INVESTIGATION OF A DRY AIR COOLING OPTION FOR AN S-CO_2 CYCLE

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ABSTRACT

An investigation has been carried out of the feasibility of dry air cooling for a supercritical carbon dioxide Brayton cycle power converter for a 400 MWe (1000 MWt) Sodium-Cooled Fast Reactor as a means of directly rejecting heat to the atmospheric heat sink. It was found that dry air cooling is not practical because the carbon dioxide-to-air reject heat exchanger would need to be huge reflecting the low density of air at near-atmospheric pressure combined with the lower specific heat of air relative to water. The cost of such a heat exchanger based upon current compact diffusion-bonded heat exchanger technology would significantly raise the power plant capital cost per unit electrical power.

INTRODUCTION

Traditionally, nuclear power plants use water as an ultimate heat sink because of favorable qualities of water as a heat removal medium. In this regard, all previous analyses of the S-CO_2 cycle for nuclear applications at Argonne National Laboratory (ANL) assumed that water will serve as an ultimate heat sink environment. At the same time, increasing restrictions on water use in the world raise interest in water-free power plants. Liquid-metal cooled fast reactors with S-CO_2 energy conversion present additional benefits in this regard since they do not use water in either the reactor or balance-of-plant systems. Therefore, implementing air cooling as an ultimate heat sink will practically eliminate any significant water use by the power plant. In addition, an S-CO_2 cycle energy conversion system is being adopted for solar power plants, which are usually considered to be located in desert environments, where water cooling may not be even available as an option or would require the transport of water to the power plant. For all these reasons, an air cooling option for an S-CO_2 cycle, if proved to be feasible, will present significant benefits for the cycle range of applicability.

Cooling Media Properties

Selection of water as a cooling medium for nuclear power plants comes, on one hand, from its relatively wide availability and, on the other hand, from water's preferable properties for heat transfer as well as experience with its use as a heat transfer medium. Table 1 compares main properties of water, air, and CO_2 at conditions typically experienced in an S-CO_2 cycle cooler.

Table 1. Properties of Water, Air, and CO_2 in a Typical S-CO_2 Cycle Cooler Application

	Units	**Water** (at 1 atm, 30 ℃)	**Air** (at 1 atm, 30 ℃)	**CO_2** (at 7.4 MPa, 40 ℃)
Density	kg/m^3	998.2	1.196	224.1
Specific heat (C_p)	kJ/kg-K	4.184	1.005	3.160
Thermal conductivity	W/m-K	$598.5*10^{-3}$	$25.5*10^{-3}$	$33.4*10^{-3}$
Viscosity	Pa*s	$1.0*10^{-3}$	$18.3*10^{-6}$	$20.2*10^{-6}$

In comparison to air, water shows about 1,000 times higher density and four times higher specific heat. That means that for the same temperature rise in a heat exchanger, water mass flow rate would be four times less, while the volumetric flow rate (mass flow rate divided by density, measured in m^3/s, the quantity which directly affects pumping power) would be about 4,000 times less for water than for air. In addition, the more than 20 times higher thermal conductivity for water will significantly reduce the requirement for the surface area in a heat exchanger. Note that the three orders of magnitude higher viscosity for water will not have the same significant effect on the cooler design since viscosity affects the pressure drop through the Reynolds number which (in a turbulent regime) is usually included in the pressure drop through a one-fourth power dependency. Therefore, the difference in viscosity will not play the same role as the opposite difference in density. Overall, Table 1 demonstrates significant benefits of water compared to air as a cooling medium. Also note from Table 1 that CO_2 shows better heat exchange properties than air in all categories, such that it is expected that in a CO_2-air heat exchanger, most of the limitations to the heat exchanger will be on the air side.

Reference Cooler Design

In this work, the cooler design analysis with air cooling was carried out for an S-CO_2 cycle developed for the ABR-1000 reactor applications. The reactor power is 1000 MWth which, for calculated 40% net plant efficiency translates to 400 MWe. The S-CO_2 cycle conditions for the reference design are shown in Figure 1. These results are obtained with the steady-state part of the ANL Plant Dynamics Code (Moisseytsev and Sienicki, 2006). The same code will be used for all other analyses presented in this paper.

Particular to the cooler design, it is assumed that the cooling is accomplished by water at 30 ℃. The water flow rate was previously selected at 20,000 kg/s for the cooler heat removal capacity of 569.2 MW. The CO_2 static conditions at the compressor first stage inlet were selected to maximize the cycle efficiency and are equal to 31.25 ℃ and 7.4 MPa, which is only 0.25 ℃ above the CO_2 critical point, 30.98 ℃. Considering flow acceleration at the compressor inlet duct, pressure drops in the pipes between the cooler and the compressor, and mixing with 5% cooler bypass flow, the resulting CO_2 conditions at the cooler outlet are 32.7 ℃ and 7.627 MPa, as shown in Figure 1. These conditions are fixed by the cycle analysis and are not affected by the cooler design, as are the CO_2 cooler inlet temperature, which is defined by the low-temperature recuperator (LTR) performance, and the CO_2 flow rate in the cooler, which is calculated from the reactor heat exchanger (RHX) heat balance and CO_2 flow split between the compressors. Therefore, the only variable on the CO_2 side of the cooler is the CO_2 inlet pressure which is defined by the cooler pressure drop. In the cycle analysis calculations with the PDC, the cooler design parameters, except for the cooler length, are provided by the user in the input file. The code will calculate the required cooler length to match the CO_2 inlet temperature and outlet conditions. Another result of the cooler calculations is the pressure drop on the CO_2 side. Note that with fixed inlet and outlet CO_2 temperatures and fixed outlet pressure, the calculated cycle efficiency is mostly independent of the cooler design. It is only affected by the CO_2 pressure drop in the cooler, which is usually a small fraction of the overall pressure change in the cycle. The net plant output, however, is affected by the cooler design through the pumping power requirements for the water side, as discussed below.

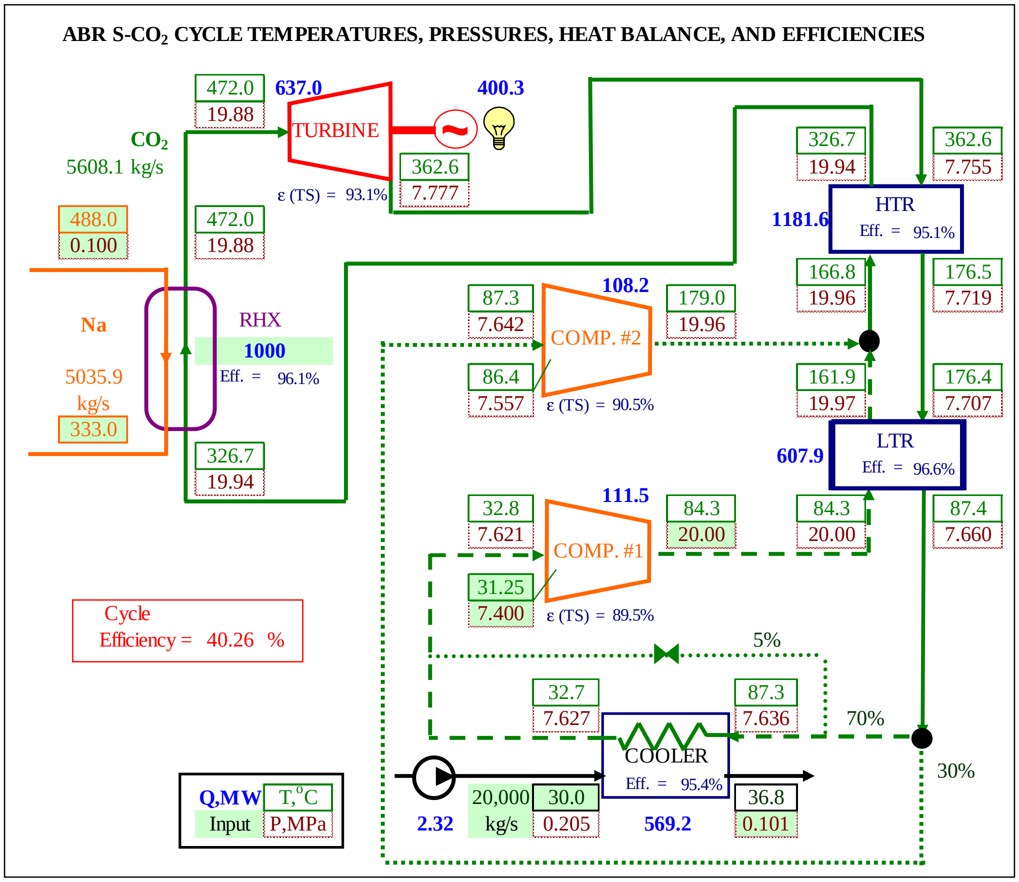

Figure 1. Reference 400 MWe S-CO_2 Cycle Conditions.

On the ultimate heat sink side, the inlet temperature (30 ℃ in Figure 1) and the outlet pressure (1 atm in Figure 1) are user-defined boundary conditions. The ultimate heat sink was assumed to be provided by water for the reference ABR-1000 plant design in Figure 1. In order to have direct comparison between air and water cooling, the same boundary conditions, i.e., 30 ℃ inlet and 1 atm outlet, will be used in this report for the air cooling calculations. The water flow rate of 20,000 kg/s was previously selected as an optimal value for this particular design. The modeling capabilities in the PDC extend only to the cooler and the water pump and do not include any other cooling circuit components; that is, the pressure drop in the connecting piping is currently ignored. The pumping power requirement is calculated as:

$$W_{water\ pump} = \frac{1}{\varepsilon_{pump}} \frac{\dot{m}_{water}}{\rho_{in}} \Delta p_{water} \quad (1)$$

where

ε_{pump} = pump efficiency (90% is assumed),

$\dot{m}_{water}$ = water flow rate, kg/s

ρ_{in} = water cooler-inlet (pump-outlet) density, kg/m^3

Δp_{water} = pressure drop on the water side of the cooler, Pa.

