Powerful Heat Transfer Solutions™ for Supercritical CO_2 Recuperators

Richard A Huntington
Partner and Chief Engineer
Powerful Heat Transfer Solutions™
Spring, Texas

Richard Huntington has over 40 years of experience in rotating equipment and gas turbine engineering in the Oil & Gas industries. His work experience includes 33 years for Exxon and ExxonMobil Research & Engineering Company, 5 years for ExxonMobil Upstream Research Company, and 7 years as an independent consultant for the Oil & Gas, LNG and power generation industries. He is the lead inventor or inventor of more than 48 granted patent families that focus on gas turbine design, combustion and control of gas turbine systems, power generation, and heat transfer systems. He is a member of the ASME PTC-10 committee that recently completed its work on a revision of this code. Mr. Huntington has a B.S. (Cum Laude) and M.S. degree in Mechanical Engineering from the University of Maryland and is a member of the American Society of Mechanical Engineers.

ABSTRACT

Supercritical carbon dioxide (sCO_2) cycles offer a promise of high efficiency and small equipment size, at least for the turbomachinery. The efficiency of these cycles is dependent on the effectiveness of the critically important recuperator. This is the heat exchanger responsible for recovering about 75% of the heat from the expander exhaust and returning it to the expander by preheating the fresh sCO_2 feed. The recuperator's task requires more heat transfer surface area than the primary heater and coolers combined. It also requires low pressure drops to avoid excess parasitic losses. Especially for open sCO_2 power cycles like the Allam Cycle, the recuperator must not be prone to fouling that may cause plugging, loss of effectiveness, increased pressure drops, and loss of efficiency and availability.

In this study, the author compares alternative heat transfer technologies to perform the recuperator duty for a relatively small 100 MWe net sCO_2 power cycle. The alternatives include a series of Printed Circuit Heat Exchangers (PCHE), Shell and Tube Exchangers (STE) and the patented recuperator designed / invented by Powerful Heat Transfer Solutions™.

For both PCHE and STE, multiple parallel heat exchangers require complex piping arrangements and equipment spacing to compensate for the very large operating temperature differences, especially for startup and shutdown conditions.

The new recuperator solves these problems by incorporating an intermediate heat transfer fluid to allow optimization of each part of the heat exchanger. This removes many size constraints, significantly reduces piping complexity, allows operating flexibility, and is less prone to fouling.

INTRODUCTION

Supercritical CO_2 power cycles have long been promoted for future renewable and nuclear energy opportunities. Whole conferences such as this 8TH INTERNATIONAL SUPERCRITICAL CO_2 POWER CYCLES SYMPOSIUM [1] hosted by SwRI have been organized to further the technologies for sCO_2 and foster their commercial applications. US DOE [2] suggests that sCO_2 cycles are applicable to generate power from a wide range of high value heat sources, including fossil fuels, geothermal, concentrated solar, and nuclear. Benefits cited include thermal efficiencies of 50% or more and compact power plants with much smaller footprints and lower capital costs. The sCO_2 turbomachinery (e.g., turbines, pumps and compressors) are extremely compact when compared to, for example, steam power plants. Much of this compactness is due to the sCO_2 cycle that operates between high pressures of 200-300 bar or higher to low pressures of not less than 75 bar. This compares to a steam Rankine cycle that operates at similarly high pressures and temperatures but has a low pressure range that is well into vacuum with pressures measured in mmHg absolute. The high pressures throughout the sCO_2 cycle and the high fluid densities mandate the use of smaller equipment at similar mass and energy flow rates.

These are the pros generally cited. A major con has been the design of the recuperator that is critical to achieve the high efficiencies mentioned above. For an example application, sCO_2 at 250 bar is heated to 700°C and then expanded through a turbine of 85% adiabatic efficiency to 85 bar. This lower pressure is selected to remain above the critical pressure of CO_2 of about 74 bar. At the turbine outlet, the sCO_2 temperature is about 565°C. In a competing steam Rankine cycle, low pressure turbine exhaust is at vacuum pressures and may be close to ambient temperatures. It is ready to be further cooled and condensed to water and then pumped back to a high pressure. For the example sCO_2 cycle, the 565°C holds far too much energy to simply reject to the environment and a recuperator (a heat exchanger) is required to take heat from the turbine outlet at about 85 bar and deliver it to the cooled sCO_2 that has already been pumped/compressed to about 250 bar or more. The recuperator preheats this high pressure sCO_2 from circa 70°C to a temperature close to the turbine outlet. In a sCO_2 cycle that may achieve about 100 MWe power delivery at 50% cycle efficiency, about 200 MW of high value heat must be delivered to the circa 250 bar sCO_2 turbine inlet stream, a net of 100 MW may be converted to power, 100 MW of energy is rejected to the environment through a cooler, and 500 to 600 MW of heat must be transferred from the turbine exhaust to the cool very high pressure sCO_2. That is, the recuperator is responsible for 70 to 75% of the heat delivered to the turbine inlet.

The recuperator selection may be a difficult application due to the combination of a high pressure difference (170 bar or more) and the high temperature difference (about 500°C) between the hot and cold streams of the heat exchanger. This is in addition to the large relative heat duty of such a recuperator.

Printed Circuit Heat Exchangers (PCHE) have been a typical selection for sCO_2 power cycles for many years [3,4]. Conventional shell and tube exchangers (STE) may also be considered with some limitations. Micro channel shell and tube exchangers (MSTE) have also been promoted because of their better capabilities for sCO_2 recuperator applications versus conventional STE [5]. Typical PCHE have semi-circular circa 2mm diameter channels while MSTE have tubes of 1-3 mm [6]. According to Kwon [5], PCHE and MSTE have limitations due to potential fouling and high pressure drop. Fouling potential may not be a major concern for closed sCO_2 power cycles, but it is for open loop sCO_2 cycles, like the Allam Cycle [7], that combust fossil fuels to create CO_2. Fouling from carbonaceous and other particulate matter formed in even a very well mixed near-stoichiometric flame has the potential to cause slow but constant fouling of such small passages [8]. This fouling will directly impact cycle efficiency, plant availability and reliability, and operating cost.

Recently, Powerful Heat Transfer Solutions™ developed a new recuperator system [9] that employs an Intermediate Heat Transfer Fluid (IHTF) to separate the heat duties of the sCO_2 turbine exhaust and the high pressure sCO_2 feed to allow optimization of heat exchangers for each to have low pressure drop and minimal fouling potential.

SIZING OF PCHE AND STE

To evaluate either PCHE or STE for a particular application, bases for sizing the exchangers are needed along with plant simulations to account for the hot-side and cold-side pressure drops that directly impact plant performance and recuperator pressures and temperatures.

For PCHE, Jiang [4] provides the method to estimate the Nusselt Number (Nu) and pressure drops for various PCHE designs. For this study, the high-angle zigzag channel design with a semi-circular shape and 2 mm width is selected.

(1) $Nu = 0.0845Re^{0.721}Pr^{1/3}$ $\quad$ $Re = Reynolds\ No.$ and $Pr = Prandtl\ No.$

(2) $f = 1.336Re^{-0.1268}$ $\quad$ $f = Darcy\ Friction\ Factor$

Nusselt Number and Reynolds Number are calculated using the hydraulic diameter of the channel.

