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OPTIMIZATION OF OPERATING PARAMETERS OF A RECOMPRESSION sCO_2 CYCLE FOR MAXIMUM EFFICIENCY

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ABSTRACT
Supercritical CO_2 power systems offer significant power density advantages along with high efficiencies, compared to traditional Rankine or Brayton cycles. Of the several viable configurations, the recompression cycle has higher efficiency compared to the simple recuperated cycle for source temperatures above 500°C. It also provides a good trade-off between efficiency and plant complexity. This paper explores the dependence of critical operational parameters on source and sink-temperature, which is then used as a means to generate guidelines for developing recompression sCO_2 power plants. The maximum source temperature in the analysis is restricted to 565°C to take advantage of the existing materials and technologies associated with industrial steam turbines. However, the methodology described herein is applicable for any other source temperature range.

An important source of thermal efficiency degradation in power plants is attributable to heat exchangers. Analysis presented in this work directly relates the optimum operational parameters of the recompression cycle to the operation of the low temperature recuperator. Thermodynamic analysis confirms that a recompression fraction of 0.25 and pressure ratio of 2.5 is as an optimum design point for the recompression cycle. The penalty in efficiency and power while operating the plant in off-design conditions for a fixed recompression fraction and pressure ratio is highlighted.

NOMENCLATURE

1-8	states on ideal thermodynamic cycle
′	states on real thermodynamic cycle
η_{th}	thermal efficiency
ϕ	recompression fraction
h	specific enthalpy (kJ/kg)
HT	high temperature
HTF	heat transfer fluid
LT	low temperature
p	pressure (bar)
T	temperature (°C)
T-sink	main compressor inlet temperature (°C)
T-source	turbine inlet temperature (°C)
$\dot{w}$	rate of work (W)
w_{sp}	specific work (kJ/kg)

INTRODUCTION
Supercritical carbon dioxide (sCO_2) based thermodynamic cycles provide efficient means of converting heat to mechanical energy. Higher plant efficiency results in lower operating costs, while, higher power density leads to compact equipment and footprint thus reducing the capital expenditure. Among the several viable configurations, recompression cycle promises higher efficiencies in excess of 50% [1]. In addtiona, the

recompression cycle also provides a good trade-off between efficiency and plant complexity.

Thermodynamic studies on recompression sCO_2 Brayton cycle have found considerable interest in the past and significant information exists in the literature. In a notable work, Dostal [2] shows that the recompression cycle provides the highest efficiency compared to other cycle configurations for turbine inlet pressures in excess of 200 bar. Dostal [2] finds that the optimal pressure ratio is about 2.6 and any deviation from the optimal pressure ratio results in a sharp drop in thermal efficiency. This is in contrast to findings by Feher [3] where cycle efficiency is insensitive to pressure ratio. This anomaly arises as Dostal's findings are reported for fixed heat exchanger geometries compared to an optimally varying heat exchanger configuration considered by Feher. The recompression fraction of 0.37 to 0.42 is optimal as reported by Dostal [2]. The variation in cycle efficiency due to recompression fraction is attributable to a swing in the effectiveness of the High Temperature (HT) recuperator. The effect of increasing minimum cycle temperature (main compressor inlet temperature) and corresponding optimum pressure ratio is to reduce the cycle efficiency. This result however, is due to varying heat exchanger geometry.

A further extension of this work highlighting the importance of the heat exchanger (recuperator) in a recompression cycle is reported in Dyreby *et. al.* [4]. The authors here explored the recompression cycle design as a function of total available recuperator conductance as opposed to effectiveness. Results similar to Dostal are obtained by varying pressure ratio, recompression fraction, sink and source temperatures. For smaller recuperator geometries, Dyreby *et. al.* conclude that the recompression cycle efficiencies are lower than corresponding simple recuperated cycle.

There have been a number of papers related to optimizing the operating parameters of sCO_2 cycles [5, 6]. Most of these use Pareto or genetic algorithm techniques for multi-objective optimization. In a recent paper by Garg *et. al.* [5], a two-dimensional path function coordinate system is used to provide insight into the control of compressor discharge pressure for a host of source temperatures for maximizing either power or efficiency.

The current paper provides a comprehensive design space exploration for maximizing recompression cycle efficiency and augments it with off-design analysis for varying sink temperature.

sCO_2 SYSTEM DESIGN CONSIDERATIONS

It is evident from thermodynamics that higher turbine inlet temperatures result in higher cycle efficiencies. For industrial steam turbines in the range, 10-100 MW, 565°C is the general inlet temperature limit. This enables the use of conventional steel alloys for the turbine components. Generation III concentrating solar power (CSP) tower technology is anticipated to generate temperatures in excess of 700°C. However, current parabolic trough CSP steam turbines operate at around 400°C. Continuous improvements in concentrating solar parabolic trough collector technologies are making it possible to realize higher temperature limits. In the near future sCO_2 power systems may be able to benefit from the advances in material development [7] for ultra-super critical steam turbines; however, the use of lower turbine temperatures and existing designs will improve the speed at which a new system can be commercialized.

