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Economic analysis of SCO2 cycles with PCHE Recuperator design optimisation

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Abstract

Supercritical Carbon dioxide (SCO_2) fluid has been extensively studied for efficient power generation, refrigeration and heat pump systems. Its use in closed loop Brayton cycles is being actively considered for higher efficiency power generation systems as well as bottoming cycles for waste heat recovery systems. Small scale test loops and demonstration plants are currently running at various National laboratories and companies in the US. A proper economic feasibility study of these cycles is part of the path to commercialisation of SCO_2 Brayton cycles, particularly when scaling up small scale laboratory loops to viable full scale plants.

The higher efficiency of Brayton cycles using SCO_2 is attributed to the recuperation and transferring of heat from a higher temperature but low pressure flow exiting the turbine to the high pressure fluid exiting compressor at low temperature. This reduces the amount of additional heat required to bring the high pressure fluid to a higher temperature just before expanding in the turbine. Therefore, high effectiveness printed circuit heat exchangers (PCHE) that work with a very close temperature approach are proposed as recuperators to achieve higher cycle efficiency. A balanced plant design can be commercially achieved by optimising the high effectiveness recuperators and assessing cost sensitivity in relation to balancing an efficiency gain with capital and operating cost.

In general, the surface area required for an exchanger increases exponentially as the effectiveness of the heat exchanger is approaching one. This increases the capital cost and weight of the exchangers. In order to realise targets of equipment capital cost to the levels suggested by industrial developers for commercialising SCO_2 cycles, there is the need for understanding the controlling parameters of recuperator design and their influence to the economics of the entire cycle. There are also several assumptions made by researchers and cycle developers when considering cost of recuperators for different applications some of which are not accurate due to overlooking on economies of scale.

This paper has discussed design features of PCHE recuperators and relationship of recuperator effectiveness, cost and cycle efficiency.

1. Introduction

There is an increasing need for clean energy systems with minimal environmental effects. In addition, competitive energy market and environmental legislations put a demand for increasing efficiency of current energy systems. The development of new generation advanced nuclear reactors as well as high efficiency power plants with solar and fossil heat sources are all targeting the use of supercritical CO_2 due to properties obtained at relatively low temperature and pressure. For instance, closed recompression Brayton cycles have been studied and tested at national labs such as Sandia targeting application of high efficiency cycles primarily for nuclear power plants but extended to CSP and fossil.

Supercritical CO_2 Brayton cycles for advanced power cycles have been seriously considered since the 1960's, however, their commercialisation did not gain momentum despite their promising benefits reported in several publications. Recently, there are increased activities by industries to take these beyond research study.

The heat recovery potential of using supercritical CO_2 is related to the enthalpy change of CO_2 with temperature at high pressure resulting in single phase SCO_2 heat release curve with ability to closely match temperature profile of the heat source. In contrast, steam plants have constant temperature boiling causing a pinch with heat source temperature profile at saturation point, which restricts maximum heat recovery. This characteristic combined with sudden property change near the critical point, resulting in relatively low compression work, are main contributors to high efficiency gain without getting to very high temperature.

There are several different layouts of supercritical CO_2 power cycles both open and closed systems that utilise supercritical properties of CO2. The most common SCO2 Brayton cycles include recompression and reheat cycles. Angelino (1969) presented several modified cycles which were later studied by Dostal et al (2002) for advanced nuclear reactor applications. These cycles include pre-compression, partial cooling, recompression and partial cooling with improved regeneration. Comparative study of these different cycles indicated that recompression cycle is known to provide better efficiency, small foot print and simplicity when considering advanced nuclear reactors, Dostal et al (2004). Most of these cycles rely on high effectiveness heat exchangers for allowing high recuperative heat exchange with close temperature approach.

A study using a base case of 550 ^{0}C turbine inlet temperature and 20 MPa compressor outlet showed that SCO_2 direct cycles can achieve ~45% efficiency and power plant cost reduction by about 18% compared to conventional Rankine cycles. With increasing turbine inlet temperature to 700 ^{0}C, the efficiency can grow to 53%, further increasing the cost saving potential compared to conventional steam plant. Additionally, the turbines and compressor designs for SCO_2 are so efficient and compact that they offer a very high power density. Figure 1, shows a comparison of SCO2 Brayton cycle efficiency with other power cycles at different turbine inlet temperature.

On the other hand reheat cycles are predominantly considered for indirect cycles such as those being used for bottoming cycles in waste heat recovery applications using exhausts from simple gas turbines, diesel engines and other relatively low energy sources.