(3) $D_h = \frac{\pi D_c}{2+\pi}$ $\quad$ $D_h = Hydraulic\ Diameter$ (m)
$D_c = Channel\ Width$ (m)

The cold-side & hot-side Nu are each calculated and the heat transfer coefficients, h (W/m²/°C), are found from the Nu as:

(4) $h = \frac{k_f Nu}{D_h}$ $\quad$ $k_f = Fluid\ Thermal\ conductivity\ (W/m^2/°C)$

(5) $HTC = \left(1/h_c + 1/\left(h_h R_p\right) + t_e/k_w\right)^{-1}$ $\quad$ $HTC = Overall\ Heat\ Transfer\ Coef$
$h_c = Cold\ Side\ heat\ transfer\ coef$
$h_h = Hot\ Side\ heat\ transfer\ coef$
$R_p = Number\ of\ hot\ plates\ per\ cold\ plate$
$t_e = Equivalent\ wall\ thickness$ (m)
$k_w = Thermal\ conducitivity\ of\ wall$

Assuming that the wall thickness of the plate after etching and the ridge width between passages on a plate are both equal to $0.2D_c$, then the equivalent wall thickness is:

(6) $t_e = 0.373D_c$ (m)

For Type 316 stainless steel, the thermal conductivity of the wall is approximately:

(7) $k_w = 13.189 + 0.0153T_w$ (W/m/°C) $\quad$ $T_w = Wall\ temperature\ (°C)$

From the heat duty (Q, Watts) required for the recuperator (or a part of it) and the log mean temperature difference (LMTD, °C) between the hot and cold sides over that part, the area required to transfer that heat from the hot side to the cold-side of the recuperator is:

(8) $A = Q(LMTD)/HTC$ (m²)

From the area, the channel length may be found.

(9) $A = \frac{1}{(1+R_p)} N_c N_p (1 + \frac{\pi}{2}) D_c L_c$

$N_c = Number\ of\ Channels\ per\ plate$
$N_p = Total\ number\ of\ plates\ (hot + cold)$
$L_c = Length\ of\ a\ portion\ of\ a\ channel\ (m)$

The pressure drop across a portion of the channel may be calculated by (Darcy):

(10) $\Delta P = \frac{1}{2} \rho V^2 f\, L_c / D_h$

$\Delta P = Pressure\ drop\ along\ part\ of\ a\ channel\ (Pa)$
$\rho = fluid\ density$ (kg/m^3)
$V = Average\ velocity\ in\ the\ channel\ (m/s)$

The above is not intended to be a full description of the PCHE design calculations. Refer to [4] for details.

For a STE, the sizing and design equations are well known and will be briefly described in the following. Hewitt [10] was used as the basis for both bare tube and finned tube STE calculations. Nusselt number, Darcy friction factor, and pressure drop for tube-side are:

(11) $Nu = 0.0225 Re^{0.795} Pr^{0.495} exp[-0.0225(\ln Pr)^2]$

(12) $f = 0.184 Re^{-0.2}$

(13) $\Delta P = \frac{1}{2} f \rho V^2\, L / D_i$

$L = Length\ of\ a\ parallel\ tubes$
$D_i = Inside\ diameter\ of\ a\ tube$

For cross flow over the outside of a bare tube or tube bundle, the relevant equations are:

(14) $Nu_D = 0.3 + \frac{0.62 {Re_D}^{0.5} Pr^{1/3}}{\left[1+(0.4/Pr)^{2/3}\right]^{0.25}} \left[1 + \left(\frac{Re_D}{28200}\right)^{5/8}\right]^{0.8}$

$Nu_D = Nusselt\ No.\ over\ cylinder$
$Re_D = Reynolds\ No.\ over\ cylinder$

(15) $\Delta P = \frac{1}{2} \rho V_{max}^2 \left(1 + \sigma^2 + N_r K_f\right)$

$V_{max} = Maximum\ average\ velocity\ \ (m/s)$
$\sigma = \frac{Minimum\ Free\ Flow\ Area}{Total\ Frontal\ Area}$
$N_r = Number\ of\ tube\ rows\ in\ a\ pass$
$K_f = Crossflow\ friction\ factor\ (See\ [10])$

The crossflow heat transfer coefficient is found from the Nusselt number as before: $h_D = k_f Nu_D / D_o$ where D_o is the outside diameter of a tube in the bundle.

For extended finned tubes, rather than evaluating specific fin designs, a typical factor of 6 is used to estimate the heat transfer coefficient from the bare tube calculations. The pressure loss across a bundle of finned tubes is estimated using Eqn. 15 with the following assumptions and simplifications:

(16) $K_f = 4.567 Re^{-0.242} \left(\frac{A}{A_T}\right)^{0.504} \left(\frac{P_1}{D_o}\right)^{-0.376} \left(\frac{P_2}{D_o}\right)^{-0.546}$ See [10] for details.

Assuming a triangular tube arrangement and tube pitch equal to 1.5D_o, $K_f \sim 5.52 Re^{-0.242}$, this leads to the following estimated pressure drops across this finned tube bundle:

(17) $\Delta P = \frac{1}{2}\rho {V_{max}}^2[1.61 + 5.52 N_r Re^{-0.242}]$

(18) $V_{max} \sim \frac{2W}{\rho D_o L N_t}$ $\qquad W = mass\ flow\ rate\ (kg/s)$

(19) $Re = \frac{2W}{\mu L N_t}$ $\qquad L = Length\ per\ pass\ (m)$

$N_t = Number\ of\ tubes\ per\ row$

SIMULATION OF ALTERNATE APPLICATIONS

To evaluate alternate recuperators, a similar power plant simulation was applied to each potential recuperator design. This simulated plant produces about 100 MWe net power. The nominal conditions applied to each simulation are as follows:

TABLE 1 – Common Operating Conditions		
Motive Fluid	Supercritical CO_2	
Expander Flow Rate	1001.7	Kg/s
Expander Inlet Pressure	235	bara
Expander Inlet Temperature	700	°C
Cooler Inlet Pressure	85	Bara
Cooler Pressure Drop	2	Bar
Cooler Outlet Temperature	34.5	°C
Turbomachinery Efficiency	85%	Adiabatic Efficiency
Pipe Losses (various)	1.5	bar

The sCO_2 cycle selected is the split recuperator type that includes a High Temperature Recuperator (HTR) and a Low Temperature Recuperator (LTR). A typical arrangement is shown in Figure 1.