For the conditions in Figure 1, the water pump power consumption is 2.32 MW. The assumptions about the pump efficiency and cooling circuit pressure drop, along with the Equation (1) for the pumping power, will be retained for the analysis of the air cooling option.

The cooler design parameters used in the reference cycle calculations in Figure 1, along with major results for the cooler from the calculations are shown in Table 2. The Printed Circuit Heat Exchanger™ (PCHE) developed by Heatric (Heatric, 2014) is used as the reference heat exchanger design for the cooler for the ABR-1000 S-CO_2 cycle application. The semi-circular channel diameter of 2.0 mm was selected for both the CO_2 and water sides. There are 288 blocks, each measured 0.6 m wide by 0.6 m high with a 0.945 m calculated length resulting in a 98.0 m^3 total heat exchanger volume. It is assumed in the calculations that the cooler PCHE will have a "platelet" design, where the headers and flow distribution ports are integrated into the PCHE plates. Thus, a distinction is made in Table 2 and in the rest of this report between the total cooler (or plate) length and the heat transfer length. The former includes the latter and two headers (flow distribution regions) on each end of the plates.

Table 2. Reference Cooler Design with Water

Items	Spec. (SI)	Unit	Remarks
Type	PCHE		
Quantity	288		All parameters below are per unit
Heat transfer capacity	1.98	MWt	
Heat transfer area	149.2	m^2	
Unit width	0.6	m	
Unit height	0.6	m	
Unit length	0.945	m	
Heat transfer length	0.725	m	
Plate material	SS316		
Number of plates	170		Each side
CO_2 side channel diameter	2	mm	Semi-circular channel
CO_2 side channel pitch	2.4	mm	
CO_2 side plate thickness	1.66	mm	
CO_2 side number of channels	204		Per plate
CO_2 side channel length	0.837	m	Heat transfer region
CO_2 side channel angle	60	deg	
H_2O side channel diameter	2	mm	Semi-circular channel
H_2O side channel pitch	2.4	mm	
H_2O side plate thickness	1.66	mm	
H_2O side number of channels	214		Per plate
H_2O side channel length	0.8	m	Heat transfer region
H_2O side channel angle	60	deg	
Void fraction	36.3	%	From channels
CO_2 temperature inlet	87.3	℃	
CO_2 temperature outlet	32.65	℃	
CO_2 pressure inlet	7.636	MPa	
CO_2 pressure outlet	7.627	MPa	
CO_2 flow rate	12.9	kg/s	
CO_2 side pressure drop	9	kPa	
Water temperature inlet	30	℃	
Water temperature outlet	36.8	℃	
Water pressure inlet	0.205	MPa	
Water pressure outlet	0.101	MPa	
Water flow rate	69.4	kg/s	
Water side pressure drop	104.1	kPa	
Water pump power	2.324	MW	Total all units
Efficiency	95.4	%	

INVESTIGATION OF AIR COOLING AT THE REFERENCE CYCLE CONDITIONS

The analysis of the air cooling for the S-CO_2 cycle was started from the cooler design conditions developed for water cooling. At the first stage, the cooler design was preserved and the water was simply replaced with air. From the comparison of specific heats in Table 1, it was estimated that the air flow rate should be about 4 times higher for air than for water for the same temperature conditions. So, an air flow rate of 80,000 kg/s was initially selected (compared to 20,000 kg/s for water). The results of these calculations confirm that the temperature profiles and inlet/outlet conditions are close to those obtained for water (Figure 1). However, the calculated cooler length becomes about two times larger than for water cooling, reflecting the much smaller thermal conductivity of air compared to water, even with the higher mass flow rate.

The most important result, however, was that even though the temperature results are at least comparable for air and water cooling, the pressure drop and the resulting pumping power are huge in the case of air cooling. The calculated air compressor work (7,693 MW) exceeds by far any reactor or turbine power in Figure 1. These clearly unacceptable results are a combination of the higher mass flow rate and lower air density, resulting in both a much higher pressure drop and much higher volumetric flow rate, compared to water. Even without the pressure drop considerations, which could be addressed by the cooler design, a four times higher mass flow rate and about one thousand times lower density would result in a 4,000 times higher air volumetric flow rate with an equal increase in pumping power, according to Equation (1).

Therefore, before any cooler design modifications are introduced, a minimum value of the air flow rate has to be found. However, even with an 80,000 kg/s air flow rate, the minimum temperature approach in the cooler was less than 3 ℃ as a result of signif icant specific heat variations on the CO_2 side. Any reduction in the air flow rate though would only reduce the minimum temperature approach, resulting in an even larger cooler.

Selection of Air Flow Rate

The same calculations were repeated with 60,000 kg/s, 40,000 kg/s, and 30,000 kg/s air flow rate. As the flow rate decreased, the temperature approach (pinch point) in the cooler decreased as well, resulting at longer cooler, as shown in Table 3. Even with smaller flow rates, the circulating power requirements are still several orders of magnitude higher than for water and still exceed the power output from the generator. The example of temperature profiles and approach temperatures in cooler for 40,000 kg/s air flow rate is shown in Figure 2. As Table 3 demonstrates, reducing air flow rate below 40,000 kg/s results in significant increase in cooler length (and volume) increase. These results suggest that a 40,000 kg/s or so air flow rate is the minimal value for the air flow for practical considerations. This limit is also independent of the actual cooler design since it results from the CO_2 and air properties variation at the cooler conditions.

Because the air flow rate could not be practically reduced below 40,000 kg/s, the only remaining way to decrease the air compressor power is to reduce the pressure drop in the cooler by, for example, increasing the number of PCHE blocks, or increasing the air channel size. Since the air volumetric flow rate at 40,000 kg/s is still about 2,000 larger than for water at 20,000 kg/s, a very significant increase in the heat exchanger volume could be expected to obtain an air compressor work comparable to the water pumping power.

Table 3. Variation of Air Flow Rate at Reference Conditions and Cooler Design

Air Flow Rate, kg/s	Temperature Approach, ℃	Cooler Heat Transfer Length, m
80,000	2.5	1.7
60,000	2.0	2.0
40,000	1.3	3.8
30,000	1.0	9.6
Water @ 20,000	2.5	0.73

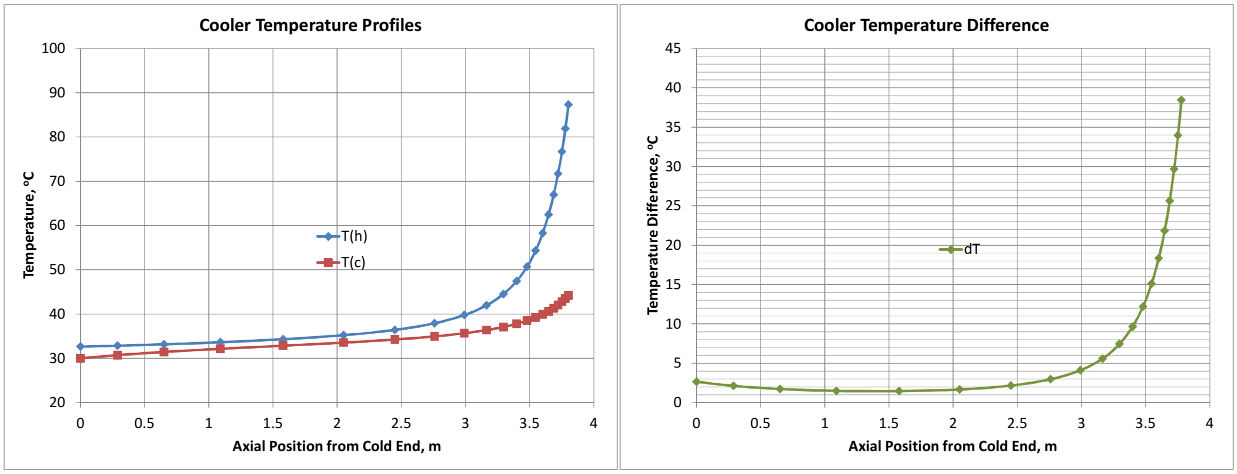

Figure 2. Results with Air Cooling with Reference Cooler Design and 40,000 kg/s Air Flow Rate.

These design modifications, presented below, were implemented independently for PCHE and shell-and-tube cooler designs, the only two options currently available for a cooler in the PDC code.

Printed Circuit Heat Exchanger Design for Air Cooling

Variation of Cooler Size

For the PCHE, the heat exchanger volume could be increased by increasing the number of HX blocks. With larger flow area, the pressure drop on the air side is reduced, thus reducing the air compressor power with increasing the number of PCHE units (blocks). Even though a noticeable power reduction could be achieved with this approach, a significant number of PCHE units would be needed for any acceptable compressor power requirements – even with 50,000 units (compared to 288 for the water cooler), the compressor power is still calculated to be higher than 25% (100 MW) of the total plant output.