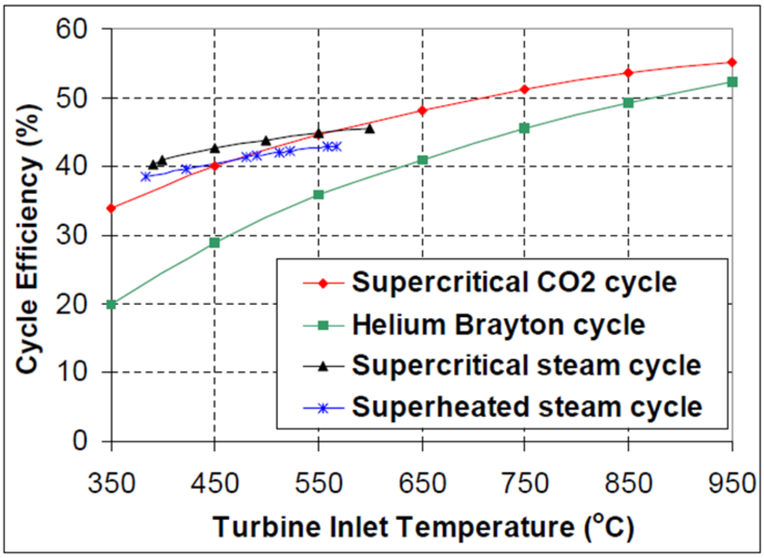

Fig.1, Cycle efficiency comparison of different advanced cycles at different turbine inlet temperature, Dostal et al. 2004

It can be observed that the supercritical CO_2 cycle mostly outperforms other cycles particularly as turbine inlet temperature increases beyond 550 ^{0}C. For instance, to obtain the same efficiency as SCO_2 cycle, helium Brayton cycle will require a much higher turbine inlet temperature. Taking into account other plant losses, a net efficiency comparison would show SCO_2 cycles with more attractive net values due to less auxiliary losses in relative to those that require boilers and high pressure feed water systems. Due to material capability and design limits, the economical application range for SCO_2 closed Brayton cycle is suggested to be between 500 ^{0}C to 700 ^{0}C, Dostal et al. 2004. Whereas open cycles that involve oxyfuel combustion and supercritical CO_2 capture and/or recirculation can utilise much higher turbine inlet temperature due to their direct link to high combustion exit temperature, Allam, et al. 2012.

In all the above cases, high effectiveness heat exchangers with high mechanical integrity and performance are a key part of developing these cycles. In particular, printed circuit diffusion bonded heat exchangers (PCHE) are often used in SCO_2 cycle implementations as recuperators and coolers.

PCHEs are high effectiveness heat exchangers with a high integrity diffusion bonding construction. Diffusion bonding is a solid state joining process with strength of a bond-line same as parent metal strength. Consequently, PCHEs have superior mechanical capability making them highly suitable for high and low temperature recuperators. Fluid flow channels are made using chemical etching providing flexibility in designing several configurations in which pressure drop and heat transfer can be optimised to give high efficiency.

As mentioned above, the efficiency of power plants is directly influenced by the enthalpy of the fluid at the inlet to the turbine, which in turn is related to the inlet temperature and pressure. In an effort to increase the efficiency of plants, the temperature and pressures are getting higher stretching capability of common materials. Consequently, in some cases, there becomes a need to use high strength materials with higher nickel content. When evaluating plant economics, particularly choosing the right turbine inlet temperature, one should not only target higher temperatures and heat sources for seeking minimal efficiency gains but also consider the capital cost associated with the use of expensive materials.

2. HEAT EXCHANGERS FOR SUPERCRITICAL CO2 CYCLES

In this section, parameters that influence the cost and size of heat exchangers will be discussed in attempt to provide guidance when specifying operating and design conditions for recuperators.

Due to the high recuperative heat duty and influence on efficiency, heat exchanger type, design and cost are key elements of achieving optimised supercritical CO_2 power cycles. In general, requirement of high effectiveness for achieving high efficiency targets means that the recuperators are designed to meet large number of NTUs. This makes it impractical to use typical shell & tubes with the need to use several shells connected in series (mainly when considering size, cost and complexity of piping.) Adding to this, recuperators are expected to meet high differential pressures with the higher pressure side typically ranging between 260-330 barg. These conditions exclude other compact heat exchangers including plate fins, plate & frame, shell & plate and other brazed technologies. When considering steady state and transient thermal stress issues related to temperature gradients and plant start-up requirements, diffusion bonded heat exchangers become the only available proven technology with higher technology readiness level. Hence, the reason for their use by most SCO_2 test loops and demonstration plants. They have also been studied for use in SCO_2 cycles particularly in relation to nuclear power by researchers at Argonne National Laboratories (ANL), Massachusetts Institute of Technology (MIT), Sandia National Laboratories (SNL), Idaho National Laboratories (INL), Korean Advanced Institute of Science and Technologies (KAIST) and Tokyo Institute of Technology (TIT). With flexibility of chemical etching providing various possibilities of channel layouts, PCHEs can be specifically optimised to meet particular needs of individual SCO_2 cycles.

In meeting the heat duty for a specified pressure drop, a design that gives higher overall heat transfer coefficient and maximise the available mean temperature difference will require less surface area and is cost effective, see equation (1).

$$Q=U*A*TD \tag{1}$$

Where, Q is heat duty; U is overall heat transfer coefficient and TD is mean temperature difference.

The logarithmic mean temperature difference (LMTD) based on terminal temperature is multiplied by other factors to account for the shape of heat release curves, fluid flow configuration, temperature short circuit factor and other conduction losses.

$$TD= F_{hc}*F_{geom}* F_{sc}*F_{long\text{-}c}* LMTD \tag{2}$$

The heat curve factor (F_{hc}) is to account for the shape of the heat release curve (temperature profile of the hot and cold fluid). This is calculated against terminal-based LMTD using integral averaging of temperature difference along the profile by subdividing it into zones or points. The geometry factor (F_{geom}) accounts for contact penalty between streams as a result of flow configurations (counter flow, co-flow and cross flow). In case of multi-pass configurations with several passes arranged side by side, short circuit factor (F_{sc}) takes account of any heat flow between passes. Longitudinal conduction factor ($F_{long\text{-}c}$) is to account for any longitudinal conduction leak through the solid material without contributing to the heat transfer between streams. This is mainly significant in designs with large temperature span but short travel length due to pressure drop limitations and when they are combined with poor heat transfer coefficient in the core.