Another important parameter affecting the sCO_2 efficiency and power output is the pressure ratio. From the equipment design and plant economics perspective, higher operating pressures are not desirable. Since the minimum inlet pressure is 73.8 bar for sCO_2, higher optimum pressure ratio makes equipment design challenging.

Finally, higher sink temperatures in tropical climates need consideration in sCO_2 cycle design and operation. Dry cooling is increasingly becoming common due to water scarcity, which further elevates the sink temperature from those used in previous studies.

METHODOLOGY

The analysis presented in this paper considers a notional power output of 10 MW with a maximum operating pressure restricted to 250 bar. The operating limits for the design are constrained to leverage commercially available industrial steam turbine technologies.

Figure 1 is a typical schematic of a recompression cycle whose state points are as per the T-S diagram in Figure 2. The cycle has two pressure levels. The main compressor (1-2′), recompressor (8-3′) and turbine (5-6′) operate at the same pressure ratio. Sensible heat at the turbine exhaust is recovered in HT (6′-7) and LT (7-8) recuperators, thus reducing the heat addition in the main heat exchanger (4-5). The flow is split before the gas cooler (8-1) and a fraction of the total flow (recompression fraction ϕ) enters the recompressor. The flow then merges before entering the HT recuperator. The main heat exchanger and gas cooler are excluded from the analysis. Hence, in this analysis, source temperature (T-source) refers to the turbine inlet temperature and sink temperature (T-sink) refers to the main compressor inlet temperature.

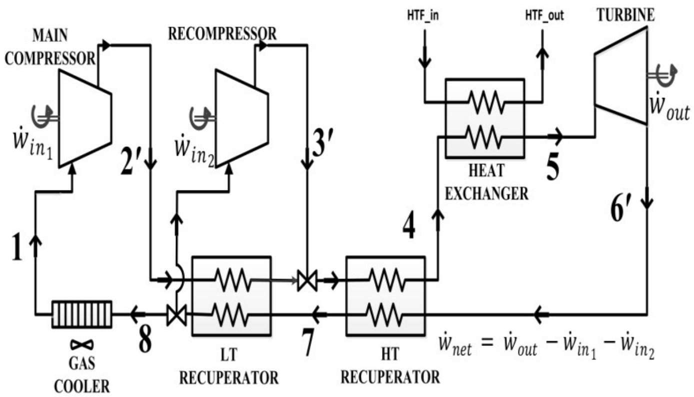


Figure 1: Schematic of recompression cycle power plant

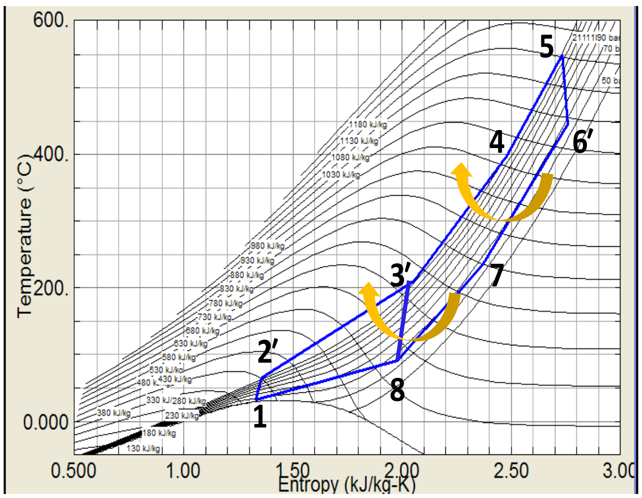


Figure 2: Typical T-S Diagram of Recompression Cycle

The thermal efficiency of the recompression cycle as stated in Equation (1) accounts for the recompression fraction ϕ.

$$\eta_{th} = \frac{(h_5-h_6')-(1-\phi)\,(h_2'-h_1)-\phi(h_3'-h_8)}{(h_5-h_4)} \quad (1)$$

The specific work output from the cycle is shown in Equation (2).

$$w_{sp} = (h_5 - h_6') - (1-\phi)(h_2' - h_1) - \phi(h_3' - h_8)\text{ , kJ/kg} \quad (2)$$

This paper is divided into two sections; design space exploration and off-design analysis. A Matlab® program linked to the REFPROP database [8] has been developed to compute the design state-points of the recompression cycle. CO_2 properties are estimated from Refprop 9.1, using the equation of state by Span & Wagner [9]. Inputs to the code are the low side pressure, main compressor inlet temperature, turbine inlet temperature, pressure ratio, recompression fraction, desired output power, LT and HT recuperator minimum temperature difference, and turbine & compressor isentropic efficiencies. The outputs include, but are not limited to, the cycle efficiency and net specific work.

Key assumptions for design calculation are listed below:
- Low side pressure is fixed at 76.5 bar
- Isentropic efficiency of the turbine is 90%
- Isentropic efficiencies of the compressors are 85%
- Minimum temperature difference across the hot and cold side of the recuperator is 5°C
- No pressure drop in the recuperators were considered during design space exploration, however, the effect of pressure drop in the recuperators have been incorporated for the off design analysis.

Even though some assumptions appear restrictive, they do not affect the qualitative nature of the design space exploration In the operating range of this analysis, it has been verified that the temperatures are all above pseudo-critical temperature, because of which pinch does not occur within either HT or LT recuperator.

