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The Selection, Evaluation and Rating of Compact Heat Exchangers (SEARCH) Software Suite – Code Capabilities and Experimental Comparison

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ABSTRACT

The optimal design of compact heat exchangers typically requires a combination of analytical performance estimation, computational fluid dynamics, and finite element modeling, with each design iteration taking hours to days at a time. To simplify this traditional design process Sandia National Laboratories in collaboration with Vacuum Process Engineering has developed an efficient, flexible, and comprehensive microchannel heat exchanger (MCHE) design tool. This code implements a sub-heat exchanger thermodynamic model, American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code mechanical constraints, and a thermal-hydraulic solver within Engineering Equation Solver (EES) with capabilities to model any combination of liquid, gas, two-phase, and supercritical fluid using over 400 pure fluids and mixtures. This paper describes the core solution algorithm as applied to a water-water recuperator, recent performance testing undertaken at the Nuclear Energy Systems Laboratory heat exchanger test loop, and a comparison between design code expectations and measured heat exchanger performance.

INTRODUCTION

Heat exchangers have continued to receive significant research and development support because they are a key enabling technology for the commercialization of supercritical carbon dioxide (sCO2) Brayton cycles, with the Department of Energy (DOE) Office of Fossil Energy (FE) recently awarding a 9.5 M$ grant to a team led by Thar Energy and including the Southwest Research Institute (SwRI), Oak Ridge National Lab (ORNL), and the Georgia Institute of Technology for the development of recuperators in addition to 10's of M$ awarded for many previous projects led by Sandia, SwRI, Thar Energy, Altex Technologies, Brayton Energy, Oregon State University, the University of Wisconsin – Madison, The Ohio State University, Argonne National Laboratory, and Purdue University.

The recent workshop on heat exchangers for sCO2 power cycles hosted after the Electric Power Research Institute (EPRI) Conference on Corrosion in Power Plants showed a clear consensus that the technical readiness of heat exchangers, with the exception of high-temperature units like the primary heat exchanger, is not an issue with several vendors confident in their proposed solutions. However the manufacturing readiness level (scalability) and cost loom large as new technologies have not been proven at scale and mature technologies have larger than desired cost due to both capital expense (CAPEX) and operating expense (OPEX) as a result of high pressure drops.

Sandia has been involved in a Cooperative Research and Development Agreement (CRADA) with Vacuum Process Engineering (VPE) since May of 2014 in an effort to advance the manufacturing readiness level and cost of PCHEs manufactured in the United States. This paper summarizes recent progress under this CRADA to develop a flexible microchannel heat exchanger (MCHE) design software tool and validate it using both thermal-hydraulic and mechanical test data.

MCHE DESIGN ALGORITHM

Overview of the Algorithm

The Selection, Evaluation, And Rating of Compact Heat exchangers (SEARCH) design tool was developed by Sandia National Laboratories in collaboration with Vacuum Process Engineering (VPE) in order to automate and simplify the design of conventional printed circuit heat exchangers (PCHEs). The algorithm contains three key modules to evaluate the thermodynamic, mechanical, and thermal-hydraulic constraints on a units design. By implementing these modules in Engineering Equation Solver (EES) the same code can be used to evaluate a single design point, rate a device based on its design, perform parametric studies of design options, and perform multi-objective optimization studies by leveraging capabilities built-in to the EES platform. EES also provides integrated fluid thermodynamic and transport properties, and the capability to couple with Refprop in order to calculate the properties of pure fluids and preset or custom mixtures.

The Sub-Heat Exchanger Module

The sub-heat exchanger module is an implementation of the algorithm described by Nellis and Klein [1] to apply the effectiveness-NTU solutions incrementally within a heat exchanger to capture the effects of variable properties on performance with minimal computational cost as depicted in the diagram in Figure 1. This algorithm is effective for two-fluid heat exchangers that are mostly counterflow, but with suitable design margin can also account for small sections of cross-flow like those found in PCHEs. As discussed by Nellis and Klein only three to five sub-heat exchangers are required to accurately capture heat exchanger performance, but finer discretization is more effective when properties are highly variable near the critical point or under two-phase conditions.