Variation of Air Channel Size

The other way to decrease the air pressure drop in the cooler is to increase the air channel size. Since the air pressure is expected to be close to atmospheric, the required wall thickness on the air channel is small (compared to the CO_2 side). The stress calculations showed that for an assumed maximum air pressure of 0.2 MPa (2 atm), a 0.1 mm wall thickness is sufficient for 2 mm semi-circular channels. For the wall thickness between the air channels, a 0.1 mm wall for 2 mm channels corresponds to a pitch-to-diameter ratio of 1.05. Since the minimum wall thickness is expected to scale closely to linear with the channel diameter, a p/d=1.05 was selected for all variations in the channel diameter. On the other hand, in the PCHE design, the wall thickness between the CO_2 and air channels should also withstand the CO_2 pressure. For 8 MPa CO_2 pressure and 2 mm channel, the minimum wall thickness is 0.3 mm. Therefore, the plate thickness on the air side was selected to be equal to the channel radius plus 0.3 mm, t=d/2+0.3 mm. Note that these wall thicknesses on air side are smaller than the “equal sides” design assumed before (see Table 2), so the results at 2.0 mm would be better than those shown above. No modifications to the CO_2 plates were introduced, such that the channel geometry on the CO_2 side is retained as that for the reference water cooler in Table 2.

Figure 3 shows how increasing the air channels reduces the pressure drop on the air side and, therefore, the air compressor power requirements. The results in Figure 3 were obtained with 50,000 PCHE units. For channels larger than 6 mm, the compressor power requirement is reduced below 20 MW, i.e. below 5% plant gross output. For the 6 mm air channel diameter, the plate thickness is 3.3 mm and the cooler length is 0.638 m. 48,350 PCHE blocks (0.6 m x 0.6 m x 0.638 m) would be required to limit the air compressor power to 5% of the plant output (20 MW), resulting in a total cooler volume of 11,105 m^3, ***which is more than 100 times larger than the water cooler for the same conditions and 2.3 MW water pumping power***.

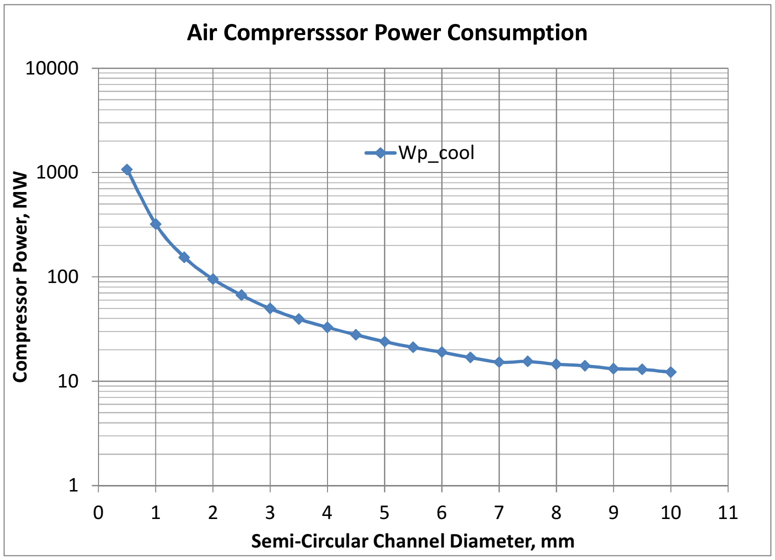

Figure 3. Variation of the Air Compressor Power with Air Channel Diameter (50,000 Blocks).

In addition to a larger cooler and higher pumping power (compared to the water case), a 6 mm channel diameter is at the upper end of current PCHE technology. Even for Heatric's Formed Plate Heat Exchanger (FPHE) design, 6 mm distance between the fins is considered to be an upper limit. To investigate if other heat exchanger option which could provide larger air channels with potential to further reduce pressure drop and compressor power on the air side, an analysis similar to that presented above was carried out for a shell-and-tube heat exchanger.

Shell-and-Tube Heat Exchanger Design for Air Cooling

The PCHE cooler design analysis presented above showed that it is beneficial to have wide air channels to reduce the air circulation work. There are, however, technological limits on the channel width with the PCHE technology. On the other hand, the shell-and-tube heat exchanger (S&T HX) type, with air flowing on the shell side outside the tubes, allows for an almost unlimited size of the air channel. In addition, due to the basically atmospheric air pressure, the shell thickness will only serve as a channel guide and will not need to withstand any significant pressure difference. Therefore, the shell thickness could be very small and is not expected to impose any additional limitations on the heat exchanger size (and, to a large extent, on the HX cost).

The CO_2 will be flowing inside the tubes, which are expected to hold the full CO_2 pressure of about 8 MPa. At this point, no stress analysis on the tubes has been carried out; it is just assumed that the tube will have inner diameter of 1 cm and thickness of 0.2 mm (resulting in an outer diameter of 1.4 cm). To enhance the heat transfer to air, it is also assumed that the tube will have longitudinal straight fins on the outer (air) surface. The fin dimensions are the subject of another optimization study; in this work it is assumed that the tubes will be arranged in a triangular lattice with six 1 mm thick fins on each tube and the fins will be (almost) touching the fins from the opposite tubes, as shown in Figure 4. For the first round of calculations, a tube pitch-to-diameter ratio of 1.7 was selected to provide wide air channels (p/d=1.7 was used for at-scale comparison of air and CO_2 channels in Figure 4).

In the PDC code, the shell-and-tube heat exchanger option calculates heat transfer in a single tube; all other tubes are assumed to be identical. Therefore, only the number of tubes matters for the heat transfer calculations; it is not important how these tubes are organized into separate HX modules. For simplicity, it is assumed that there would be only one module and the module outer (shell) diameter is the user input. The code calculates the number of tubes which can fit into the cross-sectional area and then calculates the required length of the tube to match the CO_2 inlet/outlet conditions.

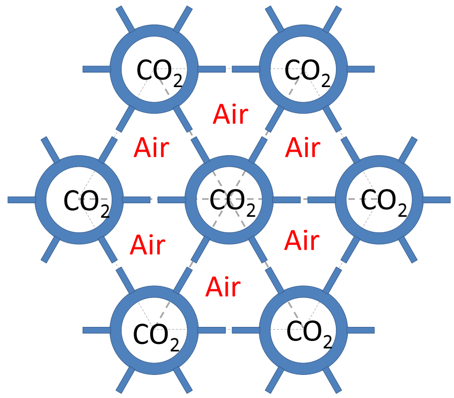

Figure 4. Shell-and-Tube Heat Exchanger Tubes with Outside Fins.

Variation of S&T HX Diameter (Number of Tubes)

The results of the calculations for a S&T HX varying diameter (or number of tubes), presented in Figure 5, show that with a sufficiently large heat exchanger, the air compressor power can be reduced to the same 20 MW (5%) as in the PCHE case. The 20 MW mark is achieved at 122 m diameter; the tube length in this case is calculated to be 11.4 m, and the total HX volume is 133,600 m^3. This volume is more than 10 times larger than for the air cooled PCHE and ***more than 1,000 times larger than the reference water cooled PCHE (still with higher pumping power)***.

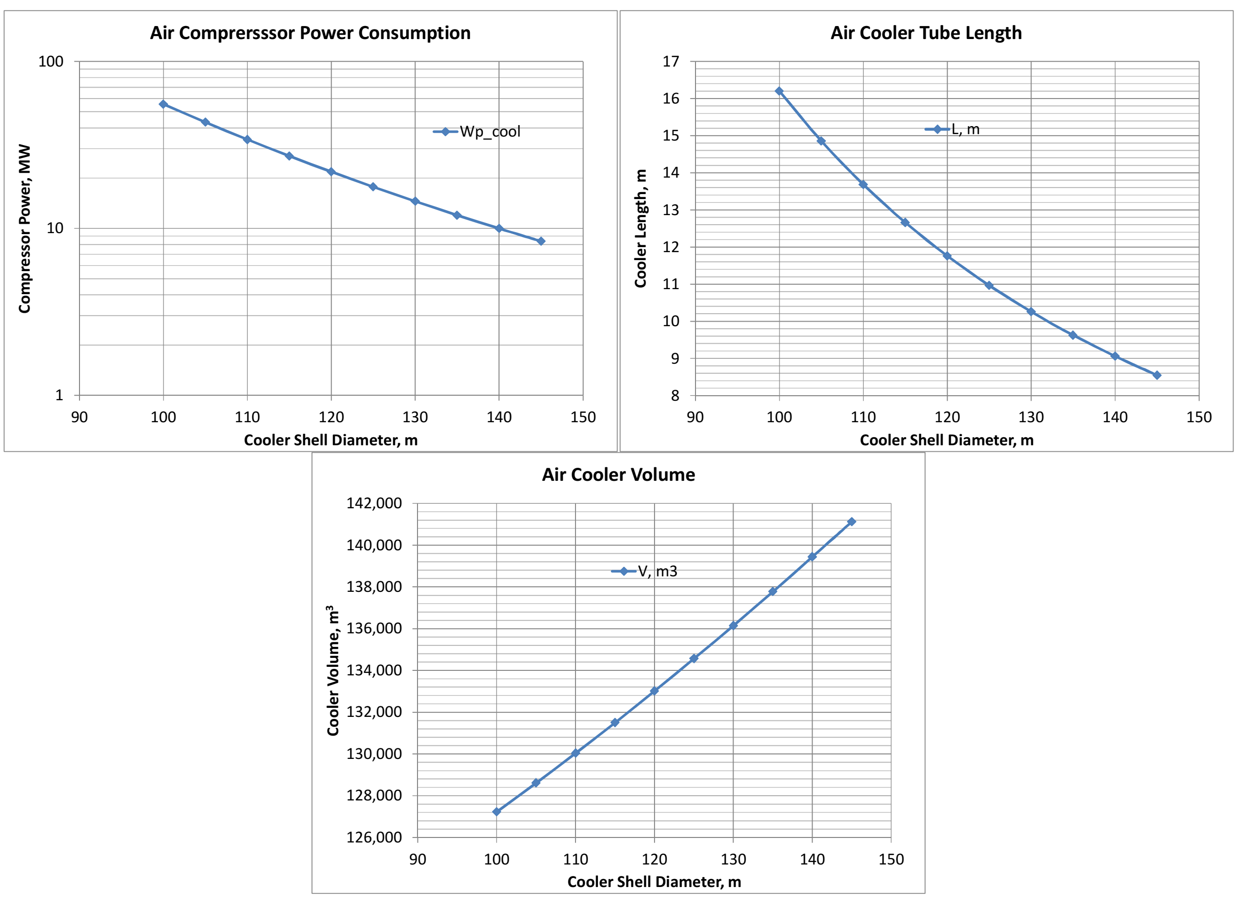

Figure 5. S&T HX Sizing for Air Cooling.