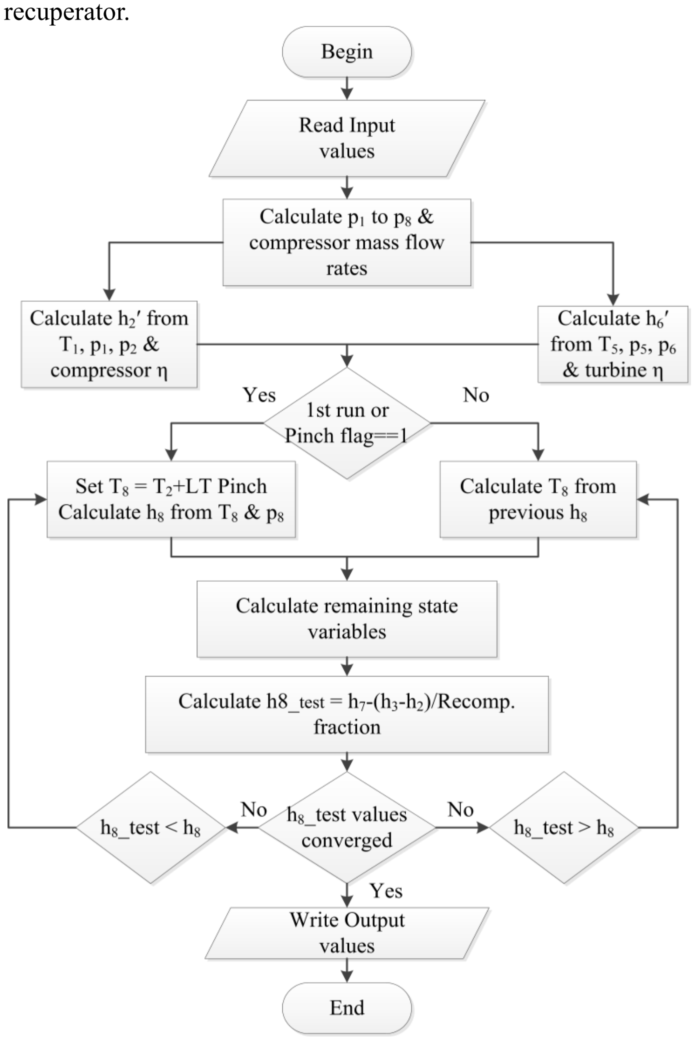


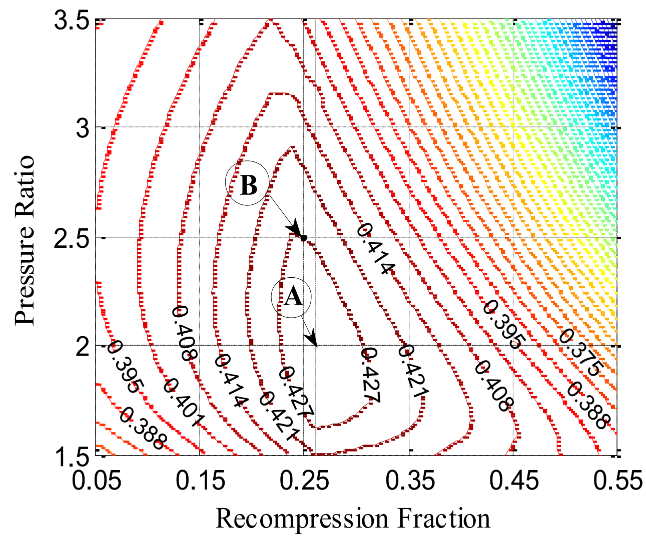
Figure 3: Flowchart for the design state-point calculation of recompression cycle

A flowchart describing the computation scheme for the recompression cycle thermodynamic state-points is shown in Figure 3. Pressure ratio and recompression fraction are two key parameters deciding the efficiency and power output of a recompression cycle. For a practical range of turbine and main compressor inlet temperatures, the control parameters are varied to explore the design space. The design analysis is carried out in a staged manner, at discrete values of T-source & T-sink, culminating in the selection of an optimum design operating point. This approach towards finding an optimum operating point provides greater insight as compared to a single multi-variable optimization.