The terminal states are first assigned or determined using the required heat exchanger duty. The total duty is then divided into increments of heat transfer each representing a “sub-heat exchanger” with intermediate states representing virtual terminal conditions of the sub-heat exchangers. The capacitance rate of each stream in each sub-heat exchanger is calculated according to Equation (1), where care must be taken with the temperature difference in the denominator in order to accommodate any combination of single- or two-phase flows correctly. The most robust method found to do this is shown where the magnitude of the temperature difference is restricted to a minimum temperature value, and then assigned the appropriate sign based on the enthalpy difference across the sub-heat exchanger. This correctly decomposes the calculation under two-phase conditions to the limit when the capacitance rate ratio C_R approaches zero and configuration is irrelevant. Effectiveness, NTU, and UA are then calculated as normal according to Equations (2) through (4), and the sum of the individual sub-heat exchanger UAs will approximate the UA of the complete unit.

$$\dot{C}_i = \dot{m}\frac{h_i - h_{i+1}}{\mathrm{MAX}\big(\text{1e-4 [K]}, |T_i - T_{i+1}|\ \mathrm{SIGN}(h_i - h_{i+1})\big)} \tag{1}$$

$$\varepsilon_i = \frac{\dot{q}_i}{\mathrm{MIN}\big(\dot{C}_{A,i}, \dot{C}_{B,i}\big)\big(T_{A,i} - T_{B,i+1}\big)} \tag{2}$$

$$NTU_i = \begin{cases} \dfrac{\ln\left[\dfrac{(1-\varepsilon C_R)}{1-\varepsilon}\right]}{1-C_R} & \text{for } C_R < 1 \\ \dfrac{\varepsilon}{1-\varepsilon} & \text{for } C_R = 1 \end{cases} \quad \text{where } C_R = \frac{\mathrm{MIN}\left(\dot{C}_{A,i}, \dot{C}_{B,i}\right)}{\mathrm{MAX}\left(\dot{C}_{A,i}, \dot{C}_{B,i}\right)} \tag{3}$$

$$UA_i = NTU_i \, \mathrm{MIN}\left(\dot{C}_{A,i}, \dot{C}_{B,i}\right) \tag{4}$$

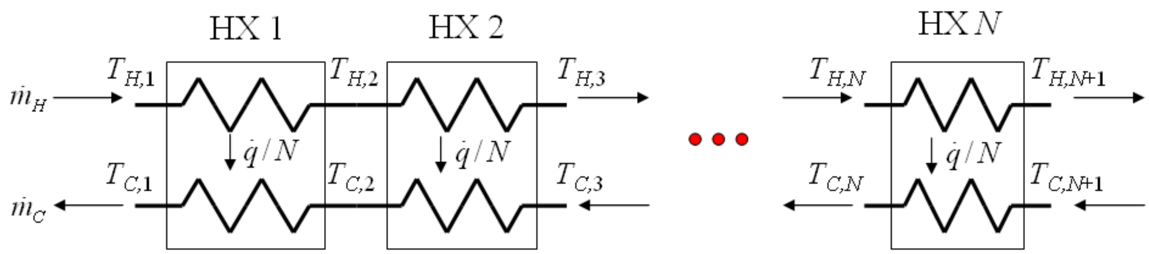

Figure 1. A diagram showing the discretization of a single heat exchanger into N sub-heat exchangers where the assumptions of the effectiveness-NTU heat exchanger model can be applied. Note that by careful implementation it is not required to know explicitly which is the hot side ("H") or cold side "C" ahead of time. From [1].

Mechanical Design Module

The mechanical design module implements the pressure containment requirements from Appendices 42 and 13 of Section VIII of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel (BPV) Code [2], with the addition of the header design requirements extracted from UG-34 of Section VIII.

The Appendix 13 equations governing pressure containment within the PCHE core were first discussed by Le Pierres [3], and were non-dimensionalized as described previously [4] and repeated here for completeness as Equations (5) through (7). These relations dictate the thickness fractions within the core based on the selected material and operating conditions of the device with terms defined according to Figure 2. By providing either the channel size or wall thickness the other can be determined, and in turn the cross-sectional geometry of the unit.

Multi-Channel Plate – Stay Plate Membrane Stress Scaling

$$\frac{t_4}{h+t_4} \geq \left(1+\frac{SE}{P}\right)^{-1} \tag{5}$$

Multi-Channel Plate – Long Plate Membrane Stress Scaling

$$\frac{t_2}{H+t_2} \geq \left(1+2\frac{SE}{P}\right)^{-1} \tag{6}$$

Multi-Channel Plate – Long Plate Total Stress Scaling

$$\frac{P}{SE} \leq 3\left(\frac{1-t_f}{t_f}+\frac{1}{AR^2}\left(\frac{1-t_f}{t_f}\right)^2\right)^{-1} \quad \text{where } t_f = \frac{t_2}{H+t_2} \tag{7}$$

