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TECHNICAL AND ECONOMIC FEASIBILITY OF DRY AIR COOLING FOR THE SUPERCRITICAL CO_2 BRAYTON CYCLE USING EXISTING TECHNOLOGY

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ABSTRACT

Potential economic and environmental benefits of the supercritical carbon dioxide (S-CO_2) Brayton cycle for energy production include broad applicability to a variety of heat sources, higher plant efficiency especially as the turbine inlet temperature is increased relative to the traditional Rankine superheated steam cycle, reduced fuel consumption and emissions, reduction in cooling water consumption, and a compact footprint and lower capital cost per unit electrical power. In particular, the S-CO_2 Brayton cycle can be used with dry air cooling eliminating the need for water cooling. The key to making dry air cooling both technically and economically feasible is identification of an effective and affordable CO_2-to-air heat exchanger technology utilizing materials compatible with S-CO_2. A suitable CO_2-to-air cooler design that is similar to that of a radiator with S-CO_2 flowing through horizontal finned tubes across and air supplied by fans upward has been identified. A vendor specification for the cooler is obtained and independently modeled using the Engineering Equation Solver (EES) to confirm the quoted heat exchanger performance and sizing. A cost based optimization procedure is employed to find the optimal cycle operating conditions with a goal of minimizing the plant \$/kWe for the 105 MWe (250 MW_{th}) Advanced Fast Reactor (AFR)-100 Sodium-Cooled Fast Reactor Small Modular Reactor. Cycle operating conditions such as minimum pressure, maximum pressure, and minimum temperature are chosen as the parameters to be optimized. The cycle minimum pressure and temperature are selected along the CO_2 pseudo-critical line to take advantage of the high fluid density near the pseudo-critical point while an increasing minimum CO_2 temperature is dictated by the ambient air temperature. The cycle maximum pressure is varied from 18 to 30 MPa in an attempt to regain some of the lost efficiency due to necessarily higher minimum CO_2 temperature. The mechanical design of the cycle heat exchangers (reactor intermediate sodium-to-CO_2 heat exchanger, high temperature recuperator, and the low temperature recuperator) and the piping are modified as per ASME code requirements to withstand the higher cycle design pressures. The associated change in cost of components is taken into account. All the cycle calculations are performed using the Argonne National Laboratory Plant Dynamics Code (PDC) in conjunction with newly developed air cooler model assuming that the air as at an ambient temperature of 30°C. The air cooler model calculations take into account the tradeoff between the cooler size and the required air circulation power. The calculations indicate that the optimum cycle conditions correspond to a minimum CO_2 pressure of 8.2 MPa, minimum CO_2 temperature of 35°C, and a maximum CO_2 pressure of 25 MPa. Corresponding to these conditions, the plant \$/kWe is only about 1-2% higher than that of a water cooled plant utilizing compact diffusion-bonded CO_2-to-water heat exchanger technology.

INTRODUCTION

The recompression supercritical carbon dioxide (S-CO_2) Brayton cycle has been gaining a lot of interest for energy conversion in nuclear power, and concentrated solar power (CSP) systems because of higher plant efficiency than the traditional Rankine superheated steam cycle, especially, as the turbine inlet temperature is increased past 500^{o}C. Liquid-metal fast cooled reactors with S-CO_2 energy conversion will also eliminate the use of water in reactor or balance-of-plant systems. Therefore, implementing dry air cooling system for this cycle will practically eliminate the need for any significant water use, thereby, increasing the range of applicability of the cycle. Therefore, it is important to investigate the techno-economic feasibility of using air as the ultimate heat sink for S-CO_2 cycles. Previous study at Argonne National Laboratory (ANL) [1] in this regard, concluded that the air cooling option is technically feasible provided that the penalty is paid in terms of both reduced plant efficiency and increased plant capital cost. The study focused on reducing the approach temperature in the cooler (and hence, cooler volume) by increasing the minimum temperature in the cycle and operating the main compressor inlet close to the pseudo-critical line. The calculations showed that increasing the minimum temperature of the cycle to 40^{o}C lead to least plant capital cost per unit electrical output ($/kWe). The previous study made two assumptions which are subject to change in the current study.

- The CO_2-to-air cooler was assumed to be based on Heatric hybrid heat exchanger technology (H^2X – FPHE configuration on the air side) [2]. This will allow for raw material savings over the printed circuit heat exchanger (PCHE) technology but the cost of such a cooler is still significantly higher compared to CO_2-to-water cooler. This is primarily due to the fact that the properties of air and water as a heat transfer media are quite different. Hence, the primary contribution of this study is to identify cheaper alternative CO_2-to-air cooler designs in an attempt to make dry cooling option more practically feasible.

- The cycle maximum pressure was assumed to be 20 MPa. With recent developments in Heatric diffusion bonding technology, PCHEs which can withstand pressures up to 40 MPa can be fabricated. Therefore, by increasing the cycle maximum pressure some of the cycle efficiency lost due to increase in cycle minimum temperature (dictated by ambient air temperature for air cooled cycle) can be recovered, thereby, reducing the plant $/kWe. This will of course increase the cost of heat exchangers, piping etc. which have to be taken into account during calculation of the plant $/kWe. In this study, the cycle conditions are altered and re-optimized to minimize the plant $/kWe in comparison to the reference water cooled plant.

Reference water cooled cycle conditions and heat exchangers design

For the current study, the cycle optimization is performed for the S-CO_2 cycle developed for 100 MWe sodium cooled Advanced Fast Reactor (AFR-100) applications. In order to have a direct comparison between the air and water cooled cycles, the operating conditions and the plant $/kWe of the water cooled cycle are used as reference conditions for the optimization study. Figure 1 shows the operating conditions of the reference water cooled AFR-100 plant. The boundary conditions for the air are assumed to be same as the water boundary conditions i.e. 30^{o}C inlet temperature and 0.101 MPa (1 atm) outlet pressure. The reference dimensions for reactor heat exchanger (RHX), high temperature recuperator (HTR), low temperature recuperator (LTR) are based on the previous cost-based optimization study [3] of water cooled S-CO_2 cycle for the reference conditions (cycle minimum pressure of 7.4 MPa, cycle minimum temperature of 31.25^{o}, and cycle maximum pressure of 20 MPa). When any of the cycle operating conditions are changed, it is required to re-optimize the flow split fraction between compressors as well as the heat exchangers design. For example, split fraction of 0.68 and the heat exchangers designs optimized for water cooled cycle may not lead to optimum $/kWe if cycle maximum pressure is changed to let's say, 25 MPa. Another important mechanical design aspect to keep in mind is that as the cycle maximum pressure is changed the cycle piping have to be-redesigned to withstand the pressure at the design temperature which will affect the capital cost of the plant. Hence, the goal of the current study is to investigate the effect of cycle minimum pressure, minimum temperature, and maximum pressure on the power plant $/kWe and find the optimum cycle conditions to minimize $/kWe.

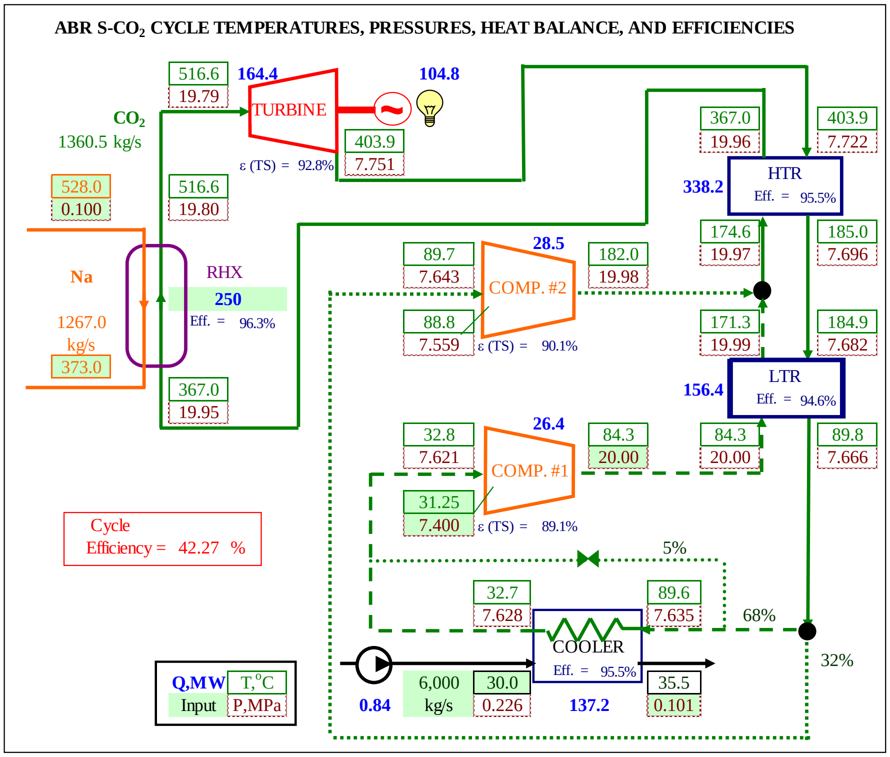

Figure 1. Reference AFR-100 water cooled S-CO_2 cycle calculations

SELECTION OF CO_2-TO-AIR COOLER

As mentioned previously, use of compact diffusion bonded heat exchangers as CO_2-to-air cooler significantly increases the capital cost of the cooler. In an effort to reduce the plant $/kWe, alternative options for the CO_2-to-air cooler were explored. In this section, details of the selected cooler design and the cooler calculations for the reference cycle conditions are summarized. Figure 2 shows the CAD model of a cooler module selected for the current study. As can be seen, the CO_2 and air are setup in cross flow arrangement with CO_2 flowing inside the finned tubes and the fans blow air over the tubes in a fashion similar to that of a car radiator. For each cooler module, CO_2 undergoes three passes with two mixing chambers to ensure uniform flow temperature between the passes. The design uses three fans per module and it is assumed that these fans distribute air flow uniformly throughout the cooler module. This assumption is required for modeling of the cooler module and the calculations later confirmed that it was indeed a fair assumption. A quotation for this cooler was obtained from Harsco Industrial Air-X-Changers [4] for the reference cooler conditions in Figure 1 and a model was developed in Engineering Equation Solver (EES) [5] to confirm the performance and sizing of the cooler. The EES code in conjunction with the ANL PDC is used for the cost-based optimization that will be described here. The basic idea of CO_2-to-air cooler modeling is to discretize the heat exchanger module into number of nodes for each row and apply the friction factor, and Colburn j-factor correlations for air and CO_2 sides to calculate the pressure drop and outlet temperature of both the streams. The detailed description of the correlations used, discretization procedure, and solution procedure is provided in the ANL internal report [6] prepared as part of this work. Table 1 compares the vendor quoted cooler specifications and the calculations from the EES model for the reference cooler conditions. It is evident that the EES calculations overall matches well with the vendor quotation with small differences in calculated air and CO_2 outlet temperatures. Figure 3 shows an example of CO_2 and air temperature profiles in each cooler module for the reference cooler conditions. One should notice that the significant amount of CO_2 temperature reduction happens in the first two passes of the module. As the bulk temperature of CO_2 approaches the pseudo-critical temperature (defined as the temperature at which specific heat reaches a maximum value for a given pressure), significant increase in the number of cooler modules or flow rate of air is needed in order to remove rest of the waste heat from the CO_2. This means that even a 0.5°C increase in the minimum temperature of the cycle at reference minimum pressure (7.4 MPa) would reduce the number of cooler units by roughly 33%. Of course by doing this one would have to pay penalty in the form of reduced cycle efficiency. Hence, the $/kWe metric needs to be more carefully investigated around the pseudo-critical point for any potential savings.

