The 4th International Symposium - Supercritical CO2 Power Cycles
September 9-10, 2014, Pittsburgh, Pennsylvania

COMPARISON OF MEASURED AND ANALYTICAL PERFORMANCE OF SHELL-AND-TUBE HEAT EXCHANGERS COOLING AND HEATING SUPERCRITICAL CARBON DIOXIDE

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ABSTRACT

As interest in the use of supercritical carbon dioxide (S-CO2) in power cycles increases, it is important to determine if commonly used design and analysis techniques are applicable to S-CO2. The thermophysical properties of S-CO2 can vary greatly, especially near the pseudo-critical point. This can result in large property variations through process equipment such as heat exchangers, particularly at low temperatures. Bechtel Marine Propulsion Corporation (BMPC) is currently operating a 100 kW closed Brayton test loop using S-CO2 as the working fluid. The Integrated Systems Test (IST) loop contains two shell-and-tube heat exchangers. The waste heat exchanger, or precooler, consists of two units with the S-CO2 on the shell side and chilled water counter flowing through the tubes. The Intermediate Heat Exchanger (IHX), which serves as the heat source, has the S-CO2 flowing though the tubes and a heated mineral oil heat transfer fluid on the shell side. These heat exchangers were modeled with Xist® shell-and-tube heat exchanger design software from the Heat Transfer Research Institute (HTRI). Heat transfer data from IST operations are compared to model predictions and conclusions are made as to the applicability of this tool for S-CO2 analysis.

BACKGROUND

Commercial interest in S-CO2 power cycles is increasing, which in turn will lead to increased demand for hardware designed specifically for use with this working fluid. One particular area of interest for S-CO2 power cycle development is heat exchangers. S-CO2 presents a challenge in that the thermophysical properties can vary greatly, particularly near the critical point. It is of interest to determine if standard design and analysis techniques are applicable to S-CO2.

Bechtel Marine Propulsion Corporation (BMPC) is currently operating a 100 kW closed Brayton test loop using S-CO2 as the working fluid. A schematic of the IST is shown in Figure 1. The Integrated Systems Test (IST) loop contains two shell-and-tube heat exchangers. The waste heat exchanger, or precooler, consists of two units with the S-CO2 on the shell side and chilled water counter flowing through the tubes. The Intermediate Heat Exchanger (IHX), which serves as the heat source, has the S-CO2 flowing though the tubes and a heated mineral oil heat transfer fluid on the shell side.

While shell-and-tube heat exchangers have a power density that is low for S-CO2 applications, they have been studied extensively and many analysis techniques for them exist. Therefore, they offer an opportunity to use mature design and analysis tools to model the heat exchangers and compare the results of the model predictions with experimental data.

DISCUSSION

The IST heat exchangers were modeled with Xist® shell-and-tube heat exchanger design software from the Heat Transfer Research Institute (HTRI). The geometry for the IHX model was developed from the manufacturer's drawings and rating specification sheet. The properties of the MultiTherm PG-1 heat transfer fluid was entered into the software using the user specified grid option. Xist® comes with the VMGThermo suite of fluid properties. BMPC also has a license for the NIST REFPROP property package, which Xist® can also utilize for fluid properties. To explore the difference between the two fluid property packages, the IHX test data will be compared to model predictions using both fluid property data sets.