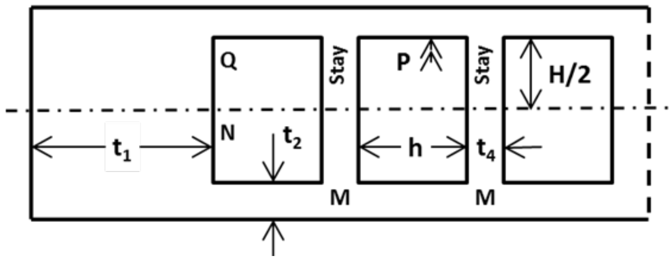

Figure 2. A dimensioned diagram of a multi-channel plate pressure vessel geometry extrapolated from ASME BPVC VIII-1-13-2(a)(8) [2]. Sections of thickness t1 and t2 form the primary pressure boundary, while sections of thickness t4 (stay plates) act as stay members.

The half-cylindrical shell portion of the header is designed according to ASME BPVC VIII-1-13-13(a) for cylindrical vessels stayed with a diametral plate where the PCHE core block is treated as the diametral staying plate shown in Figure 3. Where both compartments are pressurized equally, as is the case with the two opposing headers on a PCHE block, the design equations are given by VIII-1-13-13(b) and reproduced below as Equations (8) through (15).

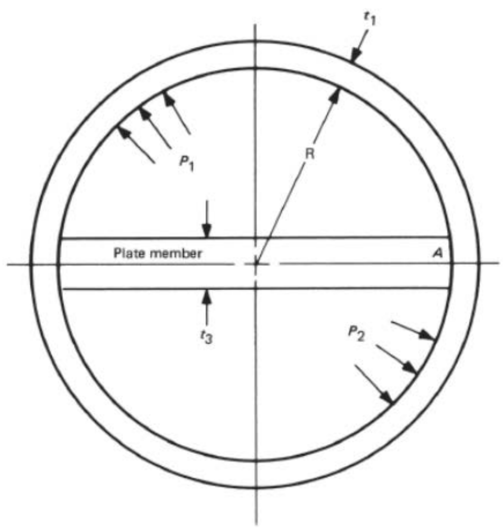

Figure 3. A cylindrical vessel with a central dividing stay plate from ASME BPVC VIII-1-13-2(c) [2].

Design Equations for the Half-Cylindrical Shell

$$S_m \leq SE \tag{8}$$

$$S_t = S_m + S_b \leq 1.5SE \tag{9}$$

$$S_{m,shell} = \frac{P_1 R}{t_1} \tag{10}$$

$$S_{m,plate} = \frac{2\pi P t_1^2}{3Rt_3(\pi^2 - 8)} \quad \textbf{(11)}$$

$$S_{b,shell} = \frac{c}{I_1}\left[\frac{2Pt_1^2}{3(\pi^2 - 8)}\right] \quad \textbf{(12)}$$

$$S_{b,plate} = 0 \quad \textbf{(13)}$$

$$I_1 = \frac{bt_1^3}{12} = \frac{t_1^3}{12} \quad \textbf{(14)}$$

$$c = \frac{t_1}{2} \quad \textbf{(15)}$$

These equations can be non-dimensionalized into the three unique constraint equations as shown in Equations (16) through (18). One thing to note at this point is that Equation (18) specifies a minimum core block thickness relative to both the pressure and header shell thickness fraction, which for traditional straight and Z-side flow paths restricts either the minimum core width or length.

Non-Dimensionalized Design Equations for the Half-Cylindrical Shell

$$\frac{t_{m,shell}}{R} \geq \frac{P}{SE_{shell}} \quad \textbf{(16)}$$

$$\frac{t_{m,plate}}{R} \geq \frac{2\pi}{3(\pi^2 - 8)}\frac{P}{SE_{shell}}\left(\frac{t_{shell}}{R}\right)^2 \quad \textbf{(17)}$$

$$\frac{t_{t,shell}}{R} \geq \left[1.5\left(\frac{P}{SE_{shell}}\right)^{-1} - \frac{4}{\pi^2 - 8}\right]^{-1} \quad \textbf{(18)}$$

Flat end closures for vessels of non-circular cross-section defined in appendix 13 are defined by ASME BPVC VIII-1 UG-34, with the equations referenced reproduced below as Equations (19) and (20). The value of C is defined explicitly in VIII-1-13-4(f) as 0.20 regardless of the values otherwise given in UG-34, leading the product of Z and C (ZC) to reduce to 0.44 in all cases for a typical PCHE with semi-circular channels where d/D is always 1/2. The constraint equations relating pressure containment to end closure thickness fraction can therefore be non-dimensionalized as the simple power-law relationship given in Equation (21).