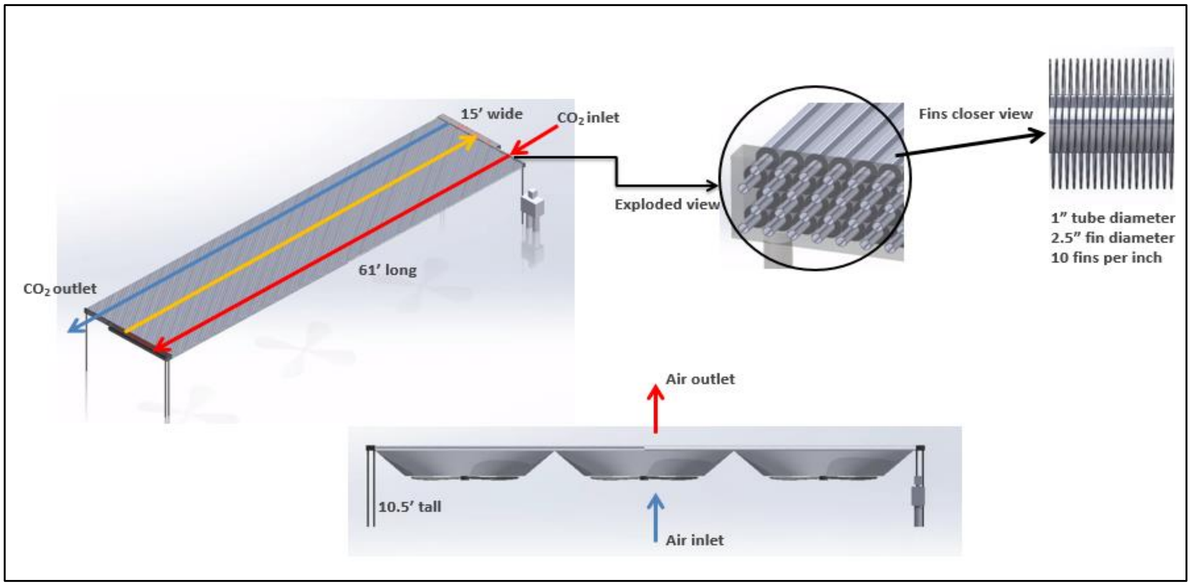

Figure 2. CAD model of the selected CO_2-to-air cooler, a six-foot tall person is shown for reference

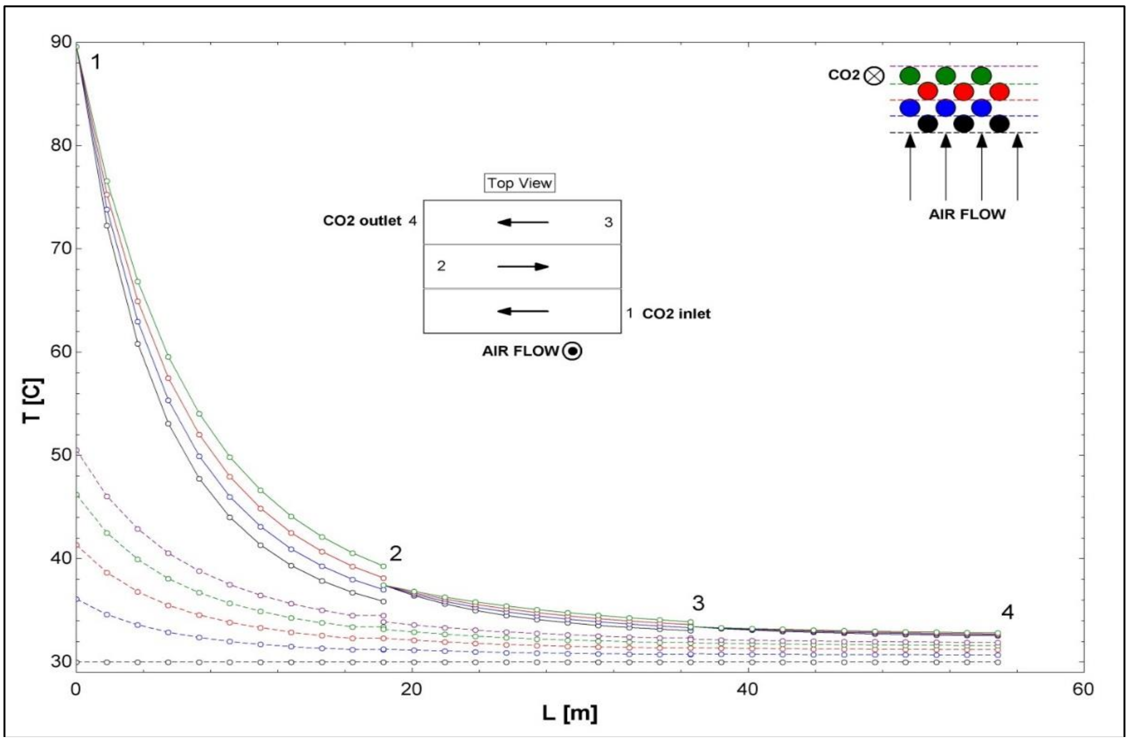

Figure 3. An example of CO_2 and air temperature profiles in each cooler module, solid lines represent CO_2 and dashed lines represent air

Table 1. Comparison of manufacturer specifications and model calculations for reference cooler conditions

	Variable	Harsco Industrial Air-X-Changers	Calculated (EES model)	Calculated (EES model)
Inputs	Number of cooler units	86	86	86
	CO_2 flow rate per unit [kg/s]	10.22	10.22	10.22
	CO_2 inlet temperature [^oC]	89.61	89.61	89.61
	CO_2 inlet pressure [MPa]	7.736*	7.736*	7.635
	Air flow rate per unit [kg/s]	317.2	317.2	317.2
	Air inlet temperature	30	30	30
Outputs	Heat transfer capacity [MW_{th}]	1.691	1.696	1.61
	CO_2 outlet temperature [^oC]	32.7	33.12	32.64
	CO_2 pressure drop [kPa]	6.895	6.645	6.802
	Air outlet temperature [^oC]	52	51.2	51.11
	Air pressure drop [kPa]	Not provided	-	0.1112

*The CO_2 inlet pressure provided in the Harsco quotation didn't match the reference inlet pressure

For the calculations presented in Table 2, number of cooler units is used as the input and the CO_2 outlet temperature is calculated for verification of the code. However, for rest of the study, the developed EES code is used to calculate number of required cooler units ($\mathrm{N_{cooler,units}}$) using CO_2 inlet temperature and pressure, CO_2 outlet temperature as the inputs. The cost of a Harsco heat exchanger unit, $\mathrm{cost_{cooler,unit}}$ is obtained from the manufacturer quote and is assumed to be constant in further calculations in this report. Once the number of required cooler units is calculated, power consumption for the cooler and total cost of the cooler are calculated using Equations (2) and (3) respectively.

$$P_{fans} = 3 \cdot \dot{W}_{per,fan} \cdot N_{cooler,units} \quad (2)$$

$$cost_{cooler} = cost_{cooler,unit} \cdot N_{cooler,units} \quad (3)$$

COMPONENTS DESIGN AND COST METHODOLOGY

As mentioned in the introduction, the cycle minimum pressure, minimum temperature and maximum pressure are chosen as the variables for optimization of the S-CO_2 cycle using modified CO_2-to-air cooler design described in the previous section. For an air cooled cycle, cycle minimum temperature is generally dictated by the ambient air temperature of the power plant location. For a given minimum temperature, the cycle minimum pressure is selected near the pseudo-critical pressure to exploit the high fluid density during compression process. For cycle minimum temperatures > 31.25^{o}C, the current study showed that it is more economical (lower $/kWe) to select the cycle minimum pressure that is slightly greater than the pseudo-critical pressure. The cycle maximum pressure is limited by the pressure containment capability of the cycle heat exchangers and piping. All the cycle heat exchangers excluding the air-to-CO_2 cooler are envisioned to be diffusion bonded compact heat exchangers from Heatric [2]. Heatric mentions that for a maximum operating temperature of 500^{o}C, PCHEs can handle pressure differentials as high as 400 bar (40 MPa). For the current study, 300 bar (30 MPa) was chosen as the upper limit for the cycle maximum pressure and Table 2 summarizes the selected conditions for the cost-based optimization study.

Table 2. Selected cycle conditions for the optimization study

Minimum pressure (MPa)	Pseudo-critical pressure (MPa)	Minimum temperature [^{o}C]	Maximum pressure [MPa]
7.4	7.422	31.25	18 – 30
7.628	7.628	32.5	18 – 30
8	8.040	35	18 – 30
8.864	8.864	40	19 – 30
9.688	9.688	45	20 – 30

As the cycle maximum pressure is increased, the mechanical design of the following components needs to be changed to withstand the design pressure.

- Reactor heat exchanger (RHX)
- High temperature recuperator (HTR)
- Low temperature recuperator (LTR)
- S-CO_2 cycle piping
- Turbomachinery components (compressors and turbine) – Not implemented in the current study

PCHE design methodology and cost estimation

Design modifications to be heat exchangers (RHX, HTR, and LTR) are implemented as per the ASME 13-9 code requirements and the design equations are summarized in the Heatric paper [7]. The commercially available Heatric PCHE units are fabricated by chemically etching semicircular channels with zig-zag pattern on a substrate plate and the plates are then diffusion bonded together to form a monolithic block. In order to simplify the mechanical design process, the channels are approximated as rectangular channels. Figure 4 shows the approximated rectangular channels along with the nomenclature.

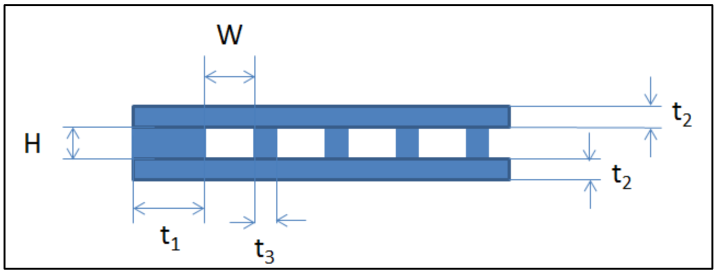

Figure 4. Approximated PCHE channels for mechanical design and the associated nomenclature

In the figure, t_2 represents the thickness of the plate after etching, t_3 represents the ridge width, $W=d$ is the channel width, $H=d/2$ is the channel depth assuming that the channels are semi-circular, and t_1 is the edge thickness. Where, d is diameter of the semicircular channels. The procedure for calculation of the edge thickness (t_1), plate thickness (t_2), and ridge thickness (t_3) involves calculation of the membrane stress (S_m) and bending stress (S_b) experienced by these members when subjected to the design pressure. These equations can be found in either ASME section 13-9 or Heatric paper [7]. Once the membrane stress and bending stress are calculated, the total tress is known from Equation (4).

$$S_T = S_m + S_b \tag{4}$$

Design pressure used to calculate the stresses is selected to be 10% greater than the cycle maximum pressure to ensure safety margin at a particular design temperature. For the current study, the dimensions of the channels for all the heat exchangers are kept same as their respective reference designs in Table 1. The mechanical design is considered to be successful when the following criteria are met.

$$S_m \leq SE \tag{5}$$
$$S_T \leq 1.5SE \tag{6}$$

Where, E is the joint factor and is 0.7 for the diffusion bonded block based on Heatric's conservative assumption, S is the maximum allowable stress of the heat exchanger material (in this case it is 316 stainless steel) and is a function of the design temperature.