A design that is able to provide higher geometry factor as a result of lower contact penalty by using a more counter flow arrangement is normally the target for high effectiveness heat exchangers. Figure 2 shows different flow configurations of PCHE (such as co-current, cross or counter-current flow arrangements) to best optimise the size of the heat exchanger.

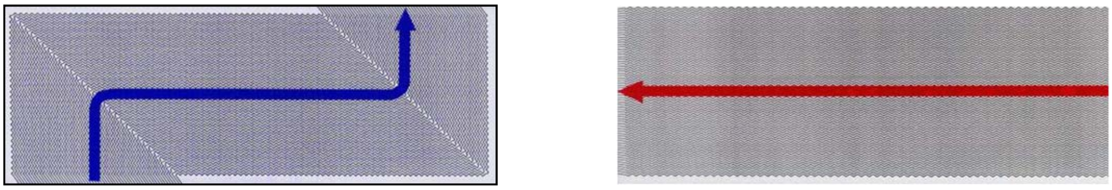

Fig. 2a, Counterflow example with side-side and end -end flows.

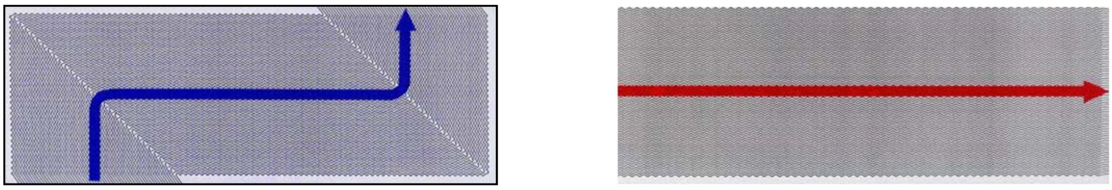

Fig. 2b, Concurrent parallel flow example with side-side and end-end flows.

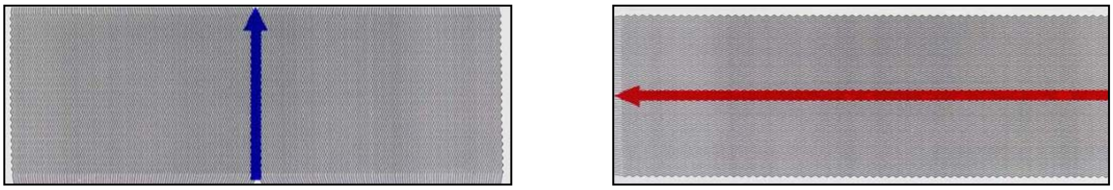

Fig. 2c, Cross flow example with end-end flows.

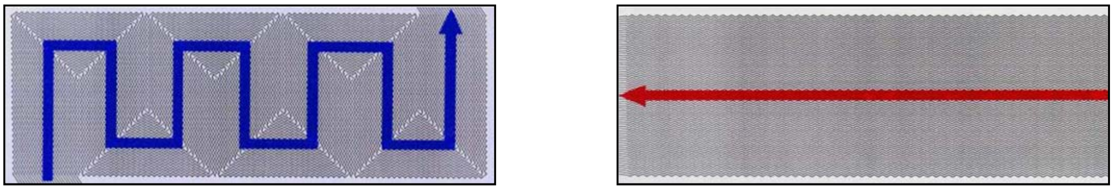

Fig. 2d, Cross counter flow example with multi-pass on one side.

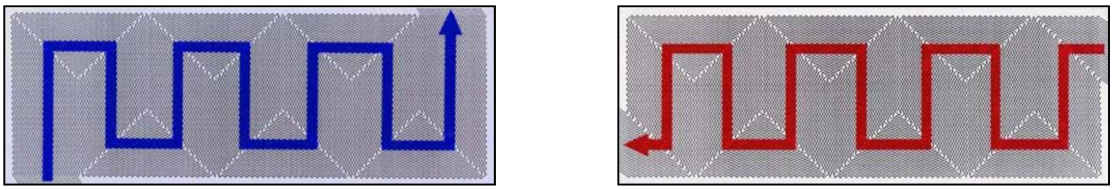

Fig. 2e, Multipass counter flow example with equal multi passes on both sides.

PCHES can be configured to have a higher counter flow region to better match large NTU with the pressure drop available.

$$NTU = \max(\Delta T_{hot}, \Delta T_{cold})/(F_{hc} * LMTD) \tag{3}$$

where, $\Delta T_{hot} = |T_{hot\text{-}in} - T_{hot\text{-}out}|$ and $\Delta T_{cold} = |T_{cold\text{-}out} - T_{cold\text{-}in}|$

In relation to placing stream headers on exchanger cores while utilising most of the surface area for heat transfer, there are different arrangements that can be used for recuperator designs. Optimum header location varies depending on process mechanical design conditions, pressure drop and design type used to meet those conditions. There are designs that are suited for supercritical CO2 recuperators. Figure 3, shows a few examples of typical PCHE header configurations for SCO2 recuperators.