Design Equations for End Closures

$$t_{cap} \geq d\sqrt{\frac{ZCP}{SE_{cap}}} \quad \textbf{(19)}$$

$$Z = 3.4 - 2.4\frac{d}{D} \quad \textbf{(20)}$$

Non-Dimensionalized Design Equations for End Closures

$$\frac{t_{cap}}{R} \geq \sqrt{\frac{44}{100}\frac{P}{SE_{cap}}} \text{ or } \frac{P}{SE_{cap}} \leq \frac{100}{44}\left(\frac{t_{cap}}{R}\right)^2 \quad \textbf{(21)}$$

Thermal-Hydraulic Module

The thermal-hydraulic module implements a variety of single and two-phase heat transfer and pressure drop correlations in order to determine the length of each sub-heat exchanger based on each UA value calculated in the sub heat-exchanger module. Combined with the cross-sectional geometry determined in the mechanical design module this allows for a complete design of the heat exchanger. The heat transfer coefficients and metal thermal conductivity shown in Equation (22) are calculated based on average properties in each sub-heat exchanger. This calculation can be explicit for simple single-phase flow conditions; however for some supercritical and two-phase conditions the correlations are necessarily implicit in order to converge on heat transfer coefficients and surface temperatures simultaneously.

$$\Delta x_i = UA_i\left(\frac{1}{h_{A,i}N_{ch,A}p_{ch,A}} + \frac{R^{"}_{f,A,i}}{N_{ch,A}p_{ch,A}} + \frac{t_m}{k_{m,i}W} + \frac{1}{h_{B,i}N_{ch,B}p_{ch,B}} + \frac{R^{"}_{f,B,i}}{N_{ch,B}p_{ch,B}}\right) \quad \textbf{(22)}$$

PROTOTYPE PCHE DESIGN

Primary Design for Water Testing

A 100 kW_{th} prototype PCHE was designed and fabricated primarily for thermal-hydraulic testing in a water-water test loop constructed in 2015 at the Nuclear Energy Systems Laboratory (NESL) at Sandia National Labs. This unit was designed for at least a 15 K inter-stream temperature difference in order to accurately measure the duty of the device, as well as approximately 60 kPa of pressure drop as shown in the abbreviated heat exchanger data sheet shown as Table 1. Straight channels were also chosen for this unit in order to provide straightforward validation of the SEARCH design code correlations.

Table 1. An abbreviated heat exchanger data sheet for the prototype 316L PCHE.

Parameter	Unit	Side A (Straight)	Side B (Z)
Fluid	-	water	water
Mass Flow Rate	kg/s (lbm/hr)	1.5 (12000)	1.5 (12000)
Volumetric Flow Rate	m^3/s (gpm)	1.5e-3 (24)	1.5e-3 (24)
Inlet Temperature	°C (°F)	82 (180)	37 (98)
Inlet Pressure	kPa (psi)	300 (44)	300 (44)
Pressure Drop	kPa (psi)	55 (7.9)	62 (9.0)
Fouling Factor	m^2-K/W	8e-5	8e-5
MAWP	MPa (psi)	20 (2900)	
MAWT	°C (°F)	550 (1000)	

Duty	kW_{th} (Btu/hr)	103 (350000)
Height x Width x Length	m (in)	0.15 x 0.15 x 0.46 (6 x 6 x 18)
Active Surface Area	m^2 (in^2)	1.2 (13)

Design for Multiple Test Phases

In order to minimize the manufacturing costs, a single prototype heat exchanger was designed for multiple phases of validation testing of the SEARCH design code, including pressure containment, single-phase thermal-hydraulics, supercritical thermal-hydraulics, and fatigue induced by combined and varying thermal and pressure stresses. This series of test phases can be accomplished in a single unit by incrementally increasing the risk of failure of the device to the point of intentional failure in the last phase to understand failure modes and validate fatigue design predictions. Using the initial water-water single-phase thermal-hydraulic operation as a base point, elevated maximum allowable working temperatures and pressures were imposed to allow for operation up to 20 MPa and 550 °C.

While these requirements do not significantly change the core geometry of the device, they do impose relatively thick headers using 316L stainless steel of 1.5 in. This provided an opportunity to test realistic header weld processes at a small scale as the core block must be sufficiently pre-heated and maintained at temperature, as well as cooled slowly to avoid significant residual stresses and poor weld quality.

