Analysis of Heat Exchanger Pinch Point in a Waste-Heat Recovery SCO2 Rankine Cycle

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Abstract — *It has recently been recognized that a large quantity of waste heat is generated annually, and thus represents a large opportunity for energy savings. With burgeoning research in cycles which utilize super critical carbon dioxide as a unique organic working fluid. This paper focuses on the analysis of a heat exchanger for use in conjunction with a Supercritical CO2 (SCO2) Rankine low grade waste-heat energy recovery cycle. The NTU-ε method is used to design the heat exchanger. Porous media flow is used for heat transfer augmentation on the tube side of the heat exchanger. Second law analysis has been used to place a practical limitation on the results of pinch point versus heat exchanger area published in this study.*

Keywords — *Supercritical CO2, Heat Exchanger, Exergy, Renewable Energy, Porous Media Flow.*

I. INTRODUCTION

This paper presents results for the heat exchanger sizing for use in a low grade waste to energy conversion process using a Supercritical CO2 (SCO2) Rankine Cycle. SCO2 is an organic refrigerant. The studies of [1,2] present a thermodynamic SCO2 cycle analysis for waste heat recovery from low temperature (200°C to 500°C) sources using small mass flow rates (20 to 60 gm/sec). Other researchers have focused on heat exchanger design including the fundamental based study of [3], which deals with the heat transfer ins heat exchanger comprised on two cylindrical tubes.The work of [4] present the findings of a numerical study on the air side performance of a fin and tube heat exchanger configuration for a variety of different fin configurations. The work of [5] gives the results of a Computational Fluid Dynamics (CFD) based analysis of an off-the-shelf heat exchanger. Thus, the study of heat exchangers is seen to remain proliferate in today's engineering heat transfer literature. Preliminary modelling from [1] suggests that SCO2 systems are viable for low temperature waste heat recovery applications. Pinch-points in range of 3 to 10 °C in the heat exchanger analysis were reported in [1], and were shown to drive the overall size of the primary heat exchanger. The Rankine cycle considered in [1] is shown in Figure 1.

The heat exchanger of interest in the present paper is shown in Figure 1 with state-points labelled as high side: input=2, exit=3, high-side: input=5 and exit=6.

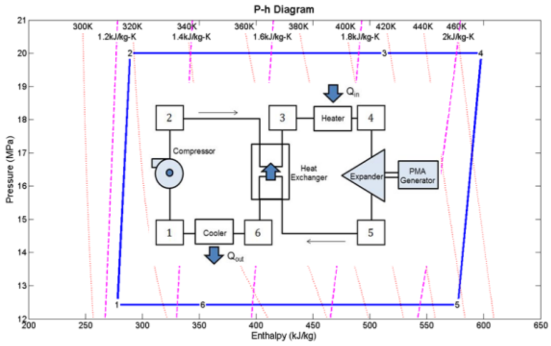

Fig. 1 Schematic of SCO2 Cycle on Mollier Diagram [1]

The current bottleneck in the commercialization of SCO2 based hardware lies in the fact that a large pinch point exists across the heat exchanger. F or a given size of heat transfer area the pinch point approaches as zero as the area becomes very large, thus making the devices very large and bulky. The concept of pinch point in a counter-flow heat exchanger is illustrated in Figure 2. Also shown in Figure 2 is the concept of the approach point which also occurs in process industry heat exchangers. The pinch point approaches zero only for an infinitely large sized heat exchanger area.

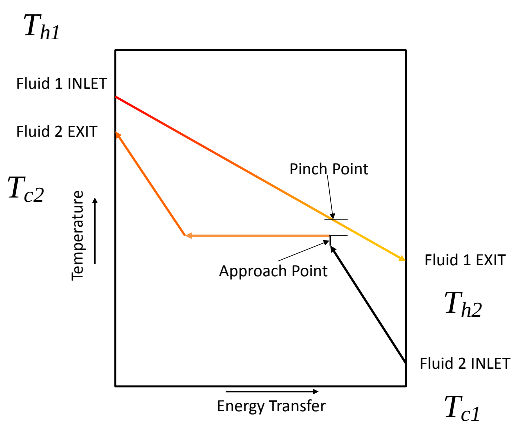

Fig. 2 Pinch point in a counter-flow heat exchanger

This is of course not realizable, and in the quest for minimizing the size of the equipment, thus the present investigation is carried out in order to determine the heat exchanger area versus the pinch point.