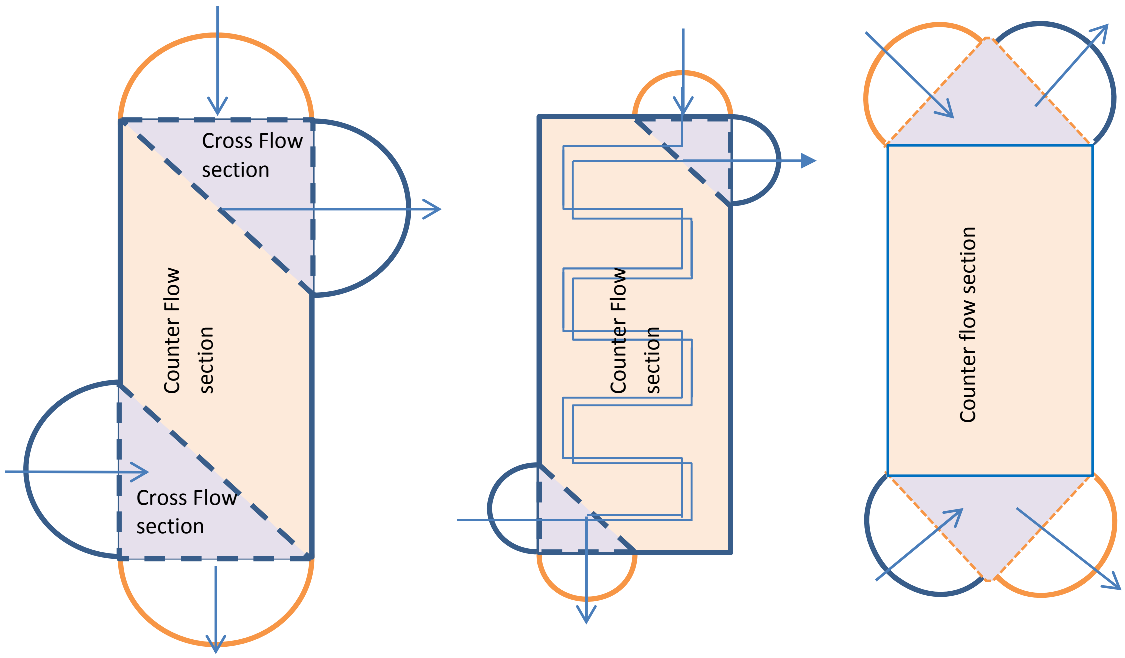

Fig. 3, Header configurations examples maximising counter flow region

The intent of different configurations is to maximise geometry factor while meeting pressure drop and mechanical design conditions. For very high design pressure, there are also ways of creating headers/ports within the block meeting mechanical limitations while maintaining greater compactness.

Additional consideration should also be taken when designing SCO_2 recuperators operating at varied compressor pressure ratios. System pressure influences the heat release curve or temperature approach to the extent of creating a pinch point within the operating temperature range. Therefore, consideration should be given by conducting pressure sensitivity of the heat exchanger performance at different operating points. Conversely, change of pressure ratios influences temperature pinch and hence driving the need for modifying simple cycles to include additional compressors and recuperators in the cycle. Pre-compression and recompression cycles introduced by Angelino are to tackle pinch problem of simple cycles as well as targeting higher cycle efficiency, Angelino, et al. (1969).

Commercialisation of both high temperature Brayton recompression cycles as well as relatively low temperature waste heat recovery SCO_2 reheat cycles is given a higher priority by industries and governments. Hence, the need for understanding the controlling parameters of recuperator design and their influence to the economics of the entire cycle is obvious. There are several assumptions made by researchers and cycle developers when considering cost of recuperators for different applications some of which are not accurate. Part of the purpose of this paper is to address misconceptions on cost of SCO2 recuperators while guiding cycle developers towards an optimised design that favour low capital cost without having significant impact on efficiency.

3. Economic feasibility – effectiveness of exchangers versus cost and cycle efficiency

Plant developers have clearly defined that in order to make the SCO_2 power cycles economically feasible and compete with current conventional power plants, capital cost of equipment needs to be reduced from current estimates. On the other hand, there is a great drive for increasing efficiency of these plants. While there are design solutions that cope with high effectiveness requirements reaching 0.99, the economics of requiring such recuperators have to be carefully studied against the benefit of low operating cost over the life time of the plant. In this section, the influence of various design conditions on the cost of recuperators and cycle efficiency is discussed.

For the analysis shown in this paper, the high temperature recuperator is selected for the case study to demonstrate relationships of cost and performance. The design temperature of the heat exchanger has been limited to Stainless Steel 316 allowable temperature. All the variables (UA, Efficiency, Cost, Payback time) are in proportion to the 18 degrees Celsius temperature approach case. The study is also based on a typical SCO2 recompression cycle.

3.1. Impact of Effectiveness on size

When developing SCO_2 cycles and deciding process conditions, it is common to focus on theoretical efficiency of the cycle, disregarding practical capital cost of the plant at the concept stage. As a consequence, often some cycle developers request design of heat exchangers that meet high effectiveness (close to unity) and/or very high temperatures, overestimating cost of the plant at feasibility stage. Hence, basing economics on such estimates of cost will give a different picture to the reality.