Unique Instrumentation Features

Instrumenting a complete PCHE is a significant challenge due to its closely-space internal geometry and thick wall sections. Thermocouples can be placed within a stack of plates during diffusion bonding in order to measure internal temperatures, and is often done to monitor and characterize a diffusion bonding process, but there is significant risk that the thermocouples will fracture during handling or short to the structure during welding and become useless. In addition this approach does not allow for the complete determination of fluid state as there is no way to measure either pressure or density.

Instead a series of temperature and pressure taps were incorporated into the etching mask patterns of several plates in order to allow access to exterior channels across the device, as well as structural temperature locations near the PCHE headers as shown in Figure 4. There are nine total temperature and pressure taps for both the straight and z-sides of the device, with five placed on each side of the unit to accommodate the headers with a typical spacing of two inches apart. The fifth tap on each side is place at the same axial location in order to provide some offset for measurements along the entire axial length of the device when it is necessary to switch from measurements on a front channel to a rear channel. In addition to the temperature and pressure taps, holes are also etched to measure five vertical positions at each corner of the PCHE core and between the T-P taps and the Z-side headers.

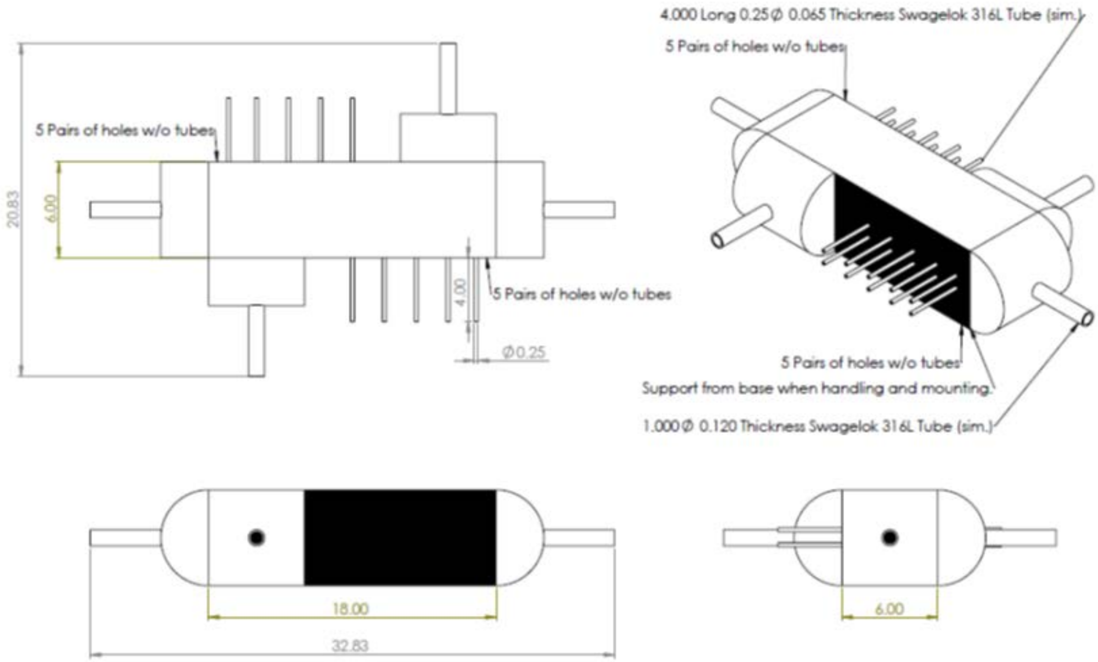

Figure 4. Exterior design features and size of the prototype PCHE.

HEAT EXCHANGER TEST PLATFORM

The heat exchanger test platform shown in Figure 5 was constructed to quickly characterize the thermal-hydraulics of heat exchangers up to 100 kW duty using two separate water loops. The heating loop includes a 100 kW_{th} Durex electric immersion heater, a Goulds Water Technology centrifugal pump driven by a Bluffton Motor Works motor and a variable frequency drive (VFD) capable of providing flow rates from 5 to 120 gpm, as well as a strainer to prevent debris from plugging the PCHE and especially the temperature and pressure taps. The cooling loop includes a 100 kW_{th} open evaporative cooler supplied but Baltimore Air Coolers (BAC), another Goulds pump, Bluffton motor, and VFD capable of 5 to 120 gpm, as well as both a large sock filter and a strainer due to the larger particulate loading on of the open loop fluid. The loop is controlled by a National Instruments compact DAQ system with a custom control system implemented in LabView.

