as a high-temperature sensible heat, and discharging in a dedicated power cycle for power boosting, as illustrated in Figure 55. This approach has a potential to overcome the drawback resulting from thermodynamic irreversibility resulting from heat transfer between phase changing steam and sensible heat storage, which generally limits application of storage options for LWR systems.

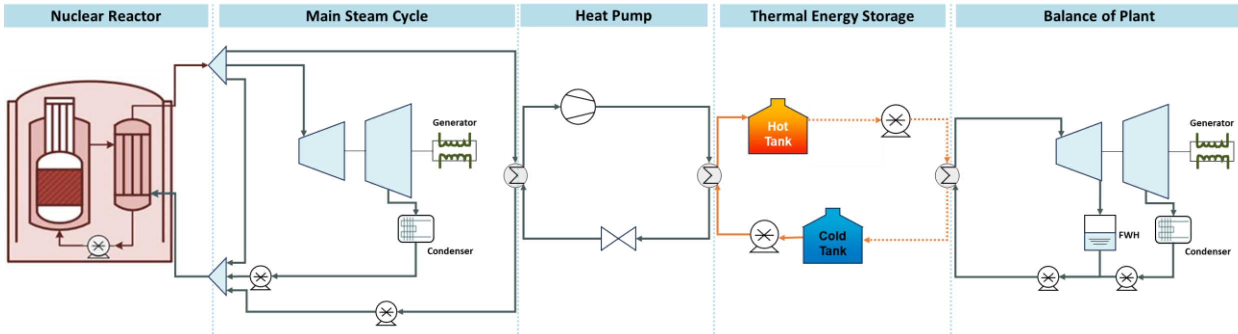

Figure 59. Conceptual layout of using heat pumping approach coupled to LWR for operation as a pumped thermal energy storage system.

7. CONCLUSION

The analysis presented in this study provides valuable insights into heat upgrade options for existing and future LWR plants, focusing on technical and thermodynamic aspects. By applying a pinch point approach and considering various performance indicators for heat pumps and heat engines, we have evaluated the feasibility and efficiency of different configurations. Since the heat source for the heat pump originates from a nuclear plant and may be used to generate electricity, an adjusted coefficient of performance (COP_{ADJ}) is developed which accounts for reductions in nuclear power production due to providing heat to the heat pump system. Heat delivery is considered for two different types of heat sinks, including sensible heating and uniform temperature heating (e.g., providing heat for a chemical reaction).

For the steam compression systems, it was found that while COP values can be relatively high, especially at lower heat delivery temperatures, technical feasibility becomes a limiting factor at higher temperatures due to extreme required pressures and temperatures. The use of advanced configurations, such as split compression and recuperation, shows potential for improving performance, but challenges remain in terms of system complexity and cost.

The Brayton cycle offers an alternative with advantages in terms of simpler operation and potentially lower costs for certain temperature ranges. While the Brayton cycle has inferior performance at low heat demand temperatures, split compression recuperated cycle slightly surpasses steam cycle for temperatures above 470°C, while operating at only moderate pressures and pressure ratios. At the maximal (uniform) delivery temperature of 550°C, the COP_{ADJ} is just below 1.3.

The study highlights the importance of considering both thermodynamic performance and technical feasibility when selecting heat upgrade options for LWR plants. Additionally, a simplified economic comparison with electrical heating provides a guidance on economic feasibility. Based on the value of clean nuclear electricity of $30/MWe, the COP_{ADJ} needs to be higher than 1.7 for heat delivery at a uniform temperature of 390°C, while as a COP_{ADJ} value as low as 1.2 is sufficient for an electricity sales price of $100/MWhe. In either case, direct electrical heating provides benefits in terms of modularity, known reliability, technical simplicity and lower upfront CAPEX. Together with limited actual industrial demand in the 300–500°C range and potential issues with actual heat delivery to the industrial process, caution is recommended before pursuing deploying heat pumps for nuclear heat-based applications.