Figure 4 shows the relation between the UA ratio and the effectiveness of the heat exchanger. The UA is a parameter calculated from the heat transfer coefficient and the surface area that indicates the amount of heat transfer area required by a heat exchanger, see equation (1). For the temperature approach analysis, a 10 MW (thermal) plant size is considered while varying the temperature approach from 2 up to 18 degrees Celsius. The base case for the UA ratio depicted in Figure 4 is the 18 degrees Celsius temperature approach. The effectiveness of a heat exchanger is a variable that expresses the actual heat transfer rate that delivers a heat exchanger in comparison to the theoretical maximum that could occur in a counter-flow heat exchanger with infinite surface area.

In its simplified form,

$$\textit{Effectiveness} = 1 - \frac{\Delta T_{approach}}{T_{hot\ in} - T_{cold\ in}} \qquad (4)$$

where minimum temperature approach, $\Delta T_{approach} = Min[(T_{hot\ in} - T_{cold\ out}), (T_{hot\ out} - T_{cold\ in})]$

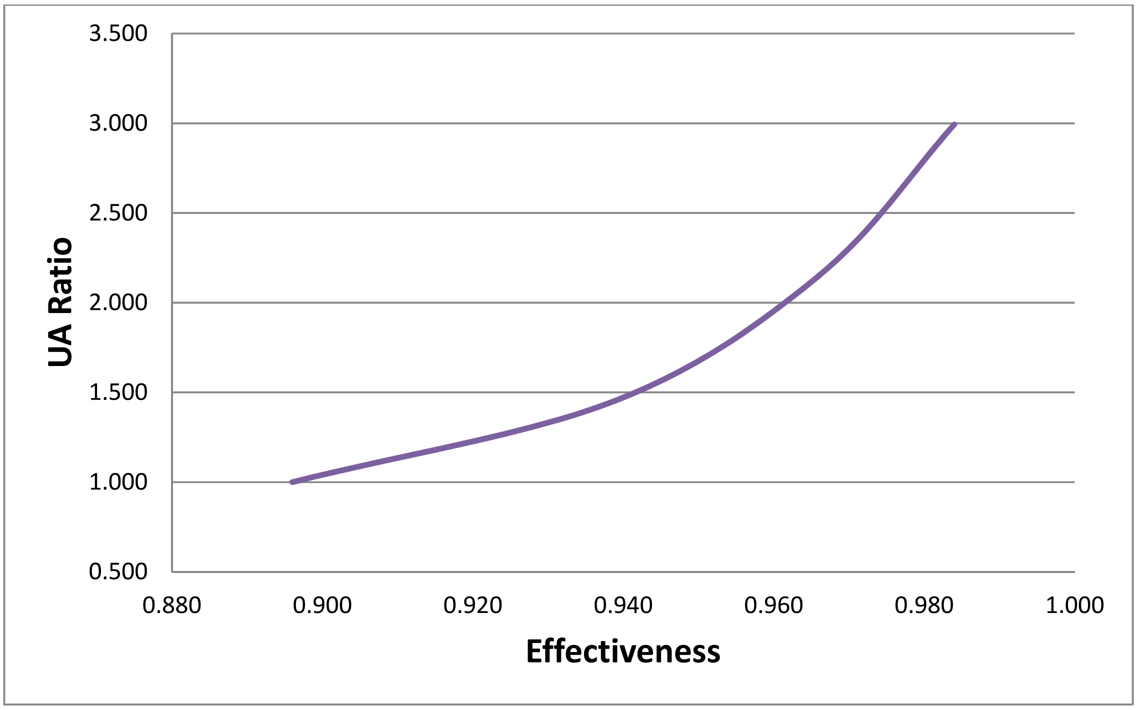

Fig. 4, UA Ratio vs. Effectiveness.

In order to reach effectiveness close to unity, the temperature difference between streams needs to be very small. This is reducing further the minimum temperature approach by getting the outlet temperature of the cold stream as close to the inlet temperature of the hot stream as possible and vice versa, resulting in small LMTDs. Therefore, the effectiveness of a heat exchanger increases when reducing the temperature approach.

As expected, the higher the effectiveness, the more heat transfer area (UA) is required by the heat exchanger. As can be seen from Figure 4, the UA grows exponentially when the effectiveness of the exchanger tends to unity. For example, with the conditions above mentioned, a heat exchanger with an effectiveness of 0.98 requires twice the heat transfer area than a heat exchanger with an effectiveness of 0.94.

This needs to be studied and analysed at different operating points when selecting the optimum point of a cycle. Ideally there needs to be an interaction between cycle developers or bodies that do heat balance of plant with heat exchanger designers at an early stage before design parameters are fixed.

The cost and size of a heat exchanger is directly proportional to its UA. In Figure 5, the Cost Ratio has been plotted against the Efficiency Ratio of a cycle. As mentioned before, both parameters have been normalised with respect to the 18 degrees temperature approach case.