Figure 5. A picture of the completed water-water heat exchanger test loop at Sandia's NESL.

ADD a P&ID DIAGRAM?

The prototype PCHE itself as shown in Figure 6 is instrumented with four 0.020" diameter miniature 100 Ohm RTD sensors provided by TC Measurement and Control and two 3 to 30 psi Kobold heavy duty differential pressure transmitters to measure inlet and outlet temperatures and differential pressure through the temperature and pressure taps. While this instrumentation is currently being used with water at near ambient pressures, it is designed to operate up to the maximum design operating temperature and pressure of the prototype PCHE. Additional pressure and temperature ports will be used in later test campaigns.

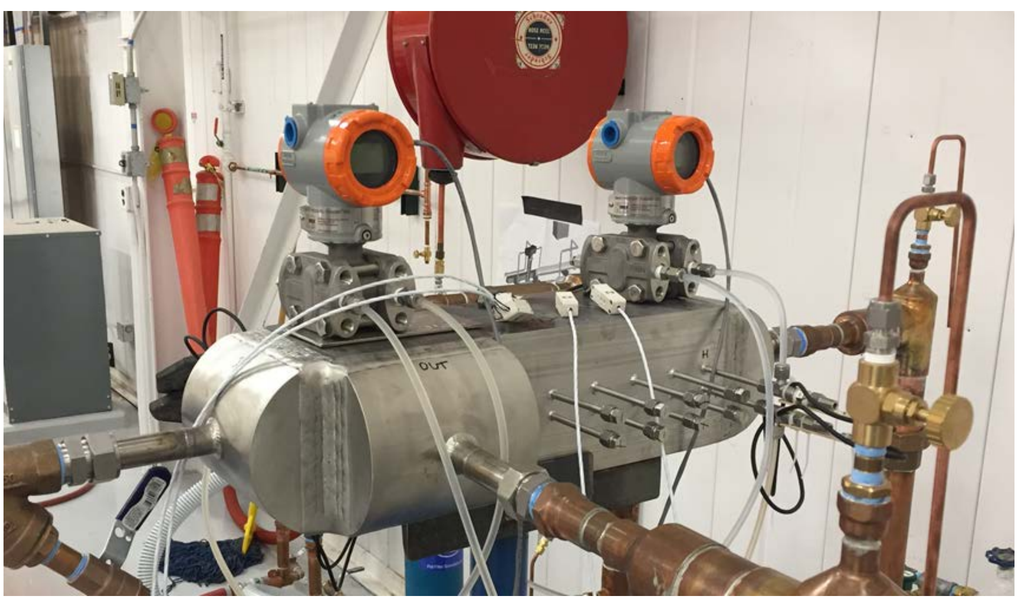

Figure 6. A picture of the instrumented prototype 316L stainless steel PCHE.

PROTOTYPE PCHE PERFORMANCE COMPARISON

Test Description

Over the course of one water test run the heating power was incremented up to 100% power as shown in Figure 7 and Table 2 up to a maximum loop temperature of 110 °F as measured by the inlet and outlet RTDs placed in the T-P taps of the prototype PCHE. For each increment of increased heater power the hot-side temperature quickly settled to steady-state temperature, however due to the duty-cycle control over heater power there was considerable short-term oscillation in temperature of approximately 3 °F. Over the course of the test the inlet pressures of the hot and cold sides of the PCHE were maintained at 62 and 38 psi, respectively, with both flow rates steady at approximately 42 gpm.

Table 2. A description of test conditions at different times.

Time Range	Description
0-750	Baseline, prepare to start test. Hot flow started first, wait to reach steady state.
750-1500	Start cooling flow; keep at maximum rate until loop below 70°F.
1500-6500	Increased heater power gradually (5-10% increments) to 100% = 110°F.
6500-7000	Shut off heater power, cooling remains on.

