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LMTD Design Methodology Assessment of Spiral Tube Heat Exchanger under the S-CO_2 cycle operating condition

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1. Introduction

Printed Circuit Heat Exchanger (PCHE) is one of the most widely accepted heat exchangers for the S-CO_2 power cycle application. The advantages of PCHE are (1) compact (2) high pressure difference endurance (3) high temperature operation. However, PCHE is quite expensive and the resistance to the fast thermal cycling is questionable. In order to overcome this problem, the Korea Advanced Institute of Science and Technology (KAIST) research team is considering an alternative for the PCHE. Currently KAIST research team is using a Spiral Tube Heat Exchanger (STHE) of Sentry Equipment Corp. as a precooler in the SCO2PE facility. A STHE is relatively cheap but the operating pressure and temperature are acceptable for utilizing it as a precooler. A STHE is consisted of spiral shaped tubes (hot side i.e. S-CO_2) immersed in a shell (cold side i.e. water). This study is aimed at whether the logarithmic mean temperature difference (LMTD) heat exchanger design methodology is acceptable for designing the S-CO_2 cycle precooler. This is because the LMTD method usually assumes a constant specific heat, but the precooler in the S-CO_2 cycle operates at the nearest point to the critical point where a dramatic change in properties is expected. Experimentally obtained data are compared to the vendor provided technical specification based on the LMTD method. The detailed specifications provided by the vendor are listed in Table 1.

Table 1. Specifications of STHE provided by the vendor

Vessel Shape	Tube	Shell
Vessel Material	316 Stainless Steel	Carbon Steel
Fluid Type	CO_2	Water
Inlet T (℃)	32.06	7
Outlet T (℃)	32	12.3
Mass Flow Rate (kg/s)	2.78	1.052
Heat Load (W)	23409.64	
Heat Transfer Area (m^2)	2.022	
Overall Heat Transfer Coefficient ($W/m^2 \cdot K$)	1656.4	
ΔT_{LMTD} (℃)	22.27	

2. Experimental Data Analysis

2.1 Overall Heat Transfer Coefficient Calculation

To calculate the overall heat transfer coefficient, *U*, from the experimental data, Eq. (1) was used. It was from a logarithmic mean heat transfer equation commonly used in heat exchanger design [1, 2].

$$U = \frac{Q}{AF_G \Delta T_{LMTD}} \tag{1}$$

therein, *Q* is the overall heat transfer rate of the heat exchanger and ΔT_{LMTD} is the logarithmic mean temperature difference (LMTD) for the two fluids. Since inside the heat exchanger, hot side and cold side have a configuration of cross-flow. This means that the geometrical modification factor F_G will be less than unity. Q for hot side was calculated with enthalpy change from each end point state obtained from the NIST property database:

$$Q = \dot{m}\left(h_{hot,in} - h_{hot,out}\right) \tag{2}$$

$$\Delta T_{LMTD} = \frac{\left(T_{hot,in} - T_{cold,out}\right) - \left(T_{hot,out} - T_{cold,in}\right)}{\ln[\left(T_{hot,in} - T_{cold,out}\right)/\left(T_{hot,out} - T_{cold,in}\right)]} \tag{3}$$

2.2 Geometric Factor Calculation

When a *U* is calculated from the experimental data, the dimensionless geometric factor F_G is needed. F_G is not explicitly provided by the vendor, but from the given technical specifications F_G can be obtained. This was done by using Eq. (1) and inserting the provided information from the vendor. The calculated F_G value is 0.314. All of the experimentally obtained *U* values were based on the calculated F_G value.

3. Results and Discussion

3.1 Comparison of Experimental U to Designed U

The overall heat transfer coefficients (*U*) were calculated from several data points which were extracted and averaged from the raw data when the experimental condition reached the steady state.

However, there were big differences between experimental and designed *U* s, as shown in Fig 1. Experimental data was grouped by the ratio of experiment to design mass flow rate of water, since overall heat transfer coefficient is a function of both hot side and cold side mass flow rate.