8. REFERENCES

[1] J. A. M. Porto, L. D. J. Alves, F. C. Nunes, E. Gross, H. H. S. Medrado, J. d. S. Rocha, M. M. Viana, A.. P. d. C. Teixeira and M. N. V. Prasad, "Chapter 6 - Nuclear Energy and its Role in Decarbonization: Scenarios and Perspectives," in Decarbonization Strategies and Drivers to Achieve Carbon Neutrality for Sustainability, Amsterdam, Netherlands, Elsevier, 2024, pp. 97-120.

[2] K. Tapia-Ahumada, M. Y. Reilly and K. Strzepek, "Deep Decarbonization of the U.S. Electricity Sector: Is There a Role for Nuclear Power? (Joint Program Report Series Report 33)," Center for Sustainability Science and Strategy, MIT, Cambridge, MA, USA, 2019.

[3] B. Hallbert, C. Barnard, C. A. Primer, R. D. Boardman, S. Lawrence, X. Chen, D. M. Osborn and J. L. Vollmer, "Chapter 12 - Light Water Reactor Sustainability Program - Enabling the continued operation of existing US nuclear reactors," in Risk-Informed Methods and Applications in Nuclear and Energy Engineering - Modeling, Experimentation, and Validation, Washington, DC., USA, Academic Press, 2023, pp. 249-257.

[4] K. A. McCarthy and B. J. Adams, "Research, Development and Demonstration (RD&D) Needs for Light Water Reactor (LWR) Technologies (INL/EXT-16-38305)," Idaho National Laboratory, Idaho Falls, ID, USA, 2016.

[5] M. Lipka and A. Rajewski, "Regress in Nuclear District Heating. The need for rethinking cogeneration," Progress in Nuclear Energy, vol. 130, p. 103518, 2020.

[6] IAEA, "Opportunities for Cogeneration with Nuclear Energy - IAEA Nuclear Energy Series (No. NP-T-4.1)," IAEA, Vienna, Austria, 2017.

[7] A. Al-Othman, N. N. Darwish, M. Qasim, M. Tawalbeh, N. A. Darwish and N. Hilal, "Nuclear Desalination: A State-of-the-art review," Desalination, vol. 457, no. 1, pp. 39-61, 2019.

[8] CEZ Group, "Emission-free heat is already flowing from Temelin to Ceske Budejovice," CEZ Group, 18th Oct. 2023. [Online]. Available: https://www.cez.cz/cs/pro-media/tiskove-zpravy/bezemisni-teplo-uz-proudi-z-temelina-do-ceskych-budejovic-183168. [Accessed 11th Aug. 2024].

[9] I. Oliker, "District Heating Supply from Nuclear Power Plants," POWER, 1st Feb. 2022. [Online]. Available: https://www.powermag.com/district-heating-supply-from-nuclear-power-plants/. [Accessed 11th Aug. 2024].

[10] NuScale Power, "A Preliminary Assessment of NuScale Steam Production Rates for Industrial Applications (WP-139434, Rev 3)," Office of Technology, NuScale Power, Portland, Oregon, 2023.

[11] WNN, "Start up of Chinese industrial nuclear steam project begins," WNN, 4th Mar. 2024. [Online]. Available: https://www.world-nuclear-news.org/Articles/Start-up-of-Chinese-industrial-nuclear-steam-proje. [Accessed 11th Aug. 2024].

[12] World Nuclear Association, "Interview: Zhang Xianggui, Deputy Plant Manager of Units 3&4, CNNC Jiangsu Nuclear Power Station," World Nuclear Association, [Online]. Available: https://world-nuclear.org/our-association/publications/world-nuclear-performance-report/tianwan-nuclear-power-plant-steam-supply. [Accessed 11th Aug. 2024].

[13] Nuclear Engineering International, "China launches nuclear steam project at Tianwan," Nuclear Engineering International, 6th Mar. 2024. [Online]. Available: https://www.neimagazine.com/news/china-launches-nuclear-steam-project-at-tianwan-npp-11576695/. [Accessed 11th Aug. 2024].

[14] K.-M. Adamson, T. G. Walmsley, J. K. Carson, Q. Chen, F. Schlosser, L. Kong and D. J. Cleland, "High-temperature and transcritical heat pump cycles and advancements: A review," Renewable and Sustainable Energy Reviews, vol. 167, p. 112798, 2022.