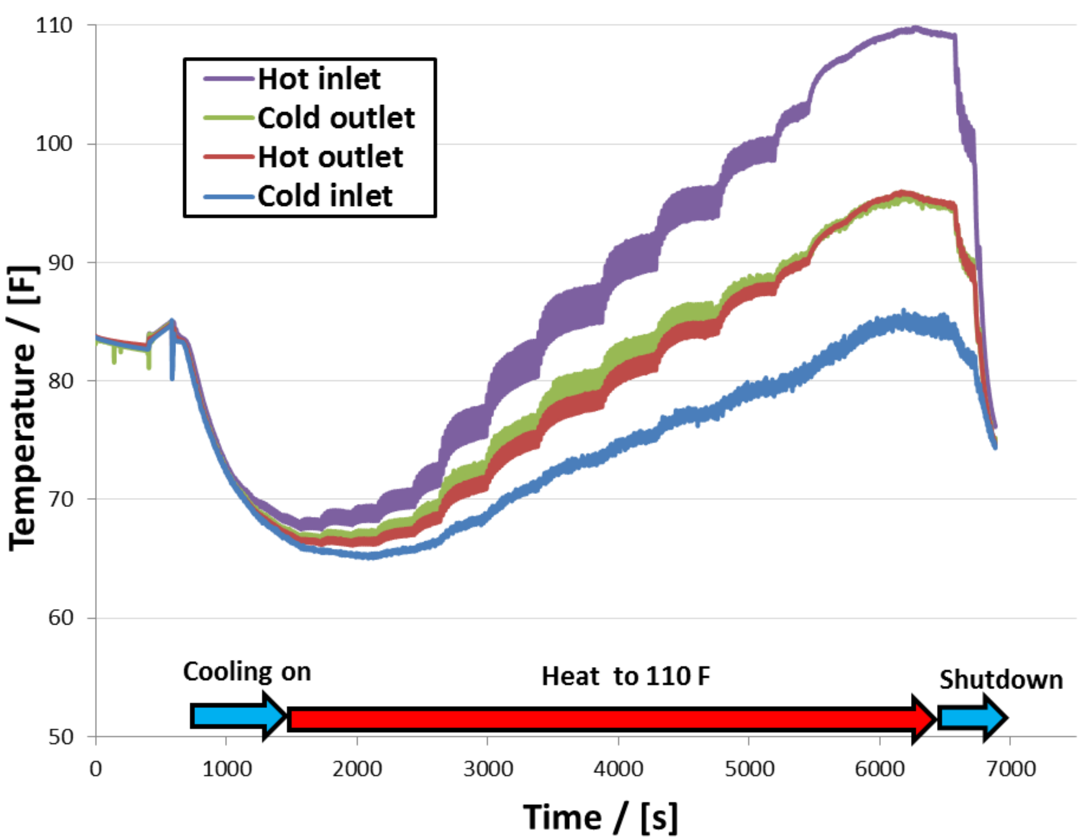

Figure 7. Prototype PCHE inlet and outlet temperatures over time.

Comparison with SEARCH Rating Calculations

In addition to running as a design tool, SEARCH can be used to perform rating calculations for an existing PCHE design by using the exact same equation set and simply changing which values are selected as inputs. This allows comparisons between the rating calculations and measured performance to also provide validation of the design capability in SEARCH as the same equation set is used for both.

Using the measured operating conditions described previously at several times as shown in Table 3, the rating results of SEARCH can be compared to calculated performance metrics of the prototype PCHE as a measure of the accuracy of the SEARCH design algorithm. Note that data is averaged over several periods to avoid any error due to the oscillatory steady-state temperatures caused by the duty-cycle heater controller. Based on this comparison SEARCH is generally conservative, under-predicting the heat transfer by at least 10%, the UA by at least 25%, and the effectiveness by at least 10%. Possible sources of this discrepancy include inaccuracies in instrumentation, the simplicity of the SEARCH algorithm compared to a complete fluid-structural numerical simulation, and non-uniform flow distribution within the prototype PCHE. However these results are promising and will be reinforced by additional testing in the coming year as well as improvements to the heat exchanger loop instrumentation, characterization of instrument uncertainties, and added capability in the SEARCH algorithm.

It should be noted that the effectiveness of this prototype is intentionally much lower than is typical in most PCHE designs in order to increase the terminal temperature differences to more accurately characterize performance and validate the SEARCH design algorithm. A high-effectiveness PCHE would have a terminal temperature difference of only a few °F requiring very low temperature measurement uncertainties. Instead as shown in Figure 7 the prototype has terminal temperature differences between the hot outlet and cold inlet, as well as the cold outlet and hot inlet, which range from 5 to 10 °F depending on the power level.

Table 3. SEARCH predictions for several heat exchanger performance metrics over time throughout the test and the percent difference in the calculated value from measurements.