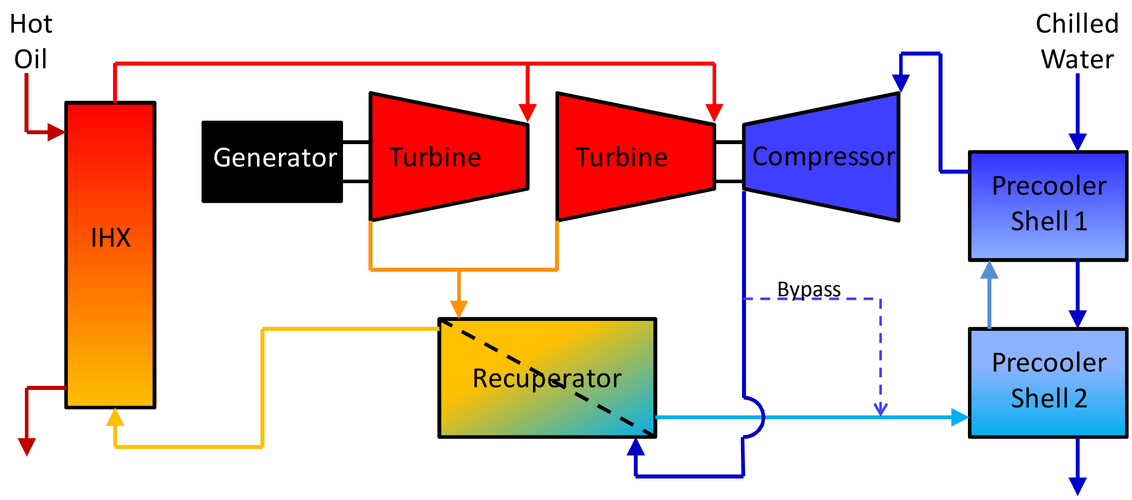

Figure 1. Schematic of the IST S-CO2 Brayton Test Loop.

For the IHX analysis, Xist® was run in simulation mode using the program default calculation options. The inlet S-CO2 and heat transfer oil temperatures were prescribed from the test data. It was found that this configuration provided the most reliable convergence for the IHX model. Key results for comparison to test data were predicted for overall heat transfer and pressure drop on the S-CO2 (tube) side.

The precooler consists of two identical shells in series. Xist® has the ability to solve shells in series and, in fact, had better convergence for the series configuration than for the individual shells for the model options chosen. The VMGThermo properties package was used for the water (tube) side properties.

With the precooler case, the model solution was only able to converge using the REFPROP fluid property package for S-CO2. It is not certain whether this was due to the widely varying fluid properties at lower temperatures and pressures, or if the appropriate software options were not configured. Again, Xist® was run in simulation mode for this analysis. However, in this case the inlet water temperature and outlet S-CO2 temperature were prescribed. Not only did this provide the best convergence, but also most accurately represents how the precooler is operated.

The IST is well instrumented to obtain heat transfer and pressure loss information. For the IHX, S-CO2 pressure is measured in the inlet piping of the tube side, and the pressure differential is measured directly across the heat exchanger. Similarly, the pressure is measured upstream of the shell side for the oil and the pressure differential is measure directly. All absolute pressure and differential pressure transducers are Rosemount 3051S series with a measurement accuracy ±0.025% of span. Temperatures are measured in the inlet and outlet piping of both fluids using Type T Special Limits of Error (SLE) thermocouples with a published accuracy of ±0.4% of the reading.

Oil flow is measured directly with an Endress and Hauser Promass 83F coriolis mass flow meter, with an accuracy of ±0.1% of the reading. However, the mass flow meter in the oil system develops an erroneous reading above 450°F. Flow above this temperature is regulated by controlling the pump speed to a frequency predicted by the pump performance curves. In this region mass flow is controlled within ±1 lbm/s. So, for a desired flow rate of 50 lbm/s this represents a ±2% uncertainty.

The S-CO2 flow in the IHX can be determined by summing the mass flows across the two turbines downstream of the heat exchanger or by subtracting bypass flow from the total loop flow upstream of the compressor. However, each of these methods will be affected slightly by the leakage of S-CO2 through the turbomachinery shaft seals that are between the flow meters and the IHX. The leakage is calculated as the difference between the total loop flow upstream of the compressor and the sum of the turbine and bypass flows. It is then assumed that 1/3 of the leakage occurs at each shaft seal, so that the IHX flow is the sum of the turbine flows plus 2/3 of the leakage. The mass flow at each location is measured with MicroMotion Elite series coriolis mass flow meters with an accuracy of ±0.35%. This estimated leakage flow distribution could be off by as much as 25% of the actual flow distribution, but this error is small because the total leakage flow is less than 5% of the IHX mass flow.