[15] C. Arpagaus, F. Bless, M. Uhlmann, J. Schiffmann and S. S. Bertsch, "High temperature heat pumps: Market overview, state of the art, research status, refrigerants, and application potentials," Energy, vol. 152, no. 1, pp. 985-1010, 2018.

[16] J. Spale, A. J. Hoess, I. H. Bell and D. Ziviani, "Exploratory Study on Low-GWP Working Fluid Mixtures for Industrial High Temperature Heat Pump with 200 °C Supply Temperature," Energy, p. 132677, 2024.
[17] Industrial Efficiency & decarbonization Office, US Department of Energy, "Manufacturing Energy and Carbon Footprints (2018 MECS)," Industrial Efficiency & decarbonization Office, US Department of Energy, Dec. 2021. [Online]. Available: https://www.energy.gov/eere/iedo/manufacturing-energy-and-carbon-footprints-2018-mecs. [Accessed 11th Aug. 2024].
[18] C. McMillan, "Manufacturing Thermal Energy Use in 2014," National Renewable Energy Laboratory, 2019. [Online]. Available: https://data.nrel.gov/submissions/118. [Accessed 11th Aug. 2024].
[19] G. Angelino and C. Invernizzi, "Supercritical heat pump cycles/Les cycles supercritiques de pompes à chaleur," International Journal of Refrigeration, vol. 17, no. 8, pp. 543-554, 1994.
[20] G. L. Eaton and C. A. Porter, "Critical Temperatures of Petroleum Oils," Industrial & Engineering Chemistry, vol. 24, no. 7, pp. 819-822, 1932.
[21] J. Jiang and J. M. Prausnitz, "Critical temperatures and pressures for hydrocarbon mixtures froman equation of state with renormalization-group-theory corrections," Fluid Phase Equilibria, vol. 169, no. 2, 2000.
[22] C. Zamfirescu, G. F. Naterer and I. Dincer, "Upgrading of Waste Heat for Combined Power and Hydrogen Production with Nuclear Reactors," Journal of Engineering for Gas Turbines and Power, vol. 132, no. 10, p. 102911, 2010.
[23] V. Singh, i. Dincer and M. A. Rosen, "Investigation of new mechanical heat pump systems for heat upgrading applications," International Journal of Energy Research, vol. 42, no. 9, pp. 3078-3090, 2018.
[24] C. Zamfirescu and I. Dincer, "Performance investigation of high-temperature heat pumps with various BZT working fluids," Thermochimica Acta, vol. 488, no. 1-2, pp. 66-77, 2009.
[25] L. B. Yeatts Jr., D. N. Hess, H. F. McDuffie and W. T. Rainey, Jr., "KINETICS OF THE THERMAL DECOMPOSITION OF BIPHENYL (ORNL-3602)," Oak Ridge National Laboratory, Oak Ridge, TN, USA, 1964.
[26] S. Gallarini, A. Spinelli, L. Lietti and A. Guardone, "Thermal stability of linear siloxanes and their mixtures," Energy, vol. 278, p. 127687, 2023.
[27] M. Arjmand, L. Liu and I. Neretnieks, "Exergetic efficiency of high-temperature-lift chemical heat pump (CHP) based on CaO/CO2 and CaO/H2O working pairs," International Journal of Energy Research, vol. 37, no. 9, pp. 1122-1131, 2012.
[28] M. Schmidt, M. Gollsch, F. Giger, M. Grun and M. Linder, "Development of a moving bed pilot plant for thermochemical energy storage with CaO/Ca(OH)2," in SOLARPACES 2015: International Conference on Concentrating Solar Power and Chemical Energy Systems, Cape Town, South Africa, 2015.
[29] A. Gupta, P. Sabharwall, P. D. Armatis, B. M. Fronk and V. Utgikar, "Coupling Chemical Heat Pump with Nuclear Reactor for Temperature Amplification by Delivering Process Heat and Electricity: A Techno-Economic Analysis," Energies, vol. 15, no. 16, p. 5873, 2022.
[30] A. Gupta, P. Sabharwall and V. Utgikar, "Preliminary Pilot-scale Study and Techno-economic Analysis of Chemical Heat Pump with Conventional Nuclear Reactors (INL/RPT-22-67366-Rev000)," Idaho National Laboratory, Idaho Falls, ID, USA, 2022.
[31] F. Joseck, V. Novotny, E. A. R. Molina, K. P. Sweeney, E. K. Worsham, B.-H. Choi, S. J. Root, J. Toman, N. Popli, W.-C. Cheng, K. Vedros, T. Knighton and R. M. Saeed, "Thermal Integration of Advanced Nuclear Reactors with a Reference Refinery, Methanol Synthesis, and a Wood Pulp Plant (INL/PRT-23-03951)," Idaho National Laboratory, Idaho Falls, ID, USA, 2023.
[32] V. Novotny, J. Kim, S.-B. Cho and R. M. Saeed, "Balancing Industrial Energy Park Demands Using Nuclear Energy and Thermal Energy Storage (INL/RPT-24-77366)," Idaho National Laboratory, Idaho Falls, ID, USA, 2024.

