Experimental Validation of a Compact Double-Walled Twisted-Tube Heat Exchanger Concept

Reactor Concepts Research Development and Demonstration (RCRD&D)

Edward Blandford
University of New Mexico

Melissa Bates, Federal POC
Chris Grandy, Technical POC
Experimental Validation of a Compact Double-walled Twisted-Tube Heat Exchanger Concept

Edward D. Blandford¹
Amir F. Ali¹
Joel Hughes¹
Maolong Liu¹
Bryan Wallace¹
Denise Chaves¹

&

Matthew D. Carlson²

¹ The University of New Mexico
² Sandia National Laboratory

August 2019
Executive Summary

This report summarizes the results of a 3-year NEUP 15-8667 project sponsored by the U.S. Department of Energy to Experimentally Validate the Compact Double-walled Twisted-Tube Heat Exchanger Concept to support the development of the Fluoride-salt-cooled High-temperature Reactor (FHR). The NEUP activities led by the University of New Mexico (UNM) in collaboration with Sandia National Laboratory (SNL) are split into two major parts, Part-I Simulation for optimized heat exchanger design and Part II Experimental evaluation of the HX performance. The executive summary lists the major findings of the simulation and experimental activities.

PART I - Simulation for Optimized DT-HXR Design

A computational thermal-hydraulic study performed utilizing RELAP5-3D in conjunction with MATLAB to investigate the decay heat removal performance of the DRACS when partial blockages in the primary system and DRACS are considered. The study chose two redundancy schemes, the first nominally requiring two out of three (2/3) DRACS to operate to provide adequate cooling to the reactor and the second requiring three out of six (3/6) operating DRACS to provide cooling. In the study, basic event trees (ETs) and fault trees (FTs) developed for the FHR were used to identify an important potential failure mode of the DRACS: flow channel blockage occurring in either the DRACS loop itself or in the DHX branch (located in the primary system). One important simplification to the model is that the thermosyphon-cooled heat exchanger (TCHX) which normally rejects heat from the DRACS loop in the Mk1 PB-FHR design was modeled as an idealized boundary condition in the simulation, and was assumed to be a simpler natural draft heat exchanger (NDHX) with direct heat rejection to air in the ET and FT analyses. The methodology and results of investigating the performance impact from these scenarios was discussed.

A number of insights resulted from this study. Firstly, the rather high level of robustness of the decay heat removal systems to point blockages was unanticipated and lends support to the concept of DRACS as a robust emergency system in general. In the future, it is recommended that this work be expanded to investigate blockages occurring over a finite length, especially in areas of high risk for freezing (i.e., the salt side of the NDHX or TCHX, whichever is used). Secondly, large behavior differences were noted in the core outlet temperature time histories for blockages occurring in the DHX branches versus the DRACS loops. These differences indicate that PCOT may be insufficient on its own from a risk metric standpoint, especially given that metallic creep will be a transient process dependent on the thermo-mechanical and thermo-hydraulic responses of the system. Additionally, this difference in behavior for blockages occurring in different systems highlighted the importance of the primary system having access to a large amount of thermal inertia. With high levels of redundancy, careful sizing of the systems must be performed to balance the risks associated with near term overheating and mid-to-long term overcooling. This information should prove useful to future decision makers in the design and development of the FHR.

The second computational application presented in this report is a study focused on the economic optimization of a salt-to-salt IHX. The study used metaheuristic search algorithms to rapidly minimize a cost function for an IHX design for both shell-and-tube and twisted-tube concepts. The optimization variables for the shell-and-tube IHX were the outer tube diameter, baffle spacing, and inner shell diameter, while the optimization variables for the twisted-tube IHX were the maximal
outer tube diameter, the modified Froude number (dictates relative tube twist), and the inner shell diameter. The conclusions from the study were that twisted-tube heat exchanger technology likely presents a good technological fit for the challenges of salt-to-salt heat exchangers, and substantial cost savings could be effected by using twisted tubes over conventional shell-and-tube heat exchanger designs, particularly when taking into account operating costs.

The results from these computational studies indicated that twisted-tubes enjoy a significant advantage over traditional shell-and-tube heat exchangers from a cost perspective as well as a length perspective. The study has limitations and could be improved to better deal with design constraints such as matching baffle spacing directly to shell-diameter and investigating multi-pass and multi-shell exchanger designs. Additional experimental data at lower modified Froude numbers and Reynolds numbers and smaller tube diameters could be very useful in determining the limits of twisted-tube performance. From an algorithm perspective, both FFA and CSA performed essentially equivalently, although FFA had a faster run time due to fewer function calls in the implementation.

**PART II – Experimental evaluation of the HX performance**

The HTF, a unique heat transfer facility, was constructed at UNM with the purpose of performing exploratory and validation data collection for different heat exchanger designs. The facility has capabilities for studying natural circulation and bi-directional forced circulation of the simulant fluid Dowtherm A in the primary loop, and forced circulation flow in the secondary loop, while covering a range of Reynolds and Grashof numbers important to heat exchangers in the FHR. Heat transfer data was collected for a bayonet-style twisted and plain tube bundle heat exchanger, focusing on heat transfer performance especially on flow regimes relevant for decay heat removal heat exchangers, but also extending into the lower ranges relevant for exchangers transferring heat to the power conversion cycle or intermediate loop. Data for the buoyancy affected regimes in particular was exploratory, but propagation of errors from the data and parameter inputs is also suitable for certain types of validation studies. As a validation metric, MARE and MaxRE were used to compare the predicted values of the data using correlations with the actual measured data. Agreement for the correlations developed here was generally in the single percentage digits, except for maximum relative errors. Tube-data was observed to follow two regimes: inertial dominated flow correlated to Reynolds number and buoyancy dominated flow correlated to Rayleigh number. The shell-side data was found to correlate well to Reynolds number when the estimated Richardson number was < 1. When the estimated Richardson number exceeded unity, downflow correlated well to simultaneous functions of both Reynolds and Rayleigh numbers. Interestingly and somewhat surprisingly, when downflow Richardson numbers exceeded 30, the data again correlated well to Reynolds number alone. For up-flow, the data correlated well to Reynolds number throughout the full range of estimated Richardson numbers. The measured heat transfer performance of the twisted tube and plain tube bundle HX. These figures show the heat transfer enhancement gained by utilizing the twisted tubes over plain tubes in both natural and forced circulation regimes. For natural circulation, an improvement of 55% increase in the heat transfer performance at Re of 120 and a maximum of 250% increase at the max Re achieved for forced circulation experiment.

The single-assembly double-wall heat exchanger was used to validate a tool for predicting the performance for a three-fluid parallel stream heat exchanger with two thermal communications. The three-fluid heat exchanger tool can be used to understand and predict the temperature
distributions for any parallel stream three fluid heat exchanger. The tool predicted the outlet temperatures of the triple fluids with a maximum uncertainty of 18% in selected cases. The code is as a quick tool for designing a triple flow experiment to optimize the performance-based on the fluids’ properties and experimental conditions. It can be part of a larger scope code for integrated system analysis where the triple flow HX is utilized.

The triple flow HX performance was tested at SNL for SCO2 facility. Dowtherm to SCO2 experiments were partially conducted, and data was collected. Further experiments are required to enable the full analysis of the HX performance for utilizing the triple flow HX in coupling FHR to SCO2 power conversion cycle.
Publications


Joel Hughes, “Heat exchanger challenges and opportunities for the fluoride salt-cooled high-temperature reactor (FHR), Ph.D. dissertation, 2017, University of New Mexico, USA.


Joel Hughes, and Edward D. Blandford,” Compact Double-Wall Twisted-Tube Heat Exchanger for the Fluoride Salt-Cooled High-Temperature Reactor (FHR) with Implications for In-Service Inspection,” ANS winter meeting, November 9–13, 2014, Anaheim, CA


Acknowledgements

We would like to thank (1) the U.S. Department of Energy Office of Nuclear Energy for their support of this project through the Nuclear Energy University Program, (2) Sandia National Laboratory for modifying and utilizing their Super critical Carbon-dioxide (SCO2) loop for conducting experiments.
Project details and objective

Experimental Validation of a Compact Double-walled Twisted-Tube Heat Exchanger Concept

Project Title: Experimental Validation of a Compact Double-walled Twisted-Tube Heat Exchanger Concept

Date of Report: August 31, 2019

Recipient: University of New Mexico
Department of Nuclear Engineering
Albuquerque, New Mexico 87131
Phone: (505) 845-9119, Fax: (505) 844-9010
E-mail: ne@unm.edu

Award Number: NEUP15-8667

Principal Investigator: Edward D. Blandford, edb@unm.edu

Co-PIs: Amir F. Ali, amirali@unm.edu
Matthew D. Carlson, mcarlso@sandia.gov
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21. Schematic diagram of the modified HTF including the single assembly DWHX [4].

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23. The temperature effectiveness of fluid 2 decreases as the flow of water in the annular increases [4].

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PART I - Simulation for Optimized DT-HXR Design

1. Introduction

The Fluoride Salt Cooled High-Temperature Reactor (FHR) is a class of generation IV reactor concepts which the Department of Energy is interested in funding research to support. The FHR generally features a high temperature low pressure fluoride salt as a coolant, an advanced power generation cycle, a tri-structural isotropic (TRISO) fuel particle, and a pool-type passive decay heat removal system. The combination of these systems is seen as a large boost to system safety, reliability, and efficiency. Figure I-1 depicts a proposed FHR thermal fluid scheme. Several thermodynamic cycles are available for power production but, only a few are used in large scale power plants, with the two major cycles being the Rankine cycle, air Brayton, and the S-CO$_2$ Brayton cycle.

![Figure I-1: A proposed design for the Fluoride Salt-Cooled High-Temperature Reactor [1].](image)

Enhanced salt-to-gas heat exchangers (HXs) is a needed technology to ensure the highly efficient energy transfer and hence the energy conversion for coupling the FHR and air or S-CO$_2$ Brayton cycles. The enhanced HX fall into several categories but can be described as exchangers that seek to improve the heat transfer rate beyond what is typically possible using just plain tubes and surfaces. Generally, they seek to increase heat transfer coefficients to reduce either the heat exchanger size, capital cost, or operating cost. The passive enhanced HX that do not require additional power input to employ include modifications to the surface geometries as well as modifications to the working fluids. Geometrical enhancements include modifications to the surface roughness, surface coatings, use of extended surfaces such as fins, addition of different types of displacement inserts, increasing the swirl component of the flow in the system (i.e.,
through twisted-tape inserts or twisted tubes), and increasing secondary flow strength through coiled tubes.

An important example of passively and doubly enhanced exchangers is the twisted tube heat exchanger, which is similar to shell-and-tube heat exchangers but utilizes tubes that have been twisted along their axial length with an elliptical cross-sectional profile. A small section of this type of heat exchanger is illustrated in Figure I-2. As shown, the twisted-tube bundle has an interesting property: it is self-supporting due to regular tube-to-tube contact between each tube and its immediate neighbors along the axial length of the bundle. This is a significant advantage over traditional shell-and-tube heat exchangers, as it allows the bundle to be constructed without the use of baffles and tube support plates. This, along with the integral tube-to-tube contact for much higher bundle stiffness, and the parallel flow along the axial length of the bundle mean that this design is inherently superior to traditional shell-and-tube bundle with regards to flow vibration [2]. In the context of the FHR this is a very important aspect, as previous experience with the shell-and-tube primary heat exchanger in the MSRE indicated that flow-induced vibrations problems will be very important to design against in future concepts [3].

![Image](image.png)

**Figure I-2:** Rendering of a twisted-tube section.

The flow field within a twisted-tube exchanger is more uniform than a shell and tube exchanger due to lack of baffles and crossflow. A uniform flow field has the advantage of improving temperature uniformity between the tubes, helping to protect against tube-to-tube sheet failures and tube buckling (a challenge also for SFRs [4]). It additionally reduces the hot spots that occur at localized stagnation points near baffles, which in turn reduces (although it does not eliminate) the potential for fouling within the exchanger [5]. Different methods for restoring (removing the scale/fouling) for twisted-tubes have been developed, with chemical cleaning-in-place (CIP) being one option with good effectiveness [6]. Finally, the heat transfer performance of twisted-tube bundles is significantly improved over shell-and-tube exchangers, although the manufacturing cost is generally greater [7].

A swirl velocity component is introduced on both the tube side and the shell side, increasing heat transfer coefficients for both working fluids. This is advantageous for designs where enhancement is desired for both streams, with one example being a salt-to-salt Intermediate Heat Exchanger (IHX) in the FHR. Cost and performance data published by [7] has shown for various
process applications that the heat transfer surface area in twisted tube exchanger can be reduced on average by 43% compared to shell-and-tube exchangers and that the per unit area costs of the twisted-tube exchangers are 31% higher. This results in a net capital cost reduction of 26%. The lower required surface area is an additional advantage for space-restricted applications. The FHR incorporate heat exchangers within the reactor vessel, size restrictions will play an important role in their design.

Optimization of the salt-to-S-CO₂ heat exchanger is well underway. The following sections describe simulation work that is ongoing in support of this NEUP. The first section focuses on several aspects of simulated performance of the heat exchanger itself, while the second portion focuses on system-scale simulation work which will better inform how the proposed heat exchanger will interact with the rest of the system and help ensure optimal design.

I2. Heat Exchanger Simulation Work

I2.1 Optimization of salt-to-salt intermediate heat exchanger

Because of the additional complexity associated with modeling of compressible fluids (especially supercritical fluids) and double-wall heat exchangers, the initial step in optimization of the IHX has been to start with a single-wall salt-to-salt heat exchanger. Currently, a study investigating the techno-economic performance of twisted-tube heat exchangers and shell-and-tube heat exchangers is being written up for archival publication. The pre-publication results of this study are provided here in their preliminary form.

A literature review was performed on modern ways of optimizing heat exchangers with an emphasis on metaheuristic algorithms. The use of metaheuristic search algorithms in heat exchanger economic optimization goes back more than a decade. Metaheuristic algorithms are loosely defined as random walk heuristic searches including some situational awareness component in the algorithm to find an optimal solution to multidimensional optimization problems in greatly reduced time. Typically, these algorithms are inspired by some natural phenomenon such as natural selection or behavior of some type of wildlife. One of the most famous examples is the genetic algorithm, which has seen widespread use across scientific fields. While in general metaheuristic searches are not guaranteed to find the global optimum, they have the advantage of being very fast running, and can search a large parameter space and generally arrive very close to the global optimum in a short amount of time [8].

Since early investigations of optimization of shell-and-tube heat exchangers using genetic algorithms, a large body of articles have been published on related metaheuristic algorithms which may improve the optimization search. Some example algorithms used for shell-and-tube heat exchangers include the genetic algorithm[9], particle swarm optimization[10], biogeography based optimization[11], imperialist competitive algorithm[12], cuckoo search algorithm[13], artificial bee colony[14], firefly algorithm[15], global sensitivity analysis and harmony search algorithm[16], gravitational search algorithm[17], bat algorithm[18], and Tsallis differential evolution[19]. Three of these algorithms were selected for this study due to their good performance in the literature: the genetic algorithm (GA), cuckoo search algorithm (CSA), and firefly algorithm (FFA).
A second literature review was performed to determine the availability of heat transfer and friction factor correlations for twisted tube heat exchangers. A large number were found, though it is clear the experimental work covered under this NEUP is required to cover the appropriate parameter ranges and geometries necessary for a finalized double-wall twisted tube heat exchanger design. Table I-1 below summarizes the shell-side heat transfer correlations, Table I-2 the tube-side heat transfer correlations, and Table I-3 the friction factor correlations found in this review. Of note is the lack of heat transfer correlations available for modeling an annulus between a twisted and a straight tube, and the lack of correlations available for small tube pitches at which the heat transfer coefficient is maximized.

Table I-1. Shell-side heat transfer correlations found in the literature review.

<table>
<thead>
<tr>
<th>Shell-side Nu correlations</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Nu_{df} = 6.05 \times 10^6 Fr_{M}^{-2.494 + 0.235 \log(Fr_{M})} \cdot Re_{df}^{n} \cdot a \log(Re_{df}) \left(\frac{T_w}{T_f}\right)^{-0.55} \cdot Pr^{-0.4}$</td>
<td>[20]</td>
</tr>
<tr>
<td>$n = -1.572 Fr_{M}^{-0.1661 - 0.04373 \log(Fr_{M})}$</td>
<td></td>
</tr>
<tr>
<td>$a = 0.269 Fr_{M}^{-0.01490 - 0.01040 \log(Fr_{M})}$</td>
<td></td>
</tr>
<tr>
<td>$63.6 \leq Fr_{M} \leq 1150$</td>
<td></td>
</tr>
<tr>
<td>$2000 \leq Re_{df} \leq 30000$</td>
<td></td>
</tr>
<tr>
<td>$\left(\frac{T_w}{T_f}\right) \leq 1.75$</td>
<td></td>
</tr>
<tr>
<td>$Nu_{df} = 0.023 \cdot Re_{df}^{0.8} \cdot Pr^{0.4} \cdot (1 + 3.6 \cdot Fr_{M}^{-0.357}) \cdot \left(\frac{T_w}{T_f}\right)^{-0.55}$</td>
<td>[21]</td>
</tr>
<tr>
<td>$232 \leq Fr_{M} \leq 2440$</td>
<td></td>
</tr>
<tr>
<td>$2000 \leq Re_{df} \leq 50000$</td>
<td></td>
</tr>
<tr>
<td>$Nu_{df} = 0.0521 \cdot Re_{df}^{0.8} \cdot Pr^{0.4} \cdot \left(\frac{T_w}{T_f}\right)^{-0.55}$</td>
<td>[21]</td>
</tr>
<tr>
<td>$Fr_{M} \sim 64$</td>
<td></td>
</tr>
<tr>
<td>$2000 \leq Re_{df} \leq 50000$</td>
<td></td>
</tr>
<tr>
<td>$Nu_{df} = 83.5 Fr_{M}^{-1.2} \cdot Re_{df}^{n} \cdot Pr^{0.4} \cdot (1 + 3.6 \cdot Fr_{M}^{-0.357}) \cdot \left(\frac{T_w}{T_f}\right)^{-0.55}$</td>
<td>[21]</td>
</tr>
<tr>
<td>$n = \begin{cases} 0.212 Fr_{M}^{0.194}, &amp; Fr_{M} &lt; 924 \ 0.8, &amp; Fr_{M} \geq 924 \end{cases}$</td>
<td></td>
</tr>
<tr>
<td>Transitional</td>
<td></td>
</tr>
<tr>
<td>$Nu = 0.2379 \cdot Re^{0.7602} \cdot Fr_{M}^{-0.4347} (1 + 3.6 Fr_{M}^{-0.357}) \cdot Pr^{0.33}$</td>
<td>[22]</td>
</tr>
<tr>
<td>$231 \leq Fr_{M} \leq 392$</td>
<td></td>
</tr>
<tr>
<td>$2000 \leq Re \leq 10000$</td>
<td></td>
</tr>
</tbody>
</table>

The heat exchanger analysis was performed using the log-mean temperature difference (LMTD) method. This method solves the heat transfer in an exchanger using the basic equation below[23]:

$$Q = UA_o \Delta T_{lm}$$  \hspace{1cm} (1)

where $\Delta T_{lm}$ is the LMTD. Thermal circuit theory dictates that the heat transferred across the series resistances is conserved across each individual thermal resistance:

$$\frac{\Delta T_{lm}}{R_o} = \frac{\Delta T_t}{R_t} = \frac{\Delta T_{f_{out}}}{R_{f_{out}}} = \frac{\Delta T_w}{R_w} = \frac{\Delta T_{f_{outs}}}{R_{f_{outs}}} = \frac{\Delta T_s}{R_s}$$  \hspace{1cm} (2)
and the overall thermal resistance $R_o$ of series resistances is simply the summation of these resistances:

$$R_o = R_t + R_{foul} + R_w + R_{foul_s} + R_s$$  \hspace{1cm} (3)

where $R_t$ is the thermal resistance of the tube-side film, $R_{foul}$ is the thermal resistance of the tube-side fouling, $R_w$ is the thermal resistance of the tube wall, $R_{foul_s}$ is the thermal resistance of the shell-side fouling, and $R_s$ is the thermal resistance of the shell-side film.

In these equations the film thermal resistances $R_t$ and $R_s$ for twisted tube heat exchangers will be determined using the heat transfer coefficient correlations defined in the tables here. In the literature, the thermal resistances due to the tube wall and fouling have usually been neglected, simplifying the analysis. In a real heat exchanger, however, these thermal resistances (especially fouling) can have a major impact on the equipment performance. It was therefore desirable to make some estimate for their values.

Table I-2. Tube-side heat transfer correlations found in the literature review.