II. Heat Exchanger Analysis

The analysis herein is based on the Number of Transfer Units- Effectiveness, NTU-ε method of heat exchanger sizing [6]. The analysis of the heat exchanger was carried out using Mathcad and NIST REFPROP [8]. Recall, use of the Log-Mean-Temperature-Difference (LMTD) method is convenient when the fluid inlet temperatures are known, and the outlet temperatures are known or prescribed, otherwise, the NTU-ε method is preferred to avoid trial and error of using the LMTD method [6]. Referring to Figure 2 the First Law of Thermodynamics for heat transfer from the hot fluid to the cold fluid reads

$$Q = C_h\left(T_{h1} - T_{h2}\right) = C_c\left(T_{c2} - T_{c1}\right) \qquad (1)$$

Where the capacity rate is defined as the product of the flow rate and the specific heat as follows

$$C = \dot{m}c_p \qquad (2)$$

In the NTU-ε method:

$$Q = \varepsilon C_{min}\left(T_{h1} - T_{c1}\right) \qquad (3)$$

where $C_{min} = min\left(C_c, C_h\right)$ is the minimum of the two capacity rates. The Number of Transfer Units is defined as follows:

$$NTU \equiv \frac{UA}{C_{min}} \qquad (4)$$

while the effectiveness is a metric of the efficiency of the heat exchanger and is given as a functional relationship below:

$$\varepsilon = f\left(NTU, C_R = \frac{C_{min}}{C_{max}}\right) \qquad (5)$$

Various handbook formulations and charts of the functional relationship given in Eqn. (5) are available in the heat transfer literature ([6],[7],[9]-[11]). In the present analysis, the effectiveness is computed from the results for a given set of parameters. For this exercise, a shell and tube heat exchanger was considered. The heat exchanger sizing was carried out using Mathcad and NIST REFPROP in the range of heat recovery (aka lift) of 1 < Q < 5 kW for 20 < $\dot{m}$ < 60 grams/sec flow rate of SCO2. The analysis herein assumes state-points 2 and 5 of Figure 1 are fixed. The overall heat transfer coefficient, U (W/m^2-K) of the heat exchanger was developed using the Dittus-Boelter [6] correlation was adopted as the baseline. For external air flow over the tubes of the heat exchanger, the Zhukauskas correlation [6] was adopted as the baseline.

III. Results and Discussion

A. Thermophysical Properties

Figures 4, 5, and 6 show the thermo-physical properties obtained from NIST REFPROP [8] for SCO2 as a function of temperature and pressure. Figure 4, 5, and 6 show the density, enthalpy and entropy of SCO2, respectively.

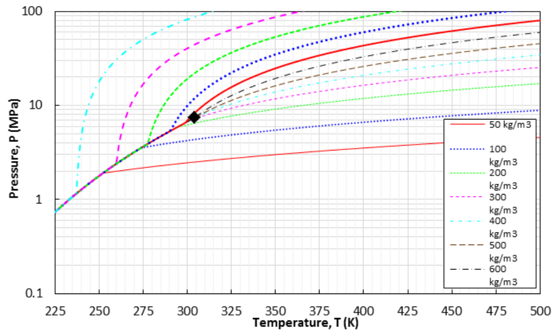

Fig. 4 SCO2 Density

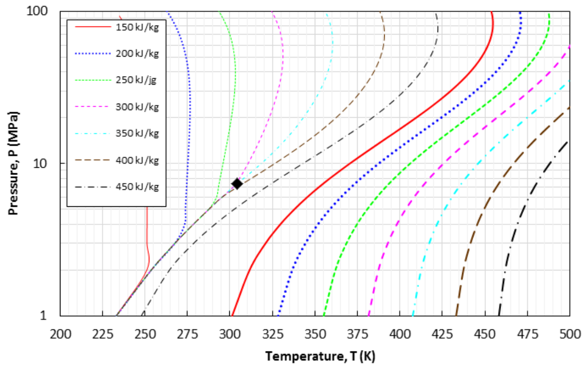

Fig. 5 SCO2 Enthalpy

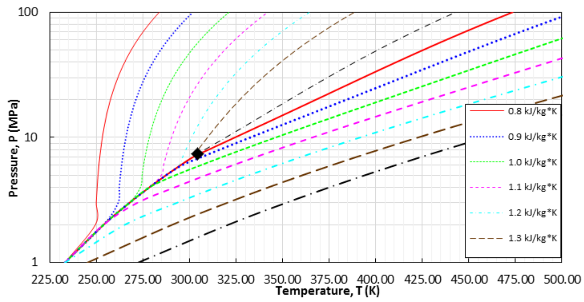

Fig. 6 SCO2 entropy

B. Heat Exchanger Pinch-point Plots

Figures 7 through 11 show the heat exchanger model simulation results for the pinch-point for various values of the flow rate.