[33] I. H. Bell, J. Wronski, S. Quoilin and V. Lemort, "Pure and Pseudo-pure Fluid Thermophysical Property Evaluation and the Open-Source Thermophysical Property Library CoolProp," Industrial & Engineering Chemistry Research, vol. 53, no. 6, pp. 2498-2508, 2014.
[34] T. Westover, H. R. Fidlow, H. Gaudin, J. Miller, G. Beimark and N. Richards, "Impacts of Extracting 30% of Reactor Power from a Pressurized Water Reactor (INL/RPT-23-74666 Rev.0)," Idaho National Laboratory, Idaho Falls, ID, USA, 2023.
[35] A. Maekawa, "Evolution and Future Trend of Large Frame Gas Turbine for Power Generation," Journal of Power and Energy Systems, vol. 5, no. 2, 2011.
[36] SIEMENS Energy, "We Power the World with Innovative Gas Turbines," [Online]. Available: https://p3.aprimocdn.net/siemensenergy/f31eafe1-17e9-4ffd-961b-b0b300c968a1/GT-Portfolio-Brochure-2023-update_20231031_144ppi-pdf_Original%20file.pdf. [Accessed 11th Aug. 2024].
[37] J. Inigo-Labairu, J. Dersch and L. Schomaker, "Integration of CSP and PV Power Plants: Investigations about Synergies by Close Coupling," Energies, vol. 15, no. 19, p. 7103, 2022.
[38] Y. Zhao, J. Song, M. Liu, Y. Zhao, A. V. Olympios, P. Sapin, J. Yan and C. N. Markides, "Thermo-economic assessments of pumped-thermal electricity storage systems employing sensible heat storage materials," Renewable Energy, vol. 186, pp. 431-456, 2022.
[39] M. Geyer, F. Trieb and S. Giuliano, "Repurposing of existing coal-fired power plants into Thermal Storage Plants for renewable power in Chile," Deutsche Gesellschaft für Internationale Zusammenarbeit (GIZ) GmbH, Bonn, Germany, 2020.
[40] EIA, "Cost and Performance Characteristics of New Generating Technologies, Annual Energy Outlook 2022," EIA, Washington D.C., USA, 2022.
[41] EIA, "Capital Cost and Performance Characteristic Estimates for Utility Scale Electric Power Generating Technologies," EIA, Washington D.C., USA, 2020.
[42] NREL, "Natural Gas Plants," [Online]. Available: https://atb-archive.nrel.gov/electricity/2019/index.html?t=cg. [Accessed 11th Aug. 2024].
[43] S. Basu and A. K. Debnath, "Chapter 13 - Advanced Ultrasupercritical Thermal Power Plant and Associated Auxiliaries," in Power Plant Instrumentation and Control Handbook (Second Edition) - A Guide to Thermal Power Plants, Academic Press, Washington D.C., USA, 2019, pp. 893-988.
[44] B. Wehrmann, "First phase-out auction for German hard coal deemed success, modern plants go offline," Clean Energy Wire - Journalism for the Energy Transition, 1st Dec. 2020. [Online]. Available: https://www.cleanenergywire.org/news/first-phase-out-auction-german-hard-coal-deemed-success-modern-plants-go-offline. [Accessed 11th Aug. 2024].
[45] The Technology Collaboration Programme on Heat Pumping Technologies by IEA (HPT TCP), "Steam compressor for steam recycling at pulp drying with pressurized," July 2022. [Online]. Available: https://heatpumpingtechnologies.org/annex58/wp-content/uploads/sites/70/2022/07/spillinghthpannex58democaseifinal-1.pdf. [Accessed 11th Aug. 2024].
[46] Hanwha Power Systems, "Centrifugal air & Gas Compressors - Process & Gas Solutions," [Online]. Available: https://www.hanwhapowersystems.com/product/product_view.asp?idx=6981. [Accessed 11th Aug. 2024].
[47] Atlas Copco, "Mechanical Vapor Recompression (MVR)," [Online]. Available: https://www.atlascopco.com/pressurepoints/en/categories/mechanical-vapor-recompression. [Accessed 11th Aug. 2024].
[48] Atlas Copco, "Driving Centrifugal Compressor Technology," [Online]. Available: https://www.atlascopco.com/content/dam/atlas-copco/compressor-technique/gas-and-process/documents/driving-centrifugal-compressor-technology-new.pdf. [Accessed 11th Aug. 2024].
[49] Leheng, "Pipeline Booster Centrifugal Steam Compressor," [Online]. Available: https://www.lhevaporator.com/products/turbo-compressor/pipeline-pressurized-centrifugal-steam-compressor. [Accessed 11th Aug. 2024].
[50] ATLAS COPCO, "Industrial piston compressors," [Online]. Available: https://www.atlascopco.com/content/dam/atlas-copco/compressor-technique/industrial-