<table>
<thead>
<tr>
<th>Tube-side $Nu$ correlations</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>( Nu = 0.021 \left( \frac{S}{A} \right)^{-0.91} \left( \frac{R_e}{T_f} \right)^n )</td>
<td>[24]</td>
</tr>
<tr>
<td>( n = -0.17 - 0.27 \times 10^{-5} \left( \frac{x}{d_e} \right)^{1.37} \left( \frac{S}{d_e} \right)^{2.1} - 109.6 )</td>
<td>( 7000 \leq Re \leq 200000 ) ( 6.2 \leq \left( \frac{T_w}{T_f} \right) \leq 12.2 )</td>
</tr>
<tr>
<td>( Nu = 0.034 \left( \frac{S}{d_e} \right)^{-0.590} \left( \frac{S}{d_e} \right)^{-0.165} )</td>
<td>[25]</td>
</tr>
<tr>
<td>( 5000 \leq Re \leq 20000 ) ( 0.2 \leq s \leq 0.4 ) ( 0.0198 \leq A_i \leq 0.0218 ) ( 0.0058 \leq B_i \leq 0.0094 )</td>
<td></td>
</tr>
<tr>
<td>( Nu = 0.019 \left( \frac{S}{A} \right)^{-0.83} )</td>
<td>[26]</td>
</tr>
<tr>
<td>( 6000 \leq Re \leq 100000 ) ( 6.2 \leq \left( \frac{S}{A} \right) \leq 16.7 )</td>
<td></td>
</tr>
<tr>
<td>( 0.396 \left( \frac{S}{d_e} \right)^{0.161} \left( \frac{S}{A} \right)^{-0.519} \left( \frac{S}{B_i} \right)^{0.161} )</td>
<td>[22]</td>
</tr>
<tr>
<td>( 1000 \leq Re \leq 17000 ) ( 6.86 \leq \left( \frac{S}{A} \right) \leq 11.9 ) ( 0.144 \leq s \leq 0.250 )</td>
<td></td>
</tr>
<tr>
<td>( Nu = 3.66 + 0.512 \left( \frac{B_i}{A_i} \right)^{1.532} \left( \frac{S}{B_i} \right)^{0.609} )</td>
<td>[27]</td>
</tr>
<tr>
<td>( 100 \leq Re \leq 500 ) ( 10 \leq \left( \frac{S}{A_i} \right) \leq 17 )</td>
<td></td>
</tr>
</tbody>
</table>
The thermal resistance of a plain tube is given in most major heat transfer textbooks, and is determined from solving the steady state heat conduction equation with no internal heat generation in cylindrical coordinates [23]:

\[ R_w = \frac{1}{2\pi \lambda_w L} \ln \left( \frac{r_{out}}{r_{in}} \right) \]  

(4)

Similarly, the thermal resistance of a twisted tube was estimated in this analysis by using the conduction shape factor formula for confocal elliptical surfaces (derived in cylindrical elliptical coordinates) [28]:

\[ R_{w,twist} = \frac{1}{2\pi \lambda_w L} \ln \left( \frac{r_{max, out} + r_{min, out}}{r_{max, in} + r_{min, in}} \right) \]  

(5)

In reality, this thermal resistance is an approximation of the idealized case from which it was derived. Specifically, it assumes that the inner and outer surfaces of the twisted tube are isothermal (although a similar assumption is made for plain cylindrical tubes). Secondly, it assumes that the inner and outer profiles of the tube are confocal ellipses. Manufacturers have indicated that the tube thickness is maintained constant in the tube-forming process [29], which violates the confocal condition. However, it is a modest violation and it is noted that conduction shape factors vary only slightly with changes in the geometry [30].

Table I-3. Friction factor correlations found in the literature review.

<table>
<thead>
<tr>
<th>Shell-side f correlations</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>( f = 0.3164 \text{Re}_{df}^{-0.25}(1 + 3.6 \text{Fr}<em>M^{-0.357}) ) ( \text{Re}</em>{df} \geq 800 ) ( \text{Fr}_M \geq 100 )</td>
<td>[31]</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tube-side f correlations</th>
<th>Reference</th>
</tr>
</thead>
<tbody>
<tr>
<td>( f = 0.82 \left( \frac{S}{A} \right)^{-0.63} \text{Re}^{-0.18} ) ( 7000 \leq \text{Re} \leq 200000 ) ( 6.2 \leq \left( \frac{S}{A} \right) \leq 12.2 )</td>
<td>[24]</td>
</tr>
<tr>
<td>( f = 0.316 \left( 1 + 3.27 \left( \frac{S}{A} \right)^{-0.87} \right) \text{Re}^{-0.25} ) ( 6000 \leq \text{Re} \leq 100000 ) ( 6.2 \leq \left( \frac{S}{A} \right) \leq 16.7 )</td>
<td>[26]</td>
</tr>
<tr>
<td>( f = 10^{a_1 + a_2 \log(\text{Re}) + a_3 (\log(\text{Re}))^2} ) ( a_1 = -19.70 + 4.90 \left( \frac{S}{A} \right) - 0.22 \left( \frac{S}{A} \right)^2 ) ( a_2 = 10.52 - 2.66 \left( \frac{S}{A} \right) + 0.12 \left( \frac{S}{A} \right)^2 ) ( a_3 = -1.47 + 0.36 \left( \frac{S}{A} \right) - 0.016 \left( \frac{S}{A} \right)^2 )</td>
<td>[22]</td>
</tr>
</tbody>
</table>
\[ 1000 \leq Re \leq 17000 \]
\[ f = 0.71497 Re^{0.07777} Pr^{-1.03974} \left( \frac{A_i}{B_i} \right)^{-0.076212} \left( \frac{s}{d_e} \right)^{-0.33393} \]
\[ 7900 \leq Re \leq 26500 \]
\[ 0.160 \leq s \leq 0.250 \]
\[ 0.024 \leq A_i \leq 0.026 \]

All available correlations presented in Tables I-1, I-2, and I-3 are collected together and presented in Appendix I-A at the end of the report.

Next, because the nature of the formation and evolution of heat exchanger fouling is extremely complex, a greatly simplified approach was performed for the scoping study by adjusting shell-and-tube fouling factors for the elliptical shape by the following equations:

\[ R_{f\text{oult},twist} = R_{f\text{oult}} \left( \frac{R_{w,twist}}{R_w} \right) \]  
\[ R_{f\text{ouls},twist} = R_{f\text{ouls}} \left( \frac{R_{w,twist}}{R_w} \right) \]  

where,

\[ \frac{R_{w,twist}}{R_w} = \ln \left( \frac{r_{\text{max.out}} + r_{\text{min.out}}}{r_{\text{max.in}} + r_{\text{min.in}}} \right) \]
\[ \frac{r_{\text{out.e}}}{r_{\text{in.e}}} \]

and \( r_{\text{out.e}} \) and \( r_{\text{in.e}} \) are equivalent radii for a cylindrical tube with the same perimeter as a twisted tube.

With this information, it is possible to formulate the solution for the overall heat transfer coefficient for a single-wall twisted tube heat exchanger:

\[ \frac{1}{U} = \frac{1}{h_t} \left( \frac{\Pi_{\text{out}}}{\Pi_{\text{in}}} \right) + R_{f\text{oult},twist} \left( \frac{\Pi_{\text{out}}}{\Pi_{\text{in}}} \right) + \frac{\Pi_{\text{out}}}{2\pi\lambda_w N} \ln \left( \frac{r_{\text{max.out}} + r_{\text{min.out}}}{r_{\text{max.in}} + r_{\text{min.in}}} \right) + R_{f\text{ouls},twist} + \frac{1}{h_s} \]  

where \( \Pi_{\text{out}} \) and \( \Pi_{\text{in}} \) are the outer and inner perimeters of a twisted elliptical tube, respectively.

The analysis presented here focuses on a single wall heat exchanger for initial simplicity and benchmarking of the design optimization procedure but will be expanded for a double-wall twisted tube heat exchanger with inner cylindrical wall. To analyze such a system, the overall thermal resistance equation is easily expanded with additional terms to:

\[ R_o = R_{t,i} + R_{f\text{oult},i} + R_{w,i} + R_{f\text{ouls},i} + R_{s,i} + R_{t,o} + R_{f\text{oult},o} + R_{w,o} + R_{f\text{ouls},o} + R_{s,o} \]  

which includes additional subscripts \( i \) and \( o \) specifying the inner or outer tubes, respectively.

The optimization was performed using published sources of the metaheuristic algorithms (see Figure I-3 for the basic methodology of the design procedure). The GA is included in the MATLAB optimization toolbox [32], while the CSA and FFA algorithms are available for download from the MathWorks website as files authored by the inventor of the algorithms (see [8]).
for print published versions of the MATLAB functions or check MathWorks File Exchange for updated versions). To work, they require a “fitness function” which is the optimization problem posed as a function to minimize. In this analysis, the fitness function is the total present cost $C_{tot}$. The total present cost is a summation of the capital (initial) cost $C_i$ and the total present operating cost of the equipment lifetime $C_{oD}$ [9]:

$$C_{tot} = C_i + C_{oD} \quad (10)$$

The capital cost function has been assumed of the following form [33]:

$$C_i = MF_p F_M F_L C_B \quad (11)$$

where $C_B$ is the base cost of the exchanger, $F_L$ is a correction factor for the length of the heat exchanger, $F_M$ is a correction factor for the materials of construction, $F_p$ is a correction factor for the pressure differential, and $M$ is a correction factor for the fabrication cost of twisted-tube heat exchangers compared to shell-and-tube heat exchangers. A small amount of public data is available on the cost of twisted tube heat exchangers, which indicate a roughly 30% increase in the cost per unit area. For this analysis, $M$ has been assumed as either 1.0 or 1.3 and is labeled appropriately in the results.

The base cost of a carbon steel shell-and-tube heat exchanger with a floating head is [33]:

$$C_B = e^{11.667 - 0.8709 \ln (S_{imp}) + 0.09005 (\ln (S_{imp}))^2} \quad (12)$$

Figure I-3. The basic methodology of the cost optimization procedure for the heat exchangers, adopted from [9].
where $S_{imp}$ is the surface area of the exchanger in square feet. The correction factor for pressure is [33]:

$$F_p = 0.9803 + 0.018\left(\frac{P_{imp}}{100}\right) + 0.0017\left(\frac{P_{imp}}{100}\right)^2$$

(13)

where $P_{imp}$ is the pressure rating in pounds per square inch. The material correction factor for stainless steel tubes and shell is [33]:

$$F_M = 2.70 + \left(\frac{S_{imp}}{100}\right)^{0.07}$$

(14)

The length correction factor is [33]:

$$F_L = \begin{cases} 2.156L_{imp}^{0.6557} + 0.6984, & L_{imp} < 20 \\ 1, & L_{imp} \geq 20 \end{cases}$$

(15)

where $L_{imp}$ is the length of the exchanger in feet. Shorter heat exchangers require a cost estimate increase due to the larger fraction of capital cost of the tube sheets and ends.

To estimate the operating cost of the heat exchanger on a per year basis, the pump efficiency $\eta$, the mass flow rates $G_t$ and $G_s$, the densities $\rho_t$ and $\rho_s$, and the pressure drops $\Delta P_t$ and $\Delta P_s$ must be known. The pumping power can then be estimated with [9]:

$$P = \left(\frac{1}{\eta}\right)\left(\frac{G_t}{\rho_t}\Delta P_t + \frac{G_s}{\rho_s}\Delta P_s\right)$$

(16)

Then, the annual operating cost is estimated as [2]:

$$C_o = P \left(\frac{C_E}{1000}\right) H$$

(17)

and the discounted total present operating cost is estimated as [2]:

$$C_{oD} = \sum_{k=1}^{ny} \frac{C_o}{(1 + adr)^k}$$

(18)

With this information, it is possible to use the metaheuristic search algorithms to optimize the heat exchanger based on input parameters. The default search parameters for each algorithm were used, as shown in Table I-4. As the results will show, this setup was adequate in finding optimal solutions using each algorithm. Three geometric parameters were used to optimize the heat exchangers. For shell-and-tube exchanger, the outer tube diameter $d_o$, the baffle spacing $B$, and the shell diameter $d_{shell}$ were varied. For the twisted tube exchanger, the maximum outer tube diameter $d_{max,out}$, the modified Froude number $Fr_M$, and the shell diameter $d_{shell}$ were varied. For both cases the tube thickness was assumed as 1 mm. The modified Froude number basically accounts for the relative tube pitch and swirl enhancement and is defined as [20]:

$$Fr_M = \frac{s^2}{d_{max,out}d_e}$$

(19)
Table I-4. Parameters chosen for the metaheuristic search algorithms.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Population type</td>
<td>Double vector</td>
</tr>
<tr>
<td>Population size</td>
<td>50</td>
</tr>
<tr>
<td>Initial population</td>
<td>Default</td>
</tr>
<tr>
<td>Initial scores</td>
<td>Default</td>
</tr>
<tr>
<td>Initial range</td>
<td>Default</td>
</tr>
<tr>
<td>Scaling function</td>
<td>Rank</td>
</tr>
<tr>
<td>Selection function</td>
<td>Stochastic uniform</td>
</tr>
<tr>
<td>Elite count</td>
<td>0.05 x population size</td>
</tr>
<tr>
<td>Crossover fraction</td>
<td>0.8</td>
</tr>
<tr>
<td>Mutation function</td>
<td>Constraint dependent</td>
</tr>
<tr>
<td>Migration direction</td>
<td>Forward</td>
</tr>
<tr>
<td>Migration fraction</td>
<td>0.2</td>
</tr>
<tr>
<td>Migration interval</td>
<td>20</td>
</tr>
<tr>
<td>Non-linear constraint algorithm</td>
<td>Augmented Lagrangian</td>
</tr>
<tr>
<td>Initial penalty</td>
<td>10</td>
</tr>
<tr>
<td>Penalty factor</td>
<td>100</td>
</tr>
<tr>
<td>Hybrid function</td>
<td>None</td>
</tr>
<tr>
<td>Maximum generations</td>
<td>300</td>
</tr>
<tr>
<td>Maximum time limit</td>
<td>None</td>
</tr>
<tr>
<td>Fitness limit</td>
<td>None</td>
</tr>
<tr>
<td>Stall generations</td>
<td>50</td>
</tr>
<tr>
<td>Stall time limit</td>
<td>None</td>
</tr>
<tr>
<td>Stall test</td>
<td>Average change</td>
</tr>
<tr>
<td>Function tolerance</td>
<td>1e-6</td>
</tr>
<tr>
<td>Constraint tolerance</td>
<td>1e-3</td>
</tr>
</tbody>
</table>

Genetic algorithm

Cuckoo search algorithm

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of nests</td>
<td>25</td>
</tr>
<tr>
<td>Discovery rate</td>
<td>0.25</td>
</tr>
<tr>
<td>Total iterations</td>
<td>1000</td>
</tr>
</tbody>
</table>

Firefly algorithm

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of fireflies</td>
<td>20</td>
</tr>
<tr>
<td>Number of iterations</td>
<td>500</td>
</tr>
<tr>
<td>Randomness $\alpha$</td>
<td>0.5</td>
</tr>
<tr>
<td>$\beta_{min}$</td>
<td>0.2</td>
</tr>
<tr>
<td>Absorption coefficient $\gamma$</td>
<td>1</td>
</tr>
</tbody>
</table>

Pre-publication results have focused initially on a salt-to-salt IHX for the Mark 1 PB-FHR assuming an intermediate loop filled with flinak. The flow and duty parameters for the heat exchangers are shown in Table I-5, assuming that 2 IHXs would be used for a 236 MWth reactor module (based on information from [34] and [35]).

Table I-5. Required performance parameters of a salt-to-salt IHX.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of IHXs</td>
<td>2</td>
</tr>
<tr>
<td>Shell-side fluid</td>
<td>flibe</td>
</tr>
<tr>
<td>Tube-side fluid</td>
<td>flinak</td>
</tr>
<tr>
<td>Shell-side inlet temp.</td>
<td>700 C</td>
</tr>
<tr>
<td>Shell-side outlet temp.</td>
<td>600 C</td>
</tr>
<tr>
<td>Tube-side inlet temp.</td>
<td>545 C</td>
</tr>
<tr>
<td>Tube-side outlet temp.</td>
<td>690 C</td>
</tr>
<tr>
<td>Shell-side mass flow/rate/IHX</td>
<td>489 kg/s</td>
</tr>
<tr>
<td>Tube-side mass flow/rate/IHX</td>
<td>423 kg/s</td>
</tr>
<tr>
<td>Duty/IHX</td>
<td>118 MW</td>
</tr>
</tbody>
</table>
Figure I-4a shows a plot of the total present cost estimates for shell-and-tube heat exchangers using the three algorithms for clean and fouled heat exchangers, assuming fouling factors of 0.00030 \( m^2C/W \). The results show that all three algorithms find nearly identical solutions for both clean and fouled exchangers. Figure I-4b shows the same comparison but for twisted tubes. It also includes an intermediate case for reduced fouling of 0.00015 \( m^2C/W \). The reduced fouling case was investigated due to literature reports of reduced fouling potential in twisted-tube heat exchangers. Twisted-tube heat exchangers by their nature have more uniform shell-side flow which reduces hot spots and the potential for fouling.

As the three algorithms were roughly equivalent in finding optimal solutions for both the shell-and-tube and twisted-tube heat exchangers, the firefly algorithm was utilized for the rest of the study due to having the fastest run time. Figure I-5 shows results of the firefly algorithm runs for various cases: clean shell-and-tube, clean twisted-tube with equivalent manufacturing cost (\( m = 1.0 \)), clean twisted-tube with 30% higher manufacturing cost (\( m = 1.3 \)), fouled shell-and-tube, fouled twisted-tube with \( m = 1.0, 1.3 \) and the case with reduced fouling for the twisted tube heat exchanger.
The results show that for clean heat exchangers, the use of twisted-tubes can reduce total costs ~34-47% depending on manufacturing costs. For fouled heat exchangers, the use of twisted-tubes can reduce costs ~20-44% depending on manufacturing costs and level of twisted-tube fouling, with potential overall savings on the order of a million dollars per heat exchanger (dollar values have not yet been adjusted to 2016 values). The biggest advantage of twisted tubes seems to be in reduced fouling and/or reduced pressure drop. Lower shell-and-tube pressure drops could be achieved only with unrealistically high baffle spacing (on the order of several times the shell diameter, which is unreasonable for tube support purposes), so it appears that twisted-tube are a good fit for salt heat exchangers.

Table I-6 shows the detailed information on the heat exchanger designs compared in Figure I-5. One of the interesting things to note is that the algorithms sometimes attempt to design unrealistically long heat exchangers, especially for the shell-and-tube design. For example, the fouled shell-and-tube heat exchanger design was > 80 meters long. In this case, the heat exchanger would have to be broken down into smaller exchangers placed in series. Such an arrangement would carry an economic penalty due to the additional tube sheets and exchanger joints required.
Table I-6. Detailed geometry and performance comparison of the different heat exchanger designs simulated.

<table>
<thead>
<tr>
<th>Clean</th>
<th>Twisted-tube</th>
<th>Twisted-tube</th>
<th>Twisted-tube</th>
<th>Reduced fouling</th>
<th>Reduced cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>$d_{max,out}$ (m)</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
</tr>
<tr>
<td>$d_{min,out}$ (m)</td>
<td>0.0074</td>
<td>0.0074</td>
<td>0.0074</td>
<td>0.0074</td>
<td>0.0074</td>
</tr>
<tr>
<td>$D_{max}$ (m)</td>
<td>144.7</td>
<td>144.7</td>
<td>144.7</td>
<td>144.7</td>
<td>144.7</td>
</tr>
<tr>
<td>$D_{min}$ (m)</td>
<td>100.00</td>
<td>100.00</td>
<td>100.00</td>
<td>100.00</td>
<td>100.00</td>
</tr>
<tr>
<td>$F_{T/M}$</td>
<td>0.0967</td>
<td>0.0961</td>
<td>0.0964</td>
<td>0.0963</td>
<td>0.0963</td>
</tr>
<tr>
<td>$P_{t}$ (m)</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
<td>0.0120</td>
</tr>
<tr>
<td>$C_t$ (m)</td>
<td>0.00287</td>
<td>0.00605</td>
<td>0.00605</td>
<td>0.00605</td>
<td>0.00605</td>
</tr>
<tr>
<td>$N$</td>
<td>3326</td>
<td>5839</td>
<td>1078</td>
<td>7447</td>
<td>6913</td>
</tr>
<tr>
<td>$d_t$ (m)</td>
<td>0.0095</td>
<td>0.0069</td>
<td>0.0222</td>
<td>0.0222</td>
<td>0.0222</td>
</tr>
<tr>
<td>$u_t$ (m/s)</td>
<td>0.863</td>
<td>0.816</td>
<td>0.488</td>
<td>0.359</td>
<td>0.689</td>
</tr>
<tr>
<td>$R_e$</td>
<td>3948</td>
<td>2798</td>
<td>3512</td>
<td>2135</td>
<td>2257</td>
</tr>
<tr>
<td>$P_r$</td>
<td>9.44</td>
<td>9.44</td>
<td>9.44</td>
<td>9.44</td>
<td>9.44</td>
</tr>
<tr>
<td>$h_t$ (W/m²°C)</td>
<td>3204</td>
<td>3681</td>
<td>1853</td>
<td>3252</td>
<td>3361</td>
</tr>
<tr>
<td>$x$</td>
<td>0.0416</td>
<td>0.0599</td>
<td>0.0381</td>
<td>0.0881</td>
<td>0.0955</td>
</tr>
<tr>
<td>$R_{P_t}$ (Pa)</td>
<td>75311</td>
<td>75413</td>
<td>35390</td>
<td>103326</td>
<td>88010</td>
</tr>
<tr>
<td>$a_t$ (m²)</td>
<td>0.2682</td>
<td>0.3232</td>
<td>0.2682</td>
<td>0.3232</td>
<td>0.2682</td>
</tr>
<tr>
<td>$d_{max}$ (m)</td>
<td>0.0082</td>
<td>0.0078</td>
<td>0.0172</td>
<td>0.0077</td>
<td>0.0077</td>
</tr>
<tr>
<td>$u_t$ (m/s)</td>
<td>0.851</td>
<td>0.647</td>
<td>0.706</td>
<td>0.514</td>
<td>0.350</td>
</tr>
<tr>
<td>$R_e$</td>
<td>2200</td>
<td>1597</td>
<td>3842</td>
<td>1282</td>
<td>1349</td>
</tr>
<tr>
<td>$P_r$</td>
<td>15.00</td>
<td>15.00</td>
<td>15.00</td>
<td>15.00</td>
<td>15.00</td>
</tr>
<tr>
<td>$h_t$ (W/m²°C)</td>
<td>8195</td>
<td>10665</td>
<td>5285</td>
<td>9504</td>
<td>9822</td>
</tr>
<tr>
<td>$R_{P_t}$ (Pa)</td>
<td>564625</td>
<td>61337</td>
<td>3845</td>
<td>81023</td>
<td>69686</td>
</tr>
<tr>
<td>$a_t$ (m²)</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
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<tr>
<td>$d_{max}$ (m)</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
</tr>
<tr>
<td>$u_t$ (m/s)</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
<td>0.00000</td>
</tr>
<tr>
<td>$R_e$</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
</tr>
<tr>
<td>$P_r$</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
<td>1.03588</td>
</tr>
<tr>
<td>$U$ (W/m²°C)</td>
<td>1758.9</td>
<td>1983.8</td>
<td>680.3</td>
<td>792.6</td>
<td>1188.8</td>
</tr>
<tr>
<td>$S$ (m²)</td>
<td>2559</td>
<td>2269</td>
<td>6615</td>
<td>5678</td>
<td>4022</td>
</tr>
<tr>
<td>$m$</td>
<td>1.0</td>
<td>1.3</td>
<td>1.0</td>
<td>1.3</td>
<td>1.3</td>
</tr>
<tr>
<td>$C_t$ (S)</td>
<td>8.110e + 05</td>
<td>9.360e + 05</td>
<td>2.265e + 06</td>
<td>2.469e + 06</td>
<td>1.865e + 06</td>
</tr>
<tr>
<td>$C_u$ (S/year)</td>
<td>1.470e + 05</td>
<td>3.081e + 04</td>
<td>1.875e + 05</td>
<td>4.163e + 04</td>
<td>3.549e + 04</td>
</tr>
<tr>
<td>$C_{t+u}$ (S)</td>
<td>9.031e + 05</td>
<td>1.893e + 05</td>
<td>1.152e + 06</td>
<td>2.538e + 05</td>
<td>2.181e + 05</td>
</tr>
<tr>
<td>$C_{tot}$ (S)</td>
<td>1.734e + 06</td>
<td>1.126e + 06</td>
<td>3.417e + 06</td>
<td>2.724e + 06</td>
<td>1.903e + 06</td>
</tr>
</tbody>
</table>

The twisted-tube designs were ~20 meters long, which is also quite lengthy but could be fabricated. It may still be desired to break this heat exchanger up into smaller exchangers placed in series for better manufacturability and ease of transportation. Another thing to note is that the twisted-tube design was limited by a lower limit of $d_{max,out} = 12 \text{ mm}$ and $F_{T/M} = 100$. This indicates that perhaps even better savings can be attained by using smaller diameter tubes (though manufacturing limits may apply) and a smaller tube pitch. Additional correlations are required for smaller tube pitches. One additional limitation that should be mentioned is that the algorithms are based on the assumption that the adjustment parameters can be varied continuously. In reality, it is most likely that off-the-shelf tube and shell dimensions would be used to reduce costs.