The cycle analysis results for a 122 m diameter S&T HX show that since the air flow rate and the air circulation power are the same as for the PCHE case, these cycle-level results are almost identical to those obtained with the PCHE.

As discussed above, in the PDC code, only the number of tubes matters, so a single unit of 122 m diameter is calculated. Of course, this size is not practical, and the heat exchanger will have to be divided into individual units. Preserving the tube length and the total number of tubes (or the HX total cross-sectional area), the individual unit diameter can be calculated as

$$D_N = D_1/\sqrt{N} \tag{2}$$

The results from Equation (2) for 122 m single-unit diameter are demonstrated in Figure 6. Even with 100 units, each unit will be 11.4 m long by 12.2 m in diameter.

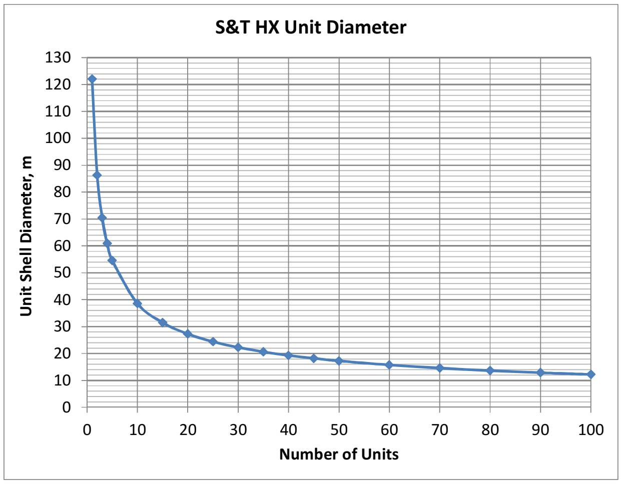

Figure 6. S&T HX Unit Diameter for Equivalent 122 m Single Unit.

Effect of Tube Fins

The outside surface fins (Figure 4) improve the heat transfer to air. At the same time, they occupy some fraction of the air flow area, thus increasing the air pressure drop. To see which effect is more important for the air cooler design, the calculations described above were repeated with bare tubes (i.e., without the fins). For the same air compressor power of 20 MW, the total HX diameter was reduced to 98 m (from 122 m) due to the larger air flow area and smaller pressure drop. However, due to reduced heat transfer, the total tube length was calculated to increase to 21.22 m (from 11.4 m with fins). As a result, the total HX volume increased from 133,600 m^3 to 160,061 m^3. Therefore, it is recommended to retain fins on the outside (air) surface of the tubes for further design analyses of S&T HX.

Variation of Pitch-to-Diameter Ratio

Similarly to the channel diameter parametric study for PCHE (Section 2.2.2), an attempt was made to find an optimal air channel side for a S&T HX. Since the air is assumed to be on the shell side, the air channel size is defined by the tube pitch-to-diameter ratio. This ratio was parametrically varied from 1.1 to 2.0. In all cases, the fin length was adjusted to the maximum value, i.e., to allow the fins from the neighboring tubes to touch each other. The relationship between the fin length (lf) and the pitch-to-diameter ratio (p/d) for the tube outer diameter of do is:

$$l_f = \frac{1}{2} d_o \left(\frac{p}{d} - 1\right). \tag{3}$$

For all considered values of the pitch-to-diameter ratio, the heat exchanger diameter was selected to preserve a 20 MW air compressor power. The results of the calculations show that no optimal value for the pitch-to-diameter ratio exists to minimize the heat exchanger volume. For practical reasons (such as tube welding to the tube sheets), a value of p/d=1.3 was selected. For this design, the tube length is 5.82 m and the shell diameter is 138.5 m, giving the total volume of 87,700 m^3.

Using Equation (2) for an individual unit size, it is calculated that 566 units will be needed for the designs with shell L/D=1. For a round number of 600 units, the shell diameter is 5.65 m. The total HX volume in all these cases is still 87,700 m^3, which is about 900 times larger than for the water PCHE cooler.

Comparison of Cooler Types

The results of the air cooling heat exchanger design calculations at the reference conditions, presented above, showed a significant increase in the cooler size compared to water cooling case. With the PCHE air cooler, the total volume increase is about 100 times, while with S&T HX, this value is around 900. Figure 7 graphically shows the comparison between the heat exchanger sizes for a water cooling PCHE and an air cooling PCHE as well as a S&T HX. For practical reasons, only 1/100 (1%) of the entire heat exchanger is shown in Figure 7; the entire plant will have 100 times the number of units shown in the figure. A six-foot (1.8 m) tall person is shown in Figure 7 for scale comparison.

It is also important to note that along with the heat exchanger size increase, the cooling medium circulation power increased from 2.3 MW for water to 20 MW for the air cases, resulting in an additional 17.7 MWe lost from the plant output, which would represent about 4.4% reduction in net electricity production.

Figure 7 clearly shows that there is a significant increase in the cooler heat exchanger volume and the number of units, if air cooling would be adopted for the reference cycle conditions. Those much larger heat exchangers will increase both the plant footprint and, more importantly, the total cost. Given the fact that there are four major heat exchangers in the S-CO_2 cycle (RHX, HTR, LTR, and cooler) and they are all of at least comparable size for the water cooled case, and assuming that the heat exchangers will represent the major portion of the cycle capital cost, increasing the HX volume by a factor of 100 will result in a 25-fold increase in the cycle capital cost. Clearly, this would be too expensive for the cycle to remain competitive.

Therefore, other options to further reduce the air cooler size need to be explored. The first candidate in this regard is to increase the temperature difference between the CO_2 and air sides in the heat exchanger. Since the air side temperature is fixed by the assumed environment conditions, increasing in the temperature approach can only be achieved by raising the CO_2 temperature. Of course, for the S-CO_2 cycle this would mean moving away from the critical point, leading to an unavoidable reduction in the cycle efficiency. The trade-off between the cycle efficiency and the cooler volume for various minimum S-CO_2 cycle temperatures is investigated in the next section.

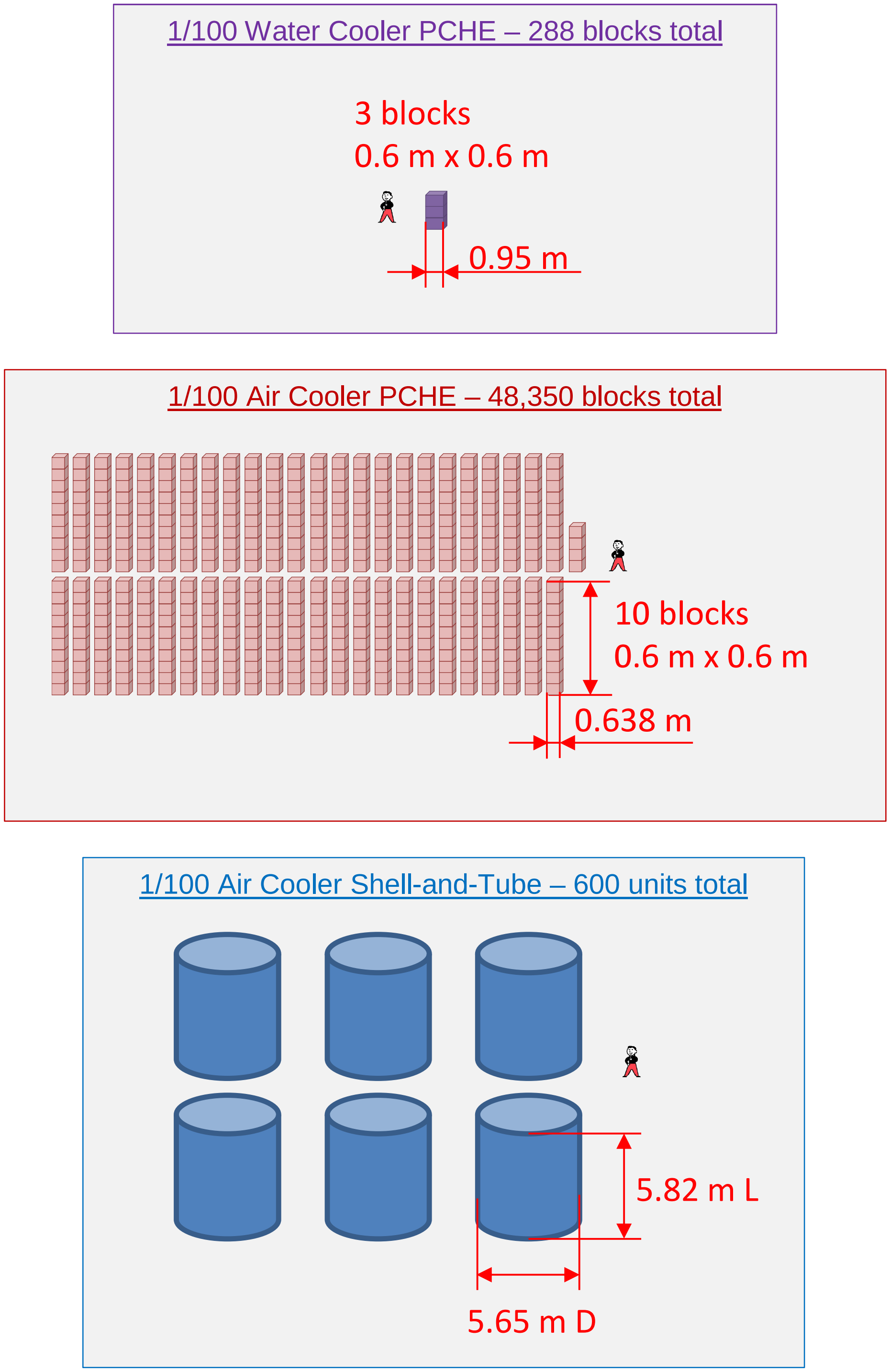

Figure 7. To-Scale Comparison of Cooler Heat Exchanger Sizes (1/100 of Units are Shown).