Table 3. Design parameters for the S-CO_2 cycle PCHEs

Parameters	RHX	HTR	LTR
HEX type	Z/I PCHE	Platelet PCHE	Platelet PCHE

Unit length (m)	1.5		0.6		0.6	
Unit width (m)	0.6		1.5		1.5	
Unit height (m)	0.6		0.6		0.6	
Design temperature (°C)	550		450		300	
Design pressure (MPa)	18 – 30		18 – 30		18 – 30	
	Hot side (Na)	Cold side (CO_2)	Hot side (CO_2)	Cold side (CO_2)	Hot side (CO_2)	Cold side (CO_2)
Channel diameter (mm)	6.0	2.0	1.3	1.3	1.3	1.3
Channel depth (mm)	4.0	1.0	0.65	0.65	0.65	0.65
Pitch to diameter ratio	1.083	Variable	Variable	Variable	Variable	Variable
Plate thickness (mm)	Variable	Variable	Variable	Variable	Variable	Variable

Table 3 shows the design parameters for the PCHEs relevant to the current study. As mentioned earlier, the channel dimensions, and design temperature are kept constant for all the conditions and the plate thickness, and ridge thickness (hence, channel pitch-to-diameter ratio) are calculated as a function of the design pressure. Channel pitch is equal to the sum of ridge thickness and the channel diameter. These values are updated in ANL PDC inputs as the cycle maximum pressure is changed. Please note that the plate thickness in table 3 is the original plate thickness before etching and is assumed continuous throughout the layer.

A proper capital cost estimation for the PCHEs is needed for the cost-based optimization technique that will be described in the next chapter. A simplified capital cost estimate procedure for the cycle PCHEs is as follows [3].

- The mass of raw material required for fabrication of each PCHE unit is calculated from the volume of the PCHE unit and the material density

$$M_{block} = V_{block} \cdot \rho_{ss316(@20^0C)}$$

- The material cost of SS316 is chosen as 7.64 $/kg in order to be consistent with previous optimization studies performed at ANL and the total material cost of one PCHE block is calculated. Please note that the current market material cost for SS316 might be different and the cost of the cycle heat exchangers (including cooler) can be different depending on when the purchase is made.

- The fabrication cost to perform chemical etching and diffusion bonding depends on the type of PCHE. For example, Z/I PCHE as in the case of RHX is less expensive than the platelet PCHE as in the case of HTR and LTR.

- The total cost of each PCHE block is the sum of material cost and fabrication cost [3]. The capital cost of full PCHE is calculated as,

$$cost_{PCHE} = cost_{PCHE,block} \cdot N_{blocks}$$

Note that PCHE in the above equation can be either RHX, HTR, LTR, or water cooler. The additional costs such as costs associated with welding of blocks and headers, as well as engineering and shipping costs are neglected as they are considered to be small compared to the fabrication and material costs.

Piping design methodology and cost estimation

Plant piping is designed as per ASME B31.1 process piping guidelines and the design equations are summarized in this section. The design process involves calculation of minimum required wall thickness for a known pipe inner (or outer) diameter, design pressure, and design temperature. The required minimum wall thickness is calculated using Equation (7).

$$t_{min} = \frac{PD_i}{2(SEW+PY)(1-UTP-CA)-P} \tag{7}$$

Where,

t_{min} = minimum wall thickness.
D_i = inner diameter of the pipe s
E = quality factor and is equal to 1 for seamless pipes (ASME B31.3 Table 302.3.4)
W = weld joint quality factor and is equal to 0.975 (ASME B31.3 Cl. 302.3.5)
Coefficient Y is equal to 0.4 (ASME B31.3 Table 304.1.1)
P = internal pipe pressure or the design pressure
S = maximum allowable stress and is a function of design temperature
UTP = Under tolerance allowance to account for manufacturing tolerances
CA = Corrosion allowances in percentage

The piping inner diameters are selected based on previous optimization study to keep the pressure drop in the piping to reasonable values [3]. The pipe lengths are selected from the S-CO_2 plant layout for AFR-100 developed at ANL [8] and is shown in Figure 5. A total of 25 pipe sections are identified and Figure 6 shows the numbering of nodes, and sections in a schematic of the layout as used in the PDC. The lengths (L_{pipe}) and inner diameters of each pipe section are summarized in Table 4. The wall thickness for these pipe sections are calculated based on the design pressure and design temperature for that particular section. The pipe lengths in the PDC are left unchanged from previous settings because it requires re-optimization of the pipe inner diameters to ensure that the cycle efficiency is unaffected by the choice of pipe inner diameters.

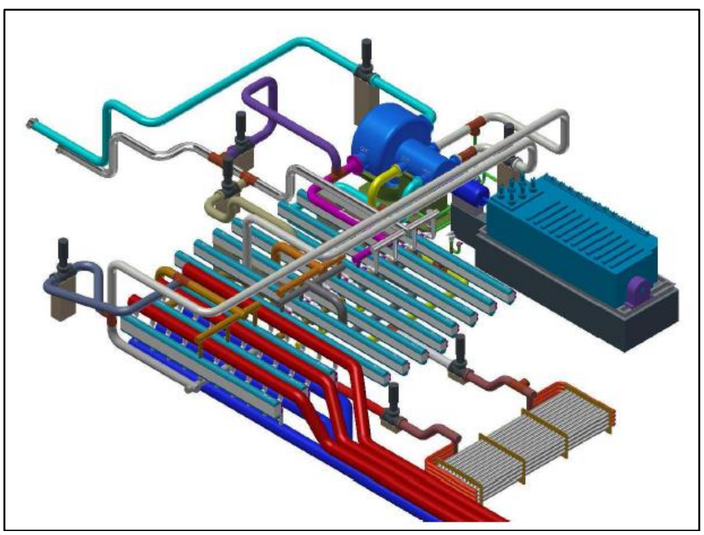

Figure 5. S-CO_2 Brayton cycle layout for AFR-100 developed at ANL [8], Aqua pipe is the inlet line from Na-to-CO_2 heat exchanger and Silver pipe is the return line to Na-to-CO_2 heat exchanger

The cost of each pipe section is calculated as follows,

$$D_o = D_i + 2t_{min}$$
$$V_{pipe} = \frac{\pi}{4}.(D_o^2 - D_i^2).L_{pipe}$$
$$cost_{pipe} = V_{pipe}.\rho_{ss316(@20^0C)}.cost_{ss316}$$

The material cost of SS316 ($cost_{ss316}$) is 7.64 $/kg. The total capital cost of the piping ($cost_{piping}$) is simply calculated as sum of cost of all the 25 pipe sections. It should be noted that the fabrication cost, costs associated with welding of pipe sections are ignored in the piping cost estimation as these costs are hard to estimate beforehand.

Turbomachinery components design methodology and cost estimation

Design procedure for the turbomachinery components is not straight forward and requires attention to lot of details. For example, the blade dimensions are not just a function of cycle maximum pressure but also depends on other details like number of stages, hub diameter etc. Moreover, the casing to withstand the high pressure will have to be designed according to the calculated blade dimensions. Due to lot of complications involved, no changes were made to the PDC turbomachinery inputs. The turbomachinery inputs to the PDC are based on previous optimization study at the ANL [3]. Consequently, cost estimation for the turbomachinery components is difficult and these costs are treated as constant in the cost-based optimization procedure described below

Section	D_i (m)	L_{pipe} (m)
1	0.68302	29
2	0.68302	2
3	0.68302	12
4	0.68302	30
5	0.5	9.25
6	0.5	5.5
7	0.5	13
8	0.5	9.25
9	0.5	5
10	0.5	55
11	0.5	21
12	0.5	5
13	0.5	10
14	0.5	10
15	0.68302	10
16	0.68302	15
17	0.68302	2
18	0.5	12
19	0.45	38
20	0.25	17
21	0.25	21
22	0.25	11
23	0.25	11
24	0.25	12
25	0.25	25

Table 4. S-CO₂ cycle piping inner diameters and lengths, based on AFR-100 layout

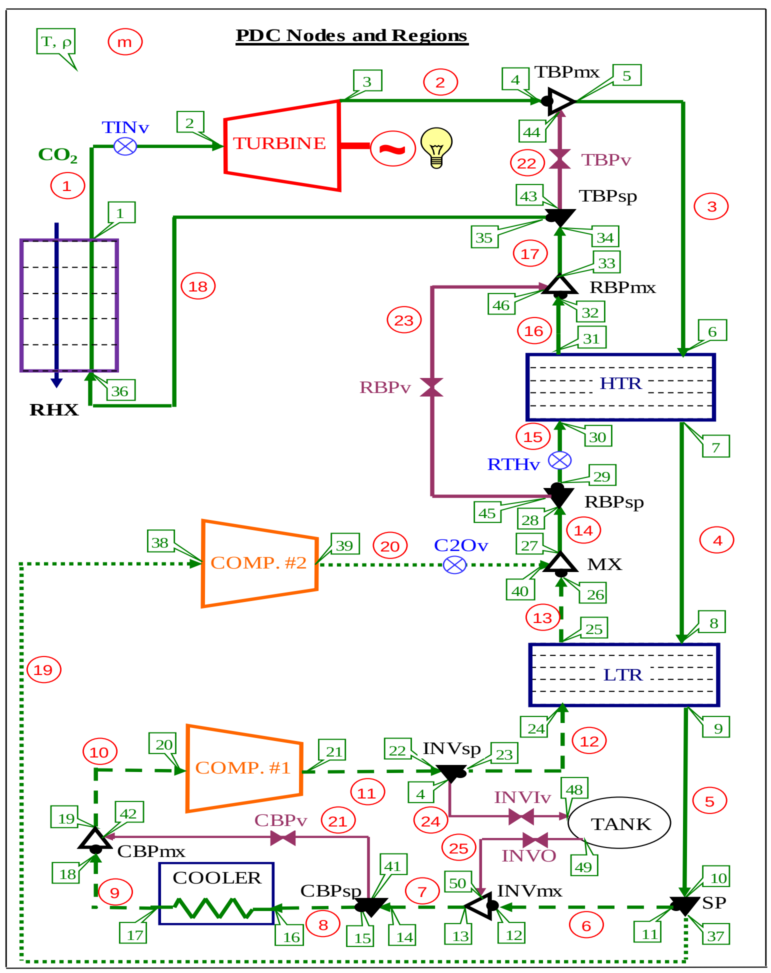

Figure 6. PDC Nodes and regions for identifying pipe sections in S-CO₂ cycle

COST-BASED OPTIMIZATION

For a power plant it is important to take into account the plant net electrical output as well as the capital cost of the plant. Usually, there is a trade-off between these two and a cost-based optimization method is employed to find the optimum operating conditions. This section summarizes details of the cost-based optimization technique employed to perform the plant optimization. The plant capital cost per unit electrical output ($/kWe) is calculated using Equation (8) and takes into account the changes in cost of heat exchangers, piping for different cycle conditions.

$$\frac{\$}{kWe} = \frac{cost_{rest} + cost_{RHX} + cost_{HTR} + cost_{LTR} + cost_{air\,cooler} + cost_{piping}}{P_{elec} - P_{fans}} \tag{8}$$

Where, $cost_{rest}$ is the capital cost of the rest of the plant, i.e. excluding the components being optimized here: RHX, HTR, LTR, cooler, and the piping. And $cost_{RHX}$, $cost_{HTR}$, $cost_{LTR}$, $cost_{air\ cooler}$, $cost_{piping}$ is the cost of reactor heat exchanger, high temperature recuperator, low temperature recuperator, CO_2-to-air cooler, plant piping respectively. P_{elec} is the work output from the cycle, and P_{fans} is the work input to operate the CO_2-to-air cooler fans.