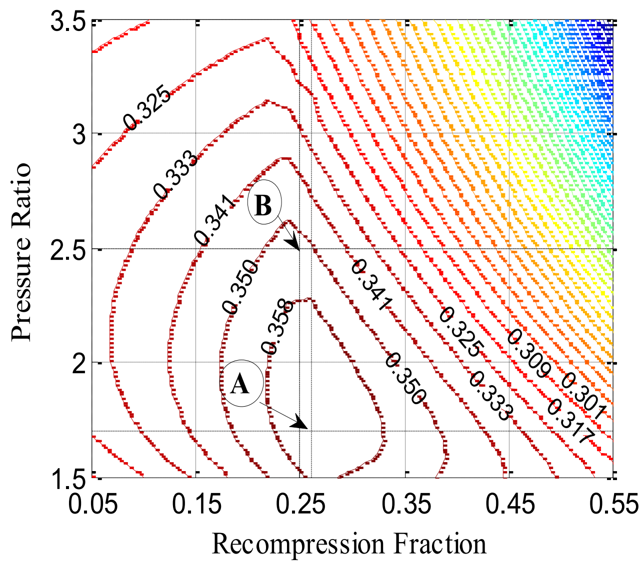
Off-design analysis of the cycle involves off-design performance calculation of individual components namely, turbine, compressors and recuperators. A sliding mode control of the inlet pressure is used for the turbine off-design performance estimation. The blading constant (a function of the turbine geometry) is evaluated at the design point. The turbine inlet pressure at off-design mass flow conditions is calculated from the blading constant. For the compressors, performance maps relating pressure ratio and corrected mass flow rate are used (Thermoflow® compressor maps are employed in this analysis). The recuperators are sized at the design point with a 3.5% pressure drop and conductance re-evaluated at off-design conditions. Pressure drop is one of the changes that differentiate the results presented in the design space exploration and the off-design analysis section of this paper. To achieve a targeted power of 10MW at the design and off-design conditions, constrained minimization methodology is employed.

DESIGN SPACE EXPLORATION

In Figures 4 & 5, pressure ratio and recompression fraction are varied to find the optimum efficiency of the recompression cycle. Maximum thermal efficiency point is marked 'A'. Point 'B' referring to a pressure ratio of 2.5 and recompression fraction of 0.25 is discussed later in this section. Decreasing the source temperature from 565°C to 450°C reduces the cycle efficiency by 7% as visualized in Figures 4a and 4b. Interestingly, it is found that optimal recompression fraction of about 0.25 remains identical while optimal pressure ratio is about 2. On the other hand, the effect of sink temperature change on optimal recompression fraction and pressure ratio is significant as shown in Figures 5a and 5b. The optimum pressure ratio decreases from 2.7 to 1.7 and optimal recompression fraction decreases from 0.4 to 0.25 for sink temperature change from 32°C to 55°C, leading to an overall efficiency difference of 7%.

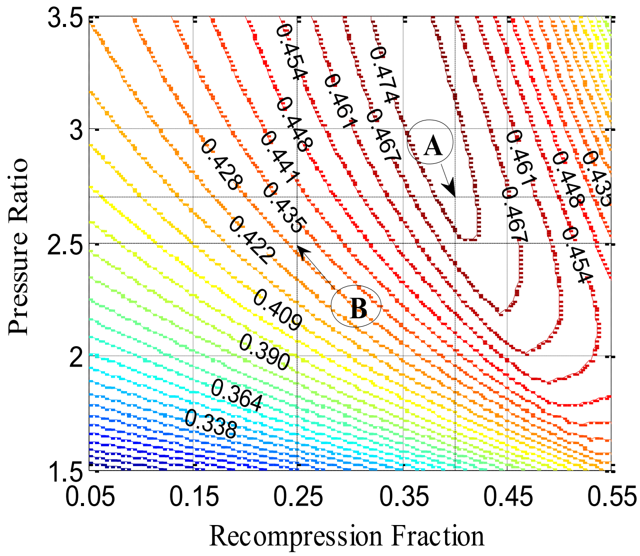


4a: T-source 565°C & T-sink 45°C

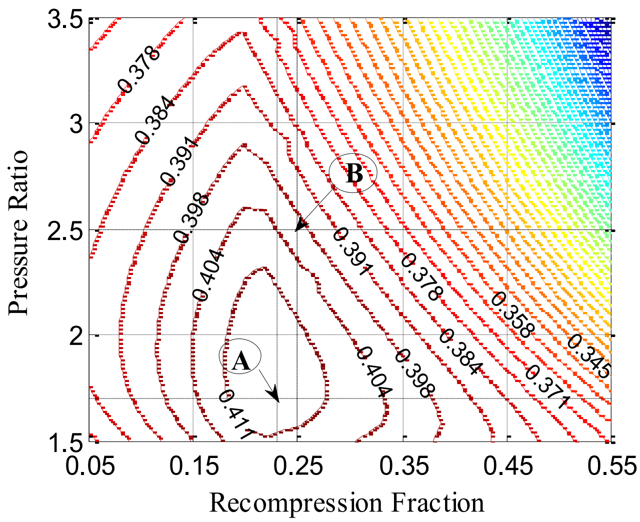


4b: T-source 450°C & T-sink 45°C

Figure 4: Recompression cycle efficiency variation with pressure ratio and recompression fraction for varying source temperature



5a: T-source 565°C & T-sink 32°C



5b: T-source 565°C & T-sink 55°C

Figure 5: Recompression cycle efficiency variation with pressure ratio and recompression fraction for varying sink temperature

There is a unique correspondence of the optimal recompression fraction and optimal pressure ratio with the sink temperature. Such a correspondence is independent of the source temperature. To understand the uniqueness of optimal efficiency, temperature difference of the LT recuperator on the cold fluid approach (T_8-$T_{2'}$) and discharge ends (T_7-$T_{3'}$) are investigated.