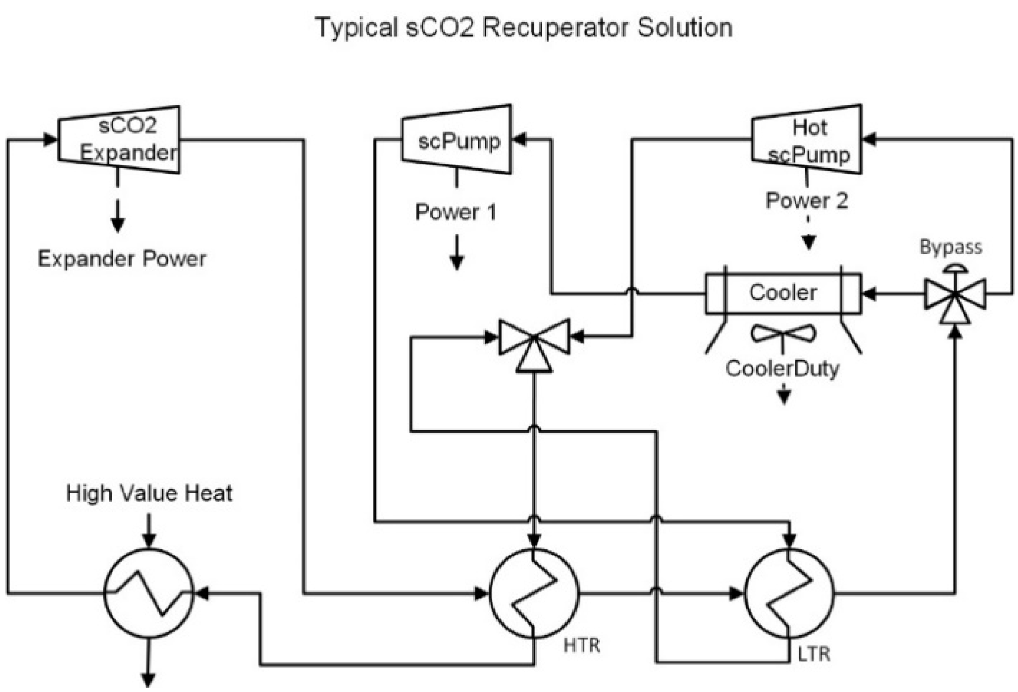

Figure 1

PRINTED CIRCUIT HEAT EXCHANGER

Ref. [4] recommended the use of high-angle zigzag channel type PCHE to achieve a combination of smaller size and lower pressure drop vs. other PCHE types. It also offered the size limitations of a single PCHE device. The suggested limit is a module size measuring 1.5 m by 0.6 m by 1.0 m with up to eight modules welded together before headers are welded to the stack. This brings the maximum dimensions of a single PCHE to 1.5 x 0.6 x 8.0 m without headers included. For a typical sCO_2 power plant as depicted in Figure 1, at least two PCHE devices are required: One for the HTR service and one for the LTR service. Finally, [4] recommends 2 hot plates per cold plate or R_p=2.

For the nominal 100MWe plant shown in Figure 1, one to five PCHE are placed in series-parallel for the HTR and LTR services. That is, for the initial simulation one largest size PCHE is used for the HTR service and one largest size PCHE is used for the LTR service. For the next simulation, two parallel strings of a 1 + 1 largest size PCHE were used for each of the HTR and LTR services. Then 3, 4 and 5 parallel strings of 1 + 1 largest size PCHE were used for the remaining simulations. As the number of parallel strings increases, the PCHE pressure drop reduces, and the overall plant efficiency improves.

For each of these simulations, DWSIM [11] was used for the plant simulation and employed the GERG-2008 equation of state for sCO_2. As DWSIM does not already include the needed design equations for PCHE devices, an Excel workbook was developed to take the HTR and LTR heat duties and temperature profiles from each simulation and perform the necessary sizing and pressure drop calculations. That information was then entered back into DWSIM to update the simulation until the heat duties, temperature profiles, and plant performances converged. All HTR and LTR heat exchanger simulations assumed a minimum approach temperature of 10°C. Furthermore, the bypass rate was adjusted in every case so that the hot and cold temperature pinches of the LTR exchangers were balanced to maximize the cycle efficiency of each plant simulation.

TABLE 2 – PCHE Simulation Results

No. of PCHE in Parallel (HTR & LTR)	Expander Outlet Pressure (bar)	Expander Outlet Temperature (°C)	HTR Heat Duty (MW)	LTR Heat Duty (MW)	HTR Hot ΔP (bar)	LTR Hot ΔP (bar)	HTR Cold ΔP (bar)	LTR Cold ΔP (bar)	Plant Cycle Efficiency
1	130.6	624.2	485	151	26.8	17.8	41.8	8.4	24.0%
2	97.0	588.8	464	140	6.7	3.8	9.0	1.8	43.7%
3	90.9	581.2	459	139	2.8	1.5	3.7	0.7	46.6%
4	88.9	578.6	456	138	1.5	0.8	2.0	0.4	47.6%
5	88.0	577.4	456	138	0.9	0.5	1.2	0.2	48.0%

Table 2 illustrates that a single maximum sized PCHE is too small for this modestly sized circa 100 MWe power plant and that multiple PCHE in parallel are necessary to avoid the exceptionally large pressure losses and the resulting loss of plant efficiency. With each additional PCHE in parallel for the HTR and LTR service, the efficiency improves and comes close to a maximum with five units in parallel for each service. Increasing the number to 10 units in parallel shows a small gain but may not be economically justified.

Figure 2 shows the Hot and Cold-side Reynolds numbers and the overall heat transfer coefficients (HTC) across the HTR heat exchanger for the single PCHE (upper group of curves) and for five

PCHE in parallel (lower group of curves). The average Reynolds number for the Cold and Hot side of the upper group is about 71,500 and 39,000, respectively, while the similar average for the lower group is about 15,000 and 8,200. The related average HTC are 5.5 and 1.8 kW/m^2/°C. This demonstrates the strong relationship between Re and HTC and the clear tradeoff between HTC, system pressure drop, cycle efficiency, plant design complexity, and cost.

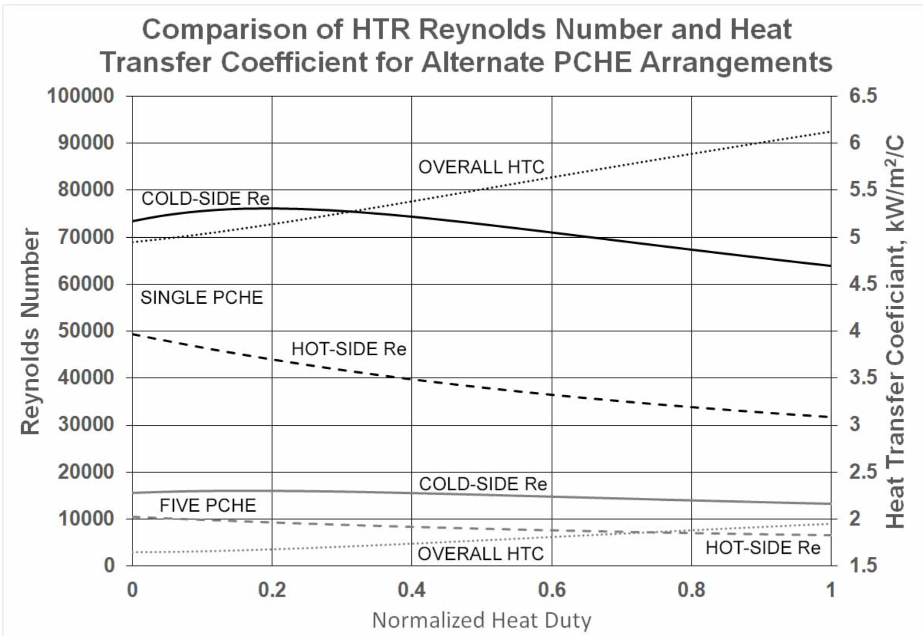

Figure 2

Another tradeoff relates to the plant layout. Equipment spacing and piping design with extensive expansion loops are necessary to accommodate multiple PCHE in a series-parallel arrangement and the extreme temperature changes that will be seen, not only from the cold to hot sides of the PCHE, but also from running to non-running conditions. Figure 3 shows a potential equipment and piping arrangement for the circa 100MWe power plant.