Several assumptions are made in the process of calculating the $\$/kW_e$ and these assumptions are as follows:

- The plant capital cost per unit electrical output, including the cost of heat exchangers and piping costs, is assumed to be equal to 4,780 $/kWe for the reference water cooled S-CO_2 plant with net electrical output of 104.8 MWe (see Figure 1).

- Rest of the plant capital cost ($cost_{rest}$) is calculated by excluding the cost of heat exchangers and piping costs for the reference plant and is assumed to be constant throughout the optimization study. This value is $4,480 $/kWe for net electrical output of 104.8 MWe.

- The model for CO_2-to-air cooler design is not yet implemented in the PDC, hence, the optimization is performed assuming that the cooler is reference PCHE design. The number of cooler units is kept constant at 72 throughout the optimization (Refer to [1] for further details). After the optimization is complete, the cooler operating conditions are exported to the CO_2-to-air cooler EES model to calculate the required fan power (P_{fans}) and cost of the cooler ($cost_{cooler}$) as a function of the number of cooler units.

A Matlab code was developed to automate the calculations during the process of optimization for different cycle operating conditions. Ideally the optimization process for all the components should be performed simultaneously, but such a process would require enormous amount of computational time. In order to reduce the computational effort and simplify the optimization process, a sequential optimization method was used. This allows for optimization of individual components. The flow chart of the Matlab code is shown in Figure 7. For the calculation of $/kWe during optimization, the code replaces fan power with pump power for CO_2-to-water cooler. Prior to beginning of the optimization, the cycle minimum pressure, minimum temperature, and maximum pressure are set as inputs to the code and the channel pitch-to-diameter ratio, and plate thickness of the PCHE heat exchangers are modified in the PDC input files. The number of heat exchanger units for each heat exchanger is reset to the reference values [1] (96 RHX units, 48 HTR units, 48 LTR units) to start the optimization with the same initial condition for all the cases.

Optimization of CO_2-to-air cooler

For the CO_2-to-air cooler optimization, the cooler conditions after optimization of other components (split fraction, RHX, HTR, and LTR) are exported from the PDC to the EES code for CO_2-to-air cooler. The code uses the CO_2 inlet temperature and pressure as well as, CO_2 outlet temperature as inputs to calculate the required fan power, the cooler cost, and the CO_2 pressure drop for a given number of cooler units. All the calculations are performed for fixed ambient air temperature of 30^{o}C and air outlet pressure of 0.101 MPa in order to be consistent with the water cooled cycles. Using smaller number of air cooler units will decrease the cooler cost but increases the required fan power (for the required heat removal rate) and vice versa. Therefore, the number of cooler units is varied and the \$/kWe is calculated to find the optimum number of cooler units.

The cost-based optimization technique introduced above is applied for different cycle conditions and the results are presented in this section. The calculations are performed for two scenarios as described below.

- In the first scenario, the PDC inputs to the turbomachinery components (turbine and compressors) are left unchanged (i.e., the number of turbine and compressor stages are kept fixed). For this scenario, depending on the cycle conditions the isentropic efficiency of the compressors and turbine can drop significantly.

- In the second scenario, the turbine design part of the PDC is skipped and a constant static-to-static efficiency (called turbine efficiency from hereon) of 93.4% is assumed for the turbine. Please note that this corresponds to total-to-static efficiency of 92.8% for the reference conditions (see Figure 1). The PDC inputs to the compressors is left unchanged. To achieve a constant turbine efficiency of 93.4% for different operating conditions, it is required to either increase/decrease the number of turbine stages or implement other modifications to the turbine. These modifications will have significant amount of cost associated with them which are not accounted for during calculation of the plant \$/kWe.

In the next two subsections the calculations for both these scenarios will be presented, followed by cost-based optimization results to investigate the sensitivity of cycle minimum pressure on the plant \$/kWe near the pseudo-critical pressure.

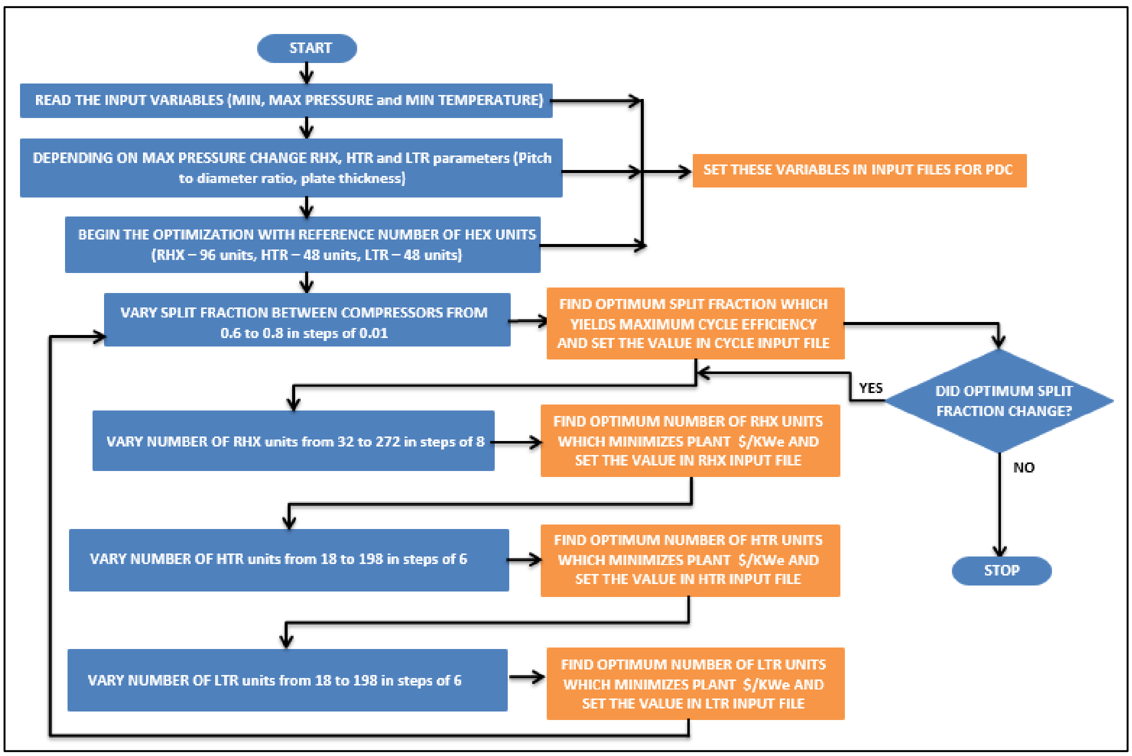

Figure 7. Matlab optimization code flow chart

Plant optimization for fixed turbomachinery inputs

The plant efficiency and the plant capital cost as a function of the cycle maximum pressure are plotted in Figure 8 for different cycle minimum pressures and temperatures listed in Table 2. The net plant electrical output (calculated based on the cycle electrical output minus the fan power consumption) is used to calculate the plant efficiency in Figure 8. The fluid density decreases as the cycle minimum temperature increases which increases the work input to the compressor, thereby, decreasing the cycle and plant efficiency as can be seen in the figure above. An exception to this is that the plant efficiency for 7.628 MPa, 32.5°C case is higher than that of 7.4 MPa, 31.25°C case. The reason for this is that the selected cycle minimum pressure is equal to the pseudo-critical pressure for the 32.5°C whereas the cycle minimum pressure for 31.25°C case is lower than the pseudo-critical pressure (see Table 2). Also plotted in the figure is the plant efficiency for the reference water cooled cycle. The plant efficiency of the air cooled cycle (for 7.4 MPa, 31.25°C case) is lower than that of the water cooled cycle due to an increase in the fan power consumption for the CO_2-to-air cooler. For the cost-based optimization described in the previous section, the optimization parameters (split fraction, and number of heat exchanger units) are changed in steps. This is the reason for rugged nature of the curves in Figure 8. Choosing smaller step sizes during the optimization will make these curves smoother but will increase the computational time.

