PRELIMINARY STRUCTURAL ASSESSMENT OF A PRINTED CIRCUIT HEAT EXCHANGER WITH S-SHAPED FINS

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ABSTRACT

A preliminary study has been performed to evaluate the mechanical integrity of a printed circuit heat exchanger (PCHE) with S-shaped fins for gas-cooled fast reactors (GFRs) coupled to a supercritical CO_2 (s-CO_2) Brayton power cycle. PCHEs are one type of compact heat exchangers that are assembled by stacking and diffusion-bonding multiple etched plates. In the application of GFRs, the PCHE serves as an intermediate heat exchanger (IHX) that transfers heat from the primary coolant helium to the working fluid s-CO_2. The s-CO_2 Brayton cycle is considered as a promising power conversion system for nuclear power plants because of its high thermal efficiency at high temperatures. With this it comes to the high operating pressure of s-CO_2, a potential challenge to the structural strength of PCHE's diffusion bonding.

ANSYS-Mechanical was used to simulate the stress distribution of S-shaped fin channels. The reference model is based on the experimentally tested S-shaped fin PCHE from the Tokyo Institute of Technology (TIT). S-shaped fin PCHEs reduce pressure drop while maintaining good heat transfer performance compared with zigzag channel PCHEs. Although thermal hydraulic characteristics have been studied, the mechanical strength of S-shaped fin PCHEs isn't as well established. In this paper mechanical stresses in S-shaped fins induced by original pressure loading were investigated. As plasticity of base material SS 316 taken into account, small portion yield at tips of the S-shaped fins. A sensitivity study was performed due to the complexity of surface geometry and computational cost. Meshing schemes, boundary conditions and alternative models were discussed to show that the reference model is capable of simulating the stress field. Local stresses from the simulation that are used to evaluate the design and service limit assure the compliance with ASME standards. Because of the differences between the reference model and real S-shaped fins, either should the base material change to alloys with higher mechanical strength or the design of fins be improved to reliably maintain the structural integrity.

KEYWORDS
PCHE, GFR, s-CO_2 cycle, S-shaped fin, stress concentration, ASME

1. INTRODUCTION

The nuclear energy has been expected to become one of the promising energy solutions in the future. The development of the Generation IV reactors as new reactor power plants design draw much attention across the world. The reactor core concepts as well as the power conversion unit configurations have been studied extensively to obtain an optimized nuclear reactor plant design. An old power cycle concept, the supercritical carbon dioxide (s-CO_2) Brayton cycle has been revived recently due to its high thermal efficiency and compactness [1]. Some of the research efforts have been made to investigate the intermediate heat exchanger (IHX) that transfers the heat from the primary coolant system to the s-CO_2 turbine power generation cycle. A joint research group at the Ohio State University (OSU) and University of Wisconsin-Madison has shown interests in developing an advanced IHX for various primary coolants, such as helium, liquid metal and molten salts. Prototypic design of an IHX for a helium gas-cooled fast reactor (GFR) coupled with an indirect s-CO_2 Brayton cycle was proposed [2]. Major focus has been placed on one particular compact IHX design, a Heatric Printed Circuit Heat Exchanger (PCHE).

Heatric is a British heat exchanger manufacturing company that developed a commercial micro-channel heat exchanger called PCHE [3]. As is known from its name, this type of heat exchanger involves a photo-chemical etching technique that is usually applied to the electric circuit boards manufacturing industry. PCHEs are essentially plate-type heat exchangers, where the micro-scale fluid flow channels are photo-chemically etched on flat metal plates. Theses etched plates are then stacked up and diffusion-bonded to form a heat exchanger block. Fig. 1 shows the traditional etched semi-circular channels and the plate configuration. It is claimed that the diffusion-bonded interface possesses the identical mechanical strength and properties as the bonded material does [4].

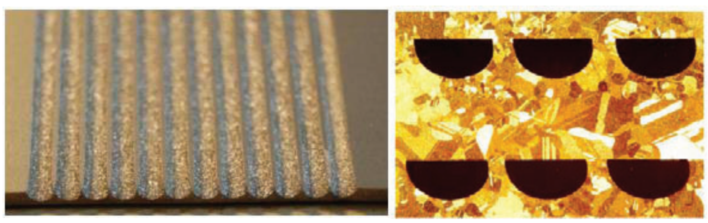


Figure 1. Etched Semi-circular Channels and Plate Configuration [4].