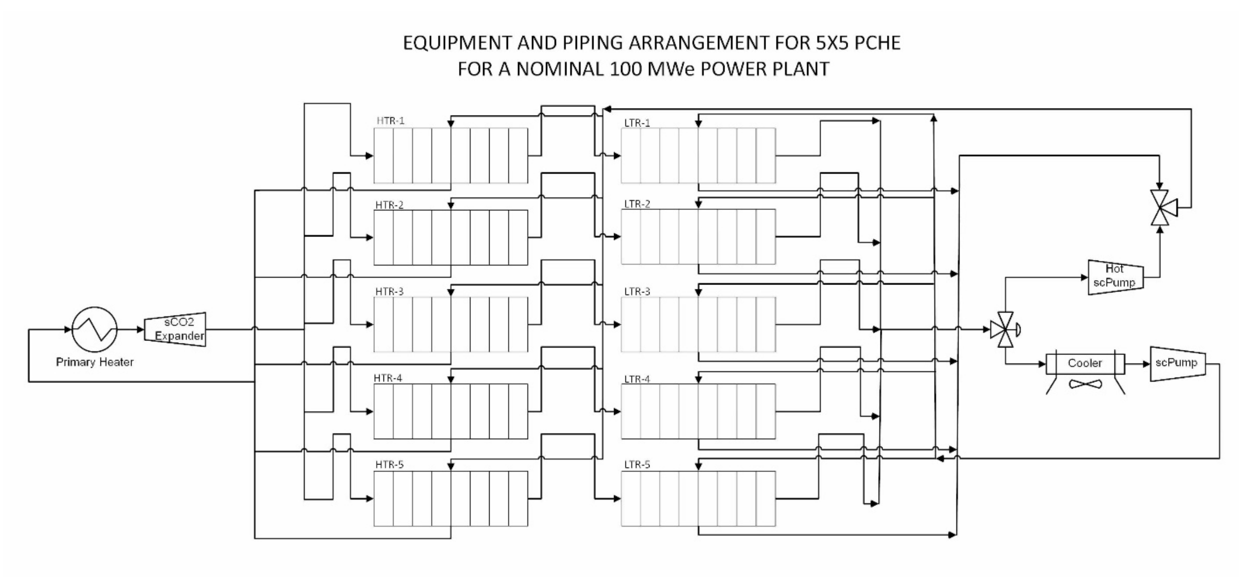

Figure 3

As can be seen from Figure 3, complex piping arrangements and widely spaced equipment will be required for any similar plant design, and 100 MWe is small for a commercial power plant. The piping and equipment layouts will be even more complex for a nominal 200 to 500 MWe power plant that may compete with current combined cycle or steam Rankine power plants.

SHELL AND TUBE EXCHANGER

There are manufacturing, economic and practical size limits for STE. For example, increasing tube-side and shell-side pressures result in heavier wall shells and thicker tube sheets that make these more difficult and more costly to manufacture and transport. This study does not intend to consider these limits. There are also many decisions with the design of STE including tube diameter, hot or cold fluid on the tube-side, and whether to fin the tubes. For this alternative study, the tube diameter selected is a nominal 10 mm with a 1.25 mm wall thickness. Both finned and bare tube exchangers were considered. In all cases considered, the shell-side of the STE is selected for the hot fluid for both the HTR and the LTR services. This selection is based on the lower pressure and higher volume flow rate of the turbine exhaust versus the cold high pressure sCO_2 streams.

The selection of a nominal 10 mm diameter is selected (vs. the 1-3 mm of MSTE) is to provide a flow area 50 times greater than the area of the 2 mm semi-circular PCHE channels with the specific purpose to minimize the fouling and blockage risks of the PCHE and MSTE devices.

The number of tubes and the tube and pass length for each STE can be selected such that the hot-side and cold-side pressure drops are equal to or less than the PCHE in the 5+5 series-parallel arrangement. This results in comparable heat duties, temperature profiles and plant cycle efficiencies with the only clear change being the surface areas for the HTR and LTR and the exchanger configuration.

The most direct way to compare the PCHE and the STE performance is to compare the heat transfer coefficients of the HTR and the LTR among the 5+5 series-parallel PCHE, a bare tube STE, and a finned tube STE. These are given in Figures 4 and 5.