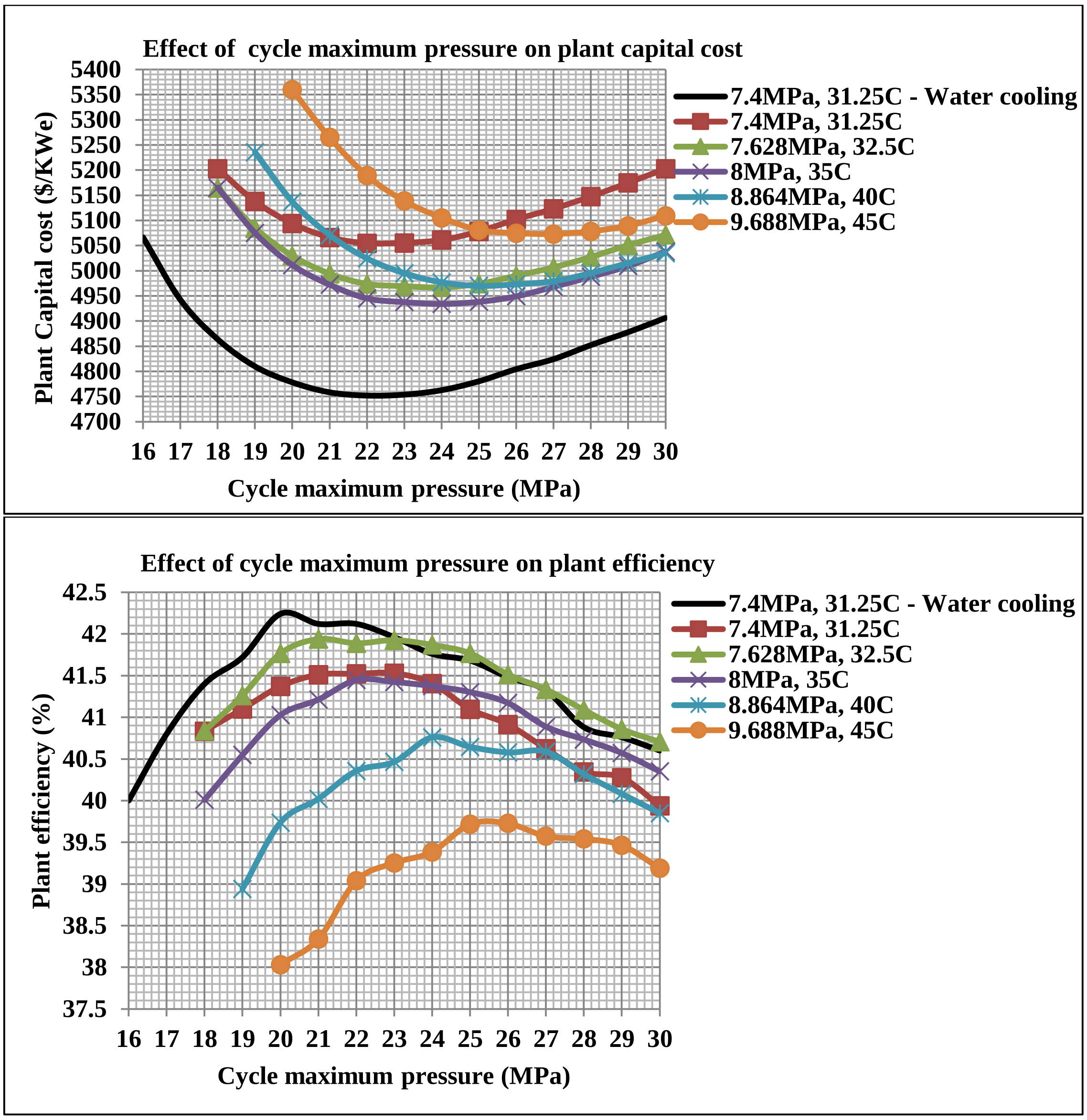

Figure 8. Effect of cycle maximum pressure on plant efficiency and capital cost for fixed turbomachinery inputs

For 7.4MPa, 31.25°C case the optimum cycle maximum pressure is around 22 MPa and the plant capital cost for the air cooled cycle increased to 5054 $/kWe. This corresponds to about 6% increase in the capital cost compared to the capital cost of the reference water cooled cycle (4780 $/kWe). For the ambient air temperature of 30°C, the plant $/kWe benefits greatly by increasing the cycle minimum temperature from 31.25°C to 32.5°C and selecting the cycle minimum pressure as the pseudo-critical pressure (7.628 MPa for 32.5°C). For this case the optimum cycle maximum pressure is around 24 MPa and the plant capital cost decreases to 4967 \$/kWe. This corresponds to nearly 2% decrease in the capital cost when compared to the capital cost of the 7.4 MPa, 31.25°C case. Out of all the conditions in Figure 8, the optimum case for the air cooled cycle is noted to be 8 MPa, 35°C case with maximum cycle pressure of 24 MPa. For this optimum case, the plant capital cost is 4934 \$/kWe which is only about 3% increase in the plant \$/kWe compared to the reference water cooled cycle. This is a significant improvement in the air cooled S-CO_2 plant economics when compared to the previous PCHE air cooler technology which resulted in about 40% increase in the \$/kWe [1]. In the results presented above, one would guess that the plant efficiency will increase as the cycle maximum pressure increases, however, it is clearly evident that the plant efficiency decreases for higher maximum cycle pressure for all the conditions. We believe that there are two possible factors to this observation as listed below.

- One reason might be related to the drop in total-to-static efficiency of the compressors and the turbine as the cycle maximum pressure is increased as can be seen in Figure 9. By increasing the cycle maximum pressure from 18 MPa to 30 MPa, the turbine efficiency drops by almost 2% and the compressors efficiency drops by almost 4%.

- Also, the decrease in plant efficiency might be a result of the cost-based optimization. For higher cycle pressures, it is more economical to use fewer number of the heat exchanger units (to reduce the capital cost of the heat exchangers) and compromise on the cycle efficiency.

In order to investigate the influence of these factors on the plant efficiency (also \$/kWe), the static-to-static efficiency of the turbine is fixed constant and the calculations are repeated again. These calculations are presented in the next section.

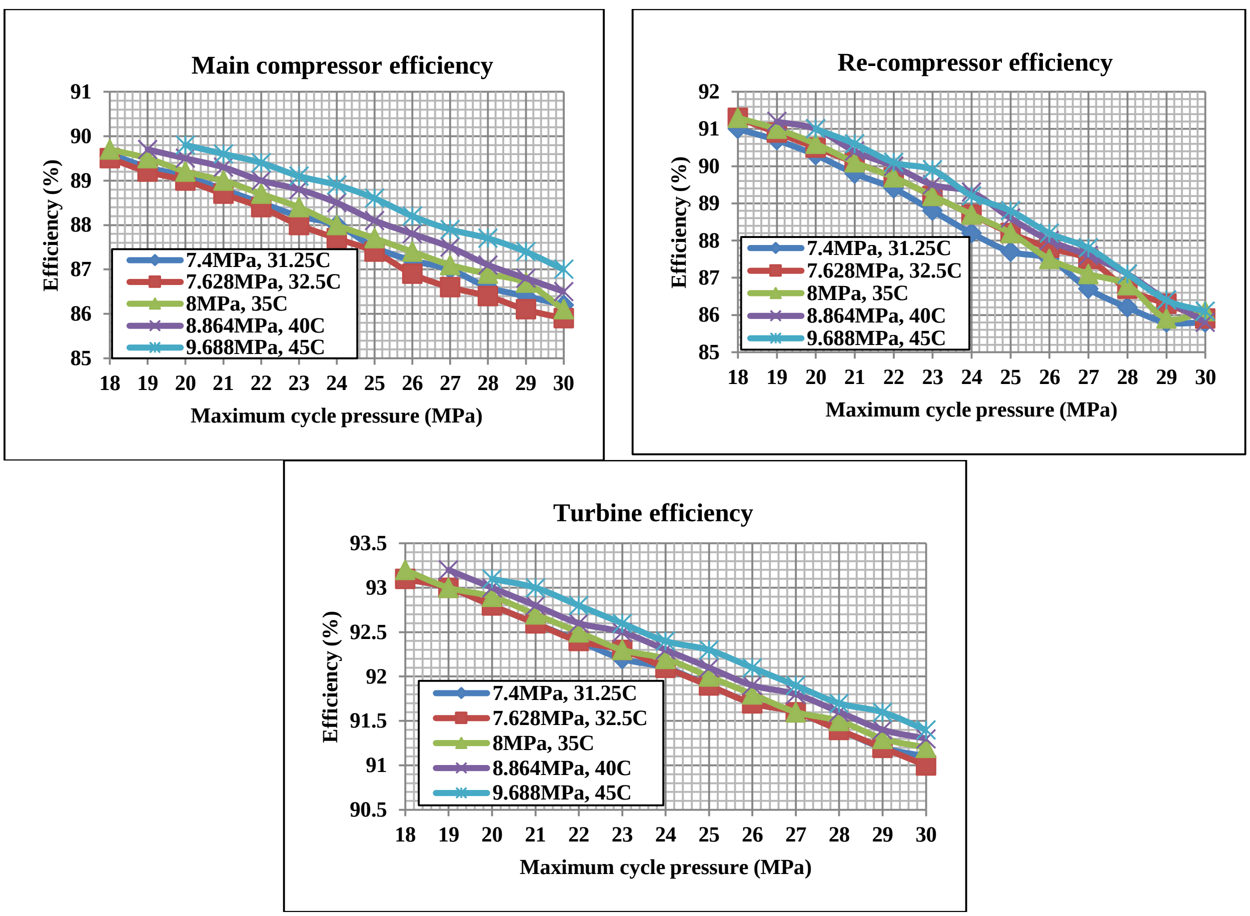

Figure 9. Drop in isentropic efficiency of compressors and turbine for fixed turbomachinery inputs

Plant optimization for fixed static-to-static turbine efficiency

Since, the turbine power output is almost 4 times that of the power input to the compressors (see Figure 9), a 2% drop in turbine efficiency in the previous section is expected to have larger impact on the plant efficiency than that of the drop in compressors efficiency. For this reason, the turbine design part of the PDC is skipped and the cost-based optimization calculations are repeated with a fixed turbine efficiency. Figure 10 presents the plant efficiency and plant capital costs calculations for a fixed static-to-static turbine efficiency of 93.4%. Similar to the previous section, there plant efficiency drops for higher cycle maximum pressures. However, drop in plant in plant efficiencies for this particular case occurs at higher pressures than the previous case. The optimum cycle conditions tend to shift to higher cycle maximum pressures which is expected due to an increase in the cycle efficiency. For example, the optimum cycle maximum pressure for 8.864 MPa, 40°C case is 25 MPa and 27 MPa for the fixed turbomachinery inputs case and the fixed turbine efficiency case respectively.