SELECTION OF THE MINIMUM S-CO_2 CYCLE TEMPERATURE

The S-CO_2 cycle benefits from CO_2 properties variations near the critical point (31 ℃, 7.4 MPa), mostly from increased density and reduced compression work at these conditions. Increasing the CO_2 temperature at the compressor inlet (which is also the minimum temperature in the S-CO_2 cycle) will decrease the cycle efficiency. Some of this loss can be recaptured, if the compressor-inlet pressure is varied simultaneously with the temperature to maintain the CO_2 density at least close to the value just above the critical point. There is, however, no simple answer what the optimal pressure is for each temperature. As a first guess, a pseudo-critical pressure (i.e., the pressure at which specific heat has a maximum value for each temperature, or vice versa) can be selected. The CO_2 pseudo-critical line is shown in Figure 8. However, the cooler size considerations are also important in selecting the minimum cycle conditions. For example, going through the pseudo-critical temperature in the cooler may increase the cycle efficiency, but will require a much longer cooler due to the increased CO_2 specific heat in this region. This effect on the cooler length was one of the reasons for selecting 31.25 ℃ (rather than 31.0 ℃) for the compressor-inlet temperature in the reference cycle configuration (the other reason was the margin above the critical point for off-design operations). Earlier cycle analysis (Moisseytsev, 2003) showed that increasing the minimum CO_2 temperature from 31.0 ℃ to 31.25 ℃ (while keeping the pressure at 7.4 MPa) allowed for a 50% reduction in the cooler volume (length). Since the cooler length and the corresponding pressure drop are even more important for air cooling, selection of “optimal” minimum cycle conditions would be even more challenging for the air cooling cases.

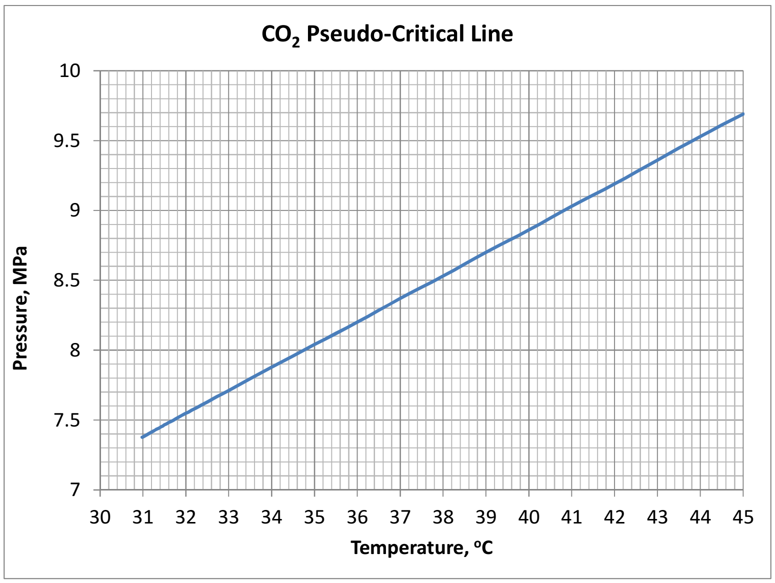

Figure 8. CO_2 Pseudo-Critical Line.

The optimization problem is even more complicated by the presence of at least two other more-or-less independent design parameters; namely, the CO_2 flow split between the compressors and the air flow rate. The former is usually dealt with by selecting the value which optimizes the cycle efficiency. Since the optimal flow split is dictated by the differences in the CO_2 properties (specific heats) in the LTR, this optimal is expected to be different for changing compressor-inlet temperatures and pressures. However, the flow split fraction will also affect the cooler design since the CO_2 flow is the same in the cooler and in the main compressor. So, selecting a higher fraction for the main compressor will result in higher CO_2 flow in the cooler which will affect the cooler heat duty and its design. For the simplicity of the calculations, this effect will be ignored in the optimization process described in this chapter; that is, the flow split fraction will always be selected to maximize the cycle efficiency for the given minimum cycle pressure and temperature.

Selection of the air flow rate is more complicated since there is no single optimization target which could be calculated by a thermal dynamics code. Reducing the air flow rate is always desirable for the air compressor power consumption. However, lower flow rate will also require a longer cooler, increasing the cooler pressure drop on the air side, and, therefore, increasing the compressor work. The latter effect is not linear due to the minimum temperature approach limitations discussed above. Therefore, the optimal air flow rate can be selected only by a trial-and-error approach, and even then there is a trade-off between the compressor power consumption and the cooler size, which cannot be resolved until a detailed plant cost analysis is completed. In this regard, an engineering judgment will be used on the selection of the air flow rate in this chapter.

A similar problem arises for the selection of the number of PCHE units (or S&T HX diameter) for the cooler due to the trade-off between the compressor power consumption and cooler size and cost. Before the cost analysis is completed, no definite answer on the optimal cooler volume could be obtained, so the results in this chapter will often be presented as a function of the number of units (or cooler volume).

In addition to these optimization challenges, there would also be a question of the optimal cooler design parameters. The design parameters, obtained in the previous optimization work, such as selection of air channel size described above, may not be optimal for different cycle conditions, and will need to be re-optimized for the final design. This issue will not be addressed in this report; all the cooler design parameters will simply be retained for the analyses presented below.

Because of the large number of the mostly independent optimization parameters, an attempt to find an optimal design for all of them at once is not expected to be successful. Instead, a gradual approach in selecting the optimal design will be implemented. At the first step, the compressor-inlet temperature will be fixed at 35 ℃ to investigate the effect of the compressor-inlet pressure, air flow rate, number of cooler units, etc. Then the results will be compared to the reference case presented above to see if there are any benefits from raising the minimum cycle temperature. All the work presented in this section will assume the PCHE option for the cooler design; no attempts will be made at this stage to optimize S&T HX design for varying cycle conditions.

Air Cooler Optimization at 35 ℃

At 35 ℃ compressor inlet temperature, the CO_2 pseudo-critical pressure is 8.04 MPa (Figure 8). To investigate the effect of the pressure selection, several values for pressures were tried around 8 MPa, - 7.4, 7.6, 8.0, and 8.5 MPa. For each pressure, the CO_2 flow split between the two compressors was optimized to obtain the highest cycle efficiency. The results show that there is a noticeable increase in the cycle efficiency if the compressor-inlet pressure is selected at or above the pseudo-critical value.

For each pressure (and the optimal CO_2 flow split), the effect of the air flow rate was then investigated. As previously, the temperature profiles in the cooler are affected by the air flow rate. Based on the calculated approach temperature and in an effort to minimize the air flow rate, the value of 20,000 kg/s was selected for air flow rate for the compressor-inlet pressure of 7.4 MPa. Similar analyses were carried out for other pressures, and the flow rates of 25,000 kg/s and 40,000 kg/s were selected for pressures of 7.6 MPa and 8.0 MPa, respectively. In addition, the calculations for 8.0 MPa are also repeated with the value of the air flow rate at 25,000 kg/s to see the effect of air flow rate selection around the pseudo-critical point.

The 8.5 MPa pressure presents an interesting case. At this pressure, the pseudo-critical temperature from Figure 8 is 37.8 ℃, which is higher than the 35 ℃ for the compressor-inlet. Therefore, CO_2 will go through the pseudo-critical point inside the cooler. The cooler temperature profiles demonstrate that transition, where the temperature approach first increases, then decreases, then increases again. With a 35,000 kg/s air flow rate, that variation in the temperature approach still stays close to the cold-end temperature difference, so this value was selected for the air flow rate.

With the air flow rate selected, the parametric study on the number of the PCHE cooler units was carried out for each pressure. For each number of units, the required cooler length, cooler pressure drop on the air side, and the air compressor power were obtained from the PDC calculations. From these values, the total cooler volume and the net plant output (and efficiency) were calculated. The HX volume includes both the number of units and the HX length. The plant efficiency includes both the cycle efficiency and the air compressor power. The comparison of these two global results between various cooler sizes for various pressures is shown in Figure 9.

One of the apparent results from Figure 9 is that there is no optimal selection of minimum pressure. As the cooler size decreases, the highest plant efficiency is achieved with lower compressor-inlet pressure. Therefore, there is no easy selection of the compressor-inlet pressure, and this parameter may need to be re-optimized, if the cost analysis results would favor smaller cooler sizes. The only exception is the pressures higher than the pseudo-critical value, represented by the “8.5 MPa” line, which show lower plant efficiency than “8.0 MPa (40,000 kg/s)” line for all cooler volumes. Overall, though, the results in

Figure 9 demonstrate that, unless a significant penalty in the cycle efficiency would be justifiable by smaller coolers, *selection of the compressor-inlet pressure at the pseudo-critical value tends to optimize the cooler volume while retaining relatively high plant efficiencies*.