	$\dot{q}$ / W		UA / (W/K)		ε	
Time / s	*SEARCH*	*Measured*	*SEARCH*	*Measured*	*SEARCH*	*Measured*
4200	44000	+7%	8100	+13%	43%	+6%
4700	44000	+12%	8200	+23%	43%	+11%
5100	54000	+13%	8400	+26%	43%	+12%
5400	61000	+14%	8500	+28%	44%	+13%
5700	67000	+14%	8600	+27%	44%	+13%
6260	67000	+16%	8700	+32%	44%	+15%

CONCLUSIONS

The preliminary results shown in this paper demonstrate that the SEARCH heat exchanger design tool is reasonably conservative in the design of PCHEs. As more test data on thermal-hydraulic performance is accumulated during this phase of testing we will be able to sufficiently characterize the inherent design margin in the SEARCH software to reduce unnecessary extra margin, with corresponding benefits in the cost and size of VPE devices.

Future phases of this CRADA program will work toward developing a more refined design algorithm allowing for better and faster optimization of PCHEs, as well as the accumulation of more thermal-hydraulic and mechanical test data for the validation of design algorithms. Additional activities are planned to reduce other contributions to PCHE cost including in the areas of both featured plate and header fabrication.

This ongoing work funded by the DOE Office of Nuclear Energy (NE) on PCHE optimization, plate fabrication, header fabrication, and failure modes in addition to the support of complimentary work being performed under NEUP, DOE-FE, and DOE-EERE funding will provide extensive knowledge and high degree of confidence in the performance and cost of PCHEs as strong technology candidate for use in a

10 MWe sCO2 Brayton cycle demonstration system under the Supercritical Transformational Electric Power (STEP) crosscut initiative.

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NOMENCLATURE

Abbreviations

ASME	= American Society of Mechanical Engineers
CAPEX	= Capital Expense
CRADA	= Cooperative Research and Development Agreement
DAQ	= Data Acquisition
DOE	= Department of Energy
EES	= Engineering Equation Solver
EPRI	= Electric Power Research Institute
FE	= Fossil Energy
MAWP	= Maximum Allowable Working Pressure
MCHE	= Microchannel Heat Exchanger
NE	= Nuclear Energy
NESL	= Nuclear Energy Systems Laboratory
OPEX	= Operating Expense
ORNL	= Oak Ridge National Laboratory
PCHE	= Printed Circuit Heat Exchanger
RTD	= Resistance Temperature Detector
sCO2	= Supercritical Carbon Dioxide
SEARCH	= Selection, Evaluation, And Rating of Compact Heat exchangers
STEP	= Supercritical Transformational Electric Power
SwRI	= Southwest Research Institute
VFD	= Variable Frequency Drive
VPE	= Vacuum Process Engineering

Symbols

AR	=	Aspect Ratio h / H
b	=	Unit-width of ASME BPVC equations, equal to Unity
C	=	A Factor in the UG-34 Equations Defined to be 0.20 by VIII-1-13-4(f)
$\dot{C}$	=	Capacitance Rate
C_R	=	Capacitance Rate Ratio
d	=	The "short span" of the Flat End Closure, equal to R
D	=	The "long span" of the Flat End Closure, equal to 2R
E	=	Weld Joint Efficiency Factor
h	=	PCHE Channel Width or Enthalpy when subscripted
H	=	PCHE Channel Height
I_1	=	Moment of Inertia of the Section of Thickness t_1
k_m	=	Metal Thermal Conductivity
$\dot{m}$	=	Mass Flow Rate
N	=	Number of Sub-Heat Exchangers
N_{ch}	=	Number of Channels
NTU	=	Number of Transfer Units
p	=	Perimeter
P	=	Pressure
$\dot{q}$	=	Heat Flow

R	=	Half-cylindrical Header Shell Inner Radius
R''_f	=	Area-specific Fouling Factor
S	=	Maximum Allowable Stress
Sb	=	Bending Stress
Sm	=	Membrane Stress
St	=	Total Stress
t_1	=	Half-cylindrical Header Shell Thickness
t_2	=	Plate Thickness Between Side A and Side B Channels
t_3	=	PCHE Core Width of Length Opposite a Half-Cylindrical Header
t_4	=	Stay Plate Thickness
t_m	=	Metal Wall Thickness
T	=	Temperature
UA	=	Conductance-Area Product
W	=	PCHE Core Width
Δx	=	Length of a Sub-heat Exchanger

Subscripts

A	=	Side A
B	=	Side B
cap	=	Flat End Closures
C	=	Cold Side
H	=	Hot Side
i	=	Sub-heat exchanger index number
m	=	Based on Membrane Stresses
plate	=	PCHE Core which is the Diametral Stay Plate of the Half-cylindrical Shell
shell	=	Half-cylindrical Header Shell
t	=	Based on Total Stress

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