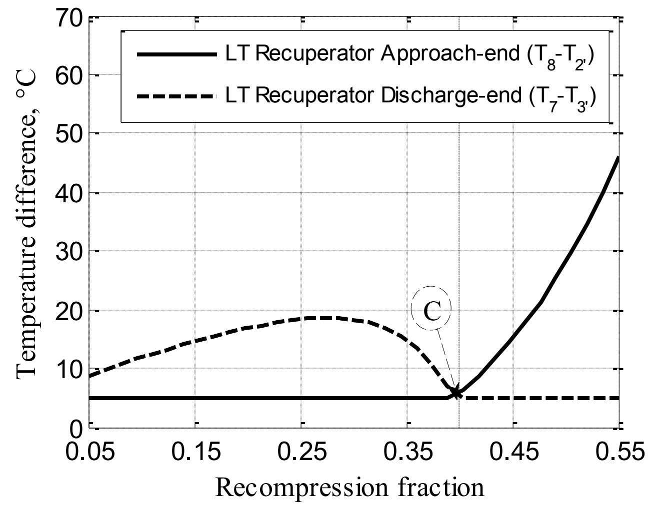


Figure 6: LT recuperator approach & discharge temperature difference for T-source 565°C, T-sink 32°C and optimal pressure ratio 2.7

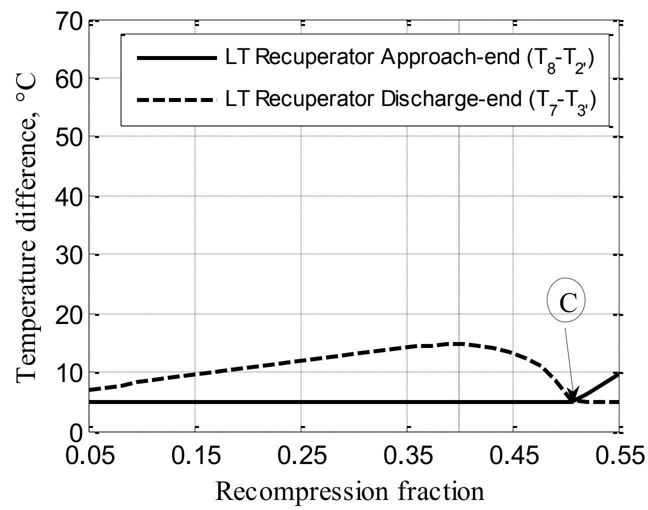


Figure 7: LT recuperator approach & discharge temperature difference for T-source 565°C, T-sink 32°C and pressure ratio 1.7

Point 'C' in Figure 6 is the intersection of the temperature difference on the approach and discharge ends of the LT recuperator. For the variation in recompression fraction from 0.05 to the point 'C', the minimum temperature difference (design set value of 5°C) occurs on the approach end and subsequently shifts to the discharge end as the recompression fraction increases beyond point 'C'. The crossover point 'C' is unique and corresponds to the optimal efficiency point of the recompression cycle (same as point 'A' highlighted in Figure 5a). At the crossover point, temperature differences on either side of the LT recuperator are equal, implying that the temperature-enthalpy behavior of the two streams is globally parallel and likely, though not guaranteed to be, in a configuration of minimum entropy generation. To emphasize the significance of crossover point, the cycle is perturbed away from its optimum by changing the pressure ratio from 2.7 to 1.7. For the recompression fraction of 0.4 and pressure ratio of 1.7, the temperature difference between the two ends is higher as seen in Figure 7. It also shows a new crossover point with a different recompression fraction (0.5), which corresponds to lower overall cycle efficiency. The new point is along the direction of slowest descent in Figure 5a.

From the equipment design perspective, it is important to fix the pressure ratio and recompression fraction. Even though there is a possibility to vary the recompression fraction during operation, it will necessitate overdesigning the LT recuperator and recompressor. The following analysis details the methodology to arrive at a design point pressure ratio and recompression fraction. A performance comparison with the simple recuperated cycle is also provided.

As shown earlier, the sink temperature change has a major influence on the optimal pressure ratio of recompression cycle. The optimal pressure ratio varies between 1.7 and 2.7 (~60% change) for sink temperature change between 32°C and 55°C. However, the optimal pressure ratio of a simple recuperated cycle varies between 1.7 and 5.5 (~320% change) for the same change in sink temperature, as shown in Figure 8. The lowest sink temperature (32°C) has the highest optimal pressure ratio in both the cycles. Moreover, at 32°C sink temperature, the recompression cycle optimal pressure ratio of 2.7 is significantly less than the simple recuperated cycle optimal pressure ratio of 5.5.