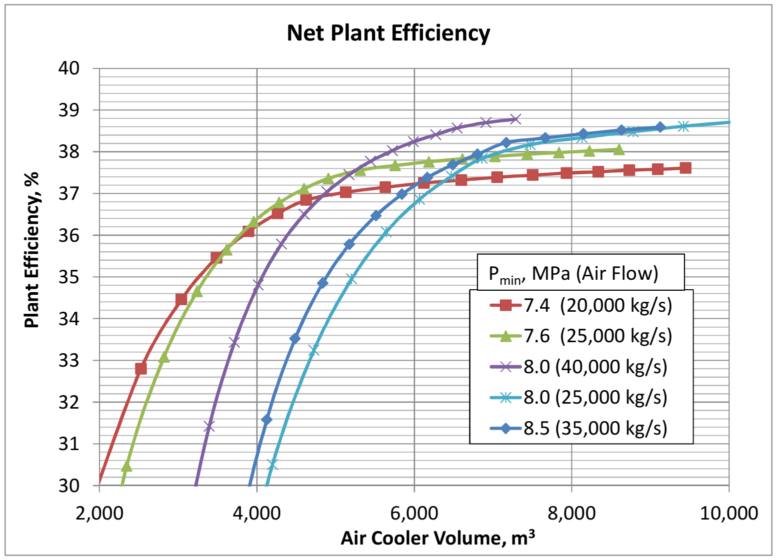

Figure 9. Plant-Efficiency versus Cooler Size for 35 ℃ and Various Pressures.

Another observation can be made from Figure 9 regarding the selection of the air flow rate. As the results for two 8.0 MPa lines show, selection of the flow rate which provides a consistent approach temperature towards the cold end of the cooler (40,000 kg/s) is more beneficial than a smaller flow rate (25,000 kg/s) at which the approach temperature initially dropped to much smaller values. Therefore, the selection of the air flow rate, which more or less maintains the approach temperature, will be retained for other analyses presented further in this report.

Lastly, the results in Figure 9 indeed show the benefits of increasing the temperature approach in the cooler. For comparison, at 31.25 ℃ and 7.4 MPa, the plant efficiency is 38.3% at 11,105 m^3 cooler volume (see Section 2). With 35 ℃ and 8.0 MPa compressor-inlet conditions, similar plant efficiency levels are achieved at about 6,000 m^3 cooler volume. So, a factor of two reduction in the HX volume can be realized without an efficiency penalty (due to the much lower air compressor power). The cycle performance results for the 8 MPa and 6,000 m^3 HX case are shown in Figure 10. Compared to the reference conditions (31.25 ℃ and 7.4 MPa), almost exactly the same net generator output of 382 MWe is calculated despite the lower cycle efficiency (39.6% vs. 40.3%) due to the lower air compressor power consumption (13.84 MW vs. 20 MW). In other words, increasing the compressor-inlet temperature and, therefore, the temperature approach in the cooler allowed regaining lost cycle efficiency from lower air circulating work. The important difference, though, is that the latter results are achieved with about half of the cooler volume.

To see if this trend in reduction of the cooler volume, while still preserving the plant net efficiency, will continue with even higher temperatures, and also to find the optimal design, the calculations were next extended to a wider range of the CO_2 compressor-inlet temperatures.

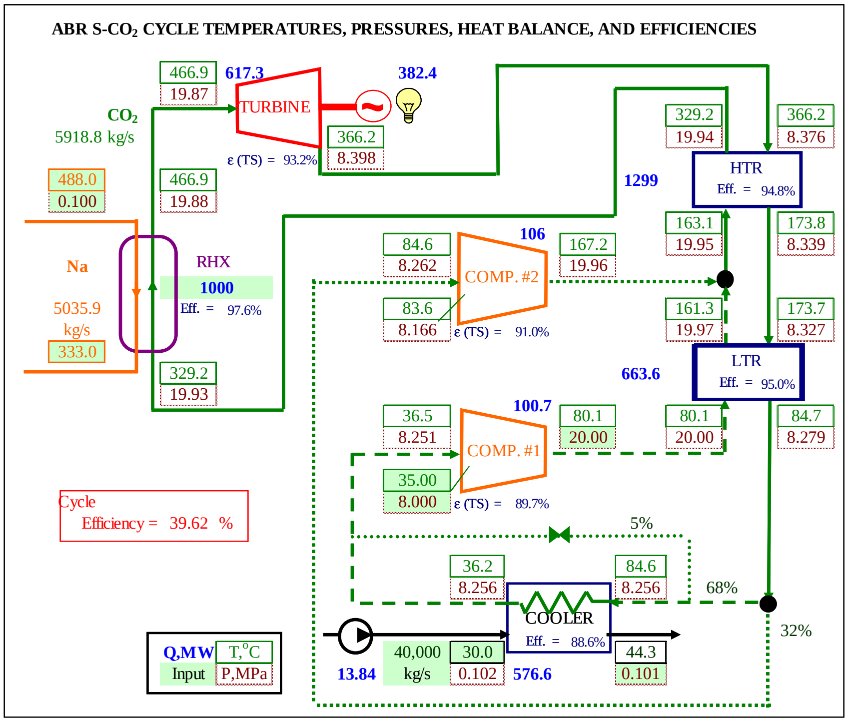

Figure 10. Cycle Results at 35 ℃, 8 MPa, and 6,000 m^3 Cooler.

Air Cooler Optimization along the Pseudo-Critical Line

The results in the previous section demonstrated that the optimal design would be at least close to the pseudo-critical line for the compressor-inlet conditions. Therefore, the analysis of the compressor-inlet temperature variation can be greatly simplified by assuming the pseudo-critical pressure at the compressor inlet for each temperature. This assumption allows separation of the cycle and cooler optimization calculations since the selection of the unique pair of temperature and pressure at the compressor inlet define the cycle performance to a large extent. The optimal cycle efficiency can now be found from variation of the flow split between the compressors. That optimal split can be assumed, with high confidence level, to be independent of the cooler design (and air flow rate), since the cycle performance will only be affected by the CO_2 pressure drop in the cooler, which is usually small (and even smaller for those large air coolers). After the cycle conditions are fixed, the effect of the cooler design on the overall plant performance can be relatively easily investigated since it wouldn't involve any optimization on the CO_2 side.

The compressor-inlet temperature was varied from 32 ℃ to 45 ℃ (in addition to the reference case of 31.25 ℃ and 74 MPa). At each temperature, the compressor-inlet pressure was selected to be equal to the pseudo-critical value from Figure 8. The CO_2 flow split fraction was then varied to maximize the cycle efficiency at each set of compressor-inlet conditions.

After the cycle conditions were optimized, the air flow rate was selected. This selection was based on the temperature approach in the cooler, as discussed in the previous section. As a general rule, the flow rate was selected at the minimal value at which the minimum temperature approach is at least 90% of the temperature difference at the cooler cold end (for example, if the temperature difference between CO_2 outlet and air inlet is 5 ℃, then the minimum temperature approach in cooler was maintained at least at 4.5 ℃).

The resulting values for the CO_2 compressor-inlet pressure, flow split fraction, and the air flow rate are shown in Table 4. Only for the reference temperature (31.25 ℃), some margin to the (pseudo) critical point was retained; in all other cases, the compressor-inlet conditions are selected to be right on the pseudo-critical line. For 35 ℃, 8.0 MPa was used (as listed in Table 4) rather than the pseudo-critical pressure of 8.04 MPa, since the results with 8.0 MPa pressure are already available from the previous section.

Table 4. Selected Conditions for Cooler Parametric Study along the Pseudo-Critical Line

T_{min}, °C	p_{min}, MPa	Flow Split	Air Flow Rate, kg/s
31.25	7.4	0.7	105,000
32.0	7.548	0.65	105,000
32.5	7.628	0.66	80,000
33.5	7.793	0.66	60,000
35.0	8.0	0.68	40,000
37.5	8.452	0.68	35,000
40.0	8.864	0.7	30,000
45.0	9.688	0.73	25,000

With the selected CO_2 conditions and the air flow rate, the calculations proceeded in the same manner as described in the previous section. The number of the cooler units was varied and the cooler length, cooler volume, air compressor power, cycle efficiency, and the net plant efficiency were recorded for each line in Table 4. These results are presented in Figure 11, which also compares those results with the reference case with the water cooler. As with the results in the previous section, no optimal cooler design and cycle conditions exist. For each given cooler volume, an optimal temperature, which gives the maximum plant efficiency, can be selected, but this choice changes with the cooler size. It is also apparent from Figure 11 that a significant penalty is imposed by air cooling compared to water cooling, either in terms of cooler size, plant efficiency, or both.

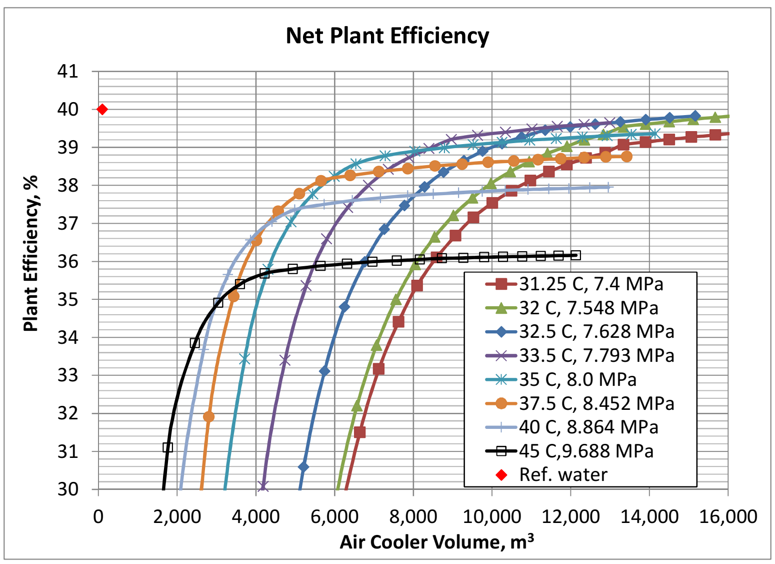

Figure 11. Plant-Efficiency versus Cooler Size along the Pseudo-Critical Line.