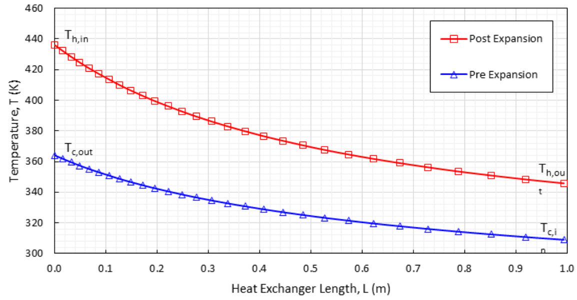

Fig. 7 Temperature profiles in counter-flow heat exchanger for 20 gm/sec flowrate.

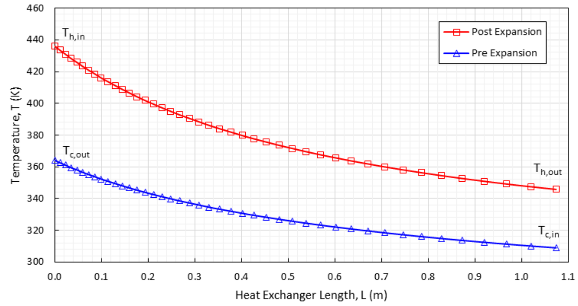

Fig. 8 Temperature profiles in counter-flow heat exchanger for 30 gm/sec flowrate.

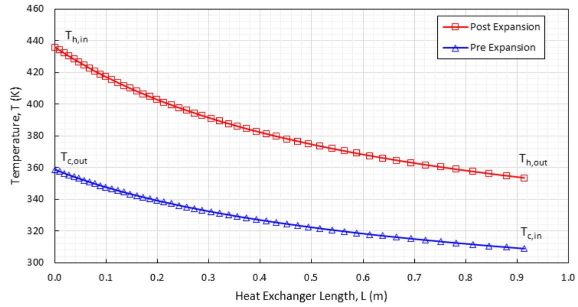

Fig. 9 Temperature profiles in counter-flow heat exchanger for 40 gm/sec flowrate.

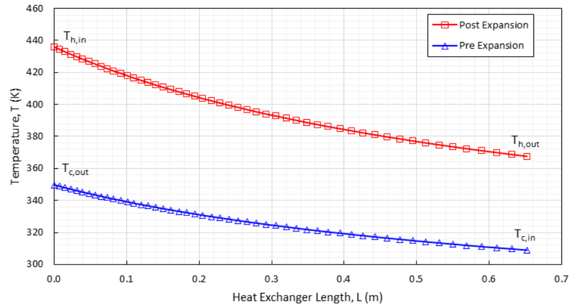

Fig. 10 Temperature profiles in counter-flow heat exchanger for 50 gm/sec flowrate.

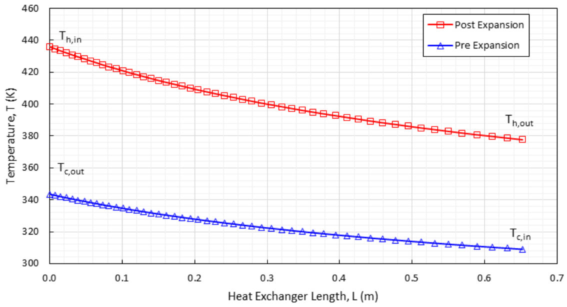

Fig. 11 Temperature profiles in counter-flow heat exchanger for 60 gm/sec flowrate.

Figure 12 summarizes the findings of the pinch-point investigation. Figure 12 plots the pinch point versus the area of the heat exchanger. This analysis has been carried out by taking into account the Second Law of Thermodynamics, so that parametric solutions which violate the second law are not shown in these results. Figure 12 shows that as the pinch point decreases, a larger heat transfer area is regarded, nearly independent of the flow rate.