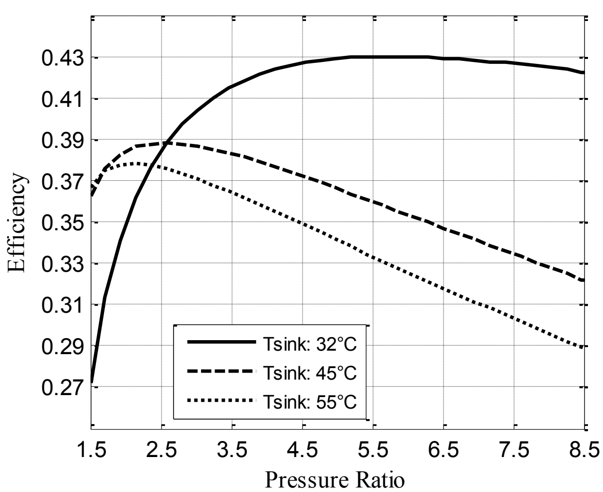


Figure 8: Simple recuperated cycle efficiency variation with pressure ratio for varying T-sink and fixed T-source 565°C

Variation of recompression cycle efficiency with recompression fraction for fixed pressure ratios of 2 and 2.5 are shown in Figures 9 & 10 respectively. The selection of an optimum design point is based on two aspects; a) the highest drop in the absolute value of the efficiency at the operating point (for given sink temperature), b) the maximum difference in the efficiency between the highest and lowest sink temperatures.

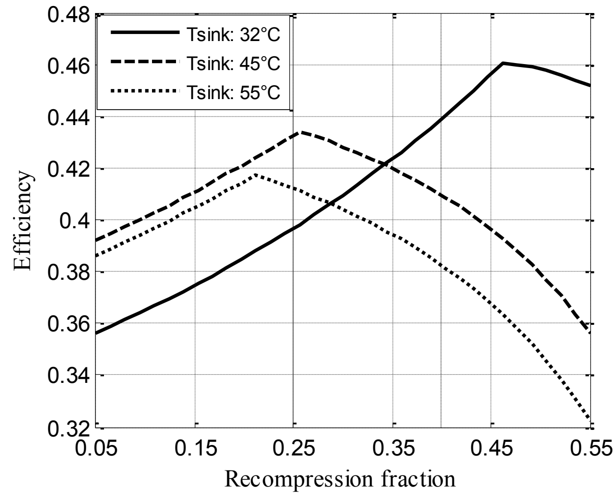


Figure 9: Recompression cycle efficiency variation with T-sink at pressure ratio 2 and T-source 565°C

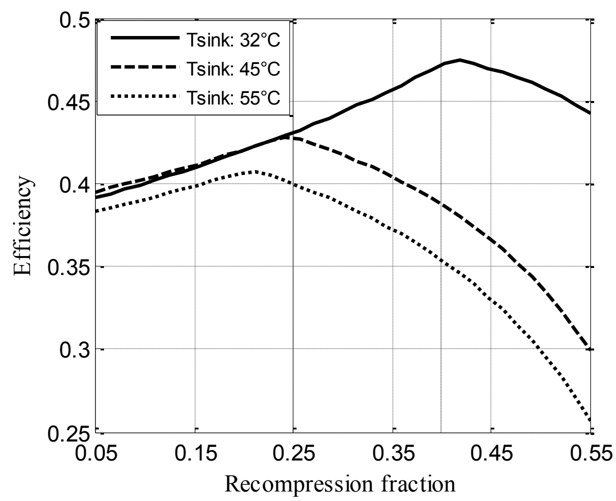


Figure 10: Recompression cycle efficiency variation with T-sink at pressure ratio 2.5 and T-source 565°C

At a pressure ratio of 2 and recompression fraction of 0.4, shown in Figure 9, the previously referred efficiency difference values are 4% and 6% respectively. Similarly, for a pressure ratio of 2 and recompression fraction of 0.25, in Figure 9, the values are 7% and 3%. For a pressure ratio of 2.5 and recompression fraction of 0.4, in Figure 10, it is 6% and 12%. Finally, for a pressure ratio of 2.5 and recompression fraction of 0.25, in Figure 10, it is 4% and 3%. Hence, a pressure ratio of 2.5 and recompression fraction of 0.25 is an optimum design point because there is very little drop in the efficiency at different sink temperatures. This design point is marked 'B' in Figures 4 & 5.

OFF-DESIGN ANALYSIS

Following the understanding wherein, sink temperature influences the optimum efficiency and subsequent design point selection, it is prudent to study the off-design performance for varying sink temperature. This section summarizes the off-design performance results using the methodology explained earlier. As mentioned earlier, off-design component efficiencies and pressure drop in the recuperators are accounted in the cycle off-design analysis. Only pressure drop contributes to the difference in design point efficiency in the 'design space exploration' section and this section. To quantify the difference in design point cycle efficiencies, pressure ratio of 2.5, with a recompression fraction of 0.25, operating at a source temperature of 565°C and sink temperature of 45°C are considered. The design-point cycle efficiency in the off-design analysis as shown in Figure 12a is 39% while in the 'design space exploration' section it was found to be 42.5% as revealed in Figure 10. The design-point cycle efficiency reduction in the range of 3% to 5% is due to the consideration of 3.5% pressure drop in the recuperators.

Figure 11 shows HT and LT recuperator behavior in off-design conditions. The minimum temperature difference of the LT recuperator shifts from approach to discharge side as the sink temperature varies from 32°C to 55°C, corroborating the earlier results from the design exploration exercise summarized in Figures 6 & 7. Note that the minimum temperature difference restriction of 5°C is removed for this off-design analysis. In the following graphs, the design point refers to a pressure ratio of 2.5.