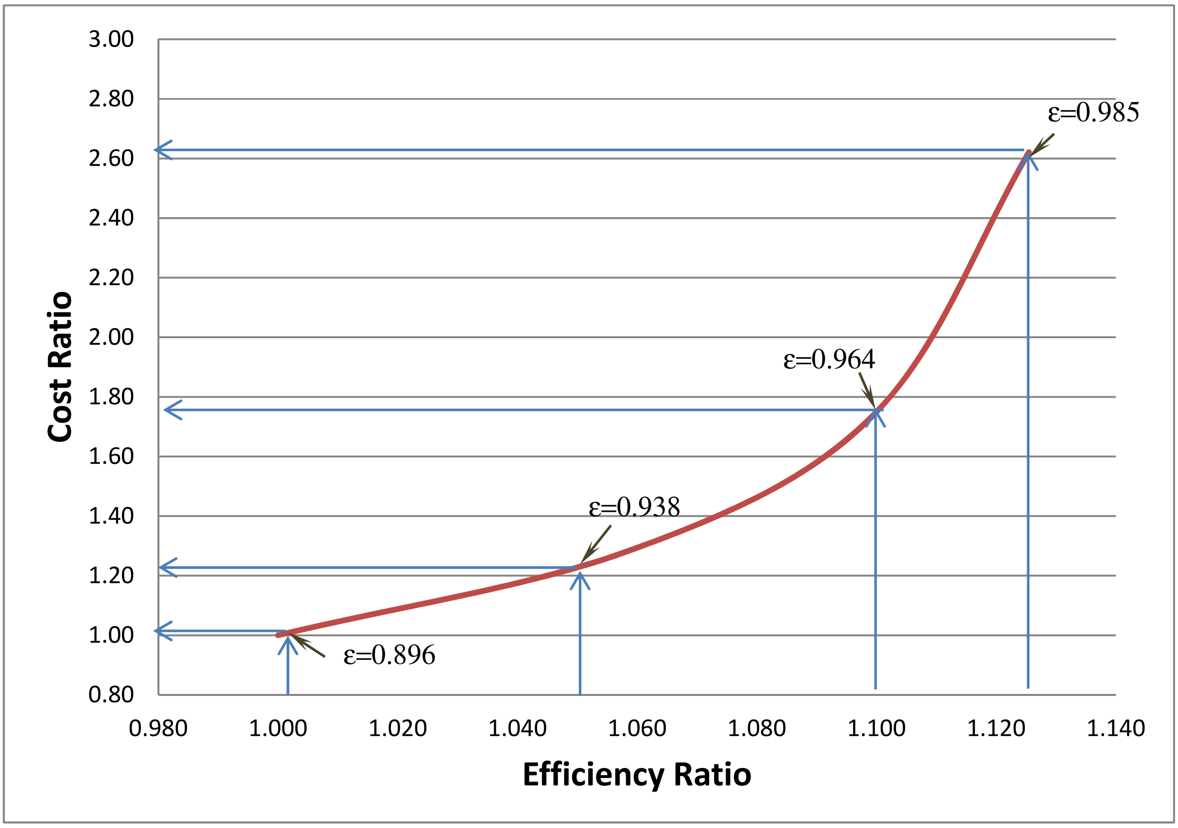

Fig. 5. Cost Ratio vs. Efficiency Ratio.

The increase in cost of equipment as cycle efficiency gain increases follows similar trend to Figure 4. The cost ratio increases exponentially with the efficiency ratio against the base case, experiencing a steeper increase at higher efficiency ratios. In Figure 5, it can be demonstrated that at relatively moderate effectiveness values in the range of ~0.9 to 0.94, a higher increase in efficiency can be gained with relatively small cost increase, whereas at higher effectiveness regions (0.96-0.99), a small gain in efficiency costs a significant capital value. For instance, an efficiency gain of 2% in the region of ~ 0.9 effectiveness requires a cost increase of only 15 %, while the same efficiency gain at higher effectiveness regions requires about 60% increase.

Of course these efficiency gains have to be evaluated against operating cost savings and payback period within the life of the power plant. Nevertheless, the message here is, a minimal further efficiency gain in a cycle already working using high effectiveness recuperators comes at the expense of significant capital cost increase. Reinforcing the point mentioned above, a detailed study and optimisation are required when selecting fixing cycle parameters because of its implication in making an economic case during commercializing.

Figure 6 shows the relation between the Cost Ratio and the Payback Time Ratio. The ratio is based on a base case of 10 MWe plant with 18 C temperature approach. Plant capital cost is estimated based on a study by Dirk Pauschert as well as from discussions with Barber Nichols. Energy process were taken from

US Energy Information Administration website. The capital cost increases with its efficiency resulting in longer operating time needed to get a return on an investment.

As already stated, the closer the temperature approach, the higher the plant efficiency. However, the capital investment required will increase at a faster rate than the plant efficiency increase leading to longer payback times. At very high efficiencies, the amount of time required to offset the initial investment with operating cost savings can become commercially prohibitive. This can be clearly seen in Figure 6.