It is also worth comparing the relative thermal resistance contributions in the different exchanger designs, which is shown in Figure I-6. Interestingly, the relative thermal resistances are similar in both shell-and-tube and twisted-tube designs, which seems to indicate that a good balance for the cost function is achieved at these ratios. For clean heat exchangers the major thermal resistance is the tube-side film, with the tube wall being a small contribution and shell-side film being a little larger. For fouled heat exchangers, a large portion of the thermal resistance is found in the fouling. As seen earlier, this greatly affects the overall cost and size of the heat exchangers.
Figure I-6. A plot of the thermal resistance contributions to the overall thermal resistance in shell-and-tube (S&T) and twisted-tube (TT) heat exchangers, where C = clean \((R_{foul[t,s]} = 0.00000 m^2C/W)\), F = fouled \((R_{foul[t,s]} = 0.00030 m^2C/W)\), LF = less fouled \((R_{foul[t,s]} = 0.00015 m^2C/W)\), and \(m\) = capital cost multiplication factor.

It is also interesting to note the vastly different tube-sheet layouts for the two types of heat exchangers. Because the pressure drop performance in the exchangers are quite different (axial flow in twisted-tube and mix of cross flow and axial flow in baffled shell-and-tube), the optimized layout included relatively few tubes in the shell-and-tube exchanger to maintain reasonable pressure drop in such a long heat exchanger, while the twisted-tube exchanger was able to pack in a very large number of tubes due to the close packing and better pressure drop performance Figures I-7, and I-8.

Because the thermal performance of a double-wall salt-to-SCO\(_2\) is anticipated to be inferior to a single-wall salt-to-salt heat exchanger, some modification to the general design of the heat exchanger may be beneficial. Discussions among the group indicate that a helical tube layout may increase the compactness of the design as well as provide additional heat transfer coefficient enhancement. The concept combines the co-axial plain inner tube and twisted outer tube with helically bended tubes to fit a longer tube inside a shorter heat exchanger. The proposed design should also increase the heat transfer coefficient in all films, which should help to maintain a compact design.
Figure I-7. Tube layout of the fouled shell-and-tube heat exchanger. This design has 1078 tubes with a pitch-to-diameter ratio of 1.25.

Figure I-8. Tube layout of the fouled twisted-tube heat exchanger with \( m = 1.3 \). The sweep of the maximum outer diameter \( d_{\text{max, out}} \) is shown as a circle which touches the six surrounding tubes. This design has 7,447 tubes in a 1.1558 m shell with 6 mm of tube-to-shell clearance.

I3. Simulation of Single Twisted-Tube Assembly

I3.1 Using Heat Exchanger Theory

As double-wall heat exchangers are a primary focus of this NEUP, understanding of the fluids or other contents within the intermediate annulus is of great interest. We are currently working with our twisted-tube heat exchanger vendor HIPEX to procure a single twisted-tube assembly heat exchanger. The purpose for this heat exchanger is to be able to test different fluids or other materials within the annulus. In the context of the FHR, these materials would likely be in the
interest of tritium management. We are estimating the performance of this heat exchanger by applying standard heat exchanger theory.

First principles require the calculation of the convective heat transfer coefficient for each of the fluids. The inner tube is a standard cylindrical tube. However, the intermediate tube is twisted. A literature review was performed in order to find correlations for determining the non-dimensional Reynolds and Nusselt numbers based on inlet temperatures and flow rates. Figure 1-9 shows the flow chart describing the iterative process for matching heat transfer rates between the primary-to-intermediate and intermediate-to-secondary fluids. Tube and shell side correlations can be found in Tables I-1&I-2.

Heat transfer rates can be found between the primary-to-intermediate and intermediate-to-secondary streams using the effectiveness/NTU method. The effectiveness of a heat exchanger can be defined as the ratio of the actual heat transfer rate $q$ to the maximum possible heat transfer rate $q_{max}$ \[^{[36]}\].

$$\varepsilon = \frac{q}{q_{max}}$$ \hspace{1cm} (20)

where

$$q_{max} = C_{min} \times (T_{h,i} - T_{c,i})$$ \hspace{1cm} (21)

and $C_{min}$ is the minimum heat capacity rate either:

$$C_h = \dot{m}_h \times c_{p,h}$$ \hspace{1cm} (22a)

or

$$C_c = \dot{m}_c \times c_{p,c}$$ \hspace{1cm} (22b)

The Number of Transfer Units (NTU) is a function of the overall heat transfer coefficient, surface area, and minimum heat capacity rate.

$$NTU = \frac{U \times A}{C_{min}}$$ \hspace{1cm} (23)

For a counterflow heat exchanger, the effectiveness $\varepsilon$ is related to the NTU by:

$$\varepsilon = \frac{1 - \exp(-NTU \times (1 - C_r))}{1 - C_r \times \exp(-NTU \times (1 - C_r))}$$ \hspace{1cm} (24)

where

$$C_r = \frac{C_{min}}{C_{min}}$$ \hspace{1cm} (25)

From primary-to-intermediate and intermediate-to-secondary, we can find the heat transfer rate by using equation 1 and the fluid outlet temperature by

$$q = \dot{m} \times c_p \times (T_i - T_o)$$ \hspace{1cm} (26)
We will then use the average temperatures to iterate on the fluid thermophysical properties. The temperature of the intermediate fluid will then be iterated on to match the heat transfer rates from the primary to the intermediate and the intermediate to the secondary.

For the single twisted-tube assembly from HIPEX we are intending to use Dowtherm A as our primary fluid, water as the secondary fluid, and the intermediate annulus consisting of fluids such as helium or air, and solids such yttrium hydride.

![Diagram](image)

Figure I-9. Flow chart describing the iterative process for matching heat transfer rates between the primary-to-intermediate and intermediate-to-secondary fluids.

I3.2 Using Computational Fluid Dynamics (CFD)

I3.2.1 Objectives:

The research team at UNM has recently initiated an exploratory CFD study to understand the advantage of utilizing twisted tube heat exchanger concept over plane tube type. The main goal of this study is to simulate single and double walled tubed heat exchanger similar to those used in the experimental course and are optimized as described in this report. The advantage of this CFD study is to explore the heat transfer and fluid flow characteristics of this new type under wide range of testing parameters that could be beyond the experimental capabilities at UNM. The overlap data collected from the experiments will be used to validate the simulation results prior expanding the simulation parameter ranges. As initial step two types of concentric tubes heat exchanger models are built and investigated. The first type uses plane inner tube and the second uses twisted inner tube to enhance heat transfer between the shell and inner fluids.
I3.2.2 Problem statement

COMSOL MultiPhysics is used to simulate the simple concentric tubes heat exchangers. All dimensions and inlet conditions for hot flow and cold flow are determined from expected experimental conditions. The outer tube has a fixed diameter and the inner tubes (plane and twisted) have equal heat transfer surface areas. The overall length of the two models is 5 ft. The shell side has Dowtherm simulant fluid and the inner tube has water. The Dowtherm and water are introduced at equal mass flow rate. The schematic below (Figure I-10) describes the model for different type heat exchangers and Table I-7 summarizes the simulation conditions.

![Diagram of heat exchangers](https://via.placeholder.com/150)

Figure I-10. Schematic of concentric tubes heat exchangers with twisted (bottom) plane (top) and inner tubes.

Table I-7. Summary of models input conditions.

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Shell side (Dowtherm)</th>
<th>Inner tube (Water)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet temperature (K)</td>
<td>375</td>
<td>345</td>
</tr>
<tr>
<td>Mass flow rate (kg/s)</td>
<td>0.25</td>
<td>0.25</td>
</tr>
</tbody>
</table>

All thermophysical properties for Dowtherm are implemented in COMSOL as function of temperature.

I3.2.3 Simulation results

The simulation results include the calculated fluid exit temperatures for both types of concentric tubes heat exchangers and the pressure drop in both shell and inner flow sides. The fluid temperature distribution for inner and shell sides are shown in Figure I-11 for plane inner tube (left) and twisted inner tube (right) heat exchangers.
The LMTD ($\Delta T_{lm}$) and overall heat transfer coefficient ($U$) can be calculated for each heat exchangers based on the difference between the inlet and exit temperatures.

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$  \hspace{1cm} (27)$$

$$U = \frac{Q \text{ (heat transfer)}}{A \text{ (area)} \times \Delta T_{lm}}$$  \hspace{1cm} (28)$$

The calculated LMTD values are 8.72°C and 11.81°C for concentric tube heat exchanger utilizing inner twisted and plane tubes respectively. The lower LMTD for twisted tube type heat exchanger clearly indicate higher heat transfer coefficient compared to plane inner tube type (24% higher heat transfer coefficient). Other advantage that the twisted tube could also minimize the pressure drop along the length of the heat exchangers. The pressure drop values are calculated for both types under the specified conditions in Table I-7. These values are 65 Pa and 475 Pa for water and Dowtherm fluids respectively in plain type heat exchanger. The pressure drop values are 60 Pa and 500 Pa for water and Dowtherm respectively for twisted tube type heat exchanger. These results show lower but close pressure drop values for both concentric tube heat exchangers under similar conditions.

### 3.2.4 Temperature contours

Figure I-11 (left) shows a smooth temperature change for plane type concentric tube heat exchanger for water and Dowtherm fluids. The calculated temperature for Dowtherm flow in the shell side of the twisted inner tube heat exchanger experience a fluctuation. The fluctuation in the calculated temperature could be attributed to the velocity fluctuation around the twisted surface. Figure I-12 shows the temperature contours for 4 sections located half-way the length of the inner twisted tube type heat exchanger. The 4 section show a full rotation cycle (360 degrees) of the twisted tube cross section (elliptical view). As the gap between the inner and outer pipes get smaller, the shell fluid velocity increases and hence the shell fluid temperature decreases. In addition, as the distance ($x$) increases the shell fluid temperature decreases and vice versa for the counter flow inner fluid (water) as shown in Figure I-12.
I4. System Modeling and Simulation

I4.1 Concept of passive decay heat removal systems of FHRs

Figure I-13 illustrate the layout and design of the Mark-1 Pebble-Bed FHR (Mk-1 PB-FHR), which is one of the FHR design concepts under investigation at the University of California, Berkeley [34]. The Mk-1 PB-FHR uses three DRACS loops to remove decay heat under emergency conditions, when the normal shutdown cooling system is not functional [34]. Each DRACS module consists of a DRACS heat exchanger (DHX) located inside the reactor vessel, a DRACS salt loop, and a salt-to-water heat exchanger outside the reactor containment, which transfers heat from the DRACS salt loop to evaporate water or a chimney [34].

Additional study was carried out to investigate concerns of overheating, overcooling, and physical access (safety and security) [37-38]. The relative high melting point of the fluoride salts makes thermal control of the coolant a challenge, in particular with respect to potential freezing during transient events [39]. In response to the concern of overcooling, and to better understand the use of alternative ultimate heat sinks, the present study proposed a DRACS design using large uninsulated water tanks as the ultimate heat sink for decay heat removal (rather than idealized salt-to-air heat transfer boundary conditions). The DRACS water tanks are designed to have a sufficient reserve volume of water to accommodate early boil-off immediately after reactor shutdown when decay heat levels are high [40]. Moreover, water tanks reduce the difficulty of implementing a reliable access control comparing to that of chimneys.
The selection of this salt-to-water heat exchanger (SWHX) is challenging due to the high temperature difference between the salt side and water side as well as the potential interaction of the salt and water due to the leakage of the heat exchanger. A double-wall shell/tube salt to water heat exchanger is selected in this study as the SWHX. Double-walled shell/tube heat exchangers have been used in a variety of applications in the nuclear industry where there are concerns about the interaction of the primary fluid with the secondary fluid. Figure I-14 shows a cross-section schematic of a double-wall tube. A 2.0 mm gas gap is located between the walls, acting as a barrier to prevent the occurrence of departure of nucleate boiling on the water side and overcooling of the salt due to high freeze temperature of the salt.

Besides of using the water tank as the ultimate heat sink, this study proposed a tiered approach to sizing the salt-to-water heat exchangers for all three DRACS, such that if freezing were to occur in the salt-to-water heat exchangers with relative bigger heat transfer area, times to freezing for the DRACS with smaller heat transfer area would be staggered.

As shown in Figure I-15, the DHX is a 2.5-m-tall shell/tube heat exchanger, located between a lower cold salt plenum and an upper hot salt plenum, to transfer heat from the primary loop to the DRACS loop. Detailed DHX parameters are provided in Table I-8. A check valve is equipped at the bottom of the DHX to limit the upward flow under forced circulation and to provide low resistance for natural circulation down flow [34].
The SWHX is located outside of the containment and transfers heat from DRACS to a water tank. Each DRACS is equipped with a water tank and a SWHX with different heat transfer area to allow incremental freeze of each DRACS while others with smaller heat transfer area continue to operate. Detailed SWHX parameters for each DRACS are summarized in Table I-9.

A RELAP5-3D nodalization diagram of the FHR cooling system and current DRACS design is given in Figure I-16. The reactor core is modeled as a one-dimensional component along a vertical flow channel and the radial power and flow distribution is not modeled. The two power conversion loops, which transfer heat from the primary loop of the reactor, are combined into one equivalent loop. The three DRACS are modeled separately to investigate the potential failure (freeze) of individual DRACS.

Figure I-15. Concept design of the DRACS (type-1).
Figure I-16. Nodalization diagram of the FHR DRACS Design 1.

Table I-8. DHX design parameters for DRACS type-1.

<table>
<thead>
<tr>
<th>DRACS</th>
<th>Tube ID (m)</th>
<th>Tube OD (m)</th>
<th>Tube height (m)</th>
<th>Tube number</th>
<th>Tube side Heat transfer area (m²)</th>
<th>Shell side Heat transfer area (m²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0109</td>
<td>0.0127</td>
<td>2.5</td>
<td>492</td>
<td>42.119</td>
<td>49.075</td>
</tr>
<tr>
<td>2</td>
<td>0.0109</td>
<td>0.0127</td>
<td>2.5</td>
<td>492</td>
<td>42.119</td>
<td>49.075</td>
</tr>
<tr>
<td>3</td>
<td>0.0109</td>
<td>0.0127</td>
<td>2.5</td>
<td>492</td>
<td>42.119</td>
<td>49.075</td>
</tr>
</tbody>
</table>

Table I-9. SWHX design parameters for DRACS type-1.

<table>
<thead>
<tr>
<th>DRACS</th>
<th>SWHX Tube ID (m)</th>
<th>Tube OD (m)</th>
<th>Tube height (m)</th>
<th>Tube number</th>
<th>Heat transfer area (m²)</th>
<th>Tank volume (m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.0109</td>
<td>0.0127</td>
<td>0.8</td>
<td>113</td>
<td>3.096</td>
<td>70.686</td>
</tr>
<tr>
<td>2</td>
<td>0.0109</td>
<td>0.0127</td>
<td>0.8</td>
<td>98</td>
<td>2.685</td>
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</tr>
<tr>
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<td>0.0127</td>
<td>0.8</td>
<td>75</td>
<td>2.055</td>
<td>70.686</td>
</tr>
</tbody>
</table>

I4.2 Transient behavior of DRACS

It is noted that the DRACS is composed of two natural circulation salt loops and a water tank as the heat sink, and the performance of each of the two loops is not only depends on itself but also upon the other two [41]. Hence, the passive decay heat removal system will need to be thoroughly
analyzed, tested, and monitored or inspected to ensure they will meet the performance goals over the lifetime of the plant [42]. This study focused on the transient behavior of DRACS of the FHRs during LOHS. Calculations performed using RELAP5-3D in this study are presented in the following sections.

4.2.1 LOHS scenario

In order to identify the performance of current DRACS design, a LOHS with SCRAM is analyzed here. The design parameters of the FHR are summarized in Table I-10. LOHS is one of the design basis accidents for FHRs, which employ the DRACS system for natural circulation decay heat removal [43,16].

Table I-10. Reactor core steady-state operating parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure (bar)</td>
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</tr>
<tr>
<td>Total thermal power (MW)</td>
<td>236.0</td>
</tr>
<tr>
<td>Primary nominal core mass flow (kg/s)</td>
<td>1084.0</td>
</tr>
<tr>
<td>Primary coolant inlet temperature (°C)</td>
<td>601.0</td>
</tr>
<tr>
<td>Primary coolant outlet temperature (°C)</td>
<td>731.6</td>
</tr>
<tr>
<td>Reactor scram time (s)</td>
<td>0.0</td>
</tr>
<tr>
<td>Primary coolant pump trip time (s)</td>
<td>0.0</td>
</tr>
</tbody>
</table>

4.2.2 DRACS transient behavior

The reactor was tripped immediately after the primary coolant pump tripped, and the pump coast-down begins. Thus, a rapid decrease of the reactor core mass flow rate is observed at the first 200.0 s, as shown in Figure I-17. In this study, two of three DRACS are assumed to be operable while the DRACS with SWHX-1 is down for maintenance. As a result, the inlet and outlet temperature of SWHX-1, shown in Figure I-18, is kept constant during the transient event. After the coast-down of the primary pump, a natural circulation flow was maintained in the core through DRACS until the second DRACS lost its function due to the salt temperature drop to its melting point (459.0 °C), as shown in Figure 17b.

Figure I-19 shows the time evolution of bulk core outlet temperature. The bulk core outlet temperature increased to its peak temperature at the first stage because the decay heat generation is higher than the heat removal from the two operable DRACS. With the increase of salt temperature in the reactor core, the natural circulation heat removal through DRACS is more effective and the reactor was cooled down gradually until one of the DRACS is overcooled and lost its function at 41.1 hours after the reactor scram. With only one DRACS working, the reactor core was heat-up again and reached and then was cooled down again. The last DRACS lost its function due to overcooling at 101.7 hours after the reactor scram.
Figure I-17. Coolant flow through the core.

Figure I-18. The inlet and outlet temperature of DHX and SWHX (a) DHX, and (b) SWHX.
I5. Summary

A computational thermal-hydraulic study performed utilizing RELAP5-3D in conjunction with MATLAB to investigate the decay heat removal performance of the DRACS when partial blockages in the primary system and DRACS are considered. The study chose two redundancy schemes, the first nominally requiring two out of three (2/3) DRACS to operate to provide adequate cooling to the reactor and the second requiring three out of six (3/6) operating DRACS to provide cooling. In the study, basic event trees (ETs) and fault trees (FTs) developed for the FHR were used to identify an important potential failure mode of the DRACS: flow channel blockage occurring in either the DRACS loop itself or in the DHX branch (located in the primary system). One important simplification to the model is that the thermosyphon-cooled heat exchanger (TCHX) which normally rejects heat from the DRACS loop in the Mk1 PB-FHR design was modeled as an idealized boundary condition in the simulation, and was assumed to be a simpler natural draft heat exchanger (NDHX) with direct heat rejection to air in the ET and FT analyses. The methodology and results of investigating the performance impact from these scenarios was discussed [37,44].