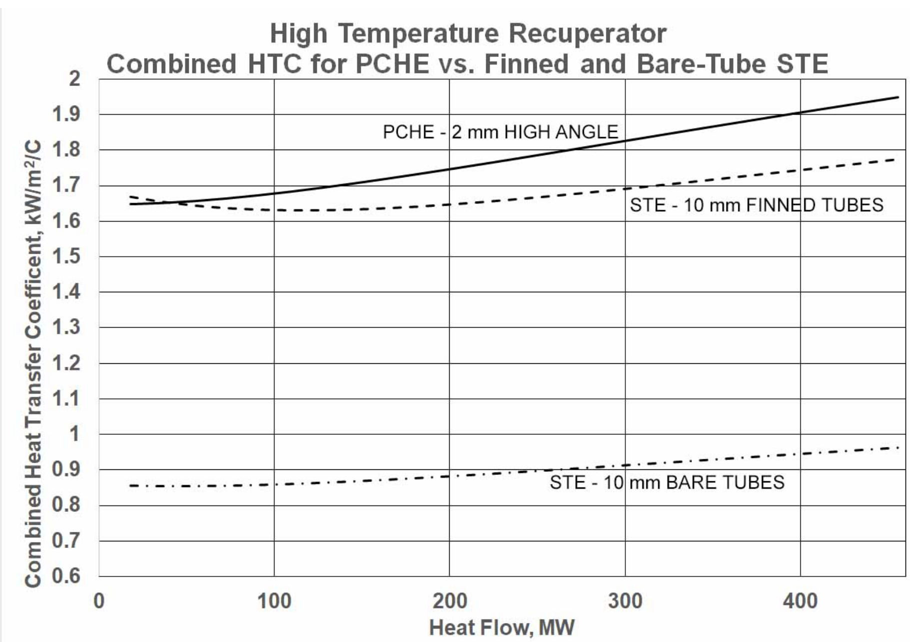

Figure 4

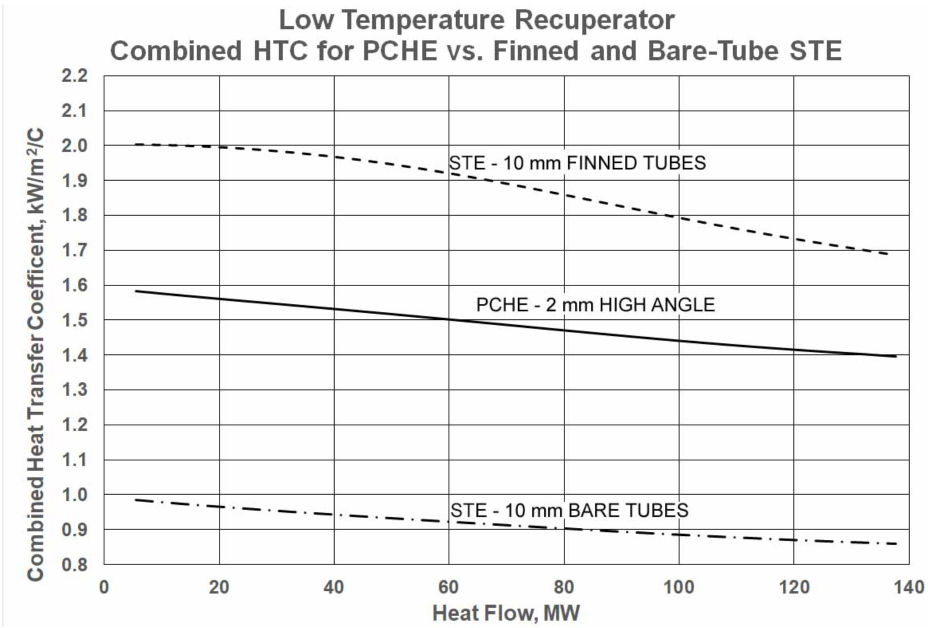

Figure 5

For the HTR shown in Figure 4, the HTC for the finned tube STE is similar but slightly lower than the HTC for the PCHE. The HTC for the bare tube STE is about half of the finned value. The average values of the HTC for the HTR are: 1.78, 1.68 and 0.90 kW/m^2/°C for the PCHE, finned STE and bare tube STE. For the finned STE in this HTR service, about 8,740 m^2 of surface area is required vs. 8,380 m^2 of surface area for the PCHE and 16,490 m^2 of surface area for the bare tube STE. For the bare tube STE, the added surface area should primarily be by the use of longer rather than more tubes.

Figure 5 gives the results for the LTR service. In this case, the finned tube STE has a higher HTC than the PCHE. This improvement has not been aided by a higher pressure drop of the STE. In fact, the cold-side ΔP is similar at 0.26 bar while the hot-side ΔP is lower at 0.22 bar. The average values of the HTC for the LTR are: 1.49, 1.87 and 0.91 kW/m^2/°C for the PCHE, finned STE and bare tube STE. For the finned STE in this LTR service, about 6,320 m^2 of surface area is required vs. 7,960 m^2 of surface area for the PCHE and 12,970 m^2 of surface area for the bare tube STE.

While it is likely that multiple shells would be required for STE in both the HTR and LTR services for a 100 MWe power plant, at least with a finned STE, the overall HTC is similar to or greater than for a PCHE and STE should be competitive while offering more manufacturing options and a much better tolerance for fouling.

NEW RECUPERATOR SOLUTION

The new recuperator solves difficult heat transfer problems, especially those that involve large temperature and pressure differences between the streams and those that require maximum assurance that contamination between the streams does not occur. For the sCO_2 recuperator system, the HTR and LTR services are further split into three heat exchangers and an IHTF is used to gather and transfer heat from one stream to another.

Figure 6 shows the arrangement of power plant and recuperator system.

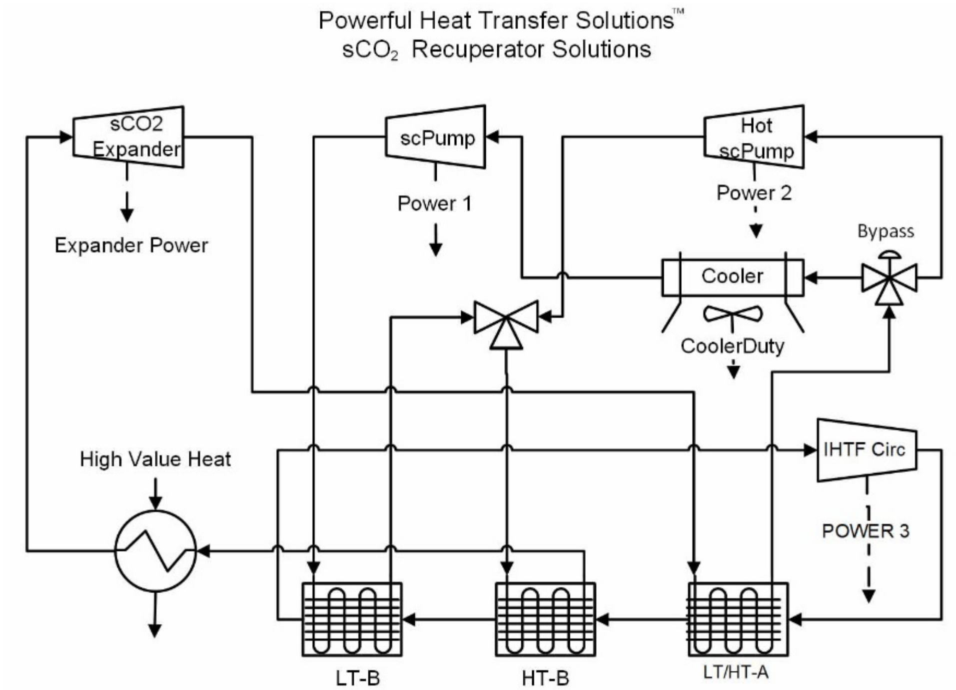

Figure 6

With this new recuperator system, the expander exhaust is directed to the LT/HT-A heat exchanger and is cooled by the recirculated IHTF. By doing this, the IHTF approaches the temperature of the expander exhaust. As with the typical split recuperator arrangement shown in Figure 1, the cooled exhaust stream is split into two streams, a Primary sCO_2 stream and a Bypass sCO_2 stream that bypasses the primary Cooler. The Bypass stream is pumped to full pressure in the Hot scPump. The Primary stream is cooled to near ambient temperature in the Cooler and then pumped to full pressure in scPump. This Primary stream is delivered to exchanger LT-B to be pre-heated. The Primary stream exiting LT-B is combined with the Bypass stream from the Hot scPump and the combined stream is directed to exchanger HT-B to heat it to near sCO_2 Expander exhaust temperature.

The IHTF is continuously recirculated through the LT/HT-A, HT-B and LT-B exchangers to first pick up heat from the expander exhaust, deliver most of that heat to the combined sCO_2 stream, and then preheat the Primary stream after it is cooled and pumped to pressure. The IHTF temperature varies as it picks up and rejects heat through the three heat exchangers. Upon leaving the LT-B exchanger, the temperature is at its lowest and is near the temperature of the Primary stream after it is cooled and pumped to pressure.

Like the typical split recuperator system shown in Figure 1, the bypass rate may be adjusted to balance the cold-end and hot-end temperature pinches of one exchanger to maximize the cycle efficiency. For the typical arrangement, the pinches of the LTR are balanced while for the new recuperator, the pinches of LT-B are balanced to maximize the cycle efficiency. Furthermore, the flow rate of the IHTF may be adjusted to balance the cold-end and hot-end pinches of the LT/HT-A exchanger to maximize effectiveness of the recuperator system.

Figure 7 is a duplicate of Figure 6 with the operating conditions of the simulation added. For this simulation, the Bypass rate is 41% and the IHTF is nitrogen (or air) near ambient pressure and the IHTF flow rate is 109.5% of the sCO_2 mass flow rate. Finally, the overall power plant efficiency is 47.6% that can be favorably compared to the 48.0% efficiency for the 5+5 series-parallel PCHE arrangement for this 100 MWe power plant. The debit is mostly due to the IHTF recirculatory power.