In addition to the capabilities to operate under such harsh operating conditions, the exhibited thermo-hydraulic performances and economic advantages are attracting. The high compactness of PCHEs reduces the size dramatically as well as the difficulties in transportation and installation of the unit, compared with the conventional shell-and-tube heat exchangers [5]. However, the performances of PCHEs are strongly dependent upon flow channel configurations and patterns. Accordingly, studies concerning the PCHE channels have been carried out in the past few years. To date, four different types of flow channels have been proposed for PCHEs, which are straight, zigzag, S-shaped fin and airfoil fin channel, as shown in Fig. 2 [6]. The straight and zigzag channels are often used in Heatric manufactured PCHEs. Bartel et al. performed a comparative analysis of PCHEs with straight and zigzag channels and found that PCHEs with zigzag channels have better heat transfer performance but larger pressure drop than those with straight channels [7]. The S-shaped fin channel concept was first introduced in 2007 by Tokyo Institute of Technology (TIT). It was found to reduce the pressure drop substantially compared with zigzag channels [8]. The airfoil fin channels were first developed in 2008 by POSTECH as an entirely-new flow channel model to address the pressure drop reduction issue [9]. Although the thermo-hydraulic characteristics of these surface geometries have been studied, relatively little attention has been given to their structural integrity. Lee et al. have investigated the structural integrity of zigzag PCHEs and found that as plasticity sufficiently lowers local stress concentration at channel tips, PCHEs that are made of SS 316 are anticipated to assure mechanical integrity in compliance with ASME standards [10]. In this paper a preliminary study on the numerical stress analysis of S-shaped fins has been performed to assess the structural integrity of PCHEs with this particular advanced flow channel and give some insights for future studies.

Figure 2. Four PCHE Surfaces: Straight, Zigzag, S-shaped Fin and Airfoil Fin (Left to Right).

2. REFERENCE MODEL

The S-shaped fins originate from sine curve flow models. Tsuzuki et al. found that the sine curve flow channels could reduce pressure drop compared with the conventional zigzag channels. However, numerical studies showed that the heat transfer performance was reduced by 16% simultaneously at the presence of the stagnant flow areas near bend corners [11]. Therefore, an innovative invention was introduced, which was to cut the bend corner and shift the down-stream fins to the center of the channels to form an offset configuration. This model could mitigate the stagnant flow and enhance the heat transfer. The individual fin was elongated to two sharp tips at the head and tail. The process of developing S-shaped fins is shown in Fig. 3. This new configuration of fins has been compared with zigzag channels with identical geometric parameters in the experimental tests by TIT, and the results showed that the pressure-drop factor of the S-shaped fin model is 4-5 times less than that of zigzag model, although Nusselt number is 24-34% less simultaneously [8]. It was found at OSU that in the optimization process of the IHX PCHE design, the reduction of the pressure drop in the micro-scale channels outweighs the enhancement of the heat transfer from economic point of view. Therefore, the S-shaped fin channels are competitive among other advanced channel configurations in the PCHE design optimization.

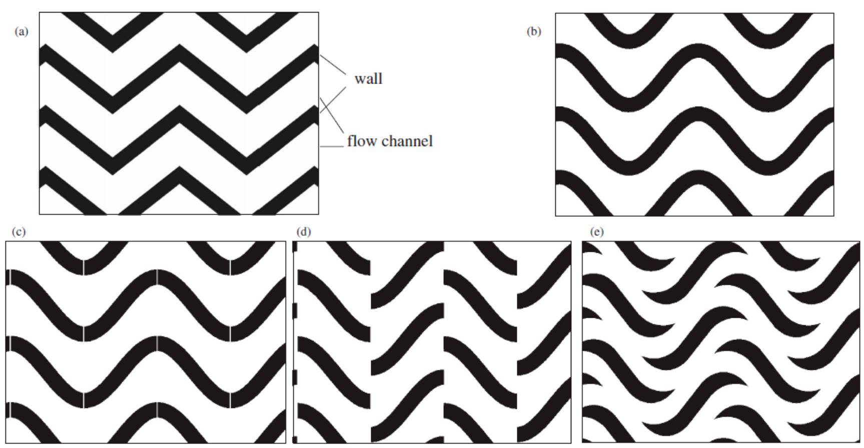


Figure 3. The Process of Developing S-shaped Fins [9].
(a) Zigzag Channels; (b) Wavy (Sine Curve) Channels; (c) Wavy Channels Cut; (d) Fin Off-set Configured; (e) Tips Elongated

Table I lists the geometric characteristics of the S-shaped fins in TIT experimental studies. Figure 4 shows some of the surface characteristic parameters. As mentioned, the S-shaped fins are essentially sinusoidal curved fins. Thus those curves are defined by two interrelated sinusoidal functions, which are characterized by three surface parameters, including the fin angle ϕ, the fin length l_f and the fin width d_f. The configuration of neighboring fin is defined by the fin gap g_p. In practice, the lateral and longitudinal pitch (p_x and p_y) are more useful.