In the precooler, mass flows of both the water and S-CO2 are measured directly with MicroMotion coriolis mass flow meters. Water system pressure is measured upstream of the precooler and differential pressure is measured across the first shell and across both shells, with the differential across the second shell being the difference of the two. Water temperature is measured upstream and downstream of each shell using standard Type K thermocouples (±1.8°F).

Absolute pressure of the S-CO2 is measured at the same end of the precooler, which is the downstream side of the shells. Differential pressures are measured between this point and the entrance to each shell. In addition, each shell of the precooler has pressure and temperature taps located at the midpoint for intermediate measurements. As with the IHX, the precooler S-CO2 thermocouples are Type T SLE.

RESULTS

The primary focus of this paper is on the ability of the analysis software to predict the behavior of the supercritical carbon dioxide, so the discussion of the results will be concentrated on the S-CO2 analysis. Data from the IST was obtained for four steady-state operating conditions, which are provided in Table 1. The range of this data spans the full range of IST operations to-date.

Table 1. Steady-state Operating Input Conditions for Heat Exchanger Modeling Cases

	IHX				Precooler (Series)			
Case	$\dot{m}$ S-CO2 (lbm/s)	T_{in} S-CO2 (°F)	$\dot{m}$ Oil (lbm/s)	T_{in} Oil (°F)	$\dot{m}$ S-CO2 (lbm/s)	T_{out} S-CO2 (°F)	$\dot{m}$ Water (lbm/s)	T_{in} Water (°F)
Cold Idle	3.5	129.2	16.7	176.5	5.6	101.2	6.9	89.5
300°F Hold	4.8	201.6	16.7	299.5	7.2	101.0	9.5	89.5
Hot Idle	4.8	429.3	50	571.3	7.6	96.8	7.1	81.5
Full Power	8.9	361.6	50	548.0	11.2	97.1	15.8	81.0

Figure 2 shows the predicted heat duty from Xist® compared with the calculated test data. The dashed line on the graph is included as a reference and represents the case where the predicted heat transfer matches the measured heat transfer. For the test data, heat duty was calculated using the tube side S-CO2 data as:

$$Q = \dot{m}(h_{out} - h_{in}) \quad (1)$$

At high temperatures and pressures the variability of S-CO2 properties is less than at low temperatures. Also, on the tube side the losses to ambient are negligible. The specific enthalpy of the S-CO2 is calculated with REFPROP using the measured temperature and pressure at the inlet and outlet. Uncertainty in the calculated enthalpy is determined using Taylor's Error Propagation Method (Equation 2) and combined with the flow meter uncertainty using the same technique.