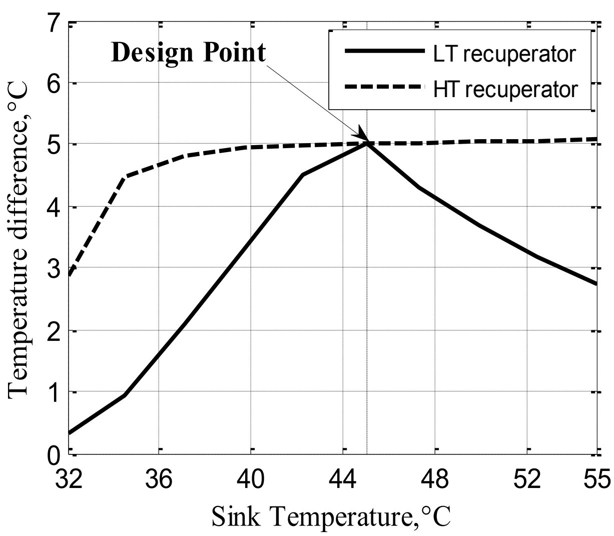
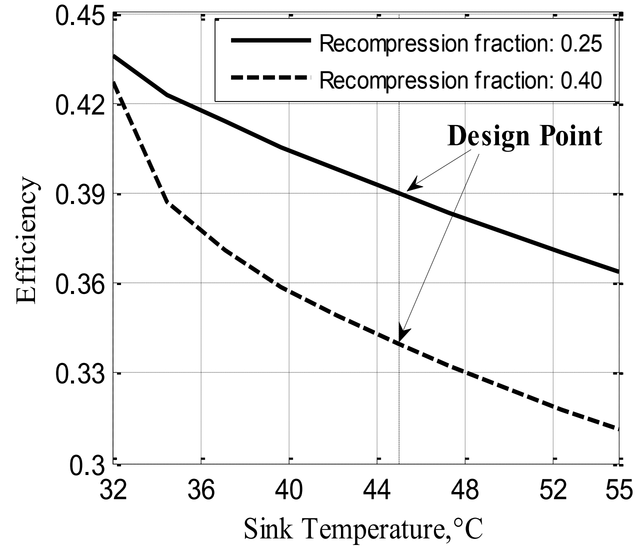
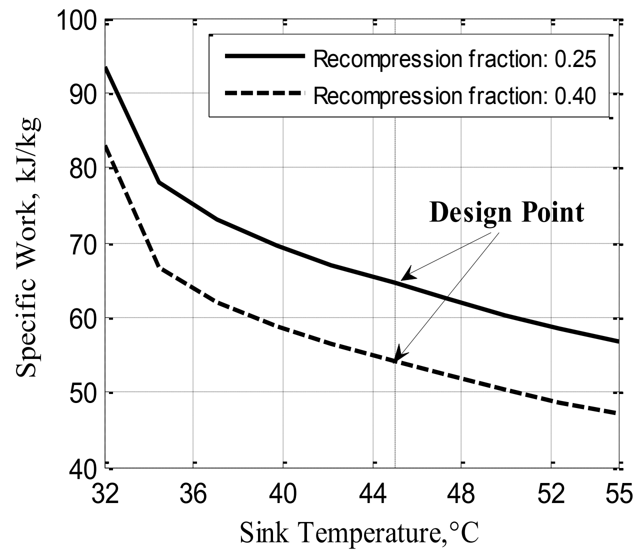


Figure 11: Recuperator minimum temperature difference variation under off-design conditions for T-source 565°C



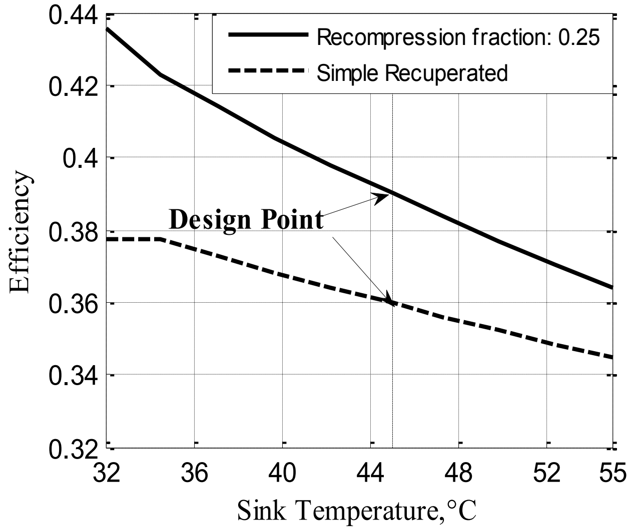
12a: Efficiency for T-source 565°C



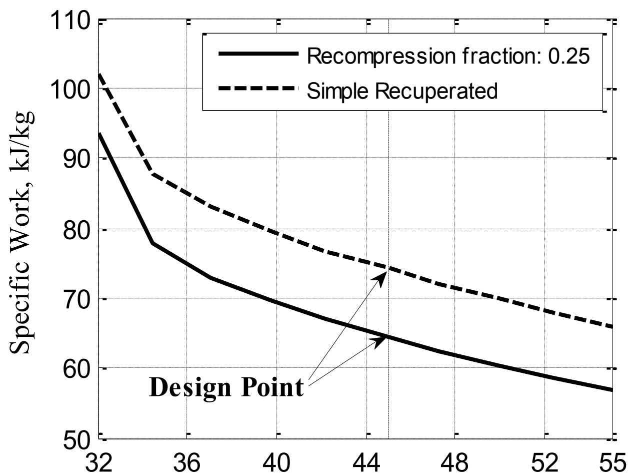
12b: Specific work for T-source 565°C

Figure 12: Off-design performance comparison of cycles with recompression fraction 0.4 and 0.25