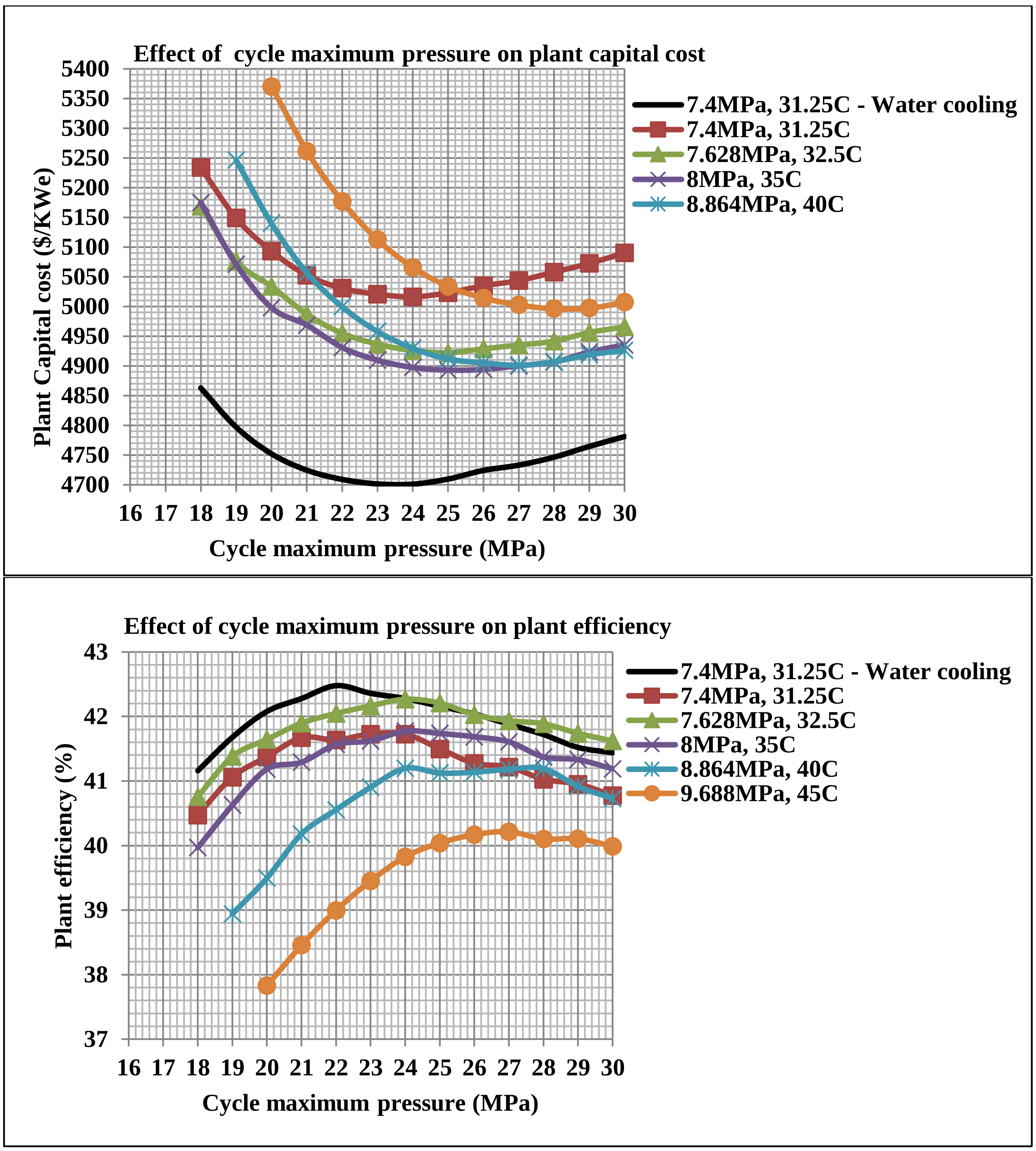

Figure 10. Effect of cycle maximum pressure on plant efficiency and capital cost for fixed turbine efficiency

Unfortunately, neither of the calculations presented in the previous section or this section take into account the turbomachinery costs to conclude which of these calculations can be considered more accurate. We believe that if the turbomachinery costs are taken into account both of these calculations will yield similar results. For the fixed turbine efficiency case, the optimum cycle conditions are 8 MPa, 35ºC with maximum cycle pressure of 25 MPa. The plant capital cost for this case is 4893 $/kWe which is about 2.5% higher than that of the reference water cooled cycle (4780 $/kWe). In the future, when a proper cost estimation procedure for the turbomachinery components is developed, it is recommended to implement the design changes to the compressors and turbine and re-optimize the cycle to obtain more accurate results for the plant $/kWe.

Selection of cycle minimum pressure around the pseudo-critical pressure

In the previous section, the optimum plant $/kWe values for the cycle minimum temperatures of 32.5ºC, 35ºC, and 40ºC are very close to each other (less than 1% difference in Figure10). In this section, the sensitivity of choice of cycle minimum pressure in the vicinity of pseudo-critical point is investigated for these three cases. These calculations will aid in proper selection of the cycle minimum pressure for a given minimum temperature.

Figure 11 presents the plant efficiency and plant $/kWe calculations for the cycle minimum temperature of 32.5ºC for various minimum pressures.

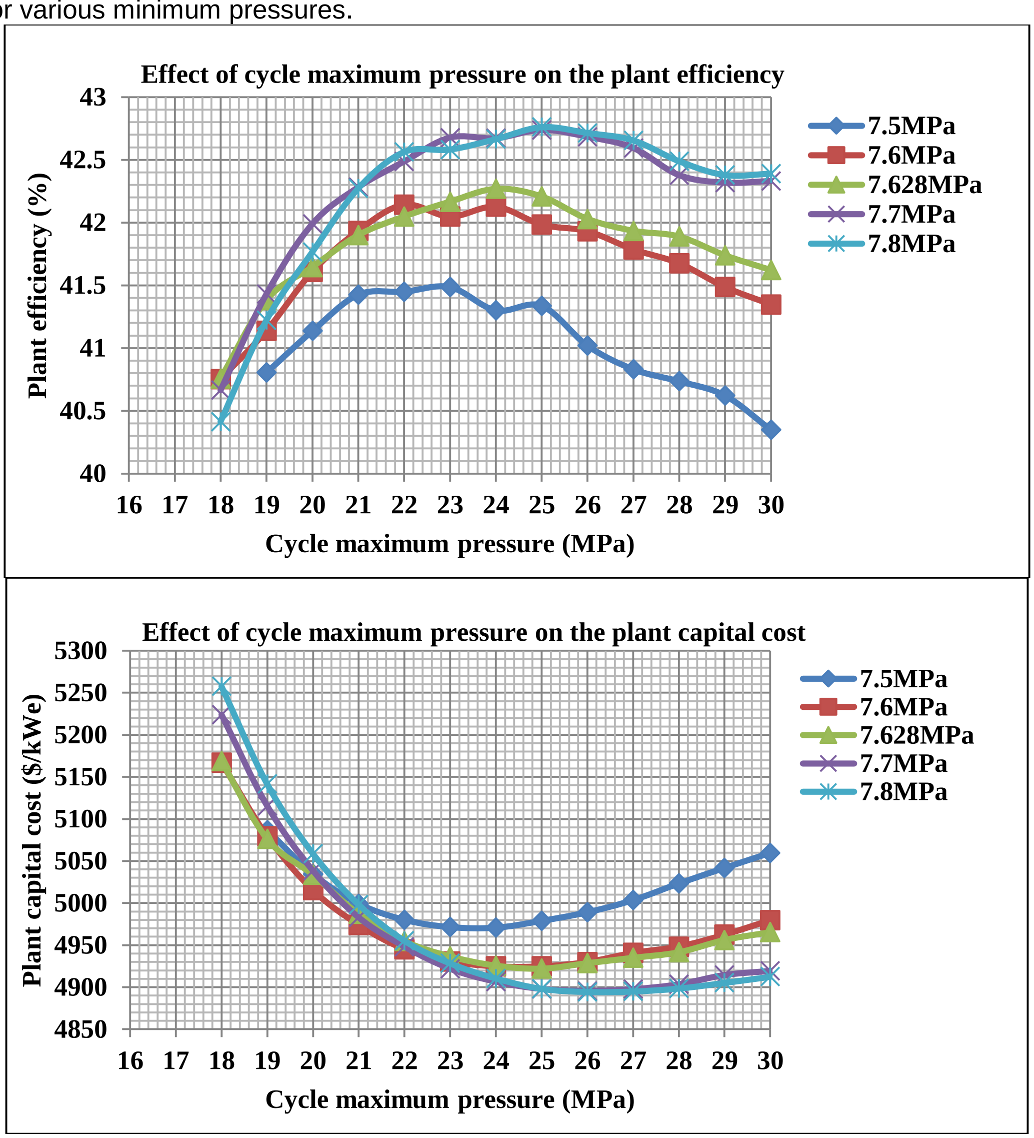

Figure 11. Effect of cycle maximum pressure on plant efficiency and $/kWe for cycle minimum temperature of 32.5ºC, different cycle minimum pressures

Please note that these results are obtained using the same cost-based optimization procedure described earlier. The plot clearly indicates that the cycle benefits greatly (due to reduction in compressor work from increased fluid density) by selecting the cycle minimum pressure greater than the pseudo-critical pressure (7.628 MPa for 32.5°C). As the cycle maximum pressure is increased, the plant efficiency for 7.7 MPa case is about 1% higher than that of the 7.628 MPa case. This increase in the plant efficiency is also reflected in the plant capital cost. By increasing the cycle minimum pressure from 7.628 MPa to 7.7 MPa the optimum plant capital cost can be reduced from 4922 $/kWe to 4894 $/kWe (about 0.5% reduction in the capital cost). Since, the results for 7.7 MPa and 7.8 MPa minimum pressures are almost identical, cycle minimum pressure of 7.7 MPa is chosen as the optimum for this case.

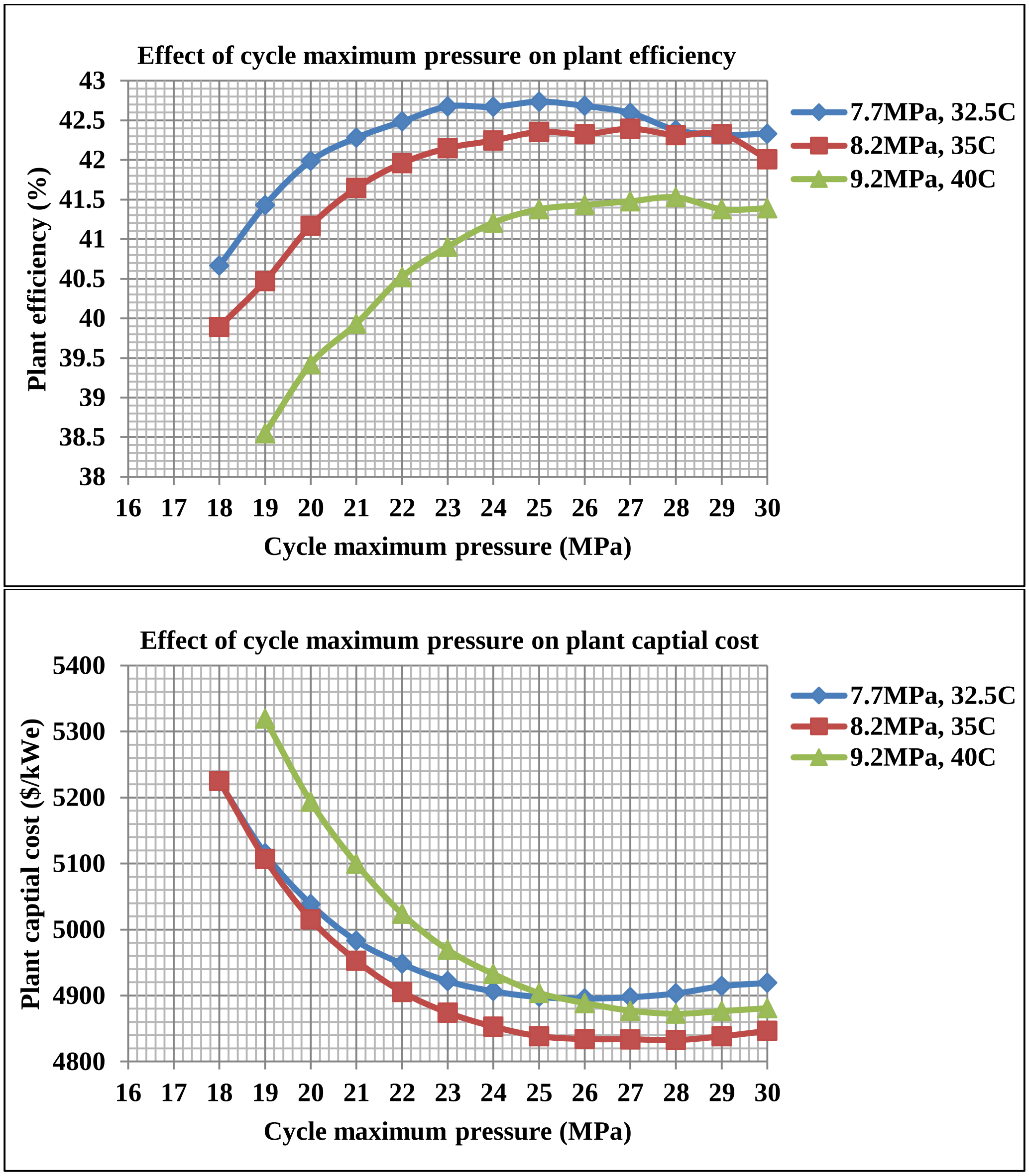

Figure 12. Effect of cycle maximum pressure on plant efficiency and $/kWe for the selected cycle minimum pressures and temperatures