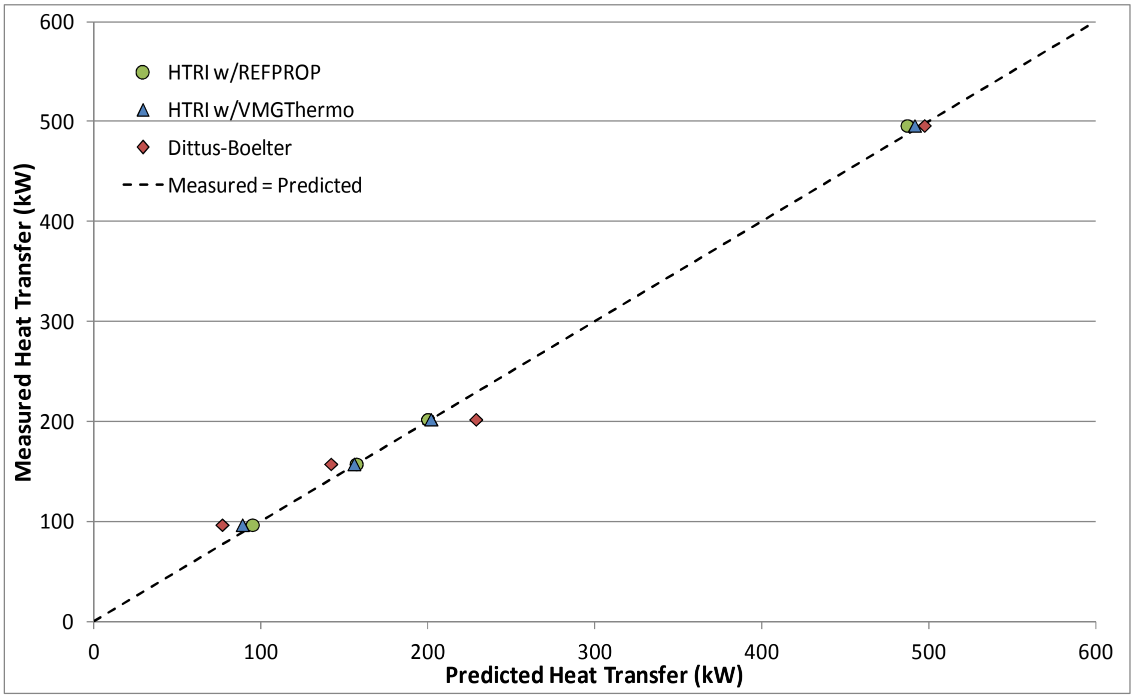

Figure 2. Measured versus Predicted Tube Side S-CO2 Heat Transfer Rates for the IHX.

$$U_{h,calc} = \pm\left(\left(\frac{\partial h}{\partial T}\right)^2 (U_T)^2 + \left(\frac{\partial h}{\partial P}\right)^2 (U_P)^2\right)^{1/2} \tag{2}$$

The data show that Xist® does a good job at predicting the heat duty of the heat exchanger. There is no significant difference between the REFPROP and VMGThermo predictions (1.5% and 2.8% average difference, respectively) which are both of the same magnitude as the uncertainty of the data (2.3%). Overlaid on the graph are the predicted heat duties for the tube side S-CO2 using the classic Dittus-Boelter equation for comparison of the data with a traditional analysis method.

$$Nu = 0.023Re^{0.8}Pr^{0.4} \tag{3}$$

To account for the error introduced by using averaged S-CO2 properties, the calculated heat transfer was normalized to the data for the full power case to enable a comparison of the trends. The predicted heat transfer trend compares well with the data, but the average error for this method was 11% even after normalization, showing that an averaged approach for S-CO2 is much less desirable than a nodalized method.

Another method commonly used to analyze heat exchangers is the log-mean temperature difference (Equation 4). While it is clearly understood that this method is only applicable to fluids with constant properties throughout the heat exchanger, it is still frequently applied.

$$Q = UA\Delta T_{LM} \tag{4}$$

Often the grouped term UA will be determined for a given heat exchanger at a known heat duty and temperature profile. This will then be used to predict the performance of the heat exchanger for another set of operating conditions. To demonstrate the inaccuracy of using this approach with S-CO2, UA was calculated for the IHX and precooler treating each of the heat exchangers as a single node. The UA for each heat exchanger was calculated using both the cold idle condition and full power condition as the baseline, and was then used to predict the performance of the heat exchangers at the other test conditions. Table 2 shows the resulting difference between the calculated heat transfer rates and the measured heat transfer rates using the single-node LMTD method. The predicted heat transfer rates had an average error of 43% when compared with the measured heat transfer.

Table 2. Difference Between Single Node LMTD Method Predicted Heat Transfer and Measured Test Data using both Cold Idle and Full Power Cases as Baselines.