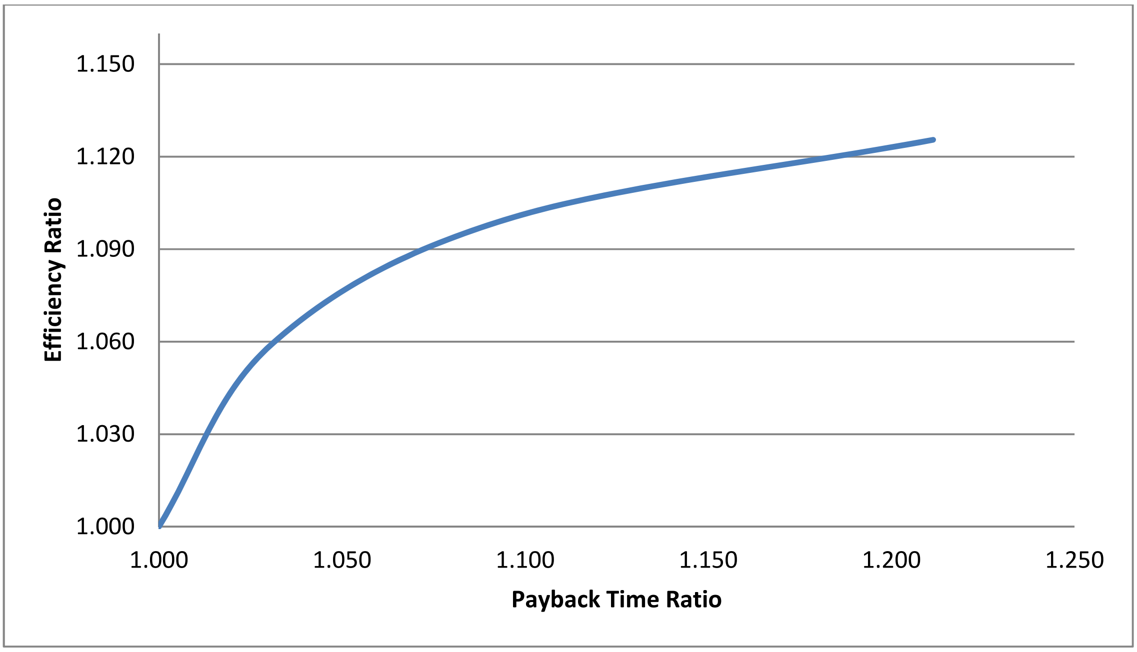

Fig. 6, Efficiency Ratio vs. Payback Time Ratio.

3.2. Economies of scale (cost per kilowatt versus Plant Size)

The cost of a PCHE is a function of several parameters: mechanical design conditions, UA requirements, and allowable pressure drops. Consequently, using a cost comparison index such as US dollar per plant thermal duty ($/kWt) is misleading. For instance, the drop in material strength as a function of design temperature is not linear and often there is a sharp drop in strength at higher temperatures – requiring higher material thickness for the same thermal duty. On the other hand, the impact of pressure drop is often overlooked. In general, there is a tendency to specify a tighter allowable pressure drop for exchangers due to unknown early assignment of high pressure drop to piping, despite sizing pumps and compressors for a higher pressure loss.

Since cost targets discussed at different venues have been set based on mainly $/kWt and for the sake of showing economies of scale $/ kW ratio is used in this discussion. For the purpose of this analysis, all of other parameters have been kept constant and only changing the duty of the plant by varying flow rates. The variable has been normalised with respect to the $/kW of the 500 MW (thermal) plant.

In Figure 7, $/kWt ratio is plotted against Plant Size Ratio (kWt ratio). The plant size has been varied from 5MWt (1 on horizontal axis), which is the base case, up to 500 MWt (100 on horizontal axis).

Figure 7 clearly shows an overestimation that can occur if one uses $/kWt established for smaller duties and linearly extrapolate it to large size duties. This is due to natural economics of scale, additional optimisation possibilities with large size, relatively low overhead contributions to cost.

It is a common confusion to believe that the cost per kilowatt of PCHE technology is fixed with respect to size. In fact, as PCHE size increases, $/kW decreases logarithmically as seen in figure 7.