Figure 11 shows the ultimate results that could be obtained with the thermal-hydraulic analysis alone. No further optimization or selection of the optimal cooler design and the cycle condition can be made without at least some sort of cost-based analysis which would compare the cooler volume with the plant electrical output.

Cost-Based Selection of Optimal S-CO_2 Cycle Temperature and Air Cooler Design

In the previous analysis, the results were obtained in the form of a trade-off between the cooler volume and the plant net efficiency (or net power output). These two figures of merit could not be compared directly. Eventually, the decision on the optimal design will be made based on the cost comparison between the cooler capital (and O&M) cost and the amount of electricity produced by the plant with a goal of minimizing the lifetime cost of the electricity production.

A simplified version of the cost analysis was previously developed for the S-CO_2 cycle components, including other heat exchangers (Moisseytsev and Sienicki, 2011). In that analysis, the component capital cost (O&M was ignored) was compared with the rest-of-the-plant cost and the effect on the plant capital cost per produced unit electrical output (measured in $/kWe) was calculated for several component design variations. The plant capital cost per unit electrical output allows for direct comparison between various component designs and various plant power levels. The component design corresponding to the minimum plant capital cost per unit electrical output was selected then as the most cost-effective (i.e., optimal) design.

A similar analysis is applied to the cooler design results obtained in the previous section. For this simplified cost analysis, several assumptions need to be made. These assumptions are kept, to the extent possible, consistent with those used in the previous analysis (Moisseytsev and Sienicki, 2011) and include:

- The capital cost of the plant, excluding the cooler, was assumed to be equivalent to 4,800 $/kWe calculated for the reference plant electrical output of 400 MWe (or 1,920 M$). This value is independent of the cooler design, and is also assumed to be independent of the minimum CO_2 temperature selection.
- The cooler PCHE cost is calculated as a sum of the material cost and the fabrication cost. Both these components are scaled from sodium-CO_2 RHX cost data from (Moisseytsev and Sienicki, 2011). The material cost is calculated based on the original (before etching) total plate mass with an assumed 316 stainless steel plate cost of 7.64 $/kg. The fabrication cost is estimated at 44,400 $/block.
- The cost variation in the air compressor for various designs is ignored.

With these assumptions, for all the points calculated for Figure 11, the cost of the cooler is calculated based on the cooler volume, converted into the material mass (with a steel density of 7,928 kg/m^3 at 20 ℃), and the number of the cooler PCHE blocks. This cooler cost is then added to the cost of the rest of the plant (1,920 M$) to calculate the plant total capital cost at each design point. Then, the plant electrical output is calculated based on the reactor power of 1,000 MW and the net plant efficiency in Figure 11. Lastly, the plant total capital cost is divided by the plant electrical output to calculate the plant capital cost per unit electrical output in $/kWe. The same calculations with the same assumptions are also repeated for the reference water cooler design for comparison. The results of the cost calculations show that, not surprisingly, due to significant cooler volume increase and/or reduced power production, the calculated plant capital costs per unit electrical output with air cooling are much higher than those with water cooling. The minimum plant capital cost per unit electrical output, 7,600 $/kWe, is calculated for the 45 ℃ at about 2,000 m^3 cooler volume, but the difference with minimum cost at 40 ℃ is very small. Even at this minimal-cost design, ***there is about 60% increase in the plant capital cost per unit electrical output***, compared to the reference water cooled design (4,800 $/kWe).

These results quantitatively demonstrate the penalties from very large cooler volume calculated with the air cooler. In fact, the additional cooler cost is calculated to be so large, that the minimum cost is calculated for high compressor-inlet temperature which minimizes the cooler volume, but results in only a 31% net plant efficiency. These results, of course, were dictated by the assumptions made for the cost analysis, including the cooler PCHE cost components. It is possible, however, that those assumptions may overestimate the cooler cost, especially for such large, multiple-unit, heat exchangers. In particular, with the PCHE technology, some of material is etched away but is included in the original material cost, thus representing wasted cost. As it was mentioned above, with the assumed 6 mm semi-circular air channel diameter, a formed-plate (FPHE) fabrication, where basically no material is wasted, is expected to be more cost-effective than etching. For 3.3 mm assumed air side plates, 1.66 mm CO_2 side plates, and a 0.3 mm minimum thickness on the air side, the total material volume is estimated to be about 50% for a hybrid configuration with FPHE on the air side and PCHE on the CO_2 side, compare to PCHEs on both sides. Note, however, that 50 % material saving from the hybrid FPHE/PCHE configuration is not necessarily converted to a 50% cost saving, since no data is available for scaling of the FPHE or hybrid HX fabrication cost.

On the fabrication side of the cost estimate used above, the block dimensions were obtained simply from the thermal-hydraulic calculations. No consideration was given to optimizing the block size or the number for blocks from fabrication point of view. Therefore, it is possible that the fabrication component of the cooler cost may also be lower than that assumed above.

Because of this heat exchanger cost uncertainty, the plant cost analysis was repeated *with an optimistic 50% reduction in the cooler cost*, representing 50% material use from FPHE technology and optimistic 50% in the fabrication cost. The results of the cost analysis with 50% reduction in cooler cost are demonstrated in Figure 12. As expected, less expensive coolers reduced the plant capital cost per unit electrical output. The minimum value in Figure 12 is now about 6,800 $/kWe (compared to 7,600 $/kWe with full cooler cost). Lower cooler cost assumptions also moved the optimal cooler design to somewhat larger units. These changes, however, were not very significant. Overall, ***there is still about a 40% penalty on the plant capital cost per unit electrical output calculated with implementation of air cooling***. Figure 12 also shows the same results of the cost analysis in terms of plant capital cost per unit electrical output versus plant efficiency for the same temperature and coolant volume curves. The optimal point is found in the 33-36% range in the net plant efficiency at 40-45 ℃ CO_2 compressor-inlet temperature.

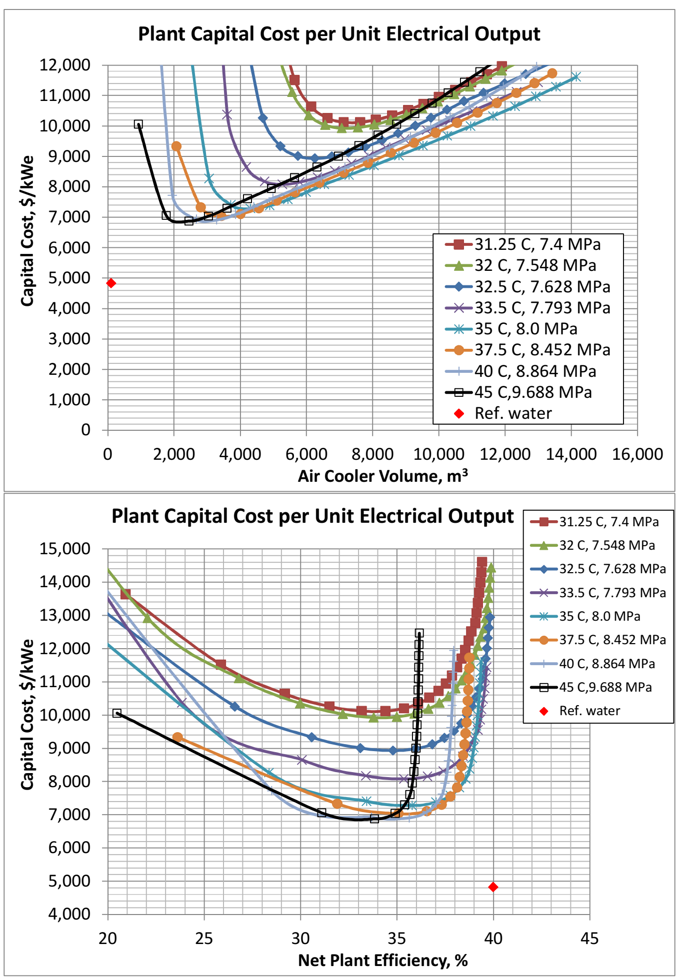

Figure 12. Plant Cost Results with 50% Cooler Cost.

Figure 13 shows the summary of the results obtaining in this analysis of dry air cooling by demonstrating how the air cooler size, obtained from the cost-based optimization (a point at 40 ℃, 3,290 m^3, and 35.6% plant net efficiency is selected), compares with the original water cooler. Still, a significant increase in the heat exchanger volume is required (although this increase is smaller than for the reference conditions in Figure 7). Figure 13 also compares the CO_2 cycle parameters and the plant performance for these two cases. Note the significant reduction in the plant net output (generator power), from 400.3 MWe with water to 356.5 MWe with air.