	IHX		Precooler (Series)	
Case	**Cold Idle** *UA*	**Full Power** *UA*	**Cold Idle** *UA*	**Full Power** *UA*
Cold Idle	-----	77.3%	-----	73.3%
300°F Hold	3.8%	83.9%	-29.0%	23.1%
Hot Idle	-35.4%	14.6%	-23.0%	33.5%
Full Power	77.5%	-----	-42.3%	-----

The measured pressure drop is compared with the predicted pressure drop calculated with the REFPROP fluid properties in Figure 3. Model results with VMGThermo properties are presented on the graph and show similar results to the REFPROP predictions. Both models over predict pressure drop by 12% at high flow rates (increased DP corresponds with increased S-CO2 mass flow rate), and under predict pressure drop by 47% at low flow rates. For comparison, a friction factor analysis (Equation 5) was performed for the tube-side S-CO2 flow.

$$\Delta p = 4\frac{L}{D_h}\left(\frac{1}{2}\rho V^2\right)c_f \tag{5}$$

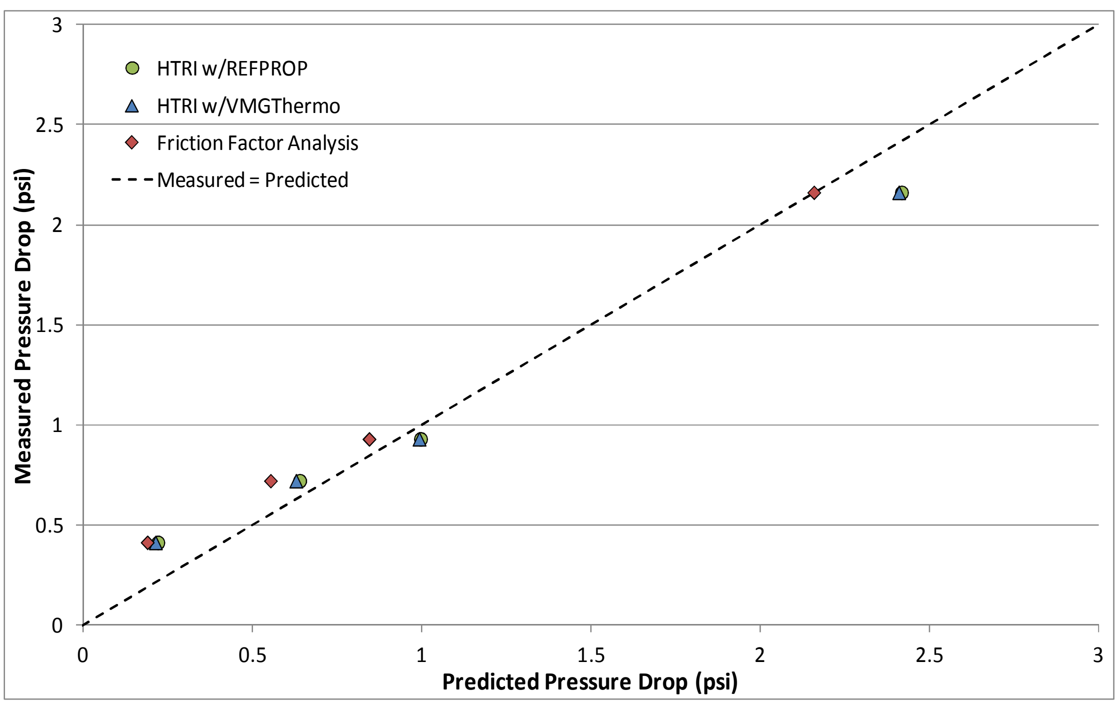

Figure 3. S-CO2 Tube Side Pressure Drop in the IHX.

Since the tube roughness is unknown, the friction factor was calculated from the data for the full power case as 0.0104, which corresponds to a relative roughness of 0.01 at the average tube Reynolds number of 118,000. The friction factor method also under predicts the pressure drop at low flow rates, but better follows the trend of the data than the model results.