A number of insights resulted from this study. Firstly, the rather high level of robustness of the decay heat removal systems to point blockages was unanticipated and lends support to the concept of DRACS as a robust emergency system in general. In the future, it is recommended that this work be expanded to investigate blockages occurring over a finite length, especially in areas of high risk for freezing (i.e., the salt side of the NDHX or TCHX, whichever is used). Secondly, large behavior differences were noted in the core outlet temperature time histories for blockages occurring in the DHX branches versus the DRACS loops. These differences indicate that PCOT may be insufficient on its own from a risk metric standpoint, especially given that metallic creep will be a transient process dependent on the thermo-mechanical and thermo-hydraulic responses of the system. Additionally, this difference in behavior for blockages occurring in different systems highlighted the importance of the primary system having access to a large amount of thermal inertia. With high levels of redundancy, careful sizing of the systems must be performed to balance the risks associated with near term overheating and mid-to-long term overcooling. This information should prove useful to future decision makers in the design and development of the FHR [37,44].
The second computational application presented in this report is a study focused on the economic optimization of a salt-to-salt IHX. The study used metaheuristic search algorithms to rapidly minimize a cost function for an IHX design for both shell-and-tube and twisted-tube concepts. The optimization variables for the shell-and-tube IHX were the outer tube diameter, baffle spacing, and inner shell diameter, while the optimization variables for the twisted-tube IHX were the maximal outer tube diameter, the modified Froude number (dictates relative tube twist), and the inner shell diameter. The conclusions from the study were that twisted-tube heat exchanger technology likely presents a good technological fit for the challenges of salt-to-salt heat exchangers, and substantial cost savings could be effected by using twisted tubes over conventional shell-and-tube heat exchanger designs, particularly when taking into account operating costs.[44]

The results from these computational studies indicated that twisted-tubes enjoy a significant advantage over traditional shell-and-tube heat exchangers from a cost perspective as well as a length perspective. The study has limitations and could be improved to better deal with design constraints such as matching baffle spacing directly to shell-diameter and investigating multi-pass and multi-shell exchanger designs. Additional experimental data at lower modified Froude numbers and Reynolds numbers and smaller tube diameters could be very useful in determining the limits of twisted-tube performance. From an algorithm perspective, both FFA and CSA performed essentially equivalently, although FFA had a faster run time due to fewer function calls in the implementation.[44]

I5. Bibliography


Appendix I-A
1 Nusselt Correlations

Shell-side: Dzyubenko (modified for liquids, derived from [Dzyubenko2006])

\[ Nu = 6.05 \times 10^6 F_r M^{-2.494+0.235 \log(F_r M) Re^n + a \log(Re) \left( \frac{\mu_w}{\mu} \right)^{-0.14} Pr^{0.4} \] (1a)
\[ n = -1.572 F_r M^{0.01661-0.04373 \log(F_r M)} \] (1b)
\[ a = 0.269 F_r M^{-0.01490-0.001040 \log(F_r M)} \] (1c)
\[ 63.6 \leq F_r M \leq 1150 \]
\[ 2000 \leq Re \leq 30000 \] (1d)

Shell-side: Dzyubenko Transitional (modified for liquids, derived from [Transfer2016])

\[ Nu = 83.5 F_r M^{-1.2} Re^n Pr^{0.4} \left(1 + 3.6 F_r M^{-0.357}\right) \left( \frac{\mu_w}{\mu} \right)^{-0.14} \] (2a)
\[ n = \begin{cases} 0.212 F_r M^{0.194}, & F_r M < 924 \\ 0.8, & F_r M \geq 924 \end{cases} \]

Transitional regime

Shell-side: Ievlev #1 (modified for liquids, derived from [Transfer2016])

\[ Nu = 0.023 Re^{0.8} Pr^{0.4} \left(1 + 3.6 F_r M^{-0.357}\right) \left( \frac{\mu_w}{\mu} \right)^{-0.14} \] (3a)
\[ 232 \leq F_r M \leq 2440 \]
\[ 2000 \leq Re \leq 50000 \]

Shell-side: Ievlev #2 (modified for liquids, derived from [Transfer2016])

\[ Nu = 0.0521 Re^{0.8} Pr^{0.4} \left( \frac{\mu_w}{\mu} \right)^{-0.14} \] (4a)
\[ F_r M \sim 64 \]
\[ 2000 \leq Re \leq 50000 \]

Shell-side: Si [Si1995]

\[ Nu = 0.2379 Re^{0.7602} F_r M^{-0.4347} \left(1 + 3.6 F_r M^{-0.357}\right) Pr^{0.33} \] (5a)
\[ 231 \leq F_r M \leq 392 \]
\[ 2000 \leq Re \leq 10000 \]

Tube-side: Si [Si1995]

\[ Nu = 0.396 Re^{0.544} \left( \frac{s}{d_{max,in}} \right)^{0.161} \left( \frac{s}{d_{max,in}} \right)^{-0.519} Pr^{0.33} \] (6a)
\[ 1000 \leq Re \leq 17000 \]
\[ 6.86 \leq \left( \frac{s}{d_{max,in}} \right) \leq 11.9 \]
\[ 0.144 \leq s \leq 0.250 \]

Tube-side: Asmantas [Asmantas1985]

\[ Nu = 0.021 Re^{0.8} Pr^{0.4} \left(1 + 2.1 \left( \frac{s}{d_{max,in}} \right)^{-0.91}\right) \left( \frac{T_w}{T_f} \right)^n \] (7a)
\[ n = -0.17 - 0.27 \times 10^{-5} \left( \frac{x}{d_e} \right)^{1.37} \left( \frac{s}{d_e} \right)^{2.1} - 109.6 \]  
(7b)

\[ 7000 \leq Re \leq 200000 \]

\[ 6.2 \leq \left( \frac{s}{d_{\text{max,in}}} \right) \leq 12.2 \]

**Tube-side: Asmantas Modified (derived from [Asmantas1985])**

\[ Nu = 0.021Re^{0.8}Pr^{0.4} \left( 1 + 2.1 \left( \frac{s}{d_{\text{max,in}}} \right)^{-0.91} \right) \left( \frac{\mu_w}{\mu} \right)^{-0.14} \]  
(8a)

\[ 7000 \leq Re \leq 200000 \]

\[ 6.2 \leq \left( \frac{s}{d_{\text{max,in}}} \right) \leq 12.2 \]

**Tube-side: Ievlev [Ievlev1982]**

\[ Nu = 0.019Re^{0.8} \left( 1 + 0.547 \left( \frac{s}{d_{\text{max,in}}} \right)^{-0.83} \right) \]  
(9a)

\[ 6000 \leq Re \leq 100000 \]

\[ 6.2 \leq \left( \frac{s}{d_{\text{max,in}}} \right) \leq 16.7 \]

**Tube-side: Yang Laminar [I2003a]**

\[ Nu = 3.66 + 0.512Re^{0.477}Pr^{0.975} \left( 1 - \frac{d_{\text{min,in}}}{d_{\text{max,in}}} \right)^{1.532} \left( \frac{s}{d_{\text{min,in}}} \right)^{-0.609} \]  
(10a)

\[ 100 \leq Re \leq 500 \]

\[ 10 \leq \left( \frac{s}{d_{\text{max,in}}} \right) \leq 17 \]

**Tube-side: Yang #1 [Yang2011]**

\[ Nu = 0.034Re^{0.784}Pr^{0.333} \left( \frac{d_{\text{min,in}}}{d_{\text{max,in}}} \right)^{-0.590} \left( \frac{s}{d_e} \right)^{-0.165} \]  
(11a)

\[ 5000 \leq Re \leq 20000 \]

\[ 0.2 \leq s \leq 0.4 \]

\[ 0.0198 \leq d_{\text{max,in}} \leq 0.0218 \]

\[ 0.0058 \leq d_{\text{min,in}} \leq 0.0094 \]

**Tube-side: Yang #2 [Yang2011]**

\[ Nu = 1.50618Re^{0.51825}Pr^{-1.2446} \left( \frac{d_{\text{max,in}}}{d_{\text{min,in}}} \right)^{1.12252} \left( \frac{s}{d_e} \right)^{-0.32367} \]  
(12a)

\[ 7900 \leq Re \leq 26500 \]

\[ 0.160 \leq s \leq 0.250 \]

\[ 0.024 \leq d_{\text{max,in}} \leq 0.026 \]

2 **Friction Factor Correlations**

**Shell-side: Dzyubenko [Dzyubenko2006]**

\[ f = 0.3164Re^{-0.25} \left( 1 + 3.6Fr_M^{-0.357} \right) \]  
(13a)

\[ Re_d \geq 800 \]
\( Fr_M \geq 100 \)

**Tube-side: Asmantas [Asmantas1985]**

\[
f = 0.82 \left( \frac{s}{d_{\text{max},\text{in}}} \right)^{-0.63} Re^{-0.18}
\]

\[7000 \leq Re \leq 200000\]

\[6.2 \leq \left( \frac{s}{d_{\text{max},\text{in}}} \right) \leq 12.2\]

**Tube-side: Ievlev [Ievlev1982]**

\[
f = 0.316 \left( 1 + 3.27 \left( \frac{s}{d_{\text{max},\text{in}}} \right)^{-0.87} \right) Re^{-0.25}
\]

\[6000 \leq Re \leq 100000\]

\[6.2 \leq \left( \frac{s}{d_{\text{max},\text{in}}} \right) \leq 16.7\]

**Tube-side: Si [Si1995]**

\[
f = 10^{a_1 + a_2 \log(Re) + a_3 \log(Re)^2}
\]

\[a_1 = -19.70 + 4.90 \left( \frac{s}{d_{\text{max},\text{in}}} \right) - 0.22 \left( \frac{s}{d_{\text{max},\text{in}}} \right)^2\]

\[a_2 = 10.52 - 2.66 \left( \frac{s}{d_{\text{max},\text{in}}} \right) + 0.12 \left( \frac{s}{d_{\text{max},\text{in}}} \right)^2\]

\[a_3 = -1.47 + 0.36 \left( \frac{s}{d_{\text{max},\text{in}}} \right) - 0.016 \left( \frac{s}{d_{\text{max},\text{in}}} \right)^2\]

\[1000 \leq Re \leq 17000\]

**Tube-side: Yang [Yang2011]**

\[
f = 0.71497 Re^{0.07777} Pr^{-1.03974} \left( \frac{d_{\text{max, in}}}{d_{\text{min, in}}} \right)^{-0.076212} \left( \frac{s}{d_e} \right)^{-0.33393}
\]

\[7900 \leq Re \leq 26500\]

\[0.160 \leq s \leq 0.250\]

\[0.024 \leq d_{\text{max, in}} \leq 0.026\]

**Tube-side: Gao [Yang2011, Gao2008]**

\[
f_t = 4.572 Re^{-0.521} \left( \frac{d_{\text{min, in}}}{d_{\text{max, in}}} \right)^{-0.334} \left( \frac{s}{d_{e,t}} \right)^{-0.082}
\]

\[5000 \leq Re \leq 20000\]
PART II – Experimental evaluation of the HX performance

II1. Introduction

A reduced-scale heat transfer facility (HTF) was constructed at UNM, beginning with light frame construction and design in fall 2012 and continuing to its current completed design iteration in 2017. The facility has operated for more than 300 hours collecting heat transfer data for twisted tubes. The design and motivation for the HTF is multifaceted, but with two main objectives: the exploration of enhanced heat exchanger performance for conditions relevant to the FHR and the collection of data to validate predictive thermal-hydraulics tools used in the FHR development community. In particular, evaluation of the performance of twisted tube heat exchangers under buoyant conditions, for both assisting and opposing flow directions, was an important goal for the facility. The instrumentation chosen for the facility was to provide validation data for system type programs (such as RELAP5-3D) as well as supporting validation data for other types of programs (including component level, such as Primex and derived codes [1-2] and potentially CFD). The data was intended to produce or aid in the creation of appropriate Nusselt number correlations for twisted tubes under low Reynolds and buoyant conditions. Finally, attempts were made to keep the HTF design modular so that other types of heat transfer problems, such as the double-wall twisted tube concept described earlier, could be easily tested using the same facility with minimal modifications.

II2. Experimental Facility

The heat transfer facility (HTF) is composed of three flow loops: the primary loop, the secondary loop, and the chilled water circuit. The purpose of the primary loop is to simulate the shell-side heat transfer of the test section. The secondary loop simulates the tube side heat transfer, and the chilled water circuit serves as a heat sink for the facility. Three flow modes are possible in the primary loop: forced up-flow, forced downflow, and natural circulation, where up-flow and down-flow refer to the direction of flow through the test section, and natural circulation can only be performed in the downflow direction. Photographs and schematic diagram of the loop as constructed are provided in Figures II-1-II-4 [3].

II2.1 Primary Loop

The primary loop provides control over the flow rate, temperatures, pressure, and flow direction of the shell-side fluid in the heat exchanger test section. The loop is heated via a custom-built electric heater designed at UNM, which can provide up to 5-6 kW of thermal energy to the loop. Flow rate in the loop is determined by a variable frequency drive (VFD) controller for the primary pump during forced circulation, and by a balance between driving and retarding forces during natural circulation. Valve throttling can also be used to supplement flow rate control during forced circulation and can serve as a method of increasing retarding forces (and decreasing flow rate) during natural circulation. The primary loop also features a number of ball valves which allow convenient switching between up-flow and downflow through the test section. During the natural circulation flow mode, the fluid traverses the rectangular section of the loop which helps to minimize unnecessary friction and form losses and maximize flow rate.
Temperature measurements are made by T-type thermocouples installed at various locations throughout the loop (inlets, and outlets). Flowrate is measured via the high-accuracy Coriolis flowmeter, and the measurements are sent to a data acquisition system which processes, displays and records the data collected during the course of an experimental run.

![Image](image)

**Figure II-1.** Photograph of the as-built heat transfer facility (HTF), prior to insulation, with the primary loop visible on the near side of the structure [3]

The LabVIEW control program was written to also provide PID control to the electric heater to maintain a predetermined average test section temperature. Pressure and liquid level control are performed via a cover gas system and measured by an Omega pressure transducer located in the surge tank.

**II2.2 Secondary Loop**

Flow rate, temperatures, and pressure of the tube-side fluid are controlled by the secondary loop. This loop is a simple uni-directional, forced circulation circuit. Heat is added to the loop in the test section and rejected by a secondary heat exchanger to the chilled water circuits. Flow rate is controlled by a VFD with supplementary valve throttling also a possibility to achieve lower flow rates. Temperature is monitored by T-type thermocouples and average temperature can be controlled by adjusting two globe valves that control the secondary heat exchanger bypass flow fraction. Lowering the flow rate through the secondary heat exchanger results in higher secondary loop temperatures, while maximizing flow rate through the secondary heat
exchanger reduces the secondary loop temperature. Pressure and liquid level control are performed by the cover gas system and monitored by an Omega pressure transducer located in the surge tank.

![Image of heat transfer facility](image)

Figure II-2. Photograph of the as-built heat transfer facility, prior to insulation, with the secondary loop visible on the near side of the structure and some of the cover gas system and auxiliary systems visible [3]

II2.3 Chilled Water Circuit

The chilled water circuit consists of a re-purposed Haskris water chiller and the supply and return lines to the secondary heat exchanger. The chiller has a variable supply set point of 55 – 90 oF and dual internal positive displacement pumps for a maximum supply flow rate of 6 GPM when run in parallel. Rotameter-type flow meters allow the operator to monitor the supply flow rate. Temperature is monitored by two T-type thermocouples: one at the secondary heat exchanger inlet and one at the outlet. An integrated reservoir provides the chiller with thermal mass and allows for fluid expansion and contraction. Ultimate heat rejection is handled by house chilled water supplied to the chiller by the university’s Ford Utility Plant. The Haskris chiller was used in all early experimental runs but the on/off temperature control was found during data analysis to introduce noise into the system. Later experimental runs bypassed the chiller and used the house chilled water supply to directly cool the secondary loop instead.
II2.4 Cover Gas System

The cover gas system provides control over liquid level, cover gas type, and cover gas pressure in the primary and secondary loops. It also handles venting and provides protection in over-pressure scenarios. When performing shake-down testing of the loop using deionized water, compressed air was used as the cover gas supply. After the switch to Dowtherm A, nitrogen was used as the cover gas to mitigate against oxidation of the hydrocarbon-based fluid at elevated temperatures. Figure II5 shows the layout of the cover gas system with the locations of the connections to the loops.
The gas can be supplied to either the drain tanks or the surge tanks. Loop filling is performed by using the compressed gas as a piston to push the liquid from the drain tanks into their respective loops. After the surge tanks are 1/3 full, the loops are considered full, and nitrogen can then be

To reduce the risk potential of the facility, engineering controls in the form of safety relief valves (SRVs) are implemented: one installed in each surge tank above the normal liquid line. The SRVs are tensioned using springs that allow the valves to open once the pressure has exceeded a certain value. For the primary and secondary loops, the SRV actuation pressures were set at 30 psig. After exceeding this pressure, the SRVs open and vent gas or liquid until the internal pressure has returned to below the actuation pressure. Because the SRVs are located in the surge tanks, they will not prevent over-pressurization of the loop due to a bad valve lineup where the flow rate in the pump is reduced to zero. Operating the pump with zero flow rate is dangerous as the liquid inside can quickly heat up and cause damage to the pump and seals. It is up to the operator to verify valve lineups according to the procedures included in Appendix D and to verify that the flow rates are above minimums specified for the pumps.
All vented gases and fluids are routed to the top of a 5-gallon stainless steel tank to cool and separate any entrained liquid. The tank has a second port on the top to vent the cooled and dried gas to an oil odor-removal filter. Finally, the gas is routed from the filter output to a dedicated laboratory exhaust intake, which maintains a negative pressure to vent fumes out of the laboratory space. This part of the system serves to maintain odor and fumes exposure control, as well as to cool the Dowtherm A vapors to below their flash point before mixing them with the oxygen in the atmosphere.

II2.5 HTF Modifications

As the HTF was designed for testing dip-type shell-and-tube heat exchangers, the facility needed to undergo a few changes in order to be able to test the single assembly DWHX. Multiple factors were taken into account when designing the modifications including cost, ease of construction, and available laboratory space. Accommodating for the length of the heat exchanger (10 feet long), and the available space in the lab, it was decided that the single-assembly heat exchanger would essentially replace the heat exchanger connecting the secondary loop to the chilled water supply. The laboratory has more vertical than horizontal space and as such, the single-assembly heat exchanger was placed vertically in an open section of the laboratory support systems. For data acquisition, a flow meter and globe valve were
installed on the chilled water supply, two globe valves and flow meters were installed for the annular fluid (one each for the air and water supply), and multiple thermocouples and pressure transducers were installed as well. The shell-side fluid for the single-assembly heat exchanger is the DOWTHERM A in the secondary loop of the facility. The intermediate fluid can be air or water depending on the test. These fluids are once-through relying on building supplied maintain steady state configuration. This loop uses a Coriolis flow meter for high accuracy of volumetric flow rate especially under natural circulation conditions (Figure II-6).

Figure II-6. Photographs of the modified HTF to accommodate for the single assembly DWHX (left) and the inner twisted tube of the DWHX (right) [3]

The primary loop functions as the shell-side of the test section which either has a plain or twisted-intermediate tube. The loop also is fitted with a set of valves to change the flow direction in the test section. These valves are also used to bypass the pump in order to reduce loop resistance during the natural circulation tests. Like the primary loop, the secondary loop can be filled with water or DOWTHERM A. The secondary loop functions as the tube-side fluid in the test section. The loop can force fluid flow in the loop over 10 gpm and can be throttled with valves down to 1.5 gpm. The fluid is heated in the test section and is subsequently cooled by a heat exchanger connected to chilled water as the heat sink. The chilled water supply initially was controlled by a chiller. The chiller turns on when the temperature of the water reaches a set-point and turns off after it has been chilled. This on and off action led to a "sawtooth" function within the chilled water which led to a large uncertainty within the data as the function cycled approximately every 20 minutes. Eventually, the chiller was bypassed, and the heat exchanger was directly connected to the building’s chilled water supply. While the reduced uncertainty from the sawtooth of the chiller, fluctuations in the supply are also a cause of uncertainty. A single line diagram of this loop is shown in Figure II-7.
II2.6 Measurements and DAQ system

The HTF loops are fitted with many measurements’ instrumentation including temperature, pressure, flow rate, and electric current and voltage. Temperature measurements are made by T-type thermocouples installed at various locations throughout the loop (inlets, and outlets). Flowrate is measured via the high-accuracy Coriolis flowmeter. Figure II-8 presents the location of different instrumentation through the primary and secondary loops of the HTF. A total of 18 thermocouples, multiple Omega pressure transducers, in addition to the electric parameters controlling the secondary loop pre-heater. All measurements are sent to a data acquisition system (DAQ) which processes, displays and records the data collected during the course of an experimental run. The LabVIEW control program was written to also provide PID control to the electric heater to maintain a predetermined average test section temperature. Pressure and liquid level control are performed via a cover gas system and measured by an Omega pressure transducer located in the surge tank. It is important to optimize the DAQ system and the PID controller gains to quantify the System Quantity Responses (SQRs) and determine the range of system variables. This step essential to determine the experiments domain and build up the testing matrix for collecting experimental data for validation.
Data acquisition for the facility is performed with a modular Compact DAQ system purchased from National Instruments (NI) coupled to a computer running NI Lab-VIEW software for data processing, display, logging, and PID control. The DAQ system is composed of a chassis (model number NI cDAQ-9178) with eight slots for different types of modules and a USB interface for communication with the computer. Two types of modules are currently utilized in the chassis for data collection: the NI 9214, which is a 16-channel thermocouple module with built-in cold junction compensation (CJC) for high accuracy, and the NI 9207, which is a combined current/ voltage measurement module that can read 8 channels of 4-20 mA signals and 8 channels of ±10 V signals. See Figure II-9 below showing the DAQ chassis with the modules.

Bulk and surface thermocouples are read into the NI 9214 modules while the flowmeter and pressure transducer signals (4-20 mA) are read into a single NI 9207 module. The LabVIEW program is run on a standard desktop computer provided for the application. The LabVIEW Virtual Instrument (VI) programmed for the facility features five major loops: a data reading and processing loop, a display loop, a logging loop, and two power control loops. The data reading and processing loop communicates with the DAQ to read in the data in real time to the...
computer’s memory and apply instrument calibration. It performs processing on a small subset of the data in real time: averaging the four temperature readings immediately surrounding the test section on the primary side (two on either side). This spatial average temperature is then processed through a time-averaging filter to reduce noise and is provided as a process variable to the PID controller VI built into LabVIEW.

Figure II-9. Photograph of the DAQ chassis and measurement modules.

The output of the PID VI, the control variable, is a current signal which is passed to one of the power controls loops. A rate limiter is provided to the PID VI to ensure that transitions between power levels are smooth, helping to reduce thermal shock to the loop (in particular, the manufacturer of the Coriolis meter recommends that the equipment not be exposed to thermal shock to increase the equipment lifespan). Values for proportional, integral, and derivative gains are passed to the VI to determine the control behavior of the system. The gains, shown in Table I-11, were empirically chosen in an effort to reduce the response time and overshoot and to increase stability. A temperature set-point is provided on the VI front panel to allow the operator to choose the average shell-side test section temperature that is desired. Due to the thermal inertia of the system, it typically takes some time to reach steady state. The correct average temperature in the test section is reached before the heat transfer rates are steady, as the rest of the loops come up to temperature. The operator must therefore observe the power level, calculated in real-time from the shell-side heat balance, to determine if the system is in steady state, as opposed to just the fluid temperatures.