Table I. Specific parameters for S-shaped fin channels

Items	TIT experimental model	Reference model
Plate thickness, t_p, mm	1.5	1.5
Longitudinal pitch, p_y, mm	7.565	7.565
Lateral pitch, p_x,, mm	3.426	3.426
Fin angle, ϕ, °	52	52
Fin width, d_f, mm	0.8	0.8
Fin length, l_f, mm	4.8	4.880
Fin height, H, mm	0.94	0.94
Hydraulic diameter, D_h, mm	1.31	1.13

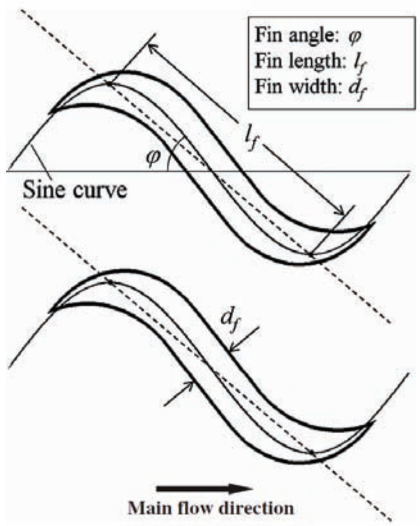


Figure 4. Surface Characteristics of S-shaped Fins [10].

The reference model is a re-constructed 3-D model that is based on the experimental S-shaped fin channel model used by TIT. Some measurements of the geometrical characteristics were performed in the 3-D modeling. Although this 3-D model is identical with the TIT experimental model in terms of the plate thickness, longitudinal pitch, lateral pitch and fin height, etc., the hydraulic diameter and fin width that is measured through 3-D model is slightly different from the experimental model. In the preliminary study, the reference model has been used.

In the numerical modeling of the stress distribution for the reference model, it is important to accurately simulate the mechanical behavior of the construction base material. Stainless steel 316 has been selected in the numerical modeling. Only mechanical stress induced by initial pressure loading has been modeled in the preliminary analysis where the thermal stress is not considered. Nonetheless, a reference temperature 550 ºC was determined based on the operating conditions of a prototypic helium-s-CO_2 IHX PCHE design, as listed in Table II. The reference temperature is essentially the average temperature in the IHX. The stress-strain relationship can be referenced to the ASME Boiler and Pressure Vessel Code (BPVC) [11].

Table II. Operating condition for a prototypic zigzag-S-shaped-fin PCHE

Item	Primary side	Secondary side
Working fluids	Helium	s-CO_2
Mass flow rate, kg/s	428	2991
Pressure, MPa	7	20
Inlet temperature, °C	800	488.8
Outlet temperature, °C	504.4	665.1

3. STRESS SIMULATION

In this stress assessment, ANSYS-Mechanical has been used to model the stress distribution. A rectangular-shape model with no fillets was chosen for the reference simulation model. The fillets of the S-shaped fin that connect the fin body and the bottom plate, as shown in the Fig. 5(a), are formed due to the nature of the photo-chemical etching. These fillets that are essentially the by-products of the etching process are beneficial because of its potential mitigation of stress concentration at the bottom plate, as is discussed in the later section. Hence, the stress analysis should focus on the stress distribution at the diffusion-bonded areas that connect the fin body and the upper plate. It is noted that in the no-fillet model a 0.1 mm roundness of tips of the fin is also included.