Both design and off-design efficiencies are high along with less variability for the recompression fraction of 0.25 when compared to the recompression fraction of 0.4. This is shown in Figure 12. Even specific work is higher for recompression fraction 0.25 as there is a reduction in recompressor work due to reduced flow. The efficiencies are also higher for the recompression cycle with recompression fraction 0.25 when compared with a simple recuperated cycle shown in Figure 13. A sharp reduction in the temperature difference of the LT recuperator at lower sink temperatures contributes to higher variability in recompression cycle efficiency than simple recuperated cycle. The specific work of a simple recuperated cycle is higher than the recompression cycle, which follows from the fact that the recompression cycle has an additional compressor.



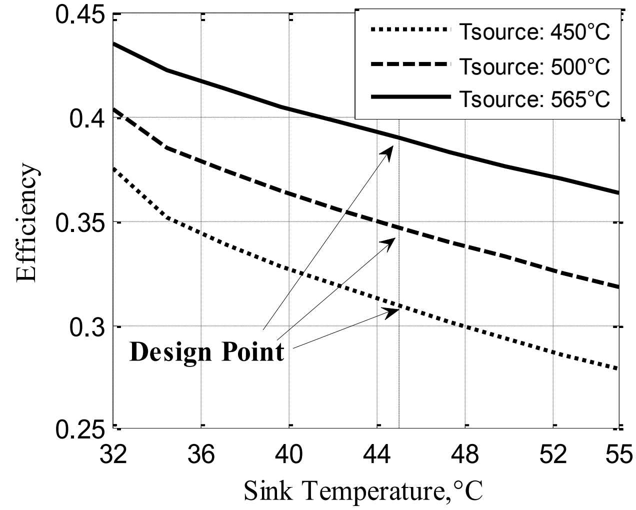
13a: Efficiency for T-source 565°C



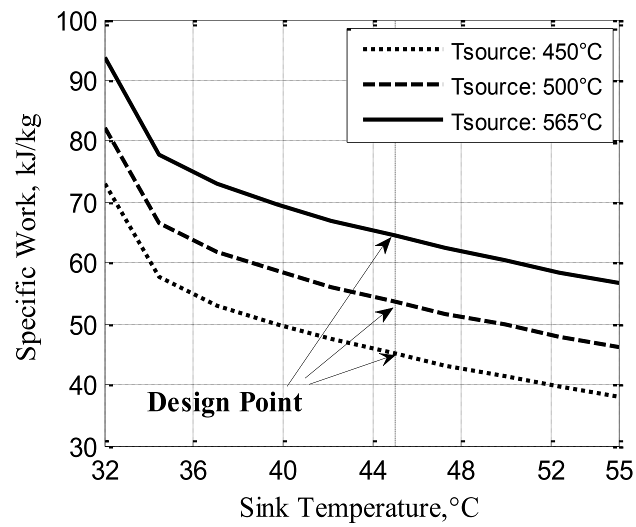
13b: Specific work for T-source 565°C

Figure 13: Off-design performance comparison of recompression and simple recuperated cycles

The absolute value of efficiency and specific work output vary with the source temperature as shown in Figure 14, but their trend across the sink temperature is similar. This again validates the observation made earlier that optimal operation of a recompression cycle is only sink temperature dependent.



14a: Efficiency at recompression fraction 0.25



14b: Specific work at recompression fraction 0.25

Figure 14: Recompression cycle off-design performance at different source temperatures

CONCLUSIONS

The present analysis on the design space exploration for a recompression sCO_2 Brayton cycle establishes optimum pressure ratio of 2.5 and recompression fraction of 0.25 compared to the recompression fraction of 0.4 reported in the literature. This design point minimizes the variation in efficiencies for the range of sink temperatures varying between 32°C to 55°C, yet maintaining an efficiency advantage over a simple recuperated cycle. Analysis shows that the optimum design point is independent of the source temperature. The crossover point, with respect to the hot and cold streams of the LT recuperator, provides an insight on entropy minimization for a recompression cycle, a key contribution of the present work.

Although the off-design analysis accounting for the pressure drop in the recuperators corroborates the inferences drawn from design space exploration which does not consider the pressure drop, further investigation is needed to confirm if recuperator pressure drop would significantly alter conclusions drawn from the present analysis.

In conclusion, the methodology for design space exploration presented in the paper enables a framework for designing recompression Brayton cycles with practical constraints on source temperature and variability in sink temperatures.

ACKNOWLEDGMENTS

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