The power control loops are responsible for communication with the Magna-Power programmable power supply. The VI package was downloaded from the Magna-Power website and includes VIs for a number of tasks: initialization, set overcurrent, set overvoltage, set current, set voltage, apply power, as well as many more. The set overcurrent and set overvoltage VIs are provided as safety mechanisms to control the trip point for the Power Supply Unit (PSU): if the applied voltage or current exceeds either of these values, the PSU will trip and disconnect power to the load. The set current and set voltage VIs communicate the desired voltage and current to the PSU, and the apply power VI tells the PSU to attempt to apply this voltage and current to the load. For standard operation of the facility, the set voltage VI sets the
maximum voltage that can be applied to the heater and is set from an option on the front panel. The PID control output provides the signal that the set current VI receives. A polling scheme refreshes the power loop containing the set current VI that allows the PSU to receive updated desired current values every 1000 ms. The PSU will then adjust to match the desired current values provided it is possible to do within the limits of the voltage setpoint and the overcurrent and overvoltage values. Finally, the power control loops also send the current and derived power (the product of output voltage and current) levels to the front panel for display to the operator.

Table I-11. The PID gains used for power supply control [3]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Proportional gain (P)</td>
<td>20</td>
</tr>
<tr>
<td>Integral gain (I)</td>
<td>1 min</td>
</tr>
<tr>
<td>Derivative gain (D)</td>
<td>0 min</td>
</tr>
</tbody>
</table>

The display and logging loops are straightforward “while” loops for displaying the desired data to the front panel: temperatures of bulk and surface thermocouples, pressure readings from the transducers, and primary and secondary flow rates. The data is displayed in numeric formatting located with each value displayed next to its respective measurement position on the loop and in graphical format for viewing time histories of the data. The graphical displays allow the operator to make a determination of the transient behavior of the system and whether it is currently in steady-state operation or is changing with time. The logging loop serves the function of saving the time histories of each measurement to disk to allow post-processing of the data at a later point.

This report summarizes the Instrumentation and DAQ system optimization for the HTF used to conduct the project experiments. The measurements’ signals and the HTF components including multiple instrumentation including thermocouples, flowmeters and pressure transducers signals were controlled, received, and recorded using integrated NI DAQ system with LabView Interface. A PID power loop controller was built and optimized in the LabView interface to control the Voltage-Current to set up the secondary loop temperature (Figure II-10). The power control loops are responsible for communication with the PSU that runs the secondary loop pre-heater and trip the electric supply for the safety of the students and HTF components including PSU and electronic instrumentations.
II3. Experimental Conditions and Results

The experimental conditions and results are presented in two different sections. The first section presents the obtained results for tube bundle of twisted tube HX and the second group of experimental results are for small scale experiment for single assembly DWHX.

II3.1 Twisted Tube Bundle HX

Figure II-11 shows a masked photograph of the two single-wall heat exchangers used for this study: one plain and one twisted. Determination of the heat transfer coefficients on the shell side and tube side was performed by using the calculated thermal duty, log-mean temperature difference, and overall heat transfer coefficient in conjunction with an estimated wall temperature through the application of thermal resistance theory. Because the test section is a two-stream heat exchanger, the determination of overall heat transfer coefficient requires both shell-side and tube-side heat transfer coefficients. For most analyses, there was a given “target” heat transfer coefficient and “non-target” heat transfer coefficient, with the target heat transfer coefficient’s value being the goal of a given analysis and the non-target heat transfer coefficient being estimated using available correlations. To reduce uncertainty in the estimation, the non-target heat transfer coefficient was typically maximized (minimizing its thermal resistance) when possible to reduce its contribution to the uncertainty in the target heat transfer coefficient.
The problem complexity was increased somewhat compared to simple single-pass shell-and-tube heat exchangers because of the dip-type design of the bundles tested. In the dip-type design, the tube-side inlet and outlet were both located at the top of the bundle. The inlet led to a down-comer tube which traversed the length of the heat exchanger and distributed its flow to the bundle tubes from a lower plenum. At the top of the bundle, the flow was collected in an upper plenum and exited the exchanger from the outlet tube which was primarily co-axial with the down-comer tube. This design increased the complexity of the analysis from two aspects: (1) the exchanger is no longer a purely counterflow or co-current-flow heat exchanger, but has two unequal tube-side passes, and (2) there are multiple heat transfer pathways in parallel. For a typical shell-and-tube heat exchanger, the heat transfer problem can be well approximated by considering only the heat transfer through the tubes; however, because the dip-type exchanger tested here was small, it was considered important to quantify the effects of the down-comer tube and the plena.

II3.1.1 Data reduction approach

Figure II-12 illustrates various forms of the thermal circuit for the heat exchanger bundles. For the thermal resistances, the subscript tot is for the total heat exchanger, o is for overall, s is for shell side, t is for tube side, w is for wall, tubes is for the tube bundle (the target of this study), dc is for down-comer, p-sides is for the plena sides, and p-ends is for the plena ends.
The plena were split into their sides and ends due to the difference in cylindrical and planar geometry. From thermal resistance theory [5], the overall thermal resistance for the four parallel paths can be expressed as:

\[
\frac{1}{R_{o,\text{tot}}} = \frac{1}{R_{o,\text{tubes}}} + \frac{1}{R_{o,dc}} + \frac{1}{R_{o,p\text{-sides}}} + \frac{1}{R_{o,p\text{-ends}}}
\]

where the individual parallel resistances can be expressed by their constituent resistances added in series:

\[
R_{o,\text{tubes}} = R_{s,\text{tubes}} + R_{w,\text{tubes}} + R_{t,\text{tubes}}
\]
\[
R_{o,dc} = R_{s,dc} + R_{w,dc} + R_{t,dc}
\]
\[
R_{o,p\text{-sides}} = R_{s,p\text{-sides}} + R_{w,p\text{-sides}} + R_{t,p\text{-sides}}
\]
\[
R_{o,p\text{-ends}} = R_{s,p\text{-ends}} + R_{w,p\text{-ends}} + R_{t,p\text{-ends}}
\]

Assuming that the mean temperature difference of the heat exchanger holds for each of the parallel heat transfer pathways, we can derive the following equations:

\[
Q_{\text{tot}} = Q_{\text{tubes}} + Q_{\text{dc}} + Q_{p\text{-sides}} + Q_{p\text{-ends}}
\]
\[
\frac{\Delta T_m}{R_{o,\text{tot}}} = \frac{\Delta T_m}{R_{o,\text{tubes}}} + \frac{\Delta T_m}{R_{o,dc}} + \frac{\Delta T_m}{R_{o,p\text{-sides}}} + \frac{\Delta T_m}{R_{o,p\text{-ends}}}
\]

Estimates for the wall temperatures for the exchanger were determined in an approximate manner by calculating, for the tube bundle, the mean temperature drop due to either tube-side or shell-side film and adding or subtracting from the arithmetic mean fluid temperature on the tube or shell side.

\[
Q_{\text{tubes}} = \frac{\Delta T_m}{R_{o,\text{tubes}}} = \frac{\Delta T_s}{R_{s,\text{tubes}}} = \frac{\Delta T_w}{R_{w,\text{tubes}}} = \frac{\Delta T_t}{R_{t,\text{tubes}}}
\]
\[
T_{w,s} = \frac{T_{s,\text{in}} + T_{s,\text{out}}}{2} - \Delta T_s
\]
\[
T_{w,t} = \frac{T_{t,\text{in}} + T_{t,\text{out}}}{2} + \Delta T_t
\]

The “target” thermal resistance due to the tube-side or shell-side film for the tube bundle can be expressed as an algebraic function of the other resistances:

\[
R_{t,\text{tubes}} = \left( \frac{1}{R_{o,\text{tot}}} - \frac{1}{R_{o,dc}} - \frac{1}{R_{o,p\text{-sides}}} - \frac{1}{R_{o,p\text{-ends}}} \right)^{-1} - R_{w,\text{tubes}} - R_{s,\text{tubes}}
\]
\[
R_{s,\text{tubes}} = \left( \frac{1}{R_{o,\text{tot}}} - \frac{1}{R_{o,dc}} - \frac{1}{R_{o,p\text{-sides}}} - \frac{1}{R_{o,p\text{-ends}}} \right)^{-1} - R_{w,\text{tubes}} - R_{t,\text{tubes}}
\]
The overall thermal resistances are defined as follows:

\[
R_{o,tot} = \frac{1}{US_{s,tot}} \\
R_{o,dc} = \frac{1}{U_{dc}S_{s,dc}}
\]

\[
R_{o,p-sides} = \frac{1}{U_{p-sides}S_{s,p-sides}} \\
R_{o,p-ends} = \frac{1}{U_{p-ends}S_{s,p-ends}}
\]

And the individual thermal resistances can be defined as well:

\[
R_{s,tubes} = \frac{1}{h_{s,tubes}S_{s,tubes}} \\
R_{t,tubes} = \frac{1}{h_{t,tubes}S_{t,tubes}} \\
R_{s,dc} = \frac{1}{h_{s,dc}S_{s,dc}} \\
R_{w,dc} = \frac{1}{2\pi \lambda_w L \ln \left( \frac{d_{dc,\text{out}}}{d_{dc,\text{in}}} \right)} \\
R_{t,dc} = \frac{1}{h_{t,dc}S_{t,dc}} \\
R_{s,p-sides} = \frac{1}{h_{s,p-sides}S_{s,p-sides}} \\
R_{w,p-sides} = \frac{1}{2\pi \lambda_w L_{\text{plenum}} N_{\text{plena}} \ln \left( \frac{d_{\text{plenum,\text{out}}}}{d_{\text{plenum,\text{in}}}} \right)} \\
R_{t,p-sides} = \frac{1}{h_{t,n-sides}S_{t,n-sides}} \\
R_{s,p-ends} = \frac{1}{h_{s,p-ends}S_{p-ends}} \\
R_{w,p-ends} = \frac{\delta_{p-ends}}{S_{p-ends} \lambda_w} \\
R_{t,p-ends} = \frac{1}{h_{t,p-ends}S_{p-ends}}
\]

And the wall thermal resistance for the twisted tubes

\[
R_{w,tubes} = \frac{1}{2\pi \lambda_w L N} \ln \left( \frac{d_{\text{max,\text{out}}} + d_{\text{min,\text{out}}}}{d_{\text{max,\text{in}}} + d_{\text{min,\text{in}}}} \right)
\]

Two simplifications were made to the above analysis: (1) the surface areas of the plena ends were both assumed to be that of a full circle, although the upper plenum has the down-comer tube extending above it, and (2) the heat transfer coefficients for the tube and shell sides of the plena ends were assumed equal to those of the plena sides. These are rather small
approximations as the surface area of each plenum was a small fraction of the total surface area of the heat exchanger.

\[
h_{t,p\text{-}ends} \approx h_{t,p\text{-}sides} \\
h_{s,p\text{-}ends} \approx h_{s,p\text{-}sides}
\]

With the geometric parameters measured, the inlet and outlet temperatures measured, and the flow rates measured, it was possible to estimate values for the thermal resistance equations above. To minimize the contribution to uncertainty, the non-target heat transfer coefficient for the bundle was maximized (thereby minimizing its thermal resistance) when possible by running the pump at a high flow rate. The non-target heat transfer coefficient for the bundle was estimated using available correlations for twisted tubes, while all of the heat transfer coefficients for the down-comer and plena were also estimated using available correlations. The correlations used for the down-comer and plena were as follows. For Reynolds numbers between 2300 and 10,000, the Gnielinski correlation was used [6], and for Reynolds numbers greater than 10,000, the Sieder and Tate correlation was used [6]. For laminar, developing flow, the Shah, Bhatti, and Hausen correlation was used [6]:

\[
Nu = 3.66 + \frac{0.0668}{x^{1/3} (0.04 + x^{2/3})}
\]

where,

\[
x^* = \frac{L}{d_e Pe}
\]

\[
Pe = Re Pr
\]

where \( L \) is the length, \( Pe \) is the Peclet number, and \( d_e \) is the equivalent hydraulic diameter.

With this information, the overall heat transfer coefficient \( U \) could be estimated along with the individual resistances. By using the measured thermal duty \( Q \) and LMTD (rather than the predicted thermal duty and LMTD), it was possible to get a reasonable estimate for the target heat transfer coefficient by estimating all others according to the equations above. However, it was still necessary to take into account effect of multiple unequal tube passes on the measured LMTD to make an estimate for the actual Mean Temperature Difference (MTD). Roetzel and Spang [7] analytically solved this problem (termed “unbalanced” heat exchangers) for two tube-side passes. Their work demonstrated that MTD could be estimated with the following equations:

\[
\frac{\Delta T_m}{T_{s,\text{in}} - T_{s,\text{out}}} = \frac{V}{\ln \left( \frac{2-(\Phi_s+\Phi_t)+V}{2-(\Phi_s+\Phi_t)-V} \right)}
\]

\[
V = \sqrt{\left( \Phi_s^2 + \Phi_t^2 \right) + 2(2\epsilon - 1)\Phi_s\Phi_t}
\]

\[
\epsilon = \frac{NTU_{t,\text{coefficient}}}{NTU_{t,\text{total}}} = \frac{(US)_{t,\text{coefficient}}}{US}
\]

\[
\Phi_s = \frac{T_{s,\text{in}} - T_{s,\text{out}}}{T_{s,\text{in}} - T_{t,\text{in}}}
\]

\[
\Phi_t = \frac{T_{t,\text{out}} - T_{t,\text{in}}}{T_{s,\text{in}} - T_{t,\text{in}}}
\]
The thermal duty was determined by the average of the estimated shell-side heat balance (corrected for heat losses through the insulation and thermocouples bias) and the estimated tube-side heat balance. The inlet and outlet temperatures on the shell side were estimated by averaging the two thermocouple readings (before and after the static mixers) on each end of the test section. The heat balances were defined by the following equations:

\[ Q_s = G_s C_{p,s} (T_{s,in} - T_{s,out,corrected}) \]
\[ Q_t = G_t C_{p,t} (T_{t,out} - T_{t,in}) \]
\[ Q = \frac{Q_s + Q_t}{2} \]

The shell-side outlet temperature was corrected for thermocouple bias and heat losses through the insulation by running the loop in up flow and downflow configurations at different flow rates and temperatures with the secondary loop drained. Heat losses as a function of temperature were estimated and then the heat loss curves for up flow and downflow were matched by adjusting slightly for directional bias in the thermocouple readings, which was most noticeable when the temperature change across the test section was very small. To compensate for heat losses, the corrected outlet temperature was estimated with:

\[ T_{s,out,corrected} = T_{s,out} - \Delta T_{bias} + \frac{Q_{loss}}{G_s C_{p,s}} \]

\( Q_{loss} \) was estimated as a linear, empirical function of the average shell-side temperature and was a particularly important adjustment at low shell-side flow rates. At very low flow rates, the adjustment could change the outlet temperatures by several degrees. In some cases, the uncorrected outlet temperature could be lower than the tube-side inlet temperature, which is unphysical at steady-state operation if there is only heat transfer between the two fluid streams. The corrected outlet temperature compensated for these losses.

For a given test, a large number of time-dependent values for temperatures and flow rates were recorded. Because only steady-state heat transfer is being considered in this work, the values used in the determination of heat transfer coefficients and in building appropriate correlations should also be steady-state. The steady-state behavior of the loop was judged from plotting the calculated power transferred through the heat exchanger and slicing the data set for periods which appeared to follow, on average, a zero slope. The data set is then reduced to several steady-state slices, each of which served to produce the measured and predicted heat transfer coefficient pairs compared using the validation metric.

A variety of uncertainty propagation methodologies are available for determining the uncertainty in the quantity of interest (in our case this includes the heat transfer coefficient in dimensional form \( h \) and non-dimensional form \( Nu \), as well as the overall heat transfer \( Q \) or overall heat transfer coefficient \( U \)). The general problem of propagating uncertainties from independent variables through non-linear functions is considered immensely complex from a mathematical standpoint; however, the uncertainties can often be readily propagated through a linearization process using only the initial linear terms of a Taylor expansion [8]. For independent variables and independent random uncertainties, the uncertainty in a function \( q \) can be expressed as [9]:

51
For the results in this report, each steady-state slice comprised 900 seconds (15 minutes) of data taken at 1 Hz and averaged to produce a single data point. For each of these points, it was possible to sample from a uniform distribution with the center defined as the measurement point and the width defined by the instrumentation uncertainty. Uncertainties were also defined for the various geometrical parameters associated with heat transfer prediction (tube diameters, length, etc.). Table I-12 defines the uncertainty widths used for some of the measurements.

From each distribution, a large number of samples were drawn (n = 100,000 for the plots presented in the results section) for each steady-state averaged measurement point. These sampled values were then added to the actual measurements’ points at each step-in time to create an array of values of length n, which represent the total variations expected based on assumed uncertainties in measurements at each step-in time. The heat transfer calculation was then performed using element-by-element operations (for a total of n computations of heat transfer performance for each steady-state point) and the target heat transfer resistance (and heat transfer coefficient) distribution was determined using Equations above. The independent non-dimensional parameters (Re, Gr, etc.) were also calculated in the process of determining the target heat transfer coefficient, each with a distribution associated with it.

Table I-12. Uncertainty widths assumed in the measurement values.

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Width</th>
<th>Source</th>
</tr>
</thead>
<tbody>
<tr>
<td>$G_s$</td>
<td>0.1%</td>
<td>Product literature</td>
</tr>
<tr>
<td>$G_t$</td>
<td>0.2%</td>
<td>Product literature</td>
</tr>
<tr>
<td>$T$ (for absolute calculations)</td>
<td>1.5 $C^\circ$</td>
<td>Product literature - accuracy</td>
</tr>
<tr>
<td>$T$ (for relative calculations)</td>
<td>0.15 $C^\circ$</td>
<td>Product literature - stability</td>
</tr>
<tr>
<td>Heat exchanger dimensions</td>
<td>See note</td>
<td>Measurement</td>
</tr>
<tr>
<td>$\lambda_w$</td>
<td>5.0%</td>
<td>Assumed</td>
</tr>
<tr>
<td>Thermophysical properties</td>
<td>None</td>
<td>Assumed</td>
</tr>
<tr>
<td>$Q_{loss}$</td>
<td>50 W</td>
<td>Estimated</td>
</tr>
</tbody>
</table>

The means, medians, standard deviations, and absolute relative deviations of the target heat transfer coefficient and non-dimensional parameters were numerically determined from the result vectors. Because one of the goals of the experiment was determining the accuracy of a given correlation, no inherent uncertainty was assumed for any correlation, and instead only the uncertainties in the input parameters were propagated. This allowed for a determination on goodness of fit between two sets of data points: experimentally derived and predicted, each with a defined mean and standard deviation both in value (dependent variable) and in independent variable. Furthermore, the reduction methodology used here ignored the inherent uncertainty in the thermophysical properties, which is an important simplification. Other simplifications include ignoring tube-side heat transfer to itself in the short co-axial section of the tube-side outlet and inlet (down-comer tube). With this information, it was possible to determine numerically the level of agreement between experiment and prediction, using only the correlations for heat transfer, thermophysical properties, and assumptions about the
distributions associated with each measured parameter. It was also possible to easily change or modify correlations and graphically and quantitatively view the effect on predictive accuracy.

Finally, with the experimental data available it was possible to perform least-squares regression to determine improved correlations, such as for substituting Reynolds dependency for forced circulation correlations with Grashof or Rayleigh number dependency to generate a buoyancy-driven correlation.

The heat transfer results from the experiment for the twisted tube bundle are divided into two groups for shell side natural and forced circulation. Each result section also summarizes the predictive accuracy of existing correlations for the flow conditions tested here. Conclusions from the results are drawn.

II3.1.2 Forced circulation results

A number of studies on the heat transfer performance of twisted tubes are available in the open literature. A summary of the available correlations is presented in Appendix I-A. The correlations generally follow some readily recognizable patterns, with the shell-side correlations following the general form:

\[
Nu_F = f \left( Re, Pr, \left( \frac{T_w}{T_f} \right), Fr_M \right)
\]

Where the modified Froude number Fr_M takes into account the twisted tube geometry and the viscosity correction factor \( (T_w/T_f)^{0.55} \) for gases. Most of correlation listed in Appendix I-A were modified by replaced the temperature correction factor by the viscosity ratio term used by Sieder and Tate correlation \( (\mu_w/\mu_f)^{0.14} \) assuming this form holds for twisted tubes. Interestingly, one of the correlations is in terms of logarithmic polynomials with respect to Reynolds number and Fr_M, while the rest take simpler linear logarithmic forms. The tube-side formulas are generally similar but use nondimensional parameters other than Fr_M to characterize the geometry. A few use the ratio of the tube twist pitch to maximal inner diameter \( s/d_{\text{max}-\text{in}} \), others use the ratio of tube twist pitch to equivalent hydraulic diameter \( s/de \), the ratio of minor to major axes (minimal to maximal inner diameter) \( d_{\text{min}-\text{in}}/d_{\text{max}-\text{in}} \), and one even uses the ratio of twist pitch to minimal diameter \( s/d_{\text{min}-\text{in}} \). It is worth noting that in some of the references, it was unclear whether the authors utilized inner or outer tube dimensions for maximal and minimal diameters. Because the correlations were developed for internal flow. It is worth pointing out some of the limitations expected of correlations available in the literature. First, the Reynolds number range tested here was lower than most other studies. Secondly, the tube pitch in this work is smaller than many of those reported in similar studies, and in particular Fr_M is significantly smaller than those used in the available studies [10-12].

Forced circulation and natural circulation heat transfer regime maps have historically been generated for plain circular cylindrical tubes. For example, Metais and Eckert [13] developed regime maps for flow through vertical and horizontal tubes correlated according to Re and Ra \( (D/L) \) and divided the map into regimes corresponding to forced laminar, transitional, and turbulent flow, free laminar, transitional, and turbulent flow, and mixed laminar, transitional, and turbulent flow. As far as the author of this dissertation is aware, such maps are not available for twisted tubes in the open literature. However, some information is available for forced and free convection in tubes with twisted tape inserts [14-15]. In general, one particularly elegant
way of estimating whether forced or free convection forces are dominant is by calculating the Richardson number:

\[ R_i = \frac{G_r}{Re^2} \]

where, for this analysis, the momentum diffusivity for the Grashof number was evaluated at the wall temperature. Kim and El-Genk [16] used a combination of Richardson number and the regime maps of Metais and Eckert [13] to successfully correlate data for slow flowing water in rod bundles (shell-side). When \( R_i < 1 \), it can generally be expected that forced convection heat transfer will tend to be of greater importance than free convection, becoming dominant when \( R_i < 0.1 \). For \( R_i > 1 \), the effect of buoyancy begins to be of similar importance to forced convection and would be expected to begin dominating when \( R_i > 10 \). So, generally one could expect the mixed convection regime to lie somewhere in the range of \( 0.1 < R_i < 10 \), depending on the circumstances of the geometry and the fluids involved [17].