Similar calculations are performed for other minimum temperatures as well and the optimum cycle minimum pressures (resulting in the least $/kWe) for each of these minimum temperatures are selected and the results are plotted in Figure 12 for the final selection of the optimum cycle operating conditions. Strictly going by these plots, the optimum cycle conditions would be minimum pressure of 8.2 MPa, minimum temperature of 35ºC, and maximum pressure of 28 MPa. However, the calculations presented in this report didn't account for lot of costs which are affected by increase in the cycle maximum pressure. For example, the 28 MPa valves can be significantly expensive than 25 MPa valves. Similarly, one should also keep in mind that the heat exchanger headers will have to be thicker and welding of the thick headers, and plant piping can be an expensive task. Higher cycle pressure also means more safety concern and one should keep in mind the costs associated with the CO_2 leak in the heat exchangers (especially in the RHX, where CO_2 interacts with Na in the case of a leak), CO_2 leak into the turbine building etc. All these costs are hard to estimate and a proper engineering judgement has to be used while selecting the maximum pressure for the cycle. For the reasons stated above, the optimum maximum pressure is chosen as 25 MPa rather than 28 MPa (For minimum pressure, and minimum temperature of 8.2 MPa, and 35ºC respectively). Moreover, the plant $/kWe value for the maximum pressure of 28 MPa is not significantly lower than that of the 25 MPa case. Detailed cycle calculations for this optimum case are presented in Figure 13. Compared to the reference water cooled cycle in Figure 1, the cycle efficiency increased from 42.27% to 42.90%. The modified CO_2-to-air cooler design utilizes 68 cooler units (details of each unit are presented in Figure 2) and consumes 1.4MW electrical power for operating the fans whereas the reference cycle consumes 0.84MW electrical power to operate the water pump. ***The capital cost of the optimized air cooled cycle is 4833 $/kWe. This corresponds to only about 1% increase in the plant capital cost compared to the reference water cooled cycle (4780 $/kWe)***. The detailed optimized designs of the RHX, HTR, and LTR are presented in Table 5.

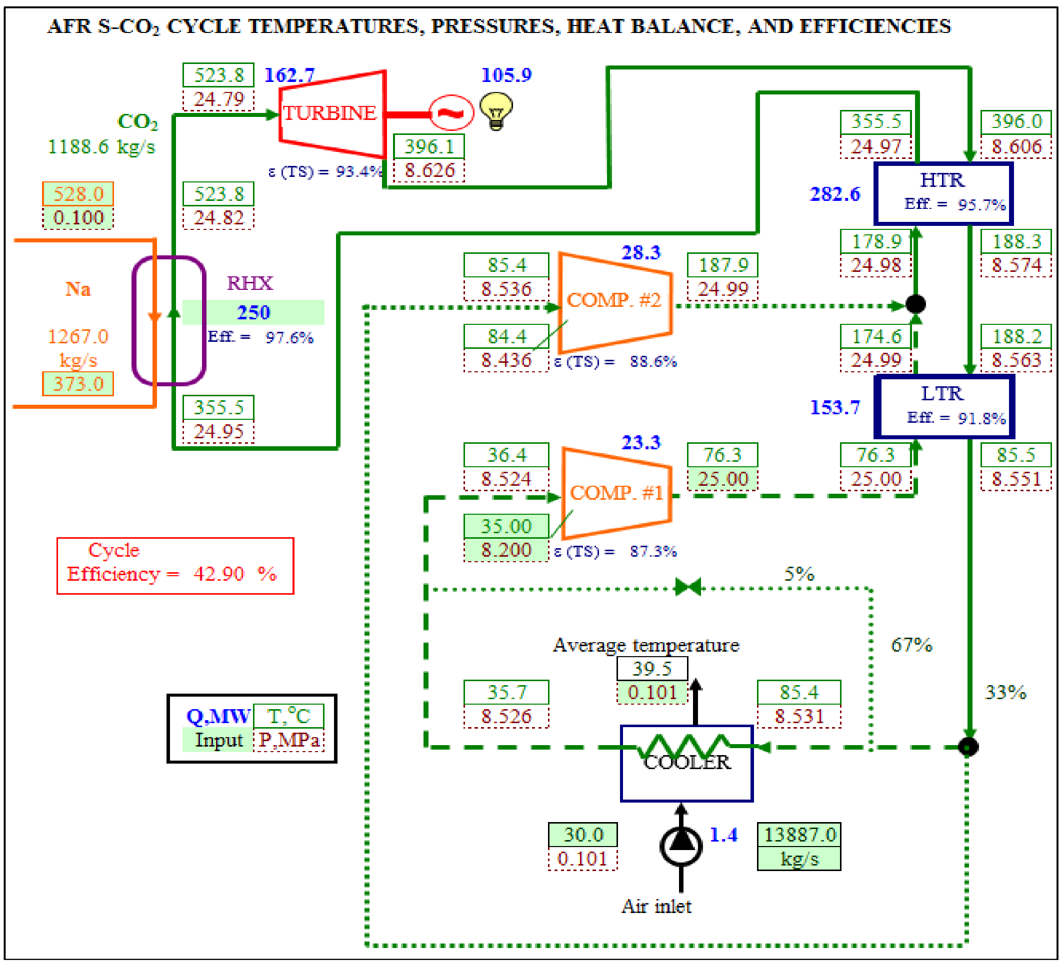

Figure 13. Cycle calculations after cost-based optimization using the CO_2-to-air cooler design

Effect of ambient air temperature

All the calculations presented till now assumed that the ambient air temperature is 30ºC. This might not be true for all the power plant locations and it is important to investigate the effect of ambient air temperature on the plant $/kWe. In this section, the preliminary calculations performed for different ambient temperatures are presented for design conditions. The maximum ambient air temperature is selected to be 40ºC. For an ambient temperature of 40ºC, the minimum temperature of the cycle has to be greater than 40ºC in order to perform the calculations. Therefore, cycle minimum temperature is selected as 45ºC and the corresponding cycle minimum pressure is chosen to be the pseudo-critical pressure (9.688 MPa for 45ºC). Figure 14 presents the calculations to show the effect of number of cooler units on the fan power consumption and the plant capital cost.

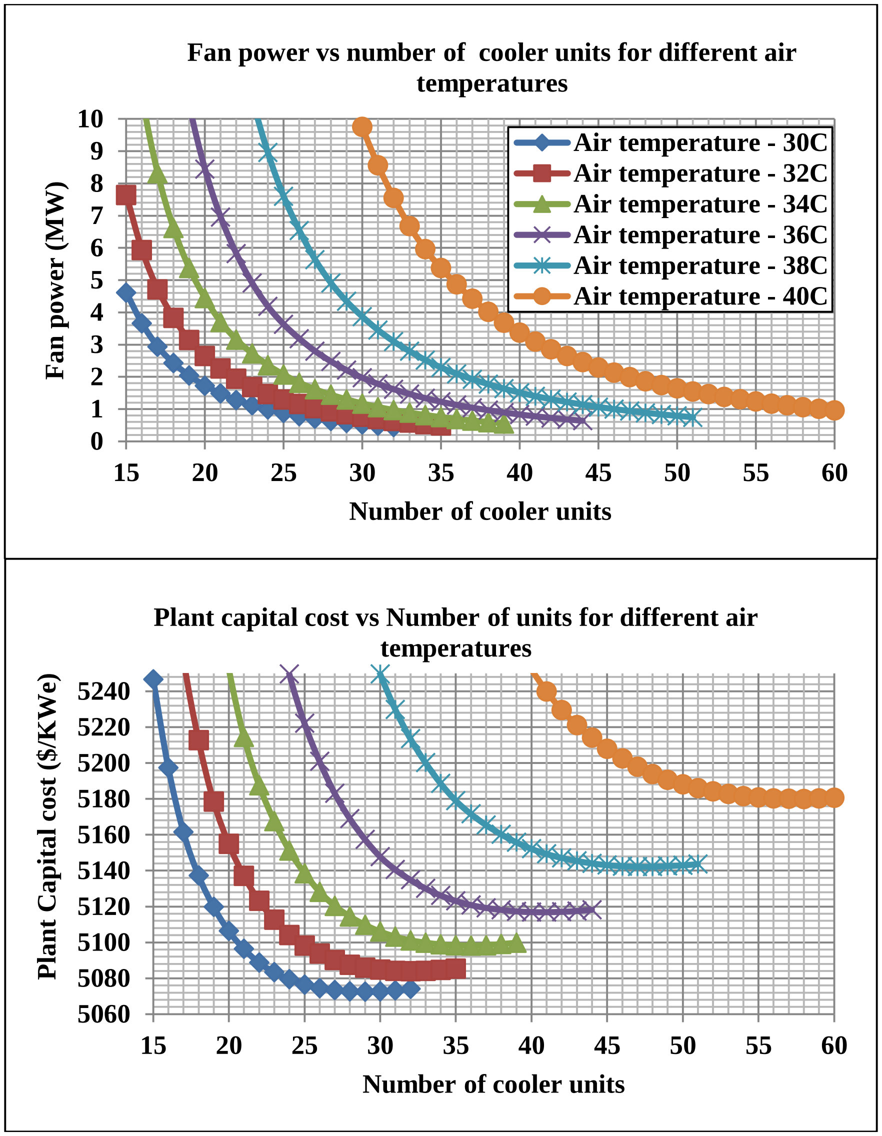

Figure 14. Effect of ambient air temperature on the required fan power and the plant $/kWe

Please note that all the curves presented in Figure 14 are for the compressor inlet conditions of 9.688 MPa, and 45ºC. As expected, the air cooler fan power consumption increases with increase in the ambient air temperature because of the reduction in the cooler approach temperature (defined as the difference in CO_2 outlet temperature and air inlet temperature in the cooler). This is especially true for smaller number of the cooler units. As the number of cooler units increase to match the heat load, the fan power consumption becomes less dependent on the ambient air temperature. Please note that in the current air-to-CO_2 the blowers have to be replaced to match the fan power curves in Figure 14. However, increasing the number of cooler units to match the heat load for different ambient air temperatures is not an economical option. Consequently, the optimum number of cooler units will have to be selected as a function of the ambient air temperature in order to minimize the plant $/kWe. The capital cost calculations presented in Figure 14 show that the optimum number of cooler units increase as the ambient air temperature increases. For example, the optimum number of cooler units increases from 29 to 58 as the ambient air temperature increases from 30°C to 40°C (100% increase in the number of cooler units, and cooler cost). However, the corresponding plant capital cost increases only from 5072 $/kWe to 5180 $/kWe as the ambient air temperature increases from 30°C to 40°C (roughly about 2% increase in the plant capital cost). One important thing to be noted here is that the results presented in Figure 14 are based only on cost-based optimization technique applied to the CO_2-to-air cooler while other parameters (such as split fraction, RHX, HTR, and LTR designs) are kept constant. Therefore, cycle minimum pressure of 9.688 MPa might not be the optimum minimum pressure for any of these ambient air temperatures and the actual increase in the plant capital cost for different ambient air temperatures can be either higher or lower than the preliminary 2% increase calculated in this section.