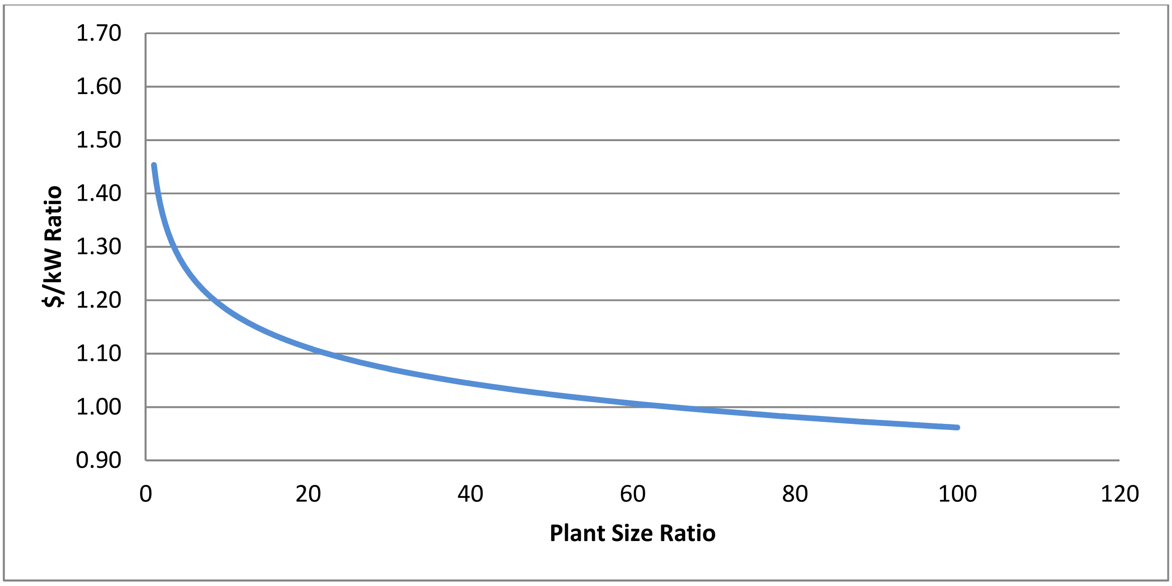

Fig. 7, $/kW Ratio vs. Plant Size Ratio.

There have been several different assumptions on actual cost of equipment. Some researchers have pointed out at various conferences that heat exchangers would take a major share of the overall capital cost of SCO2 plants, making a case for new developments to reduce cost. Estimates vary among researchers but there are cases where reports show as high as 90% of SCO2 cylce cost to be in heat exchangers, Fleming et al (2013). There are two issues associated with such estimates one being an overlooking in the economies of scale. Often estimates are based on extrapolation of cost/kWt from small duty (test loop size) to large plant sizes with hundreds of MWt. The other is underestimation of the rest of plant equipment cost due to lack of proper data (e.g. large duty SCO2 turbines, compressors) as well as other estimates like piping, controls, etc.

Further increase in efficiency is often obtained by increasing the fluid temperature at the inlet to the turbine. Although this provides attractive theoretical cycle efficiency, balancing capital investment cost with plant operating cost is necessary in selecting the right cycle design efficiency. This is also because of the fact that there is a step change in material cost and design life, when increasing maximum operating temperature. This often requires a change from using normal 300 SS grades to a high nickel alloys like Alloy 617, HR120, HR230 and Alloy 740H. Another limitation to design life in high temperature applications is the onset of creep and its interaction with fatigue, which most materials have no available data for evaluation life.

There are notions that the current price of recuperators should drop to \$25/kWt to be able to commercialise SCO_2 cycles discussed at the latest EPRI conference. These estimates can be achievable with PCHE designs depending on the conditions in which such targets are based such as design conditions and efficiency levels.

Conclusion

In this paper, recuperator design parameters and their influence on heat exchanger cost as well as cycle efficiency of a plant is discussed.

It is clearly demonstrated that an effort to gain further minimal theoretical efficiency point in a cycle already working at higher efficiency using high effectiveness recuperators comes at the expense of significant capital cost increase. Hence, it is suggested that attempts to obtain incremental efficiency gains have to be evaluated against equipment cost as well as operating cost saving and payback period should consider design life of a plant with a better forecast of future fuel cost.

A cost versus performance optimisation should be carried out in the early stages of the project in joint collaboration with equipment designers and vendors. This also helps utilising the higher degree of flexibility in PCHE design and construction.

When setting targets of cost \$/kWt in a plan to commercialise SCO2, it is crucial to acknowledge that recuperator design and cost are highly determined by effectiveness, pressure drop, product form and material design limitations rather than thermal duty. In addition, estimates of such indices should account for the influence of economies of scale on these parameters. Although \$/kWt may work for rotating equipment, it hardly defines the true cost of heat exchangers. Other parameters such as \$/kg or \$/UA are more appropriate for evaluating recuperator cost.

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Authors Biography

Dr Dereje Shiferaw leads a team of engineers working in the design and development of compact heat exchangers for new applications, mainly power generation, chemical processing, carbon capture and waste heat recovery. He is highly experienced in mechanical and thermal design of high temperature exchangers for challenging systems and process applications. Dr Shiferaw introduced several novel ideas for improving product performance and reduce cost. He has been involved in the design of several SCO2 recuperators that are in operation and is the lead engineer for the recuperators in the NetPower Demo Plant. He prepared development program for Hybrid type exchangers where formed fins and etched channels are combined to optimize performance. Dr Shiferaw is a chartered mechanical engineer and is an expert in heat transfer, fluid flow, power cycle optimization and thermal stress analysis.

Jorge Montero Carrero is part of the Design and Development Engineering Team. He has been involved in the design and optimisation of compact heat exchangers for new applications, mainly power generation, with a special focus on heat exchangers in sCO2 Power Cycles. He has experience in the mechanical, thermal and hydraulic design of high pressure and high temperature heat exchangers. He is an active member of the sCO2 community and co-presented the "SCO2 Heat Exchangers" tutorial in ASME Turbo Expo 2015."

Renaud Le Pierres is responsible for assisting in the development of Heatric business within new markets, mainly power generation, chemical processing, carbon capture, and waste heat recovery. He works within the Business Development team that, with the help of the Design Development Team, adapts existing products or develops new ones, when required, to satisfy these new markets requirements. Renaud Le Pierres has been involved in sCO2 Power Conversion Cycles projects since 2005, with successful supplies of equipment to Sandi National Laboratories, Argonne National Laboratories, GE, and ECHOGEN. In this role, he participates actively in the sCO2 community, including regularly presenting heat exchanger tutorials and chairing discussion panels at ASME Turbo Expo.