The general approach for the data reduction of the forced circulation data is summarized as follows. First, an appropriate tube-side correlation was chosen as a starting point, as the present work departed less from the literature for the tube-side conditions. The Si correlation (see Appendix I-A) was selected for its reasonable behavior in the Reynolds range and twist pitch-to-diameter ratio of this work. It is important to note that it is potentially possible for non-unique combinations of heat transfer coefficients to determine the same overall heat transfer coefficient. This is an important limitation of the methodology applied here, as the accuracy of the starting correlation used to begin the fitting determines the accuracy of the fitted correlations that follow.

Next, the shell-side Nusselt number was treated as the “target” value and was determined for forced flow (empirically estimated to be \( Re_s > 700 \), corresponding to \( R_{i,s} <1 \)) using the Si correlation as the tube-side estimate for \( Re_t > 1000 \) (corresponding to its range of validity). As the data was analyzed it was found that the forced circulation downflow data fit well as a function of Reynolds number when restricted to \( Re_s >700 \) (\( R_{i,s} <1 \)); however, the available shell-side correlations generally did not fit well, with the Dzyubenko transitional correlation over predicting by a factor of 2−3 as shown in Figure II-13.
There could be several reasons for this: (1) the tube layout was circular in this experiment rather than triangular layout for which the correlations were developed, (2) the twist pitch (represented on the shell-side by $F_{TM}$) was smaller than the experiments for which the correlations were developed, (3) the Reynolds numbers were generally low compared to available correlations except for the transitional correlation developed by Dzyubenko and Stetsyuk [18], and (4) the method of determining heat transfer was different: whereas previous experiments by Dzyubenko used resistance heated twisted tubes to heat gas (closer to a uniform heat flux boundary condition), while this work is cooling a moderately high Prandtl liquid (somewhere between idealized uniform heat flux and uniform wall temperature boundary condition).

To determine the validity of existing correlations, the predicted value of heat transfer $Q$ was compared to the measured value for the various combinations of the available correlations for the full range of data, including forced up-flow, downflow, and natural circulation. The inlet temperatures and flow rates were provided to the computational script and a combination of outer iterations to resolve outlet temperatures (and $Q$) and inner iterations to resolve the heat transfer coefficients was used to predict $Q$. As a simple validation metric, the mean absolute relative error (MARE) and maximum absolute relative error (MaxRE) were selected for determining level of agreement, as they provide a reasonable and intuitive estimate for the error between two sets of data (MARE was used in [19] and this discrete form is similar to the average relative error metric

\[
MARE = \frac{1}{N} \sum_{i=1}^{N} \frac{|y_{\text{predict}} - y_{\text{measured}}|}{y_{\text{measured}}},
\]

\[
MaxRE = \max \left( \frac{|y_{\text{predict}} - y_{\text{measured}}|}{y_{\text{measured}}} \right).
\]
defined as a continuous function in [20], which also included a definition for MaxRE:

where y is the quantity of interest (such as Q, U, or Nu) and N is the number of measurement points (274 steady-state points were collected for this study). The best combination for predicting the full range of forced and natural circulation for up-flow and downflow was found to be the Dzyubenko transitional regime correlation on the shell-side [11] and the Yang laminar correlation on the tube-side [21]. It should be noted that in this script execution mode, the predicted Q is determined using predicted MTD, which corresponds to the predicted Q and U varying by the same amount compared to the measured Q and “measured” U (since the measured U must be estimated using the predicted MTD, given the unbalanced tube passes requiring an F factor calculation).

The uncertainty bars in the plots were first calculated by numerical determination of the standard deviations for each set of data comprising a single steady-state performance point. As a metric of distribution spread, the standard deviation is not robust to outliers, so any outliers could drive the error bars to high values [22]. To help reduce the effects on the mean (a metric of the central tendency of the data) from any outliers, the median (a robust metric) was instead used in plotting the data below, in determining residuals for least squares analysis, and for calculating level of agreement between experiment and prediction. The median absolute deviation (MAD) was used instead of the standard deviation [22]:

\[
MAD = \left( \frac{1}{c} \right) \text{MED} |x_i - M_j(x_j)|
\]

where x represents the data, M represents the median, and c is a normalization constant. The MAD essentially calculates the median of the absolute deviations of the data around the median of that data. The normalization constant c has been taken to be 0.67449, which adjusts the scale of the MAD to match that of standard deviation for normally distributed data (without outliers) [28]. The error bars were calculated with 2 ×MAD, which corresponds to 2 for normally distributed results (two standard deviations).

From heat exchanger analysis, it is well understood that the overall heat transfer coefficient in a two-stream exchanger can be “controlled” by one fluid side if that side’s heat transfer coefficient is much less than that of the other fluid, due to the comprising the majority of the thermal resistance [5]. In such situation, the larger heat transfer coefficient could potentially vary by significant amounts while only having a modest effect on the overall heat transfer. Because this analysis relies on backing out the target heat transfer coefficient from the overall heat transfer coefficient, determining a much larger heat transfer coefficient (non-controlling) from a small one (controlling) is inherently a more uncertain venture, translating to the rather large uncertainty bars on the forced circulation shell-side Nusselt number measurement shown in Figures II-13. The large uncertainty bars on the shell-side Nusselt number prediction are a result mainly from the uncertainty in the twist pitch measurement.

From a validation perspective, it is apparent from the results that a new shell-side correlation could be proposed to improve experimental agreement with the data collected here. In any case, developing experiment-specific correlations for forced flow is quite helpful in determining the effect of buoyancy on the heat transfer for those experiment runs with higher Richardson numbers (even as purely a tool for decreasing uncertainty in determining these effects). To develop the new correlations, a new shell-side forced correlation of the form

\[ Nu_{F,S} = \]
was determined using the Python “Scipy” library “least-squares” function with the residuals between the predicted and measured $N_u F_S$ values being used as the function input [23]. The new forced circulation correlations could then be used as a comparative baseline in estimating the effect of buoyancy forces on heat transfer and as a starting point for fitting of buoyancy affected flow on the tube side. The shell side experimental data are presented in Figure II-14 and the best data fit correlation for forced up-flow and downflow was determined to be:

$$N_u F_S = 0.255 R e_s^{0.682} P r_s^{0.4} \left( \frac{\mu_{w,s}}{\mu_s} \right)^{-0.14},$$

assisting/opposing: $R e_s > 700 \ (R i_s < \sim 1)$

The predicted Nus in this case appears with smaller uncertainty bars because it does not take FrM as an input.

II3.1.3 Mixed and natural circulation results

While the shell-side data correlated acceptably well to Reynolds number alone for $R e_s > 700 \ (R i_s < 1)$, as the Reynolds number decreased (Richardson number increased) it became clear that the correlation was no longer adequate and in fact extremely inaccurate in many of the downflow cases. Interestingly, the up-flow data was well described by a very similar correlation based on Reynolds number (see Figure II-15):

$$N u M_s = 0.325 R e_s^{0.648} P r_s^{0.4} \left( \frac{\mu_{w,s}}{\mu_s} \right)^{-0.14}$$

opposing: $R e_s < 700 \ (R i_s > \sim 1)$

$N u_s / P r_s^{0.4}$ vs. $R e_s$

MARE = 6.8%
MaeRE = 38.0%

**Figure II-14. Comparison of experimental results and predictions for shell-side forced convection using the modified correlation and using the Si correlation for tube-side Ret > 1000.**
Figure II-15. Comparison of experimental results and predictions for up-flow shell-side mixed convection in the twisted tube bundle using the new correlation and using the Si correlation for tube-side $Re_t > 1000$

For downflow, the situation was more complex. At the very low end of the shell-side flow rate (consisting mostly of natural circulation runs), the data was surprisingly well described by Reynolds number alone. In these cases, the shell-side heat transfer coefficient was higher than up-flow by roughly a factor of two, and was reasonably well described by the following correlation:

$$Nu_{s,M} = 0.535Re_s^{0.712}Pr_s^{0.4} \frac{\mu_{w,s}}{\mu_s}^{-0.14},$$

assisting: $Re_s < 180$ ($Ra_s > \sim 30$)

For downflow (assisting), the shell-side data for $180 < Re_s < 700$ was not well correlated to either Reynolds number or Rayleigh number alone. However, similar to some of the buoyancy affected flow correlations for the twisted tape inserts [14] and some correlations for mixed convection flows in plain tubes [24], the data was found to be best correlated to a simultaneous function of both Reynolds number and Rayleigh number. The data are plotted in Figure II-16 and best fit the equation:

$$Nu_{s,M} = 0.0263Re_s^{0.132}Ra_s^{0.433} \left( \frac{\mu_{w,s}}{\mu_s} \right)^{-0.14},$$

assisting: $180 < Re_s < 700$ ($\sim 1 < Ra_s < \sim 30$)

Although static mixers were not included on the tube-side as part of the design (as the facility was originally built with the main focus on shell-side phenomenology), it was still possible to perform tests on low tube-side flow rates.
The primary loop had three thermocouples on the top horizontal section of the loop, and thermal stratification became quite obvious at low flow rates. The postulated mechanism is that hot fluid rose into the uninsulated surge tank, cooled, and returned to the pipe. This led to decreased temperature measurements in TC-BP-6 as compared to TCBP-5 and TCBP-7. This effect tended to increase with decreasing flow rate. Because only a single thermocouple was installed at the heat exchanger inlet on the secondary loop, it was unknown but considered likely the readings from this thermocouple could be affected in a similar manner as TC-BP-6. With these limitations in mind, it was observed that the low flow rate tube-side data correlated reasonably well to Rayleigh number alone for $\text{Re}_{t} < 600$ ($R_{i,t} > 1$):

$$N_{ut} = 0.276 R_{i,t}^{0.260}$$

where the results are plotted in Figure II-17. Although the flow was pumped, the heat transfer coefficient was reasonably correlated to buoyancy forces alone.

To illustrate the effect of direction on the shell-side heat transfer, $\text{Nu}_{s}$ was plotted for both up- and downflow data using the proposed correlations above and using the Si correlation on the tube-side and plotting all data for $\text{Re}_{t} > 1000$. Figure II-18 makes clear the difference in heat transfer for low shell-side Reynolds numbers, with assisted flow (downflow) having improved heat transfer over opposing flow (upflow). For the assisted flow, a transition appears to take place where both Reynolds and Rayleigh number can be used together to predict heat transfer coefficient, and the two curves converge in the range of $\text{Re}_{s} = 700 − 1000$. With the proposed correlations, the level of agreement is significantly improved. Estimated $U$ is plotted in Figure II-19 against the measured overall heat transfer coefficients.
Figure II-17. Comparison of experimental results and predictions for tube-side convection in the twisted tube bundle using the Nut = 0.276 Rat0.26 for Res > 700.

Figure II-18. Comparison of experimental results and predictions for shell-side convection in the twisted tube bundle using the new correlations and using the Si correlation for tube-side Ret > 1000.
II3.2 Single Assembly DWHX

The University of New Mexico (UNM) has acquired a tube-in-tube heat exchanger which has been denoted as a single-assembly double-wall heat exchanger. The heat exchanger contains a shell, annular tube (Twisted and plain), and inner tube (Figure II-20). This experiment aims to investigate the heat transfer performance towards the use of the DWHE to couple the FHR to the S-CO2 power cycle. The plain and twisted DWHE are investigated experimentally at UNM. The twisted inner tube DWHE will be coupled to S-CO2 Brayton cycle at Sandia National Lab. The following sections present the experimental results obtained for the single-assembly DWHE in support of this NEUP.

Figure II-20. A schematic of the single-assembly DWHE.
II3.2.1 Experimental conditions and approaches

The average temperature through the heat exchanger was set from 60°C – 90°C. The flow rate for the shell-side fluid was easily adjusted through the operating range of the pump. The intermediate and tube-side fluids (water) were not as easy to change. Relying on building supply made the design of the updates easy however, the supplied fluid flow, temperature, and pressure can fluctuate to some degree without notice. Globe valves on these streams were generally able to throttle flow to the desired flow rate. Inlet temperatures were out of our control.

This work is interested in the effects of the intermediate fluid on the effectiveness of the heat exchanger as well as validating a tool for determining the temperature distributions of each of the three fluids for predicting future performance of three-fluid heat exchangers. As such, the approach for experimental conditions were two-fold. The first, was to slow flow rates for the shell-side and tube-side fluids to achieve a temperature difference of 10°C between each fluid’s inlet and outlet temperature. This was done while varying the flow rate of the intermediate fluid. This method was chosen because a larger temperature difference gives more confidence in heat transfer results. The second method used is set the annular fluid as target fluid to investigate and the tube-side and shell-side fluids were running as fast as possible to minimize their resistance to heat transfer.

It is important to note that as testing went on, the experimental approach was changed due to a number of circumstances. The first change was due to relying on the building chilled water supply. Though typically not used for experimental work, exceptions are often made to use this water supply. While known that the system was intended for use in building cooling at the university, it was unknown at the time of experimental design that the university reduces the pressure differential of the supply during the winter. This reduced the maximum flow rate of the supply by half. While changing experimental conditions during the experiment is not ideal, this change should have little impact on the goals of this experiment.

A final change needed to be applied due to the nature of the experimental system. The double-wall test section was added to the secondary side of the HTF. This meant that in order for heat to reach the test section, it must first be added to the primary side fluid through the electrical resistance heater and then must be transferred to the secondary side through the primary heat exchanger. Due to limitations in this heat exchanger and losses through the insulation on the primary and secondary sides, the maximum heat transfer to the secondary side was significantly smaller than the facility is rated for. The primary and secondary loops were insulated in 1” thick glass insulation. The additions for accommodating the double-wall heat exchanger increased the thickness of insulation to 2” to reduce facility heat loss. In tests with high shell-side flow rate and high temperature set point with high water annular flow rate, heat lost to the chilled water and annular water were greater than could be provided through the primary heat exchanger. This led to the system never reaching the testing setpoint. In these cases, the shell-side flow rate was reduced increasing the resistance to heat transfer in the heat exchanger. These cases were significantly more prominent when the twisted annular tube was tested thus qualitatively giving us results that the twisted tube transferred significantly more heat for the same experimental conditions.

The approach for determining fluid flow direction used the 3FHE solver developed at the University of New Mexico (UNM) and provided in Appendix II-A. This solver is based off the theory found in Sokolich [25]. There are four possible flow configurations, shown in Figure II-
21, in a heat exchanger of this type. The chosen configuration is that of P3 and was chosen for two reasons. The first is that ultimately from fluid 1 to fluid 3 the heat exchanger operates in a counterflow configuration from heat source to sink. The second is that this configuration avoids a phenomenon known as temperature cross. In two-fluid exchangers this is defined to exist when the cold fluid outlet temperature is hotter than the hot fluid outlet temperature. The hot fluid is then gaining heat after the temperature cross resulting in wasted heat transfer area. The phenomenon can also happen in a three-fluid heat exchanger where the hot fluid directly transfers heat to either one or both cold fluids.

![Diagram](image)

Figure II-21. Schematic diagram of the modified HTF including the single assembly DWHX [4]

II3.2.2 Experimental results

This section shows the results of the single assembly double wall plain tube (SADWPT) and single assembly double wall twisted tube (SADWTT) heat exchangers. The results shown are based on the effectiveness of the heat exchanger. The effectiveness is heavily influenced by the outlet temperatures of the three fluids. As these fluid temperatures approach each other, the effectiveness itself will increase. Therefore, for tests utilizing the first method of experimentation with slow flow rates in the shell-side and tube-side fluids, the absolute magnitudes shown will be higher than those of the tests using the other method. The focus of this work will be on tubular heat exchangers with two thermal communications.

Figures II-22 through II-24 show the results of the plain tube tests with water as the intermediate fluid. Figure II-22 shows the effect that the change in the flow rate of the intermediate fluid has on the overall effectiveness of the heat exchanger. Similarly, in Figures II-23 and II-24 the impact of the change in flow rate on the temperature effectiveness of the intermediate and secondary fluids is shown. Figures II-25 through II-27 show the similar results of the twisted tube with water as the intermediate fluid.
Figure II-22. The overall effectiveness of the plain tube heat exchanger decreases as the flow of water in the annular region increases [4]

Figure II-23. The temperature effectiveness of fluid 2 decreases as the flow of water in the annular increases [4]
In the studied region, the temperature effectiveness of the third fluid has little response to changes in the flow of the annular water [4].

Effectiveness of the Twisted tube heat exchanger also decreased as the flow of water increased [4].

---

Figure II-24. In the studied region, the temperature effectiveness of the third fluid has little response to changes in the flow of the annular water [4].

Figure II-25. Effectiveness of the Twisted tube heat exchanger also decreased as the flow of water increased [4].
Figure II-26. The temperature effectiveness of the fluid 3 has a small response, a slight decrease in the effectiveness as the flow of water in the annular region increases [4]

Figure II-27. As expected, the temperature effectiveness of fluid 2 decreases as the flow increases [4]
III.2.3 Tool Validation

Three experimental points were chosen from the water tests performed with the plain tube heat exchanger. The testing conditions were then replicated in the 3FHE code. The three testing conditions are displayed in Table II-1. The resulting temperature distributions are shown in Figures II-28-III-30. Tables II-2 through II-4 show the comparison of the results for the experimental tests as well as those which were predicted with the energy balance equation.

Table II-1. Experimental parameters chosen for comparing predicted and measured values [4]

<table>
<thead>
<tr>
<th></th>
<th>Shell-Side Flow Rate (GPM)</th>
<th>Intermediate Flow Rate (GPM)</th>
<th>Tube-Side Flow Rate (GPM)</th>
<th>Shell-Side Inlet Temperature °C</th>
<th>Intermediate Inlet Temperature °C</th>
<th>Tube-Side Inlet Temperature °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>2.08 ± 0.02</td>
<td>4.79 ± 0.4</td>
<td>0.292 ± 0.2</td>
<td>87.8 ± 0.2</td>
<td>25.6 ± 0.2</td>
<td>8.05 ± 0.2</td>
</tr>
<tr>
<td>Case 2</td>
<td>9.66 ± 0.05</td>
<td>1.77 ± 0.4</td>
<td>3.04 ± 0.1</td>
<td>64.5 ± 0.2</td>
<td>25.8 ± 0.2</td>
<td>61.5 ± 0.2</td>
</tr>
<tr>
<td>Case 3</td>
<td>6.35 ± 0.05</td>
<td>2.34 ± 0.4</td>
<td>2.78 ± 0.1</td>
<td>82.7 ± 0.2</td>
<td>28.8 ± 0.2</td>
<td>5.97 ± 0.2</td>
</tr>
</tbody>
</table>

Table II-2. Comparison between measured and predicted values for Case 1

<table>
<thead>
<tr>
<th>Case 1</th>
<th>Shell-Side Outlet Temperature</th>
<th>Intermediate Outlet Temperature</th>
<th>Tube-Side Outlet Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured Values</td>
<td>72.5 ± 0.2</td>
<td>27.9 ± 0.2</td>
<td>18.9 ± 0.2</td>
</tr>
<tr>
<td>Predicted Values</td>
<td>76.6</td>
<td>27.2</td>
<td>15.5</td>
</tr>
<tr>
<td>Percent Difference</td>
<td>5.7</td>
<td>2.5</td>
<td>18</td>
</tr>
</tbody>
</table>

Figure II-28. Predicted temperature distribution for Case 1 [4]
Figure II-29. Predicted temperature distribution for Case 2 [4]

Figure II-30. Predicted temperature distribution for Case 3 [4]

Table II-3. Comparison between measured and predicted values for Case 2 [4]

<table>
<thead>
<tr>
<th>Case 2</th>
<th>Shell-Side Outlet Temperature</th>
<th>Intermediate Outlet Temperature</th>
<th>Tube-Side Outlet Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured Values</td>
<td>58.6±0.2</td>
<td>28.0±0.2</td>
<td>9.01±0.2</td>
</tr>
<tr>
<td>Predicted Values</td>
<td>56.8</td>
<td>32.9</td>
<td>8.3</td>
</tr>
<tr>
<td>Percent Difference</td>
<td>3</td>
<td>18</td>
<td>7.9</td>
</tr>
</tbody>
</table>
Table II-4. Comparison between measured and predicted values for Case 3 [4]

<table>
<thead>
<tr>
<th>Case 3</th>
<th>Shell-Side Outlet Temperature</th>
<th>Intermediate Outlet Temperature</th>
<th>Tube-Side Outlet Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured Values</td>
<td>77.0±0.2</td>
<td>31.3±0.2</td>
<td>9.53±0.2</td>
</tr>
<tr>
<td>Predicted Values</td>
<td>74.4</td>
<td>34.2</td>
<td>9.4</td>
</tr>
<tr>
<td>Percent Difference</td>
<td>3.3</td>
<td>9.2</td>
<td>1.4</td>
</tr>
</tbody>
</table>

As shown in Figure II-22 there is a clear trend in the overall effectiveness of the plain tube heat exchanger. As the flow of water in the annular space increases, the overall effectiveness of the heat exchanger decreases. Qualitatively, this phenomenon can be attributed to the increasing rate of the intermediate fluid removing more heat than it is passing on to the tube-side fluid. Given the limitations of our facility, we cannot slow the intermediate water any more with high confidence in our flow measurement to find a possible maximum point of effectiveness. Extra experiments were conducted replacing the intermediate water flow with air as shown in Figure II-31. In examining the figure, the opposite trend appears in the overall effectiveness. As the flow of air increases, the overall effectiveness increases as well. It appears that with a greater convective heat transfer coefficient, the air is better able to transfer heat to the tube-side fluid. The question then becomes, is there a point at which the air begins to act like the water in the annular region and the overall effectiveness will begin to decrease as the flow rate continues to increase.