The heat transfer rate in the precooler was also calculated using the tube side fluid properties with Equation (1), which in this case was the cooling water. Results presented in Figure 4 show that the model slightly under predicts the measured heat transfer at higher heat transfer rates, and over predicts the heat transfer at the lowest measured heat transfer rates, with all predicted values falling withing the uncertainty of the data. Higher heat transfer rates coincide with higher temperature differences across the heat exchanger and lower cooling water temperature, which could create larger property variations in the S-CO2. The average difference between model prediction and measured heat transfer of the precooler in series was 4%. Of this difference, the heat transfer in Shell 1 (S-CO2 out, chilled water in) was under predicted by 9% and the duty in Shell 2 was over predicted by an average of 6%.

Uncertainty in enthalpy was again calculated using Equation 2. The uncertainty of the heat transfer rate calculated from the water side test data for the precooler in series was between 9 and 17 percent. This is larger than the uncertainty in the heat transfer rate for the IHX due to the higher variability of the fluid properties at lower temperatures. Also, the accuracy of the Type K thermocouples in the chilled water system is less than the Type T Special Limits of Error thermocouples used in the S-CO2 and oil systems. The calculated uncertainty of the heat transfer rate using the S-CO2 data was between 2.5 and 10 percent. This is lower than the uncertainty calculated using the water side data, but the water side heat transfer rate was still used because the S-CO2 heat transfer also includes the heat losses to ambient and is consistently less than the predicted heat transfer.

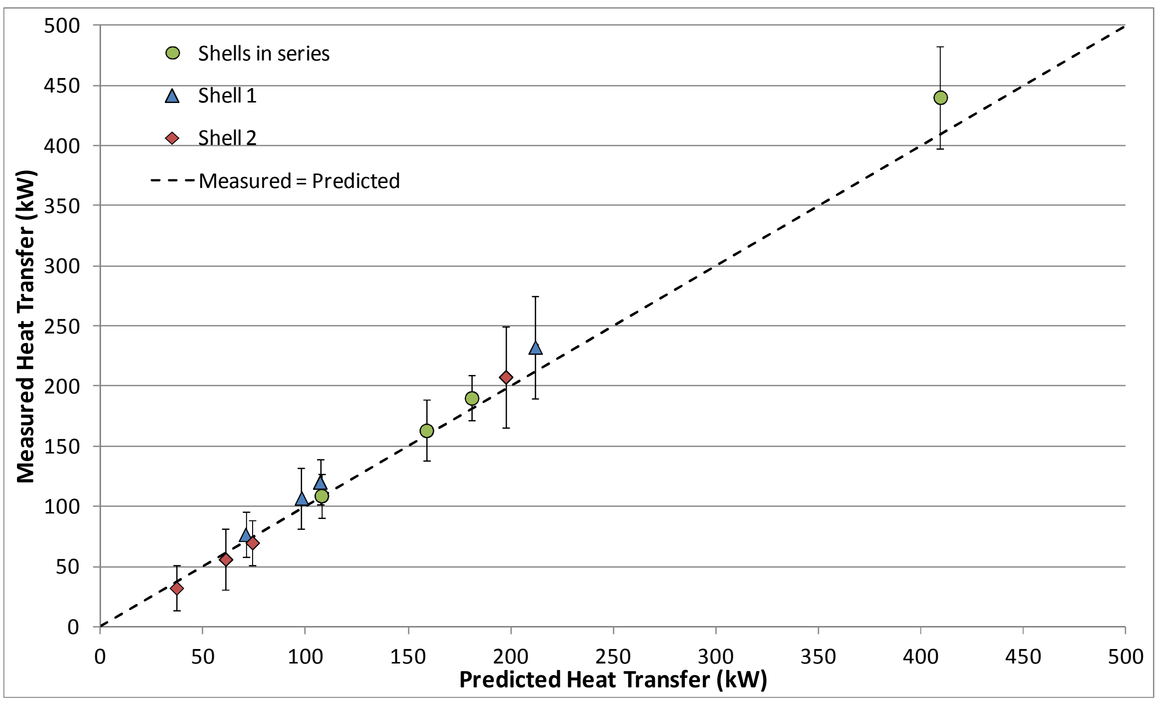

Figure 4. Comparison of Measured Heat Transfer Rates to HTRI Model Prediction for the Shell Side (S-CO2) of the IST Precooler.