U values show a growing trend in proportion to the increasing mass flow rate of both CO_2 and water as we expected. Even though the mass flow rate of both S-CO_2 and water were smaller than the design point vendor provided by 25% and 80%, respectively, surprisingly experimentally obtained *U* values were 51.32% higher than the vendor provided *U*. This means

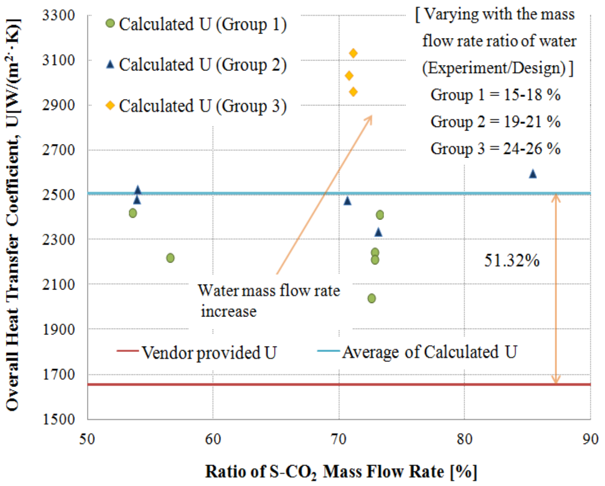

Fig 1. Calculated *U* with mass flow rate

that the LMTD method used by the vendor can be very conservative and it also implicates that the STHE can become more compact when appropriate heat exchanger design methodology is developed.

The KAIST research team thinks the reason why significant difference may exist between the experimentally measured *U* and the vendor provided *U* is because of using the LMTD method for the heat exchanger design. This will be more apparent from the following section.

3.2 Constant-pressure Specific Heat

All the extracted experimental data were in the supercritical phase region because CO_2 has the critical point at 73.8bar and 30.98℃. The constant-pressure specific heat, C_p, of CO_2 at experimental pressure and temperature from the inlet to the outlet of heat exchanger, compared to the design condition provided by the vendor, is plotted in Fig 2. It shows a maximum C_p change from the inlet to the outlet within the experimental data set and the corresponding percentage values too.

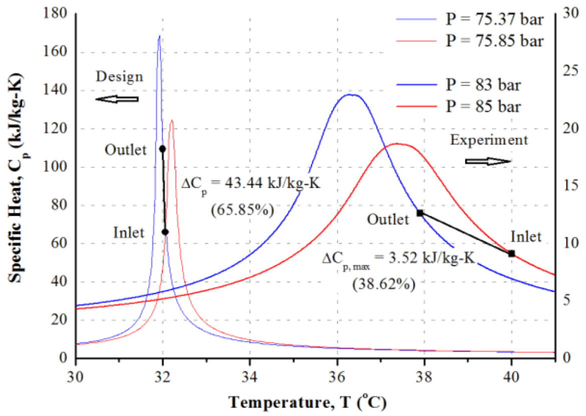

Fig 2. Change of specific heat with temperature

In Fig 2, it is clearly shown that the C_p is not held at constant for both experimental condition and the vendor provided condition. Moreover, this shows that specifications provided by the vendor are not in constant C_p region and the variation of C_p is quite large. This means that the basic underlying assumption of LMTD method is not adequate to design a heat exchanger operating near the critical point.

4. Summary and Further Works

The overall heat transfer coefficients of a STHE were evaluated from an experiment. However, the experimentally measured overall heat transfer coefficient values were significantly different from the vendor provided values. The KAIST research team thinks the reason to be inadequacy of the logarithmic mean temperature difference heat exchanger design method for a precooler in the S-CO_2 cycle due to dramatic variation of the properties near the critical point.

So far, the design methodology for PCHE is well established for the precooler in the S-CO_2 cycle. However, if the S-CO_2 cycle is to be commercialized and realized in an engineering scale, other heat exchanger options are needed. Our study clearly shows that the general heat exchanger design methodology is not suitable for non PCHE type heat exchangers and this clearly means that the development of general heat exchanger design methodology is essential to evaluate different types of heat exchangers under the S-CO_2 cycle condition. Moreover, at least in the precooler more accurate design methodology for other type of heat exchanger may result in smaller heat exchanger for this purpose. This hypothesis is based on the fact that even at lower mass flow rate of both hot and cold side the experimentally measured overall heat transfer coefficient was 50% larger than the designed value. Thus, the KAIST research team will be focused more on the development of heat exchanger design methodology for non-PCHE type heat exchangers to suggest better alternative for the cycle to enhance economy. In the near future, more experiment will be done to reach the design condition to compare the designed overall heat transfer coefficient to the experimentally measured value more accurately.

ACKNOWLEDGEMENT

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