![Overall Effectiveness vs. Fluid 2 Re](image)

Figure II-31. Overall effectiveness of the plain tube heat exchanger increases as the flow of air in the annular region increases [4]

Figure II-31 also shows the experiments for the twisted annular tube. Here we see a similar trend where the effectiveness decreases as the flow rate increases. However, looking at the absolute magnitude of the effectiveness, it is clear that the twisted tube increases the effectiveness of the heat exchanger when compared to the plain tube under similar experimental conditions.
conditions. As seen in Tables 2 through 4 above, the predicted method performed best in calculating the outlet temperature for the shell-side fluid. Predicting the outlet temperatures for the intermediate and tube-side fluids was harder, averaging 10 and 9 percent off from the actual values respectively. These high errors are notably due to the difficulty in predicting heat transfer coefficients of fluids. In a guess-and-check estimate to minimize the difference between calculated and measured values, it was found that given more accurate prediction for the fluids resulted in significantly more accurate results, under a 5 percent difference from measured values.

It is clear from the experiments that the twisted tube has shown improvement in the overall effectiveness over plain tube triple flow heat exchanger. Unfortunately, the test section was designed and assembled such that is it hard to measure the tube surface temperature during the experiment. The availability of the surface, and fluids inlet and outlet temperatures would be useful data to calculate the heat transfer coefficient and hence the Nusselt number. Currently, the experimentally measured temperature data for the inlet and outlet fluids are used to validate ongoing numerical analysis using Star CCM+. After validating the simulation, the surface temperature can be backed up, and all heat transfer parameters can be directly determined. Also, the validated numerical analysis can be extended to cover a wide range of flow rates and other parameters for providing better information regarding the twisted tube heat exchanger performance.
II3.3 SNL SCO2 Tests and Results

This section summarizes key progress to date toward building Dowtherm A, a simulant thermal oil to supercritical carbon dioxide (SCO2) heat transfer loop and collecting performance data using an existing double-walled twisted tube heat exchanger from the University of New Mexico (UNM). This report expands on previous content of UNM-millstone report “Low-Pressure Small-Scale Testing” in support of this NEUP.

II3.3.1 Heat exchanger modifications

The existing heat exchanger provided by HIPEX as shown in Figure II-32 features three concentric tubes with two headers on both ends. Nozzles 5 and 6 provide Dowtherm flow between the shell and the external surface of a twisted tube (shell side). Nozzles 3 and 4 provide flow between the interior of the twisted tube and the external surface of a central tube for the intermediate fluid (Annular flow). Nozzles 1 and 2 provide flow straight through the central circular tube in the heat exchanger (inner flow) and was originally designed for low-pressure water testing at UNM. In order to operate at the high pressures required for SCO2 the central tube needed to be modified or replaced. The original header, shown in Figure II-33, separation between the outer Dowtherm, intermediate water, and central water flow paths. This central tube was replaced with a fully welded high-pressure tube which bypassed the intermediate and central fluid o-ring seals and was instead sealed by a tube feedthrough on the outlet of the header as shown. In order to maintain the geometry of the original tube a 1” OD tube was used through the heat exchanger and welded to sections of ¾” tube which could pass through the existing header nozzles without modification. A second feedthrough was used on this new ¾” nozzle to connect again into 1” OD high-pressure tube used for the main SCO2 system as shown in Figure II-34.

The complete modified assembly, shown in exploded and detail views of Figure II-35, is able to be assembled and disassembled in-place without any modification to the existing tubes or headers. The outer main tube is first connected to one of the headers in order to provide alignment for the remaining two tubes. These tubes are slid through the first header and finally capped by the second header, with external nozzle connections completed to the rest of the system. The inner tube was replaced by a high-pressure tube to handle the SCO2 flow as shown in Figure II-36.
Figure II-32. A nozzle drawing of the existing HIPEX double-walled twisted tube heat exchanger provided by UNM.
Figure II-33. An image and cross-sectional diagram of the original HIPEX heat exchanger header.
Figure II-34. A cross-sectional diagram of the modified HIPEX heat exchanger header with a replacement high-pressure tube and feedthrough arrangement to accommodate S-CO2 flow.
Figure II-35. An expanded and exploded view of the modified HIPEX heat exchanger tube and header assembly.
Figure II-36. A drawing of the high-pressure SCO2 tube.
II3.3.2 Loop modification and setup

A Miller ProHeat 35 35 kW induction heating power supply powers a 50 ft water-cooled induction heating power cable wrapped tightly around the 410 ferritic stainless-steel heating tube in order to transfer heat into the Dowtherm. This option avoids costly, large, and leak-prone electrical immersion heaters with minimal thermal mass in order to increase system responsiveness. The power supply can operate in either direct manual heating power control mode or temperature setpoint-based control mode.

A close-coupled intermediate water loop was installed including a Goulds Model NPE stainless steel ½ hp water pump (1ST1C1E3 L96) to provide intermediate water flow to the heat exchanger as shown with an elevated expansion tank in order to allow the intermediate water loop temperature to float up and down with the Dowtherm loop temperature and avoid parasitic heat loss to the intermediate water during steady-state operation. The expansion tank is open to atmospheric pressure in order to avoid additional pressure on the water system. Bypass and throttle valves are used with the constant speed pump to provide flow control.

The SCO2 system consists of a Haskel ALG-32 air-driven gas booster pump designed to handle two-phase carbon dioxide for filling from CO2 syphon-tube bottles, a Parker Autoclave 1.5 hp Magnepump magnetically coupled high-pressure liquid circulation pump for flow circulation, several commissioning filters, and a Sentry model 12S3DC13-EW27-UCX spiral-tube and shell heat exchanger for water cooling. A close-coupled bypass valve is used in combination with a variable-frequency drive to provide for flow control.

Finally, a TrueTon 750 AC packaged chiller unit is used to provide both water flow and cooling to the SCO2 loop with ambient air as the ultimate heat sink. This unit features temperature limit control to maintain the SCO2 pump inlet temperature along with bypass and throttle valves to provide flow control as shown in Figure II-37 and II-38.

Figure II-37. Photograph of the Dowtherm to SCO2 heat transfer test rig
Figure II-38. The process flow diagram of the Dowtherm to SCO2 heat transfer test rig
### II3.3.3 Instrumentation and control

A custom control system was designed and built in National Instruments LabView programming environment with the user interface shown in Figure II-39. This system provides integrated control for the Dowtherm, water, and SCO2 subsystems with user-specified data acquisition rates, live feedback of key instrumentation, and trend plotting of all signals. A listing of the instrumentation installed, range, and uncertainties is shown in Table II-5.

Table II-5. Instrumentation listing with uncertainties.

<table>
<thead>
<tr>
<th>UUID</th>
<th>Eng Min</th>
<th>Eng Max</th>
<th>Eng Units</th>
<th>Model</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>CO2 In P</td>
<td>0</td>
<td>3000</td>
<td>psi</td>
<td>Honeywell FP2000</td>
<td>7.5</td>
</tr>
<tr>
<td>CO2 Out P</td>
<td>0</td>
<td>3000</td>
<td>psi</td>
<td>Honeywell FP2000</td>
<td>7.5</td>
</tr>
<tr>
<td>Oil In P</td>
<td>0</td>
<td>1000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>10</td>
</tr>
<tr>
<td>Oil Out P</td>
<td>0</td>
<td>1000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>10</td>
</tr>
<tr>
<td>Water In P</td>
<td>0</td>
<td>200</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>2</td>
</tr>
<tr>
<td>Water Out P</td>
<td>0</td>
<td>1000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>10</td>
</tr>
<tr>
<td>Oil Flow</td>
<td>0</td>
<td>20.4</td>
<td>gpm</td>
<td>Sur-Flo SF1015 (999.777 k-factor)</td>
<td>0.204</td>
</tr>
<tr>
<td>Water Flow</td>
<td>0</td>
<td>50</td>
<td>gpm</td>
<td>GPI (724 k-factor)</td>
<td>0.5</td>
</tr>
<tr>
<td>Oil Pump In P</td>
<td>0</td>
<td>1000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>10</td>
</tr>
<tr>
<td>Oil Pump Out P</td>
<td>0</td>
<td>1000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>10</td>
</tr>
<tr>
<td>CO2 Pump In P</td>
<td>0</td>
<td>3000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>30</td>
</tr>
<tr>
<td>CO2 Pump Out P</td>
<td>0</td>
<td>3000</td>
<td>psi</td>
<td>WIKA Model A-10</td>
<td>30</td>
</tr>
<tr>
<td>CO2 Flow</td>
<td>-10</td>
<td>10</td>
<td>lbm/s</td>
<td>Emerson F050P Coriolis Flowmeter</td>
<td>0.05</td>
</tr>
<tr>
<td>CO2 Density</td>
<td>0</td>
<td>60</td>
<td>lbm/ft^3</td>
<td>Emerson F050P Coriolis Flowmeter</td>
<td>0.125</td>
</tr>
<tr>
<td>Oil Pump Out T</td>
<td>-22</td>
<td>622</td>
<td>F</td>
<td>100 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>Oil Pump In T</td>
<td>-22</td>
<td>622</td>
<td>F</td>
<td>100 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>Oil Out T</td>
<td>-22</td>
<td>622</td>
<td>F</td>
<td>100 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>CO2 Pump Out T</td>
<td>-22</td>
<td>622</td>
<td>F</td>
<td>100 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>CO2 Pump In T</td>
<td>-22</td>
<td>622</td>
<td>F</td>
<td>100 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>Water Out T</td>
<td>-58</td>
<td>932</td>
<td>F</td>
<td>1000 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>Water In T</td>
<td>-58</td>
<td>932</td>
<td>F</td>
<td>1000 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>CO2 Out T</td>
<td>-58</td>
<td>932</td>
<td>F</td>
<td>1000 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
<tr>
<td>CO2 In T</td>
<td>-58</td>
<td>932</td>
<td>F</td>
<td>1000 Ohm Omega (Class A)</td>
<td>0.15</td>
</tr>
</tbody>
</table>
Figure II-39. The user interface for the Dowtherm to sCO2 heat transfer test apparatus.
At the start of a test campaign the sCO2 loop is first filled from a syphon tube CO2 bottle. Liquid CO2 is injected from the syphon tube bottle into the sCO2-side of the water cooler with the water running at full capacity in order to act as an ambient temperature vaporizer and reduce cold shock to the system. Once bottle pressure is achieved the gas booster pump is used transfer more liquid CO2 into the loop and pressurize the system until the desired sCO2 pressure is reached. Once at pressure the circulation pump provides sCO2 flow with both speed and bypass control to set the sCO2 flow rate.

Next, the Dowtherm system is pressurized by nitrogen ullage gas to fill the elevated tubing uniformly and avoid entrained air that could bias the measured heat transfer data and damage the pump. With both the flow and bypass valves open the nitrogen ullage gas pressure in the reservoir is slowly increased through the open oil lift gas valve with the oil lift drain valve open to monitor for oil. Once the loop is completely filled with Dowtherm it will overflow through the oil lift drain valve and the valve will be shut. Gas pressure will continue to be increased until the system pressure reaches approximately 30 psi in order to avoid cavitation in the pump.

Once filled and pressurized the pump can be started to provide Dowtherm flow with both speed, bypass, and throttle control to set the flow rate. The induction heating power supply is then activated in direct manual power control to heat up the Dowtherm to a temperature near the operating point desired, and then switched over to temperature setpoint control in order to maintain a desired oil inlet temperature to the HIPEX heat exchanger at a given flow rate. Finally, the intermediate water system pump is started with a minimal flow rate in order to provide a better thermal connection between the Dowtherm and sCO2 systems while also maintaining the largest temperature difference between water inlet and outlet temperatures possible in order to better assess parasitic heating from the water pump and heat loss from the Dowtherm loop to the water.

Under normal operation an induction heater on the Dowtherm system is used to achieve a fixed Dowtherm inlet temperature at a given flow rate while the chiller maintains a fixed sCO2 circulation pump inlet temperature and therefore heat exchanger inlet temperature. Speed, throttle, and bypass control on the Dowtherm and sCO2 systems are then used to reach a steady-state heat transfer condition and complete a single measurement. After each measurement the flow rates can be varied to vary the capacitance rate and reach different measurement conditions with the induction heating power supply and chiller scaling their thermal duty to match the test conditions. This operational approach reduces the difference in thermal mass between each dataset so that steady-state conditions can be achieved as quickly as possible.

II3.3.4 Shakedown testing

Shakedown testing was completed for the modified sCO2 system and the oil system as shown in Figures II-40 through II-43 in order to verify leak-tightness and the achievable head and flow range of both systems.

The sCO2 system was held at pressure with a leakage rate of approximately 3 psi/min, while the oil system displayed a leakage rate of less than 0.1 psi/min. The sCO2-side leakage rate is higher than desired, so after inspection of potential leakage sources it is suspected that the outlet check valve on the gas booster used to fill the loop does not seal properly. A secondary check valve with an elastomeric seal was installed at this location in order to reduce the leakage rate and avoid the need to refill the system during a test campaign. Subsequent tests have showed no appreciable pressure decay over time.
The sCO2 and oil system pumps both displayed acceptable head rise capability while mapping the system loss curves, but the sCO2 pump flow rate was artificially reduced due to the addition of an outlet filter assembly. This filtration is required in order to remove graphite dust generated during pump commissioning and bearing wear-in by the bearing which could contaminate downstream tubing and affect heat transfer measurements. If sufficient run-time is reached on the pump this filter assembly can be removed in order to increase the flow rate range.

![sCO2 System Pressure Decay Test](image1)

**Figure II-40. sCO2 system pressure decay test.**

![Oil System Pressure Decay Test](image2)

**Figure II-41. Oil system pressure decay test.**

![sCO2 System Pump Performance Test](image3)

**Figure II-42. sCO2 system pump performance test.**
II3.3.5 HX approximate model and sensitivity

In order to guide data collection and understand the sensitivity of the heat exchanger performance to operating conditions a simplified 0-dimensional model of the HIPEX heat exchanger was created in Engineering Equation Solver (EES). This model uses the as-built heat exchanger geometry and internal correlations within EES based on inlet fluid properties to calculate a series of thermal resistances from the Dowtherm through the tube walls, water, and into the CO2 in order to estimate the overall conductance-area product “UA,” the heat exchanger effectiveness, and the duty assuming pure counter-flow operation. Note that the only correlation available for annular flow heat transfer in EES assumes an adiabatic outer wall, so the heat transfer coefficient at the outer wall of the intermediate water flow passage is assumed equal to that at the inner wall.

Using the provided instrument uncertainties and nominal operating conditions at low and high flow rates the propagated uncertainty of relevant performance metrics for the heat exchanger can be calculated as shown in Table II-6. This sensitivity study demonstrates that the expected propagated uncertainty estimated for UA, effectiveness, and duty are all on the order of 1% or less, with a primary dependence on the uncertainty in the oil flow rate at low flows. Outlet temperatures also show a propagated uncertainty on the order of the instrumentation uncertainty suggesting that performance should be well-measured using the installed instrumentation.

While the previous sensitivity study accounts for measurement uncertainty, there can also be sensitivity in fluid properties within the temperature range of operation. Dowtherm and water fluid properties both have minimal sensitivity to both temperature and pressure so the operating temperature and pressure of each of these subsystems is not expected to be as important as the temperature difference and flow rate of each subsystem.
Table II-6. Propagated uncertainties using a simplified heat exchanger model at low and high flow.

<table>
<thead>
<tr>
<th>Variable</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\dot{q}$</td>
<td>$1886 \pm 15.17$ [W]</td>
</tr>
<tr>
<td>$m_{\text{CO2,LBM}}$</td>
<td>$0.16 \pm 0.0008$ [lbm/s]</td>
</tr>
<tr>
<td>$P_{\text{ambient}}$</td>
<td>$84116 \pm 2$ [Pa]</td>
</tr>
<tr>
<td>$T_{\text{CO2,in}}$</td>
<td>$299.8 \pm 0.15$ [K]</td>
</tr>
<tr>
<td>$T_{\text{el,in}}$</td>
<td>$349.8 \pm 0.15$ [K]</td>
</tr>
<tr>
<td>$T_{\text{water}}$</td>
<td>$327.6 \pm 0.15$ [K]</td>
</tr>
<tr>
<td>$V_{\text{oil,GPM}}$</td>
<td>$4.2 \pm 0.204$ [gpm]</td>
</tr>
<tr>
<td>$V_{\text{water}}$</td>
<td>$0.0001893 \pm 0.5$ [m$^3$/s]</td>
</tr>
</tbody>
</table>

Partial derivative

<table>
<thead>
<tr>
<th>Variable</th>
<th>Uncertainty</th>
<th>% of uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\frac{\partial}{\partial m_{\text{CO2,LBM}}}$</td>
<td>$-0.8358$</td>
<td>20.04%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial P_{\text{ambient}}}$</td>
<td>$1.077 \pm 28$</td>
<td>0.00%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial T_{\text{CO2,in}}}$</td>
<td>$-0.003081$</td>
<td>9.57%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial T_{\text{el,in}}}$</td>
<td>$0.0002571$</td>
<td>0.00%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial T_{\text{water}}}$</td>
<td>$0.0005607$</td>
<td>0.32%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial V_{\text{oil,GPM}}}$</td>
<td>$0.006129$</td>
<td>70.07%</td>
</tr>
<tr>
<td>$\frac{\partial}{\partial V_{\text{water}}}$</td>
<td>$0$</td>
<td>0.00%</td>
</tr>
</tbody>
</table>

- $T_{\text{CO2,out}}$ = $307.9 \pm 0.118$ [K]
- $T_{\text{el,out}}$ = $345.8 \pm 0.2185$ [K]
- $UA$ = $43.45 \pm 0.257$ [W/K]
<table>
<thead>
<tr>
<th>Variable</th>
<th>Uncertainty</th>
<th>Partial derivative</th>
<th>% of uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>$\frac{\partial}{\partial \text{Vol}_{oil,GPM}}$</td>
<td>11.3±0.204 [gpm]</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\frac{\partial}{\partial \text{Vol}_{oil,GPM}}$</td>
<td>0.0001893±0.5 [m³/s]</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
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<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
<tr>
<td>$\dot{q}$</td>
<td>$n_{\text{CO}_2,\text{LBM}} = 0.56\pm0.0028$ [lbm/s]</td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>943.3</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>1.60 %</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$n_{\text{CO}_2,\text{LBM}}$</td>
<td>0.00 %</td>
</tr>
</tbody>
</table>
SCO2 however does have significant sensitivity in fluid properties with increasing sensitivity as the pressure becomes closer to the critical pressure of 1074 psia. This confounding effect is reduced by using a high-accuracy Coriolis flow meter to measure mass flow rate, but in order to reduce the sensitivity of the specific heat capacity to temperature an appropriate operating pressure must be chosen between the critical pressure of 1074 psia and the maximum operating pressure of the loop of 2000 psig. Contours of SCO2 specific heat capacity for several pressures are shown in Figure II-44, with a minimum reasonable operating pressure of 1400 psia in order to avoid high temperature sensitivity. Higher pressures would also be acceptable, but due to small inventory of system the operating pressure is highly sensitive to operating temperature and sudden changes in heat load may produce sudden increases in system pressure.  

![Figure II-44](influence_of_pressure_on_cp.png)

Figure II-44. Influence of operating pressure on the sensitivity of SCO2 specific heat capacity to temperature.  

This simplified model was also used generate approximate contours of constant UA and duty performance for the HIPEX heat exchanger in order to guide operating flow rates for the Dowtherm and SCO2 subsystems as shown in Figure II-45. A clear trend is present demonstrating that a range of performance can be achieved by steadily increasing both flow rates, with optimal conditions expected when the ratio of the oil to SCO2 flow rate is kept similar and off-optimal conditions where one of the flow rates is much higher than the other.
II3.3.6 Test campaign

A test campaign was executed to collect data over a range of different Dowtherm and sCO2 flow rate combinations at fixed inlet temperatures based on the sensitivity of HIPEX heat exchanger performance metrics to different operating conditions investigated using the simplified heat exchanger model. Three different Dowtherm flow rates and four sCO2 flow rates were selected based on repeatable manual control positions as shown in Tables II-7 and II-8. This matrix of 12 data points was acquired at a rate of approximately one data set every 30 minutes, with 20 minutes required to transition from different steady-state operating conditions and approximately 10 minutes required to collect data over the cycling period of the chiller control system so that representative average values could be calculated. Data from each test was averaged over a complete period of the chiller control system as summarized in Tables II-9 through II-12. Pressures
measured as gage were zeroed and corrected for barometric pressure using local weather data averaged over the test period.

Table II-7. Repeatable Dowtherm flow rates using manual controls.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>0</td>
<td>0</td>
<td>4.2</td>
</tr>
<tr>
<td>40</td>
<td>90</td>
<td>0</td>
<td>7.4</td>
</tr>
<tr>
<td>60</td>
<td>90</td>
<td>0</td>
<td>11.2</td>
</tr>
</tbody>
</table>

Table II-8. Repeatable sCO2 flow rates using manual controls.

<table>
<thead>
<tr>
<th>CO2 Pump Speed [Hz]</th>
<th>Recirc Closure [deg]</th>
<th>CO2 Flow [lbm/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>60</td>
<td>0</td>
<td>0.16</td>
</tr>
<tr>
<td>60</td>
<td>30</td>
<td>0.3</td>
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<tr>
<td>60</td>
<td>90</td>
<td>0.44</td>
</tr>
<tr>
<td>75</td>
<td>90</td>
<td>0.56</td>
</tr>
</tbody>
</table>

Table II-9. Measured temperatures for each data run.