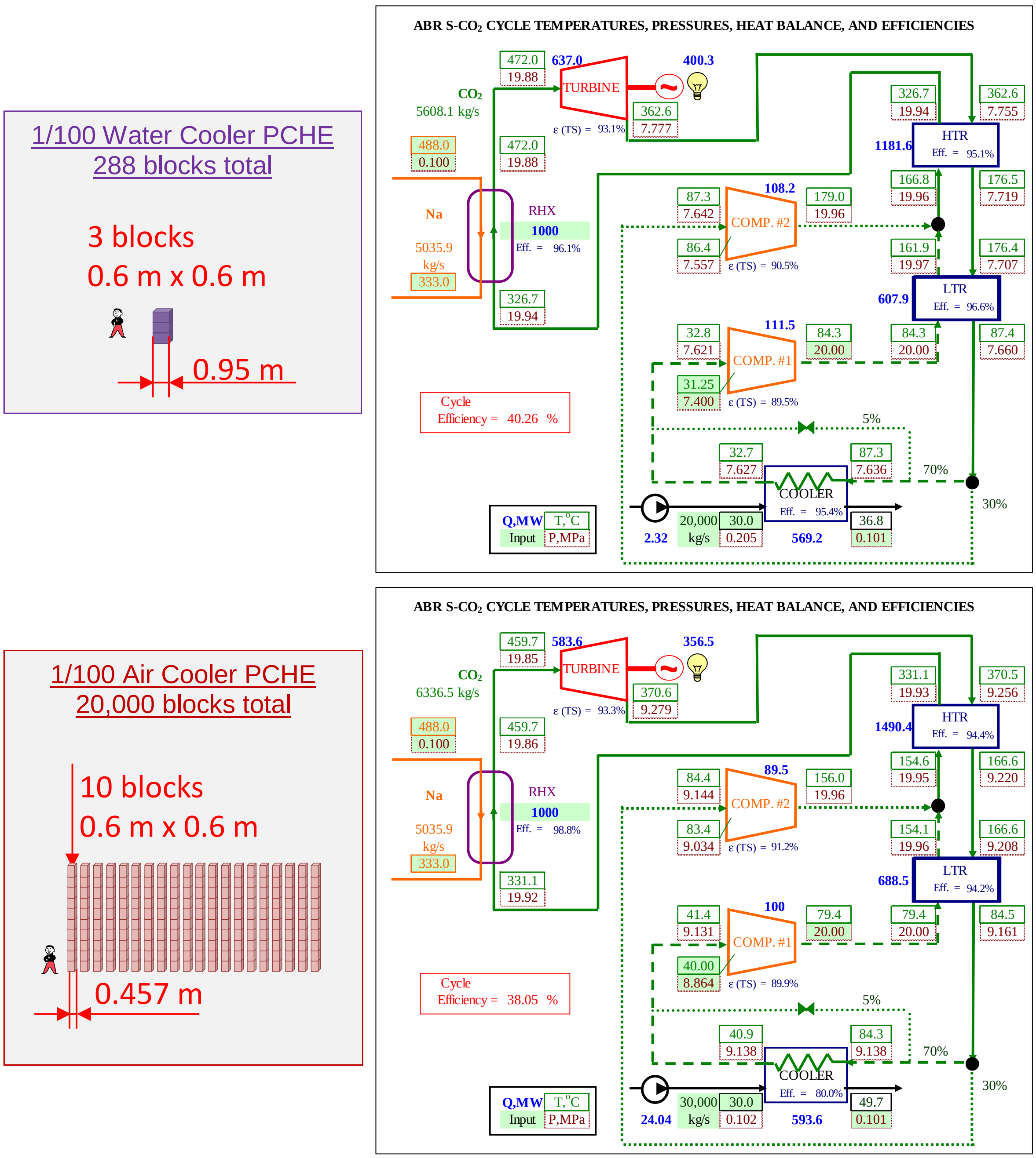

Figure 13. Comparison of the Cooler Size and System Performance for Reference Water Design and the Most Cost-Effective Air Cooler Design.

CONCLUSIONS

The work described in this paper was targeted at investigation of the dry air cooling possibility for an S-CO_2 Brayton cycle at the 400 MWe (1000 MWt) power level. If proved feasible, using air cooling instead of water as an ultimate heat sink would significantly increase the applicability range of the cycle, both by avoiding possible limitations on water use as well as by extending possibilities for plant siting.

A comparison of the heat transfer and transport properties of water and air clearly showed the traditional benefits of using water as a heat transfer medium. Most importantly, the three orders of magnitude lower air density was expected to present significant challenges for the air circulation power requirements. In addition, the factor-of-four lower specific heat for air requires four times higher mass flow rates, further increasing the compressor power. Lastly, the more than 20 times reduction in thermal conductivity leads to increased requirements for the heat transfer surface.

The air cooling calculations for the S-CO_2 cycle conditions and the cooler design optimized for water cooling increased the air compressor power requirements to an unrealistic value of 4,000 MW, which exceeds the total power produced by the plant by a factor of 10. Most of that increase comes from unfavorable air properties (compared to water) as a cooling medium. However, a specific, condensation-like CO_2 temperature profile in the cooler, also contributed to the pumping power increase by imposing addition limits on minimum air flow rate.

To decrease the air circulation power requirements, both larger cooler and larger air channels would be required. To approach a more reasonable 5% of the plant output for the air circulator power, *the cooler volume had to be increased by about 100 times for a PCHE and 900 times for a shell-and-tube heat exchanger*, compared to a water-cooled PCHE. This increase in the cooler heat exchanger would not only significantly increase the plant footprint, but would also result in about a 25-fold increase in the capital cost of the S-CO_2 cycle.

In order to reduce the cooler volume, the temperature difference between the CO_2 and air had to be increased by raising the minimum CO_2 temperature in the cycle. It was shown that the optimal cooler design is obtained for the conditions along the CO_2 pseudo-critical line. The effect of changing the minimum CO_2 cycle conditions on the cooler size and the cycle performance was investigated. The minimum cycle temperature was varied from the original 31.25 °C to 45 °C. At each temperature (except for the reference case of 31.25 °C), the pressure was selected at the pseudo-critical value. The CO_2 flow split between the two compressors was optimized to maximize the cycle efficiency at each set of compressor-inlet conditions. No other cycle and component optimization was made, such as re-optimization of compressor or recuperator designs at the new conditions. Next, for each CO_2 condition, the air flow rate was selected at the minimum value which still limits the reduction in the cooler approach temperature close to the cold end. Finally, with the selected air flow rate, the number of the cooler units was varied parametrically for each set of CO_2 conditions, and the values of total coolant volume and the net plant output (or efficiency) were recorded and compared along all CO_2 temperature points. The results have demonstrated that the cooler volume can be reduced by increasing the CO_2 minimum cycle temperature, although at the expense of lower plant efficiency.

To quantitatively characterize the trade-off between smaller cooler and lower plant efficiency and to select the optimal design conditions, a simplified cost analysis was carried out. In this analysis, the cooler PCHE cost was scaled from the estimate developed previously for other S-CO_2 cycle heat exchangers, and the total plant capital cost and the net power output were compared using the value of unit electricity cost in $/kWe. The minimum value in this cost was found for each CO_2 temperature, which would correspond to the most cost-effective design. It was observed that the unit electricity cost decreases with increasing CO_2 temperature up to about 40 °C, after which it remains about the same. Still, compared to the reference water-cooled design, *an increase in the unit electricity price of about 60% was calculated*, even for the most cost-effective air-cooled design and conditions. The calculations were repeated with an optimistic 50% reduction in the cooler PCHE cost; and the optimal point was found to move slightly towards the larger cooler. Still, *even with such optimistic reduction in the cooler cost, the unit electricity price in $/kWe was calculated to increase by about 40% compared to the reference water case*. Lastly, the size of the air cooler and the overall plant performance, selected at the most cost-effective conditions, were compared to those for the water cooler design. *The cooler size was shown to still be 33 times larger for the air cooling, while at the same time, the plant net output was reduced by 12%*.

Overall, the results of the air cooling analysis for the S-CO_2 cycle have demonstrated that even though this option might be feasible (provided that significantly larger heat exchangers can be accommodated), this approach would not be competitive with water cooling. At least a 40% increase in the electricity price could be expected from the implementation of air cooling keeping the Heatric HX technology. Therefore, the air cooler option might be considered only if a water heat sink is not available (or very restrictive or expensive). In this case, implementing liquid metal-cooled reactors with S-CO_2 energy conversion and air heat sink will practically eliminate water use completely.

The results have shown that increasing the minimum CO_2 temperature to about 40 ℃ is beneficial for air cooling. In this regime, however, the cycle operates far away from the critical point, such that the main benefits of the cycle are lost. It is also important to note that all analyses in this work assumed that the air ambient temperature is 30 ℃ (same as for the water case). If a colder air heat sink is available, it would benefit the cycle by allowing more efficient cycle operation at conditions still close to the critical point. On the other hand though, if the air temperature is hotter than 30 ℃, it will make all the problems identified in this work even worse.

The results of this study agree with some of the remarks from the (World Nuclear Association website, 2013) regarding air cooling:

> • Hardly any US generating capacity uses dry cooling, and in the UK it has been ruled out as impractical and unreliable (in hot weather) for new nuclear plants. A 2009 US DOE study says they are three to four times more expensive than a recirculating wet cooling system. All US new plant license applications have rejected dry cooling as infeasible for the site or unacceptable because of lost electrical generating efficiency and significantly higher capital and operating costs. For large units there are also safety implications relating to removal of decay heat after an emergency shutdown with loss of power. It is unlikely that large nuclear plants will adopt dry cooling in the foreseeable future.
>
> • Both types of dry cooling involve greater cost for the cooling setup and are much less efficient than wet cooling towers using the physics of evaporation since the only cooling is by relatively inefficient heat transfer from steam or water to air via metal fins, not by evaporation. In a hot climate the ambient air temperature may be 40 ℃, which severely limits the cooling potential compared with a wet bulb temperature of maybe 20 ºC which defines the potential for a wet system.

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