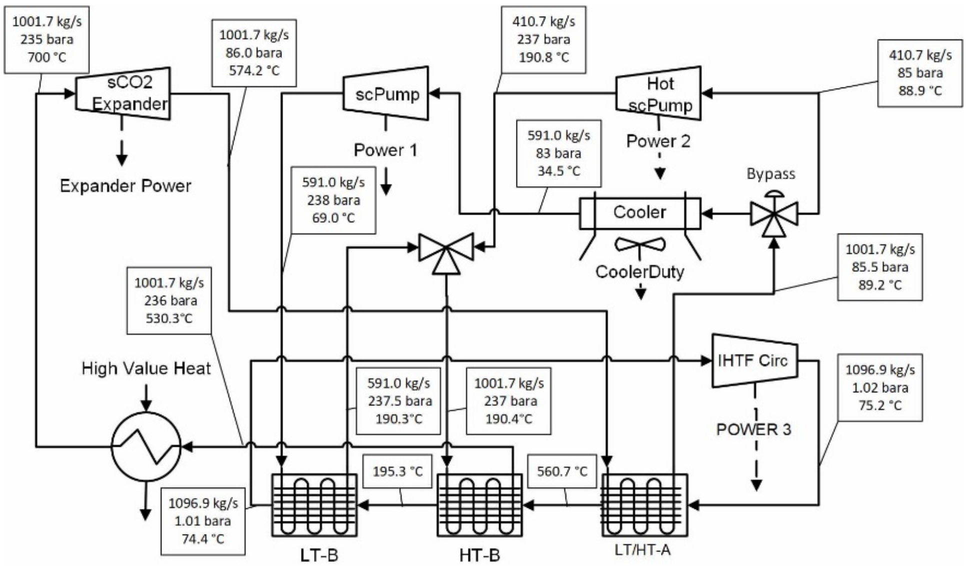

Figure 7

For the plant depicted in Figure 7, a large low pressure fan is used to circulate the nitrogen IHTF over the finned coils through which the sCO_2 flows. This fan may be an adjustable speed or blade angle fan to allow control of the IHTF flow rate. The arrangement of LT/HT-A, HT-B and LT-B coils within the flowing stream of nitrogen is like the arrangement of a heat recovery steam generator of a combined cycle power plant and can be manufactured by companies that are skilled to produce HRSG equipment. As with HRSG equipment, larger heat transfer areas may be produced simply by extending the width and length of the coils to accommodate larger power plants without the need for series-parallel arrangements needed with PCHE or STE arrangements.

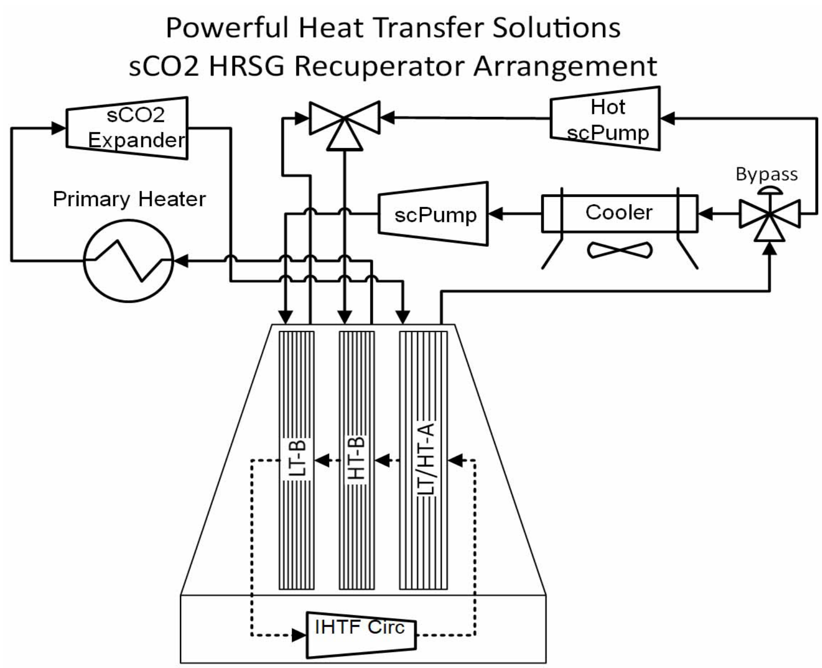

Figure 8

Like Figure 3, Figure 8 shows a potential plant arrangement with the new recuperator design using ambient pressure nitrogen or air as the IHTF as depicted in Figures 6 and 7. The recuperator system is similar in form to the HRSG of a typical combined cycle power plant. The piping arrangement for this design is much simpler than for the PCHE arrangement shown in Figure 3. Further, all individual coils are hung from the inlet and outlet headers and are free to expand with changes in operating conditions. This allows fast ramping from startup to normal operation and then to shutdown.

NEW RECUPERATOR DESIGN WITH LIQUID METALS

For these recuperator systems, the IHTF may be a gas such as the nitrogen or dry air discussed above. Small improvements of the heat transfer coefficients are possible by using helium that has a much higher conductivity than other typical gases. Argon, CO_2 and similar gases may also be used. Dry air is the most practical and is used near ambient pressure. However, liquids may also be used if a practical system to employ them can be designed. Most common liquids are not practical due to the range of temperatures required. One liquid that is very promising is a liquid metal such as a eutectic mixture of sodium and potassium, or NaK78. NaK78 is a mixture of roughly 78% potassium to 22% sodium by weight and has a normal melting point of about -13°C and a normal boiling point of about 800°C. This temperature range is more than adequate for the recuperator system. The following are properties of NaK78 [12] at low pressure.

TABLE 3 – Selected Properties of Sodium-Potassium Eutectic Mixture at Low Pressure

Temp (°C)	Specific Gravity (-)	Dyn Viscosity (cP)	Thermal Conductivity (W/m/°C)	Specific Heat (kJ/kg/°C)
-13	0.877	1.125	21.136	1.0037
0	0.874	1.002	21.400	0.995
200	0.826	0.362	24.660	0.908
400	0.778	0.234	26.160	0.878
600	0.730	0.167	25.900	0.876
800	0.682	0.134	23.880	0.893

The typical equations for Nusselt No. used to calculate tube side heat transfer coefficients have a lower bound on the Prandtl No. of about 0.3. For liquid metals, the values of Prandtl No. are much lower, and these correlations are not applicable. Shen [13] provides a correlation of Nusselt number for non-heavy liquid metals that include sodium, potassium and mixtures of sodium and potassium like NaK78. This correlation is:

$$(20)\ Nu = 10.0652(RePr)^{-0.1219} + 0.0373(RePr)^{0.7531}$$

Friction factors for tube flow pressure drop may continue to be calculated with the Darcy Eqn. (12) that is not restricted to a given range of Prandtl No. Now with Eqn. (12) and (20), sizing calculations may be performed for tubular heat exchangers with NaK78 on the tube side.

A finned tube heat exchanger with sCO_2 on the finned side and the NaK78 on the tube side provides the greatest improvement to the overall heat transfer coefficient. A plant arrangement like Figure 6 may be used when NaK78 is used as the IHTF. In this case, the IHTF circulator is a relatively low head liquid metal pump that operates at about 75°C.

Figures 9, 10 and 11 show the temperature vs. heat flow charts for the HT/LT-A, HT-B and LT-B heat exchangers in this new recuperator with NaK78 as the IHTF. For these charts, the temperature pinches are 6°C, 5°C and 5°C, respectively. This compares to the 10°C pinch used above for the PCHE and STE examples of the nominal 100 MWe power plant and compensates for the double heat transfer of the previous design.