<table>
<thead>
<tr>
<th></th>
<th></th>
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<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>110.57</td>
<td>111.47</td>
<td>87.27</td>
<td>80.90</td>
<td>133.79</td>
<td>141.70</td>
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<tr>
<td>109.06</td>
<td>109.98</td>
<td>86.05</td>
<td>80.92</td>
<td>133.07</td>
<td>141.18</td>
</tr>
<tr>
<td>114.57</td>
<td>115.46</td>
<td>87.45</td>
<td>81.28</td>
<td>140.22</td>
<td>146.03</td>
</tr>
<tr>
<td>119.68</td>
<td>120.49</td>
<td>88.57</td>
<td>81.36</td>
<td>146.60</td>
<td>151.26</td>
</tr>
<tr>
<td>121.40</td>
<td>122.19</td>
<td>89.91</td>
<td>80.92</td>
<td>147.33</td>
<td>151.83</td>
</tr>
<tr>
<td>116.79</td>
<td>117.74</td>
<td>88.60</td>
<td>81.01</td>
<td>141.77</td>
<td>147.31</td>
</tr>
<tr>
<td>120.39</td>
<td>121.31</td>
<td>91.39</td>
<td>81.04</td>
<td>143.79</td>
<td>149.05</td>
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<tr>
<td>129.52</td>
<td>130.36</td>
<td>98.63</td>
<td>82.62</td>
<td>148.44</td>
<td>152.84</td>
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<tr>
<td>115.31</td>
<td>116.36</td>
<td>87.43</td>
<td>77.66</td>
<td>140.75</td>
<td>149.24</td>
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<tr>
<td>128.29</td>
<td>129.21</td>
<td>96.03</td>
<td>78.97</td>
<td>149.53</td>
<td>156.93</td>
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<td>142.59</td>
<td>101.92</td>
<td>79.33</td>
<td>164.20</td>
<td>168.24</td>
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<td>129.54</td>
<td>130.43</td>
<td>92.14</td>
<td>78.10</td>
<td>157.57</td>
<td>162.40</td>
</tr>
</tbody>
</table>

Table II-10. Measured pressures for each data run.

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<td>19</td>
<td>4</td>
<td>6</td>
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<tr>
<td>1622</td>
<td>1613</td>
<td>12</td>
<td>17</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
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<td>1445</td>
<td>14</td>
<td>18</td>
<td>4</td>
<td>6</td>
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</table>
Table II-11. Zeroed and scaled pressures for each data run.

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<td>14</td>
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<tr>
<td>1449</td>
<td>1443</td>
<td>29</td>
<td>32</td>
<td>15</td>
<td>14</td>
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<td>1433</td>
<td>1427</td>
<td>27</td>
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<td>15</td>
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<td>1475</td>
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<td>33</td>
<td>15</td>
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<td>31</td>
<td>15</td>
<td>14</td>
</tr>
<tr>
<td>1629</td>
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<td>29</td>
<td>15</td>
<td>14</td>
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<td>1462</td>
<td>1456</td>
<td>27</td>
<td>31</td>
<td>15</td>
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</tbody>
</table>

Table II-12. Measured flow rates for each data run.

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<thead>
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<th></th>
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</thead>
<tbody>
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<td>4.16</td>
<td>3.32</td>
<td>0.43</td>
</tr>
<tr>
<td>4.16</td>
<td>3.33</td>
<td>0.56</td>
</tr>
<tr>
<td>7.42</td>
<td>3.35</td>
<td>0.56</td>
</tr>
<tr>
<td>11.32</td>
<td>3.32</td>
<td>0.56</td>
</tr>
<tr>
<td>11.33</td>
<td>3.31</td>
<td>0.44</td>
</tr>
<tr>
<td>7.42</td>
<td>2.83</td>
<td>0.44</td>
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<tr>
<td>7.43</td>
<td>2.83</td>
<td>0.31</td>
</tr>
<tr>
<td>7.44</td>
<td>2.88</td>
<td>0.15</td>
</tr>
<tr>
<td>4.12</td>
<td>3.04</td>
<td>0.31</td>
</tr>
<tr>
<td>4.15</td>
<td>2.97</td>
<td>0.15</td>
</tr>
<tr>
<td>11.26</td>
<td>2.86</td>
<td>0.15</td>
</tr>
<tr>
<td>11.26</td>
<td>2.99</td>
<td>0.30</td>
</tr>
</tbody>
</table>

II3.3.6 Data reduction

The simplified heat exchanger model can be used to verify that the actual HIPEX heat exchanger is achieving higher performance than would be expected from a plain tube heat exchanger of the same construction. Using this model and the data presented a comparison between the thermal duty calculated from measured data (“Measured Thermal Duty [W]”) and the thermal duty calculated from the simplified model (“Calculated Thermal Duty [W]”) can be made as shown in Table II-13 and Figure II-46. These results show that the measured thermal duty is consistently 7 to 20% higher than would be expected based on the simplified model, with performance at least 10% higher given the propagated uncertainties from the instrumentation used.
Table II-13. Calculated performance results and comparison with the simplified heat exchanger model.

<table>
<thead>
<tr>
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</tr>
</thead>
<tbody>
<tr>
<td>1935±111</td>
<td>1540±91.12</td>
<td>49.95±3.167</td>
<td>0.1002±0.007128</td>
</tr>
<tr>
<td>1984±113.1</td>
<td>1566±94.47</td>
<td>50.87±3.297</td>
<td>0.1029±0.0074</td>
</tr>
<tr>
<td>2536±124.1</td>
<td>2050±147.1</td>
<td>61.35±4.68</td>
<td>0.07134±0.006348</td>
</tr>
<tr>
<td>3217±171.6</td>
<td>2709±220.7</td>
<td>75.2±6.553</td>
<td>0.08846±0.009761</td>
</tr>
<tr>
<td>3142±171.2</td>
<td>2643±209.6</td>
<td>73.16±6.207</td>
<td>0.1117±0.01352</td>
</tr>
<tr>
<td>2418±122.3</td>
<td>1923±176.8</td>
<td>56.44±5.545</td>
<td>0.0859±0.01148</td>
</tr>
<tr>
<td>2298±123</td>
<td>1917±171.1</td>
<td>55.92±5.304</td>
<td>0.1228±0.01925</td>
</tr>
<tr>
<td>2055±120.1</td>
<td>1709±217.5</td>
<td>50.81±5.612</td>
<td>0.2167±0.0532</td>
</tr>
<tr>
<td>2058±117.6</td>
<td>1685±113.5</td>
<td>46.9±3.342</td>
<td>0.0997±0.01393</td>
</tr>
<tr>
<td>1936±113</td>
<td>1606±162.6</td>
<td>42.86±3.736</td>
<td>0.1864±0.04608</td>
</tr>
<tr>
<td>2861±172.5</td>
<td>2441±338.4</td>
<td>58.5±6.973</td>
<td>0.261±0.05681</td>
</tr>
<tr>
<td>3425±176.1</td>
<td>2837±254.7</td>
<td>67.42±6.374</td>
<td>0.152±0.02283</td>
</tr>
</tbody>
</table>

Figure II-46. Comparison of measured and calculated thermal duty using a simplified model for the HIPEX heat exchanger transferring heat from Dowtherm A to sCO2 with an intermediate flowing water gap.

Comparisons between the measured and calculated outlet temperatures for the Dowtherm and SC02 sides of the heat exchanger are also shown in Table II-14. The measured temperatures are generally within the error bands of the calculated values due to the large amount of propagated uncertainty in this metric and the difference between the higher performance of the HIPEX heat exchanger and what would be expected using the simplified plain tube mode.
Table II-14. Calculated temperature results and comparison with the simplified heat exchanger model.

<table>
<thead>
<tr>
<th>Measured $T_{oil, out}$ [K]</th>
<th>Calculated $T_{oil, out}$ [K]</th>
<th>Measured $T_{CO2, out}$ [K]</th>
<th>Calculated $T_{CO2, out}$ [K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>329.7±0.083</td>
<td>330.7±0.2523</td>
<td>303.9±0.083</td>
<td>302.7±0.2795</td>
</tr>
<tr>
<td>329.3±0.083</td>
<td>330.4±0.259</td>
<td>303.2±0.083</td>
<td>302.2±0.197</td>
</tr>
<tr>
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<td>334.1±0.1867</td>
<td>304±0.083</td>
<td>303±0.2609</td>
</tr>
<tr>
<td>336.8±0.083</td>
<td>337.2±0.1957</td>
<td>304.6±0.083</td>
<td>303.8±0.3497</td>
</tr>
<tr>
<td>337.2±0.083</td>
<td>337.6±0.1876</td>
<td>305.3±0.083</td>
<td>304.5±0.4753</td>
</tr>
<tr>
<td>334.1±0.083</td>
<td>334.8±0.2243</td>
<td>304.6±0.083</td>
<td>303.4±0.393</td>
</tr>
<tr>
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<td>335.8±0.2244</td>
<td>306.1±0.083</td>
<td>304.8±0.6474</td>
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<tr>
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<td>310.2±0.083</td>
<td>308.8±1.58</td>
</tr>
<tr>
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<td>334.6±0.3021</td>
<td>303.9±0.083</td>
<td>302.3±0.5374</td>
</tr>
<tr>
<td>338.4±0.083</td>
<td>339±0.3922</td>
<td>308.7±0.083</td>
<td>306.6±1.638</td>
</tr>
<tr>
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<td>312±0.083</td>
<td>310.9±2.131</td>
</tr>
<tr>
<td>342.9±0.083</td>
<td>343.3±0.2215</td>
<td>306.6±0.083</td>
<td>305.3±0.891</td>
</tr>
</tbody>
</table>

II4. Summary

The HTF, a unique heat transfer facility, was constructed at UNM with the purpose of performing exploratory and validation data collection for different heat exchanger designs. The facility has capabilities for studying natural circulation and bi-directional forced circulation of the simulant fluid Dowtherm A in the primary loop, and forced circulation flow in the secondary loop, while covering a range of Reynolds and Grashof numbers important to heat exchangers in the FHR [3]. Heat transfer data was collected for a bayonet-style twisted and plain tube bundle heat exchanger, focusing on heat transfer performance especially on flow regimes relevant for decay heat removal heat exchangers, but also extending into the lower ranges relevant for exchangers transferring heat to the power conversion cycle or intermediate loop. Data for the buoyancy affected regimes in particular was exploratory, but propagation of errors from the data and parameter inputs is also suitable for certain types of validation studies. As a validation metric, MARE and MaxRE were used to compare the predicted values of the data using correlations with the actual measured data. Agreement for the correlations developed here was generally in the single percentage digits, except for maximum relative errors. Tube-data was observed to follow two regimes: inertial dominated flow correlated to Reynolds number and buoyancy dominated flow correlated to Rayleigh number. The shell-side data was found to correlate well to Reynolds number when the estimated Richardson number was $< 1$. When the estimated Richardson number exceeded unity, downflow correlated well to simultaneous functions of both Reynolds and Rayleigh numbers. Interestingly and somewhat surprisingly, when downflow Richardson numbers exceeded 30, the data again correlated well to Reynolds number alone. For up-flow, the data correlated well to Reynolds number throughout the full range of estimated Richardson numbers [3,4].

Figures II-47 and II-48 compared the measured heat transfer performance of the twisted tube and plain tube bundle HX. These figures show the heat transfer enhancement gained by utilizing the twisted tubes over plain tubes in both natural and forced circulation regimes. For natural circulation, an improvement of 55% increase in the heat transfer performance at Re of 120 and a maximum of 250% increase at the max Re achieved for forced circulation experiment [3,4].
Figure II-47. Comparison of the measured natural circulation heat transfer performance of twisted and plain tube bundle HX

Figure II-48. Comparison of the measured forced circulation heat transfer performance of twisted and plain tube bundle HX
The single-assembly double-wall heat exchanger was used to validate a tool for predicting the performance for a three-fluid parallel stream heat exchanger with two thermal communications. The three-fluid heat exchanger tool can be used to understand and predict the temperature distributions for any parallel stream three fluid heat exchanger. The tool predicted the outlet temperatures of the triple fluids with a maximum uncertainty of 18% in selected cases. The code is as a quick tool for designing a triple flow experiment to optimize the performance-based on the fluids’ properties and experimental conditions. It can be part of a larger scope code for integrated system analysis where the triple flow HX is utilized [3,4].

The triple flow HX performance was tested at SNL for SCO2 facility. Dowtherm to SCO2 experiments were partially conducted, and data was collected. Further experiments are required to enable the full analysis of the HX performance for utilizing the triple flow HX in coupling FHR to SCO2 power conversion cycle.

II4. Bibliography


Appendix II-A
Three-Fluid Heat Exchanger
Temperature Distribution Program

# This code predicts the temperature distribution of the SADWPT heat exchanger given the inlet temperatures and flow rates of the experiment. NOTE: Predictions for Nusselt number vary based on experiment. The user must decide which Nusselt correlation to use. The code can be used for twisted tube heat exchangers as well. The twisted tube effects should be characterized within the definition of NTU

import numpy
import matplotlib.pyplot as plt
import matplotlib.lines as mlines
import csv from scipy.integrate import odeint, solve_bvp from scipy.
import optimize.import root, fsolve
import tp_properties
import Nusselt_PT

fig, ax1 = plt.subplots() ax2 = ax1.twinx()

# NUMBER OF NODES (IMPORTANT FOR CONVERGENCE)

nodes = 100000

# For air tests include pressure P =
# DEFINE ABSOLUTE TEMPERATURES

T_1i = # inlet temperatures of each fluid stream
T_2i =
T_3i =
T_sp = #temperature setpoint of experiment

# FLUID 1 IS ALWAYS POSITIVE DIRECTION

# DEFINE FLUID 2 AND 3 DIRECTIONS AS POSITIVE (+1) OR NEGATIVE (-1)
i_2 = -1
i_3 = -1

# Input flow rates
Flow1 = # gpm
Flow2 = # For air, need to change flow rate to accommodate SCFM
Flow3 =

#define Thermophysical Properties
[rho1, cp1, mu1, lambda1] = tp_properties.tp_dowtherm(T_sp)  
[rho2, cp2, mu2, lambda2] = tp_properties.tp_air(T_2i, P)
[rho3, cp3, mu3, lambda3] = tp_properties.tp_water(T_3i)

# Calculate mass flow rates mfl
mf1 = Flow1 * 0.00063 * rho1
mf2 = Flow2 * 0.00063 * rho2
mf3 = Flow3 * 0.00063 * rho3

# DEFINE HEAT CAPACITY RATES J/sK
C_1 = mf1 * cp1 * 1000 #252
C_2 = numpy.array([mf2 * cp2 * 1000]) * numpy.array([11])  # numpy.array([11])  
C_3 = mf3 * cp3 * 1000 #105

# SADWTI Areas A_1 = 0
A_2 = 0.285 #m^2

# Reynolds Calculations
Re1 = Flow1 * 0.00063 * 0.0153 / (mu1 / rho1 * 0.000189)
Re2 = Flow2 * 0.00063 * 0.022 / (mu2 / rho2 * 0.00051)
Re3 = Flow3 * 0.00063 * 0.028 / (mu3 / rho3 * 0.000616)

# Pr calculations
Pr1 = cp1 * 1000 * mu1 / lambda1
Pr2 = cp2 * 1000 * mu2 / lambda2
Pr3 = cp3 * 1000 * mu3 / lambda3
# Nu Predictions

Nu1 = Nusselt_PT.Nu_Gnielinski(Re1, Pr1, .0153, 3)
Nu2 = Nusselt_PT.Nu_DittusBoelter(Re2, Pr2)#, .022, 3) Nu3 = Nusselt_PT.Nu_StefanPreuber(Re3, Pr3, .028, 3)

# Individual heat transfer coefficient calculations from Nu predictions

h1 = Nu1*lambda1/.0153
h2 = Nu2*lambda2/.022
h3 = Nu3*lambda3/.0285

# Calculate overall heat transfer coefficients

U_1 = 1/((1/h1) + (.029/16)*numpy.log(.027/.029) + (.029/.027)*(1/h2))
U_2 = 1/((1/h2) + (.016/16)*numpy.log(.014/.016) + (.016/.014)*(1/h3))

# DEFINE NTU'S

NTU_1 = (U_1*A_1)/C_1 #.088 # DEFINED AS (U_1 A_1)/C_1 NTU_2 = (U_2*A_2)/C_2 #1.86 # DEFINED AS (U_2 A_2)/C_2

print(Nu1,Nu2,Nu3)
print(h1,h2,h3)
print(C_1,U_1,C_2,U_2,NTU_1,NTU_2,C_3) print
(Re1,Pr1,Re2,Pr2,Re3,Pr3)

# DEFINE INTERMEDIATE INLET TEMPERATURE Non-dimensional
theta_2in = (T_2i-T_3i)/(T_1i-T_3i) #0.17

# DETERMINE HEAT CAPACITY STREAM RATIOS

R_1 = (C_1*C_3)
R_2 = (C_2/C_3)

# SOLVE THE BOUNDARY VALUE PROBLEM

for i in range (0,len(C_2)):
def
myfun(x, y):
    return numpy.vstack((NTU_1*(y[1]-y[0]), i_2*NTU_1*(R_1/R_2[i])
) *(y[0]-y[1])+i_2*NTU_2*(y[2]-y[1]), i_3*NTU_2*R_2[i]*(y[1]-y

98
def bc(ya, yb):
    residual_1 = ya[0] - 1
    if i_2 == 1:
        residual_2 = ya[1] - theta_2in
    if i_2 == -1:
        residual_2 = yb[1] - theta_2in
    return numpy.array([residual_1, residual_2, residual_3])

x = numpy.linspace(0, 1, nodes)
y = numpy.zeros((3, x.size))
sol = solve_bvp(myfun, bc, x, y, max_nodes=1e6, verbose=2)

y0 = sol.sol(x)[0]  # hot temperature
y1 = sol.sol(x)[1]  # intermediate temperature
y2 = sol.sol(x)[2]  # cold temperature

ax1.plot(x, y0, 'r', linewidth=2.0)
ax1.plot(x, y1, 'k', linewidth=2.0)
ax2.plot(x, y2, 'b', linewidth=2.0)

# PRINT MEAN TEMPERATURE DIFFERENCES (DETERMINED NUMERICALLY — MUST CHECK FOR CONVERGENCE MANUALLY)

MTD_1to2_numerical = numpy.mean(y0-y1)
MTD_2to3_numerical = numpy.mean(y1-y2)
MTD_1to3_numerical = numpy.mean(y0-y2)
print("")
print("MTD_1to2_numerical \t= \{:.4f\} \t= \{:.4f\} C".format(MTD_1to2_numerical, MTD_2to3_numerical))
MTD_1to2_numerical, MTD_1to2_numerical *(T_li-T_3i)
print("MTD_2to3_numerical \t= \{4.4 f \} \t= \{4.1 f \} C".format(MTD_2to3_numerical, MTD_2to3_numerical *(T_li-T_3i)))
print("MTD_1to3_numerical \t= \{4.4 f \} \t= \{4.1 f \} C".format(MTD_1to3_numerical, MTD_1to3_numerical *(T_li-T_3i)))
print(""
print("T_1o \t\t\t\t= \{4.4 f \} \t= \{4.1 f \} C".format(y0[x.size -1], y0[x.size -1]*(T_li-T_3i)+T_3i))
T_1o = y0[x.size -1]*(T_li-T_3i)+T_3i if i_2 == 1:
    print("T_2o \t\t\t\t= \{4.4 f \} \t= \{4.1 f \} C".format(y1[x.size -1], y1[x.size -1]*(T_li-T_3i)+T_3i))
T_2o = y1[x.size -1]*(T_li-T_3i)+T_3i if i_2 == -1:
    print("T_2o \t\t\t\t= \{4.4 f \} \t= \{4.1 f \} C".format(y1[0], y1[0]*(T_li-T_3i)+T_3i))
T_2o = y1[0]*(T_li-T_3i)+T_3i if i_3 == 1:
    print("T_3o \t\t\t\t= \{4.4 f \} \t= \{4.1 f \} C".format(y2[x.size -1], y2[x.size -1]*(T_li-T_3i)+T_3i))
T_3o = y2[x.size -1]*(T_li-T_3i)+T_3i if i_3 == -1:
    print("T_3o \t\t\t\t= \{4.4 f \} \t= \{4.1 f \} C".format(y2[0], y2[0]*(T_li-T_3i)+T_3i))
T_3o = y2[0]*(T_li-T_3i)+T_3i print(""
with open('PyDoubleWallOutput.csv', 'w') as f:
    writer = csv.writer(f)
    writer.writerow(["T_li", "T_3i", 'NTU_1', 'NTU_2', 'C_1', "C_2", "C_3", theta_2in', 'T_1o', 'T_2o', 'T_3o'])
    writer.writerow([T_li, T_3i, NTU_1, NTU_2, C_1, C_2, C_3, theta_2in, T_1o, T_2o, T_3o])
R_3 = C_2/C_1 R_4 =
C_3/C_1
#R_4*(y2[0] - y2[1]) + R_3*y0[1])
stream3eff = y2[0]
stream2eff = (y1[0]−theta_2in)/(1−theta_2in)
print("stream3eff = ", stream3eff)
print("stream2eff = ", stream2eff)
effectiveness = (R_3*stream2eff*(1−theta_2in) + R_4* stream2eff)/(R_3*(1−theta_2in) + R_4)
print("overall effectiveness = ", effectiveness)

# FINISH PLOTTING

plt.rc ('text', usetex=True)
plt.rc ('font', family='serif')
ax1.set_xlabel ('$\xi$')
ax1.set_ylabel ('$\theta = \frac{T_{t(\\xi)}−T_{t, in}}{T_{s, in}−T_{t, in}}$', color='b')
ax1.set_xlim (0,1)
ax1.set_ylim (0,1)
ax1.tick_params (axis='y', color='b', labelcolor='b')
ax2.set_ylabel ('$\Theta = \frac{T_{s(\\xi)}−T_{t, in}}{T_{s, in}−T_{t, in}}$', color='r')
ax2.set_ylim (0,1)
ax2.tick_params (axis='y', color='r', labelcolor='r')
ax2.spines [ 'left '].set_color ('b')
ax2.spines [ 'right '].set_color ('r')
plt.title ('Temperature Profile of Double-wall HXR')
blue_line = mlines.Line2D ([], [], linewidth=2.0, color='blue', label='Cold-side')
red_line = mlines.Line2D ([], [], linewidth=2.0, color='red', label='Hot-side')
black_line = mlines.Line2D ([], [], linewidth=2.0, color='black', label='Intermediate')
plt.legend (handles=[blue_line, black_line, red_line], loc='best')
plt.grid (True)
plt.savefig ('double_wall_hxr.png', dpi=600, format='png') plt.show()