Table 5. Modified heat exchangers design for the optimized cycle conditions

Optimized design of the RHX

Type	Z/I PCHE	
Number of units	84	All parameters below are per unit
Heat transfer capacity	2.98 MW_{th}	
Hot side fluid	Na	
Hot side temperature inlet	528°C	
Hot side temperature outlet	373°C	
Hot side pressure inlet	0.1 MPa	
Hot side pressure outlet	0.1 MPa	
Hot side flow rate	15.1 kg/s	
Hot side pressure drop	0.1 kPa	
Cold side fluid	CO_2	
Cold side temperature inlet	355.5°C	
Cold side temperature outlet	523.8°C	
Cold side pressure inlet	24.952 MPa	
Cold side pressure outlet	27.816 MPa	
Cold side flow rate	14.2 kg/s	
Cold side pressure drop	136.4 kPa	
Effectiveness	97.6%	
Heat transfer area	169.1 m^2	
Unit width	0.6 m	
Unit height	0.6 m	
Unit length	1.5 m	
Heat transfer length	1.5 m	
Number of plates	84	Each side
Hot side channel diameter	6.0 mm	Semi-Circular channel
Hot side channel pitch	6.5 mm	
Hot side plate thickness	5.1 mm	
Hot side number of channels	84	Per plate
Hot side channel angle	0°	
Hot side channel length	1.5 m	
Cold side channel diameter	2.0 mm	Semi-Circular channel
Cold side channel pitch	2.8 mm	
Cold side plate thickness	1.6 mm	
Cold side number of channels	178	
Cold side channel angle	60°	
Cold side channel length	1.732 m	

Optimized design of the HTR

Type	PCHE	
Number of units	38	All parameters below are per unit
Heat transfer capacity	7.44 MW_{th}	
Hot side temperature inlet	396^oC	
Hot side temperature outlet	188.3^oC	
Hot side pressure inlet	8.606 MPa	
Hot side pressure outlet	8.574 MPa	
Hot side flow rate	31.3 kg/s	
Hot side pressure drop	32.3 kPa	
Cold side temperature inlet	178.9^oC	
Cold side temperature outlet	355.5^oC	
Cold side pressure inlet	24.981 MPa	
Cold side pressure outlet	24.970 MPa	
Cold side flow rate	31.3 kg/s	
Cold side pressure drop	10.7 kPa	
Effectiveness	95.7%	
Heat transfer area	281 m^2	
Unit width	1.5 m	
Unit height	0.6 m	
Unit length	0.6 m	
Heat transfer length	0.38 m	
Plate material	SS316	
Number of plates	268	Each side
Hot side channel diameter	1.3 mm	Semi-Circular channel
Hot side channel pitch	1.8 mm	
Hot side plate thickness	1.1 mm	
Hot side number of channels	715	Per plate
Hot side channel length	0.439 m	
Hot side channel angle	60^o	
Cold side channel diameter	1.3 mm	Semi-Circular channel
Cold side channel pitch	1.8 mm	
Cold side plate thickness	1.1 mm	
Cold side number of channels	715	
Cold side channel length	0.439 m	
Cold side channel angle	60^o	

Optimized design of the LTR

Type	PCHE	
Number of units	48	All parameters below are per unit
Heat transfer capacity	3.20 MW_{th}	
Hot side temperature inlet	188.2^oC	
Hot side temperature outlet	85.5^oC	
Hot side pressure inlet	8.563 MPa	
Hot side pressure outlet	8.551 MPa	
Hot side flow rate	24.8 kg/s	
Hot side pressure drop	12.6 kPa	
Cold side temperature inlet	76.3^oC	
Cold side temperature outlet	174.6^oC	
Cold side pressure inlet	24.997 MPa	
Cold side pressure outlet	24.991 MPa	
Cold side flow rate	16.6 kg/s	

Cold side pressure drop	6.0 kPa	
Effectiveness	91.8%	
Heat transfer area	293.8 m^2	
Unit width	1.5 m	
Unit height	0.6 m	
Unit length	0.6 m	
Heat transfer length	0.38 m	
Plate material	SS316	
Number of plates	273	Each side
Hot side channel diameter	1.3 mm	Semi-Circular channel
Hot side channel pitch	1.7 mm	
Hot side plate thickness	1.0 mm	
Hot side number of channels	734	Per plate
Hot side channel length	0.439 m	
Hot side channel angle	60^o	
Cold side channel diameter	1.3 mm	Semi-Circular channel
Cold side channel pitch	1.7 mm	
Cold side plate thickness	1.0 mm	
Cold side number of channels	600	
Cold side channel length	0.537 m	
Cold side channel angle	90^o	

CONCLUSIONS

The purpose of the work described in this paper is to investigate the techno-economic feasibility of dry air cooling to reject waste heat from the S-CO_2 Brayton cycle. The cycle developed for sodium cooled fast reactors (SFRs) small modular reactor AFR-100 is selected for the investigation. The previous work at ANL targeted at investigation of the possibility of using dry air cooling concluded that at least a 40% increase in the electricity price could be expected from implementation of air cooling [1]. The air cooler used in their study was based on the Heatric diffusion bonded technology and the cost of such a cooler is very high when using air as one of the heat transfer fluids. Also, the maximum cycle pressure in their study was limited to 20 MPa. Increasing this limit to higher values can regain part of the lost efficiency due to air cooling.

In an effort to reduce the cost of air cooler, an alternative air cooler option was found in the market and was chosen for the cost-based optimization study. The CO_2 undergoes three passes in each cooler module and flows inside stack of stainless steel tubes with aluminum fins to enhance the heat transfer. Each cooler module is equipped with three fans to distribute the air flow uniformly throughout the module in cross-flow arrangement. This is a very similar arrangement to that of a car radiator. A quotation from the manufacturer (Harsco Industrial Air-X-Changers [4]) was obtained for the reference cycle conditions and a model for the cooler was developed independently in EES to confirm the manufacturer quoted specifications. The EES model calculations matched well with the vendor specifications. The plant capital cost per unit electrical output ($/kWe) for the reference cycle conditions using the new air cooler is calculated. ***The calculations showed about 6% increase in the capital cost compared to the water cooled cycle for reference conditions which is a significant improvement in the plant economics compared to the previous study [1]***.

For an air cooled cycle, the cycle minimum temperature is dictated by the ambient air temperature and it is important to investigate the plant capital cost for higher cycle minimum temperatures. Increasing the cycle minimum temperature will reduce the S-CO_2 cycle efficiency and in order to regain part of the lost efficiency, the cycle maximum pressure is also increased. Therefore, three parameters namely cycle minimum temperature, minimum pressure, and maximum pressure are chosen for the cost-based optimization of the plant. The cycle minimum pressure is selected close to the pseudo-critical pressure to exploit the high fluid density during the compression process. As the cycle maximum pressure is increased the cycle components (reactor heat exchanger, high temperature recuperator, low temperature

recuperator, plant piping, and turbomachinery) design has to be modified to withstand the higher pressure differentials. Design modifications to these components, except the turbomachinery, is made as per the ASME guidelines and the cost changes associated with the modifications is taken into account during the cost-based optimization. No design modifications were made to the turbomachinery components.
For the cost-based optimization, all the heat exchangers designs are optimized individually in an effort to minimize the plant $/kWe for a given set of cycle parameters. This cost-based optimization technique is consistent with the previous optimization work at ANL. Optimization of the heat exchangers (excluding the CO_2-to-air cooler) is performed in Matlab using the ANL PDC for the cycle calculations and the final cooler conditions are exported to the EES air cooler model for selection of the optimum number of cooler units.

For the initial set of calculations, the turbine inputs to the PDC were left unchanged and a 4% drop in turbine efficiency was noticed by increasing the cycle maximum pressure from 18 MPa to 30 MPa. The cost-based optimization results showed that the optimum cycle conditions are minimum pressure of 8 MPa, minimum temperature of 35°C, and maximum pressure of 24 MPa. *For this case, the plant capital cost ($/kWe) is about 3% higher that of the reference water cooled cycle*.

For the next set of calculations, the turbine design part of the PDC is skipped and a constant static-to-static turbine efficiency of 93.4% is assumed for the calculations. The cost-based optimization results showed that the optimum cycle maximum pressure shifts slightly towards the higher value due to an increase in the plant efficiency. However, these calculations didn't account for turbomachinery costs. If these cost are accounted for, it is believed that both these set of calculations should yield similar results. For the fixed turbine efficiency case, the optimum cycle conditions are minimum pressure of 8 MPa, minimum temperature of 35°C, and maximum pressure of 24 MPa. *For this case, the plant capital cost ($/kWe) is about 2% higher than that of the reference water cooled cycle*.

In order to investigate any potential savings in the vicinity of the pseudo-critical pressure, three cases were selected (minimum temperatures of 32.5°C, 35°C, and 40°C). For all three cases, the calculations showed the plant $/kWe can be reduced by selecting the cycle minimum pressure greater than the pseudo-critical pressure (this will increase the cooler cost slightly but gain in cycle efficiency outperforms the increase in the cooler cost). Out of all the cases investigated, the optimum cycle conditions are minimum pressure of 8.2 MPa, minimum temperature of 35°C. For this case, strictly going by the obtained results, the optimum maximum pressure is 28 MPa. However, the calculations presented in this report didn't account for lot of cost changes associated with high pressure and engineering judgement was used to choose 25 MPa as optimum pressure. Moreover, the $/kWe values were not significantly different for the 28 MPa and 25 MPa cases. *For the optimum case, the plant capital cost ($/kWe) is only about 1% higher that of the reference water cooled cycle*. This of course doesn't include the increase in cost of turbine to achieve the static-to-static turbine efficiency of 93.4%.

Finally, the effect of ambient temperature on the plant capital cost was investigated. *The plant $/kWe increased by 2% as the ambient air temperature is increased from 30°C to 40°C*. These calculations were performed for the cycle minimum temperature of 45°C and the cost-based optimization technique was applied only to the CO_2-to-air cooler while rest of the parameters are fixed. In order to calculate more accurate increase in the plant capital cost, the cycle minimum pressure has be to re-optimized for different ambient air temperatures.

Overall, the results of the present analysis of the dry air cooling for the S-CO_2 cycle using the new air cooler design are very promising. Even for a worst case scenario, less than 5% increase in the plant capital cost over the water cooled plant is calculated. This has a significant impact on the applicability range of the S-CO_2 cycles.

Acknowledgements
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