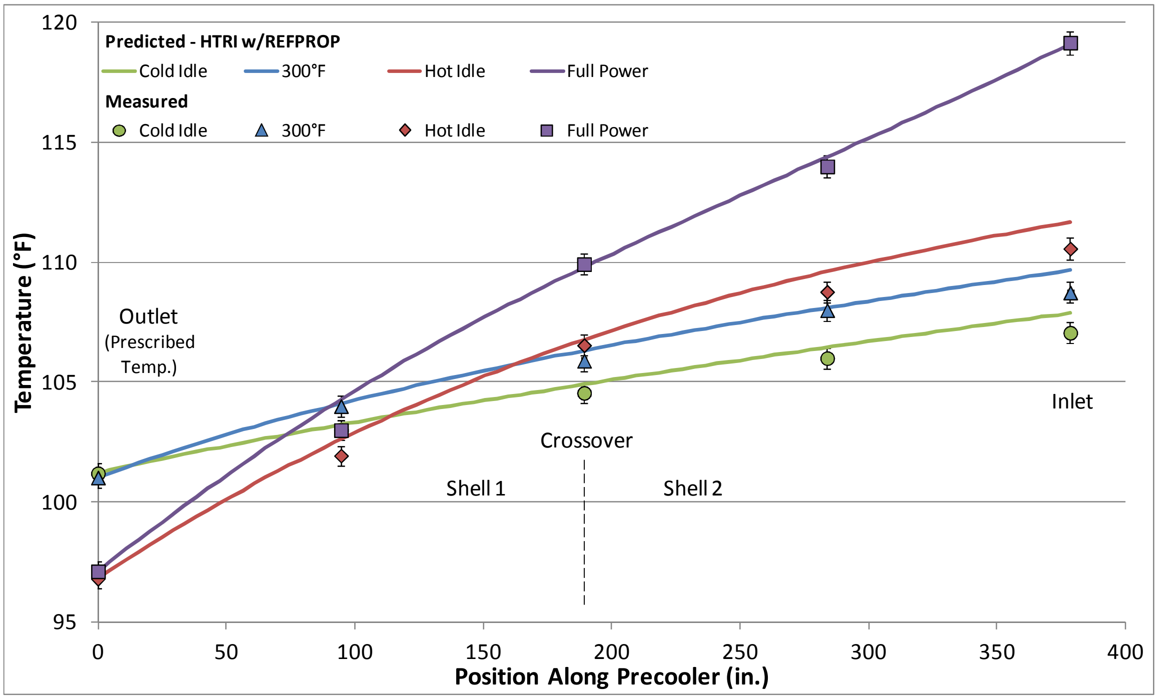

Figure 5. Precooler Shell-side S-CO2 Temperature Profiles

The calculated S-CO2 temperature profiles through the precooler in series are provided in Figure 5. Test data included on the graph show that the model does a good job at predicting the temperature of the incoming S-CO2 (recalling that the exit temperature was used as input to the simulation). The average difference in measured versus predicted inlet temperature is 1°F.

The model predicted a larger pressure loss through the precooler than the measured data (Figure 6). Individually, the model over predicted the pressure drop in Shell 2 by almost 20%, and under predicted the Shell 1 pressure drop by 9% resulting in an over prediction of 5% for the precooler series.

CONCLUSIONS

The IST shell and tube heat exchangers were modeled using HTRI's Xist® design software and the results were compared with experimental data. Results showed that the software provided accurate predictions for heat transfer rate and pressure drop on both the shell side and tube side of the heat exchangers. Development of the models showed that convergence can be an issue when using S-CO2, particularly when modeling conditions near the critical point. Methods that use averaged or constant fluid properties to predict the heat transfer and pressure drop of S-CO2 in shell-and-tube heat exchangers will not provide results as accurate as methods that use a nodalized approach.