air/documents/leaflets/compressors/le-lt-lf-lb/l-range-(2023)/LE-LT-LF-LB_EN.pdf. [Accessed 11th Aug. 2024].
[51] P. Gallick, G. Phillippi and B. F. Williams, "What's correct for my application - A centrifugal or reciprocating compressor?," in The 35th Turbomachinery Symposium, College Station, TX, USA, 2006.
[52] S. Mokhatab, W. A. Poe and J. Y. Mak, "Chapter 14 - Natural Gas Compression," in Handbook of Natural Gas Transmission and Processing (Fourth Edition) - Principles and Practice, Global Privacy Platform for Publishers, 2019, pp. 433-461.
[53] MAN Energy Solutions, "RB - Barrel type centrifugal compressors," [Online]. Available: https://www.man-es.com/docs/default-source/document-sync-archive/rb-barrel-type-centrifugal-compressors-eng.pdf?sfvrsn=a3ec4f08_1. [Accessed 11th Aug. 2024].
[54] BRONSWERK HEAT TRANSFER, "High Pressure Shell & Tube Heat Exchanger," BRONSWERK HEAT TRANSFER, [Online]. Available: https://www.bronswerk.com/high-pressure-shell-tube-heat-exchanger#:~:text=Pressure%20up%20to%201000%20bar,sufficient%20for%20practically%20all%20applications. [Accessed 11th Aug. 2024].
[55] ALFALAVAL, "Printed circuit heat exchangers," ALFALAVAL, [Online]. Available: https://www.alfalaval.us/products/heat-transfer/plate-heat-exchangers/printed-circuit-heat-exchangers/printed-circuit-heat-exchangers/. [Accessed 11th Aug. 2024].
[56] HEATRIC, "Characteristics of Diffusion-Bonded Heat Exchangers," HEATRIC, [Online]. Available: https://www.heatric.com/heat-exchangers/features/characteristics/. [Accessed 11th Aug. 2024].
[57] Britannica, "gas-turbine engine," [Online]. Available: https://www.britannica.com/technology/gas-turbine-engine. [Accessed 14th Aug. 2024].
[58] Industrial Efficiency & Decarbonization Office, US Department of Energy, "Manufacturing Energy and Carbon Footprints (2018 MECS)," Industrial Efficiency & decarbonization Office, US Department of Energy, Dec. 2021. [Online]. Available: https://www.energy.gov/eere/iedo/manufacturing-energy-and-carbon-footprints-2018-mecs. [Accessed 11th Aug. 2024].