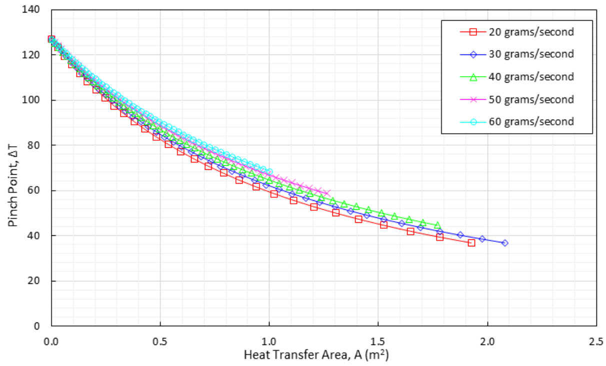

Fig. 12 Heat exchanger Pinch Point vs. Heat Exchanger Area

C. Heat Exchanger Effectiveness

Figure 13 shows the effectiveness versus lift of the heat exchanger. Figure 13 is a plot of the following correlation:

$$\varepsilon = 2\left\{1 + C_R + \left(1 + C_R^2\right)^{1/2} \frac{1 + exp\left[-NTU\left(1 + C_R^2\right)^{1/2}\right]}{1 - exp\left[-NTU\left(1 + C_R^2\right)^{1/2}\right]}\right\}^{-1} \quad (6)$$

From Figure 13, for a given value of the lift, the effectiveness ε increases as the flow rate decreases. Or for a fixed effectiveness, the range of the lift increases as the flow rate increases.

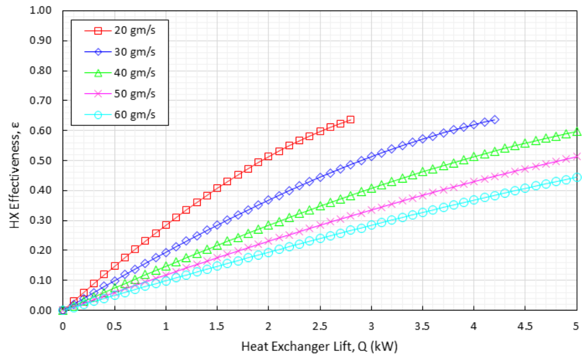

Fig. 13 Heat exchanger Effectiveness vs. Heat Exchanger Lift

D. Porous Media Heat Transfer Augmentation

Porous media cooling was introduced to cool the tube side (hot stream: post expansion fluid at 12.4 MPa) of the heat exchanger. Following [13], porous media (Silica particles) was introduced to the tube side flow for various porosities (ϕ = 0.398, 0.43, and 0.445) where the tube side heat transfer equation was modified with the apparent specific heat

$$C_{p,eff} = \phi C_{p,f} + (1-\phi)C_{p,s} \qquad (7)$$

where $C_{p,f}$ and $C_{p,s}$ denote the mean fluid specific heat, and solid (silica particle) specific heat respectively. Figure 14 shows the effect of adding porous media to the tube side flow is to lower the outlet temperature despite the porosity of the porous media used. Figure 14 illustrates that the use of porous media on the tube side for heat transfer augmentation is to decrease the tube side outlet temperature by approximately 30 °C.

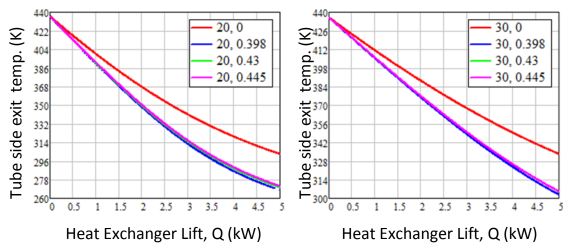

Fig. 14 Tube side outlet temperature vs. heat exchanger lift for (a) 20 gm/s (b) 30 gm/s and various porosities ϕ-0,0.398,0.43, 0.445

Figure 15 summarizes the net effect of adding porous media to augment the heat transfer on the tube side of the heat exchanger.