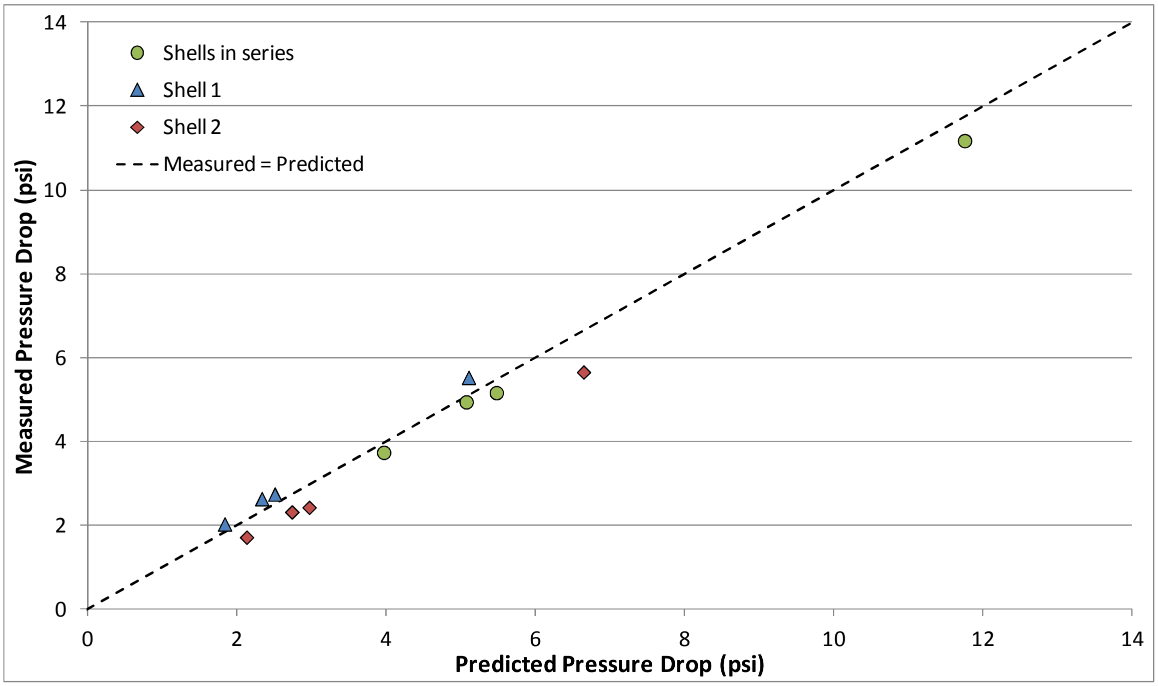

Figure 6. Pressure Drop in the IST Precooler Shell.

NOMENCLATURE

Abbreviations

BMPC	=	Bechtel Marine Propulsion Corporation
HTRI	=	Heat Transfer Research Institute
IHX	=	Intermediate Heat Exchanger
IST	=	Integrated Systems Test
NIST	=	National Institute for Standards and Technology
S-CO2	=	Supercritical Carbon Dioxide
SLE	=	Special Limits of Error

Symbols

Δp	=	Pressure drop, psi
ρ	=	Density, lbm/ft^3
c_f	=	Friction factor
D_h	=	Hydraulic diameter, ft
h	=	Specific enthalpy, BTU/lbm
L	=	Tube length, ft
$\dot{m}$	=	Mass flow rate, lbm/s
Nu	=	Nusselt number
P	=	Pressure, psi
Pr	=	Prandtl Number
Q	=	Heat Transfer Rate, W
Re	=	Reynolds number
T	=	Temperature, °F
U	=	Uncertainty
V	=	Velocity, ft/s

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