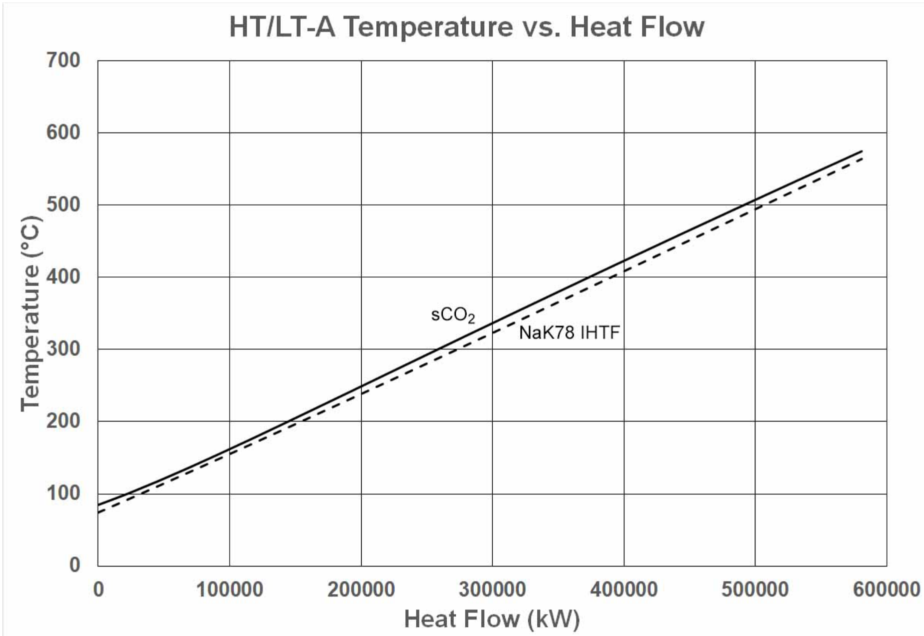

Figure 9

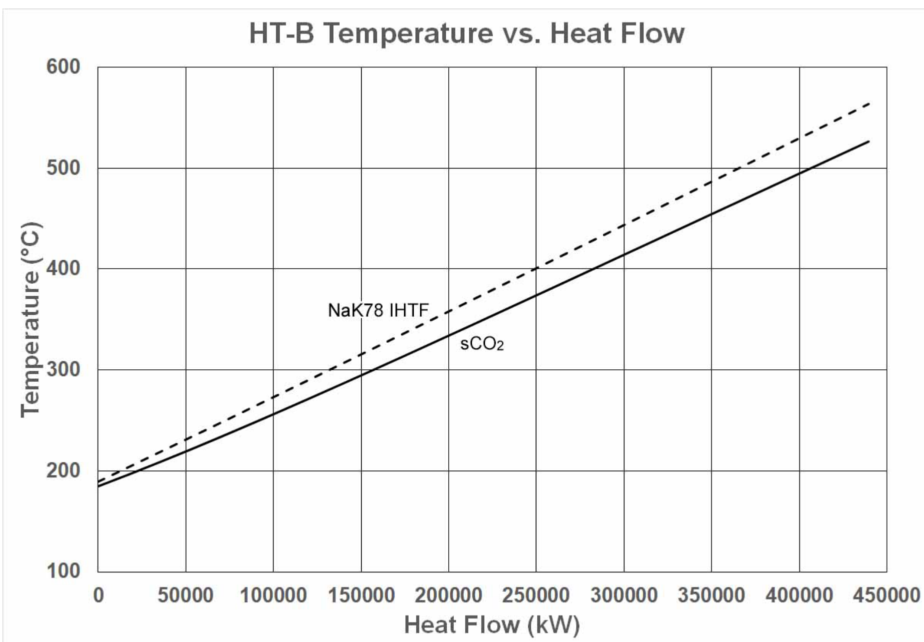

Figure 10

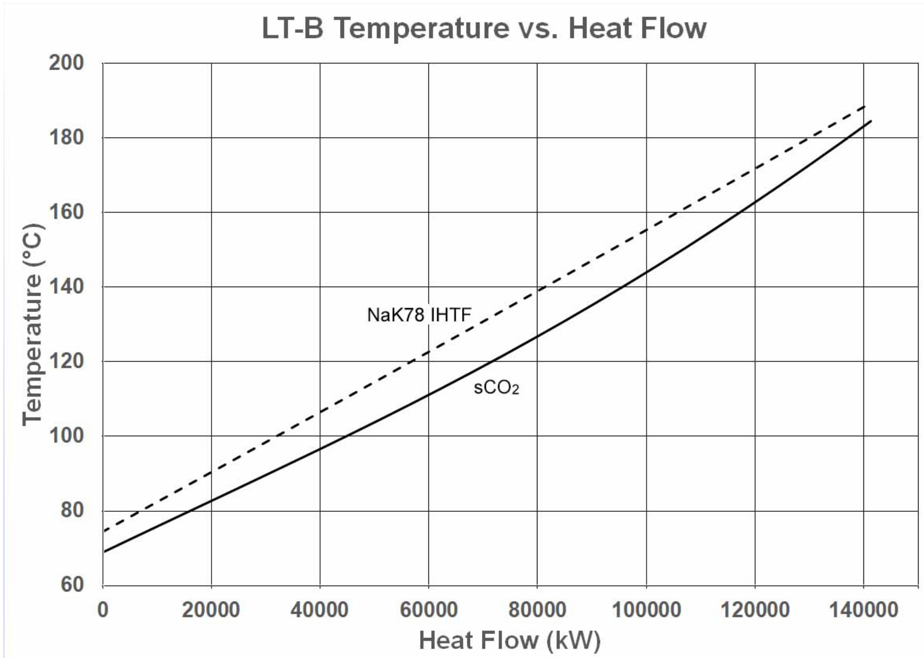

Figure 11

Figure 12 shows the overall heat transfer coefficients for the LT/HT-A, HT-B and LT-B heat exchangers shown in Figure 6.