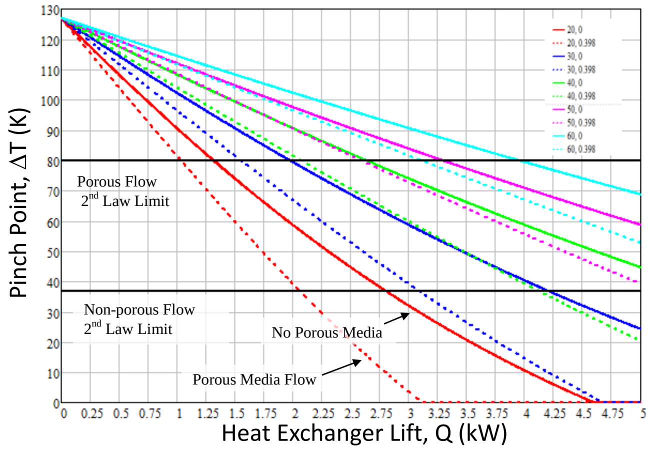

Fig. 15 Pinch Point vs Heat Exchanger Lift for flow with (ϕ = 0.398) and without (ϕ=0) porous media

Figure 15 shows that the apparent specific heat from Eqn. (7) of the porous media is the influencing parameter, since it is proportional to the capacity of the fluid stream. Also, from Figure 15 it is apparent that at a fixed value of lift, the pinch point is decreased by the augmenting the tube side flow with porous media for cooling. For instance, for a lift of 3 kW at a flow rate of 20 gm/s, the pinch point is reduced from 35 K (w/o porous flow on tube sides) to 5 K (with porous flow on tube side) for a delta of 30 K (30 °C).

E. Entropy Generation

Performance parameters such as entropy generation are useful metrics of heat exchanger performance [13]. This section of the analysis and results deals with entropy generation for the SCO2 heat exchanger studies herein. The following is taken from Cengel and Boles [13], for a steady-state system the entropy generation balance is given by:

$$\dot{S}_{in} - \dot{S}_{out} + \dot{S}_{gen} = \Delta\dot{S}_{system} \qquad (8)$$

In terms of flow rates and specific entropies at the inlets and outlets of the heat exchanger:

$$\dot{S}_{gen} = \dot{m}_{SCO2}\left(s_{hot,out} + s_{cold,out} - s_{hot,in} - s_{cold,in}\right) \qquad (9)$$

Figure 16 shows the entropy generation along the heat exchanger. The trends demonstrated in Figure 16 are in qualtitative agreement with those of [14] for a balanced heat exchanger. As the mass flow increases, the pressure drop increases from Bernoulli's eqn. as expected. This explains the trend of the lobes (location of local maxima) in Figures 16 moving from the left to right as the flow rate increases.

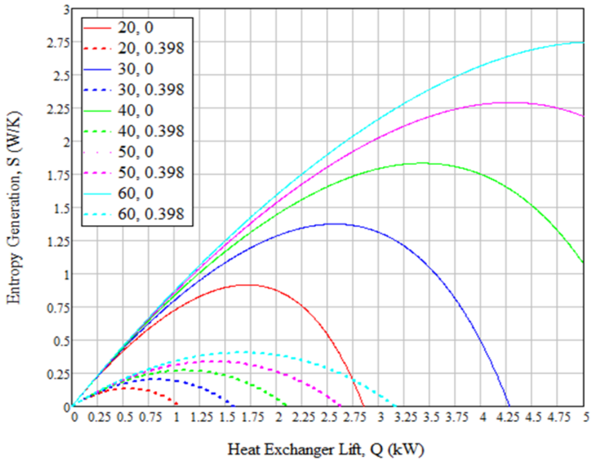

Fig. 16 Steady Flow Entropy Balance of the Heat Exchanger for flow rates of 20, 30, 40, 50 and 60 gm/s and porosities of ϕ=0,0.398

The net effect of adding in porous media to cool the tube side fluid stream is also illustrated in Figure 16. By adding in the porous media flow, the effect is to reduce the heat transfer from the fluid, since heat transfer is proportional to the product of entropy and temperature, this in effect (for a constant temperature) reduces the entropy.

IV. CONCLUSIONS

This paper has presented the thermodynamic and heat transfer modelling and analysis of a heat exchanger for use in a Supercritical CO2 (SCO2) waste heat recovery cycle. Parametric results have been presented regarding the heat exchanger area, effectiveness, mass flow rate and pinch point across the heat exchanger. The use of porous media flow has been used to augment the heat transfer in the tube side of the heat exchanger. Second law analysis has been used to place a practical limitation on the results of pinch point versus heat exchanger area published in this study. Results from this study can be used to guide the development of heat exchanger equipment to make viable the renewable energy applications using the Rankine SCO2 low grade waste-heat recovery cycle outlined in this paper.

ACKNOWLEDGMENT

This research was sponsored by funding provided to the Solar Thermal Alternative Renewable Energy Lab in the Mechanical Engineering Department of California State Polytechnic University at Pomona by Butte Industries, Inc.

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