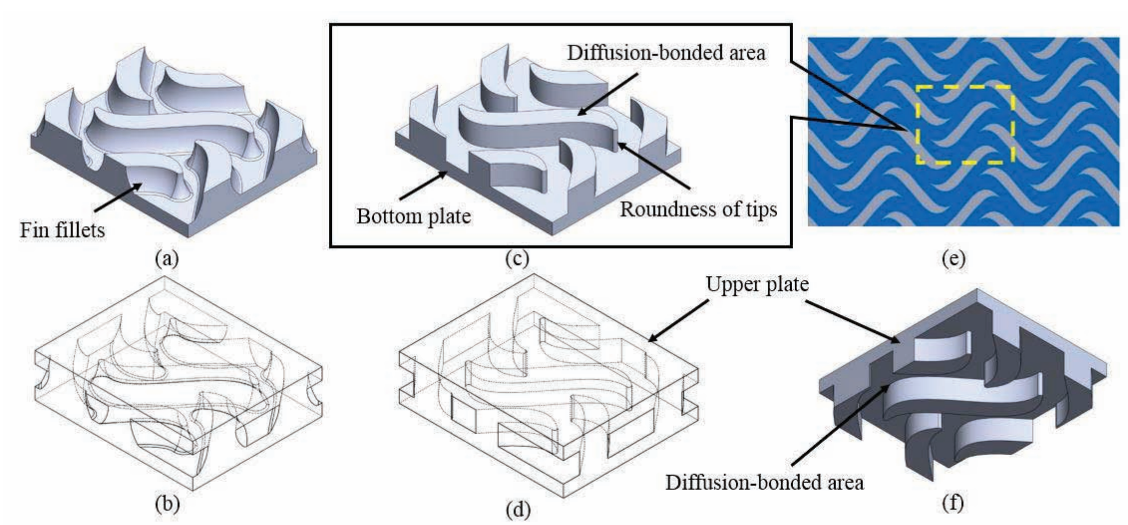


Figure 5. (a) The Actual Chemically-etched S-shaped Fin Channel Model with Fillets; (b) The Fillet Model with an Upper Plate; (c) The No-fillet Rectangular Model; (d) The Reference Model; (e) The Configuration of the Rectangular Model in the Etched Plate; (f) The Diffusion-bonded Areas at the Upper Plate in the Reference Model.

The reference model a complete S-shaped fin as well as four half-fins. As is observed from an individual S-shaped fin, the diffusion-bonded areas may experience large stresses. Especially, the tips at the head and tail of fins are expected to have excessive stress concentration and local plastic deformation. Therefore, these areas should be carefully addressed in the process of meshing. In the reference simulation model, the diffusion-bonded areas are meshed with element size 0.100 mm, while the tips of the fin are meshed with element size 0.015 mm, and the rest of the model is meshed with the maximum element size 0.520 mm, as shown in Fig. 6. Since the local plasticity is expected to occur at the tips of fins, the non-linear mode of the solver is activated. Therefore, a non-linear plastic simulation has been carried out as opposed to a linear elastic simulation for the S-shaped fin model to address local excessive stress concentration at tips of fins.

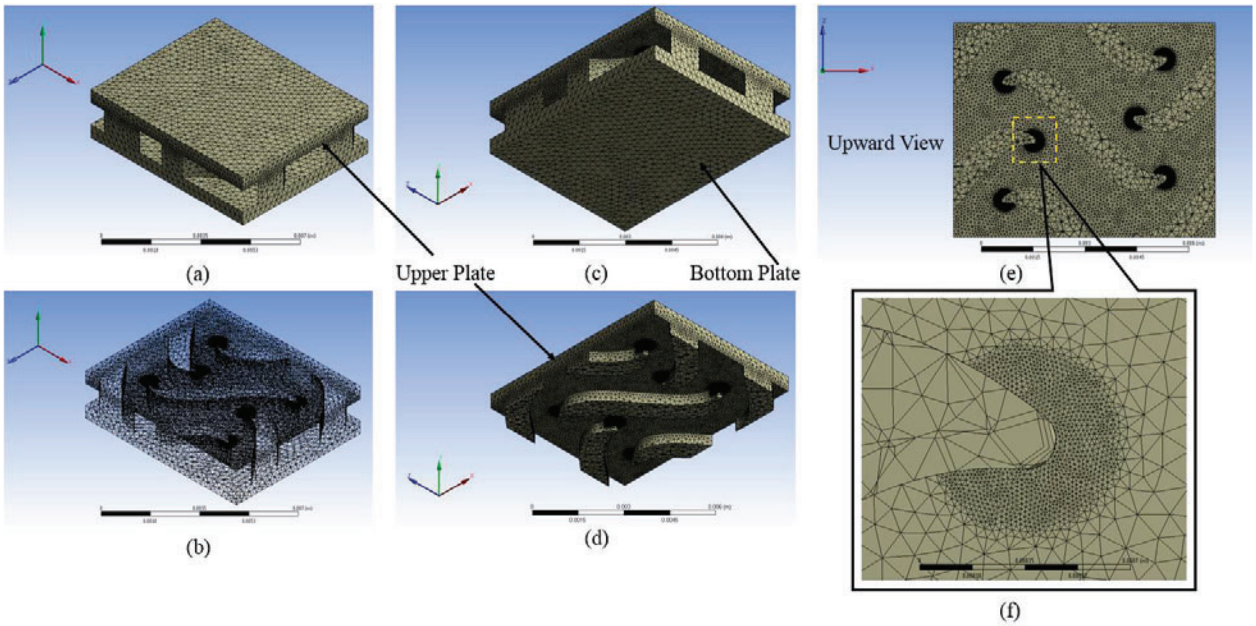


Figure 6. The Meshing Scheme for the Reference Simulation Model
(a) Downward Isometric View; (b) Mesh Formation; (c) Upward Isometric View; (d) Upward Isometric View without Bottom Plate; (e) Upward View from the Bottom; (f) Close View of Meshing at a Tip of the Fin, with Element Size 0.015 mm.