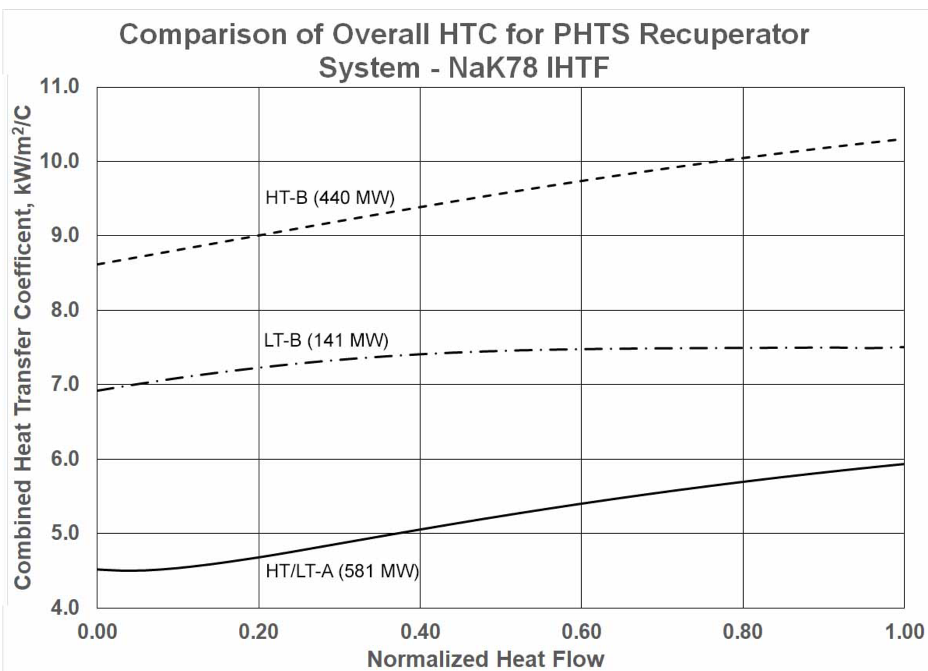

Figure 12

Comparing the HTC shown in Figures 4 and 5 with those shown in Figure 12, the NaK78 increases the HTC substantially. The average HTC for the PCHE was 1.78 and 1.49 kW/m^2/°C for the HTR and LTR services versus an average HTC for the new recuperator HT/LT-A, LT-B and HT-B of 5.2, 7.4 and 9.5 kW/m^2/°C, respectively. Even though the heat must be transferred twice from the sCO_2 to the NaK78 and back to the sCO_2, similar or less heat transfer area is required for the

NaK78 recuperator system with an identical or better plant cycle efficiency for the example 100 MWe power plant. Further, the similar surface areas are achieved with temperature pinches (Figures 9-11) roughly half the values of the PCHE at 10°C. This results from the lower pressure drop of the finned tube exchanger vs. the array of PCHE. Since the sCO_2 flows on the fin side of the heat exchanger, the risk of fouling that may result in plugging of flow passages is extremely low when compared to the 2 mm semicircular passages of the PCHE. The combination of very high HTC, low pressure drops, and substantially reduced risk of fouling demonstrate that the new recuperator is well suited to sCO_2 power plants, especially open cycles.

SUMMARY OF SIMULATION RESULTS

Selected results of the various simulations are included in Table 4. The first 3 columns provide the results with the traditional sCO_2 power cycle with a split recuperator for the example shown in Table 1. Only PCHE results with a total of 10 (i.e., 5+5) of the largest size available in a series-parallel arrangement of the HTR and LTR devices is given. In addition to the PCHE, shell and tube exchangers with both bare and finned tubes are provided. Finally, the new recuperator design results are provided for both a HRSG-like arrangement with air or nitrogen as the IHTF and one that uses a NaK78 liquid metal as the IHTF. The net power and cycle efficiency values shown in Table 4 for the new recuperator include all losses associated with the blower and pump needed to recirculate the IHTF.

TABLE 4 - Selected Results for 100 MWe Sample Power Plant

	PCHE (5+5) (2mm)	STE Bare Tube (10mm)	STE Finned Tube (10mm)	New Recuperator N_2 IHTF HRSG (10mm)	New Recuperator NaK78 IHTF (10mm)
Net Power (MW)	103.1	103.0	103.7	101.8	105.3
Cycle Efficiency (%)	48.0	48.0	48.2	47.6	48.0
Total Hot ΔP (bar)	1.45	1.58	0.82	0.5	0.42
Total Cold ΔP (bar)	1.44	1.40	0.74	1.0	0.53
Avg HTC (kW/m^2/C)	1.71	0.90	1.73	0.61	7.11
HTR or HT/LT-A Area (m^2)	8,380	16,490	8,740	162,000	11,340
LTR or HT-B Area (m^2)	7,960	12,970	6,320	29,000	2,390
LT-B Area (m^2)	-	-	-	18,000	2,120
Total Surface Area (m^2)	16,340	29,460	15,060	209,000	15,850

CONCLUSIONS

- This study has evaluated alternatives to the typical PCHE solution for sCO_2 cycles.
  - PCHE have high pressure drops and are limited in size due to their complex manufacturing processes.
  - For the relatively small 100 MWe power plant considered, many PCHE in parallel are required to achieve acceptable pressure drops and cycle efficiencies.
- The use of finned tubes with STE heat exchangers increases their effectiveness to be equal to PCHE but with limitations.
- The new recuperator incorporates an IHTF that allows optimization of each part of the heat exchanger and removes many size constraints.
- The new recuperator is less prone to fouling and loss of cycle efficiency and availability, especially for open sCO2 cycles like the Allam Cycle.
- The new recuperator may be constructed like an HRSG and has few constraints on size or surface area for a single finned coil heat exchanger.
  - All pressurized streams are contained within piping and tubing. No pressure vessels are required which removes many constraints on unit size.
  - Finned coils that hang from headers may be widened and lengthened to accommodate increased surface area requirements.
  - Inherently flexible thermal growth capability since coils hang freely in the HRSG-like structure and provides rapid start, stop and load change cycle capabilities.
- The new recuperator eliminates the need for multiple parallel trains of heat exchangers.
  - Single heat exchanger system for entire recuperator.
  - Greatly simplified piping layouts due to elimination of thermal expansion loops.
- The new recuperator, when used with liquid metals as the IHTF:
  - Achieves substantial improvement of the overall heat transfer coefficients.
  - Reduces required surface areas and recuperator footprint.

REFERENCES

[1] "8th International Supercritical CO2 Power Cycles Symposium", Feb 27-29, 2024, by Southwest Research Institute, San Antonio, TX.
[2] US DOE website: www.energy.gov/sco2-power-cycles.
[3] Dostal, V. et al, "A Supercritical Carbon Dioxide Cycle for Next Generation Nuclear Reactors", Report MIT-ANP-TR-100, 10 March 2004.
[4] Jiang, Y. et al, "Design and dynamic modeling of printed circuit heat exchangers for supercritical carbon dioxide Brayton power cycles", Applied Energy, Volume 231, 1 December 2018, Pages 1019-1032.
[5] Kwon, J.S. et al, "Compact heat exchangers for supercritical CO2 power cycle application", Energy Conversion and Management, Volume 209, 1 April 2020.
[6] Cai, H.F. et al, "Numerical investigation on heat transfer of supercritical carbon dioxide in the microtube heat exchanger at low Reynolds numbers", International Journal of Heat and Mass Transfer, Volume 151, April 2020.
[7] Allam R.J. et al, "System and method for high efficiency power generation using a carbon dioxide circulating working fluid", Patent US8,596,075B2, 3 December 2013.
[8] Huntington, R.A. et al, "Systems and methods for carbon black production with a gas turbine engine having exhaust gas recirculation", Patent US10,267,270B2, 23 April 2019.
[9] Huntington, R.A., Mittricker, F.F. and Starcher, L.K., "Multiple loop power generation using super critical cycle fluid with split recuperator", Patent US11,840,944B2, 12 December 2023.

[10] Hewitt, G.F. et al, "Process Heat Transfer", CRC Press LLC, Boca Raton & New York, 1994.
[11] "DWSIM - The Open Source Chemical Process Simulator", DWSIM Project, 2023.
[12] Foust, O. J., "Sodium-NaK Engineering Handbook", Vol. 1, US Atomic Energy Commission, 1972.
[13] Shen, Yaou et al, "Analytical Study of Heat Transfer for Liquid Metals in Turbulent Tube Flow", Nuclear Engineering and Design, Vol 373, March 2021.

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