The constant pressure load is imposed on both surfaces of the upper plate, with 20 MPa on the lower surface and 7 MPa on the other. With the assumption that the middle cross-section of an entire fin is structurally the least weak location, four sides of the reference model were set to be fixed supports as boundary conditions. The bottom plate surface was also set to be fixed support. Because the non-linear behavior of the simulation is expected, the force convergence and Newton-Raphson residuals options are activated to monitor the iterative computation process. Since in the preliminary study only stresses induced by mechanical loading were investigated, no thermal stresses are considered in the setup.

The visualized stress intensity distribution results indicate that the most severe deformation occurs at the tips of fins, with the maximum stress intensity reaching 146.4 MPa. Figure 7 shows the stress field and the mesh scheme near the tips of fins. Note that the PCHE construction material's yield strength is 116 MPa at temperature 550 °C. This indicates that a portion of the fin has yielded. Figure 8 shows five slices of planes of stress distribution at different locations of the fin. Note that plane P0 is essentially the lower surface of the upper plate in the model. It implies that the tips of fins have enormous stresses, whereas the diffusion-bonded areas along the perimeter of the fin do not experience extreme stress concentration. By checking the nodal stress intensity information of these planes, it is determined that the portion of the fin that yield accounts for less than 1% of the total elements in each plane, as listed in Table III. The elements in the entire model that undergo plastic deformation accounts for approximately 2.14%. In addition, the local stress of elements in each sliced plane was also analyzed, as listed in Table III. The elements with local stress beyond 101 MPa accounts for 3.16% of the entire model, including the elements that yield.

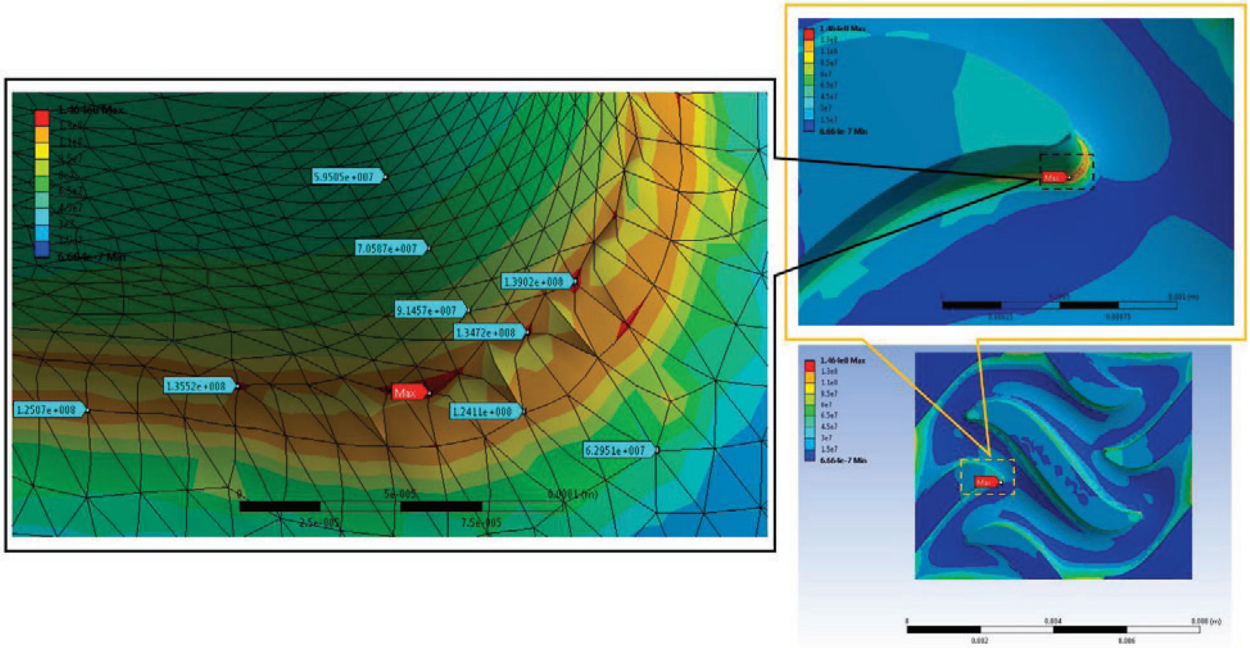
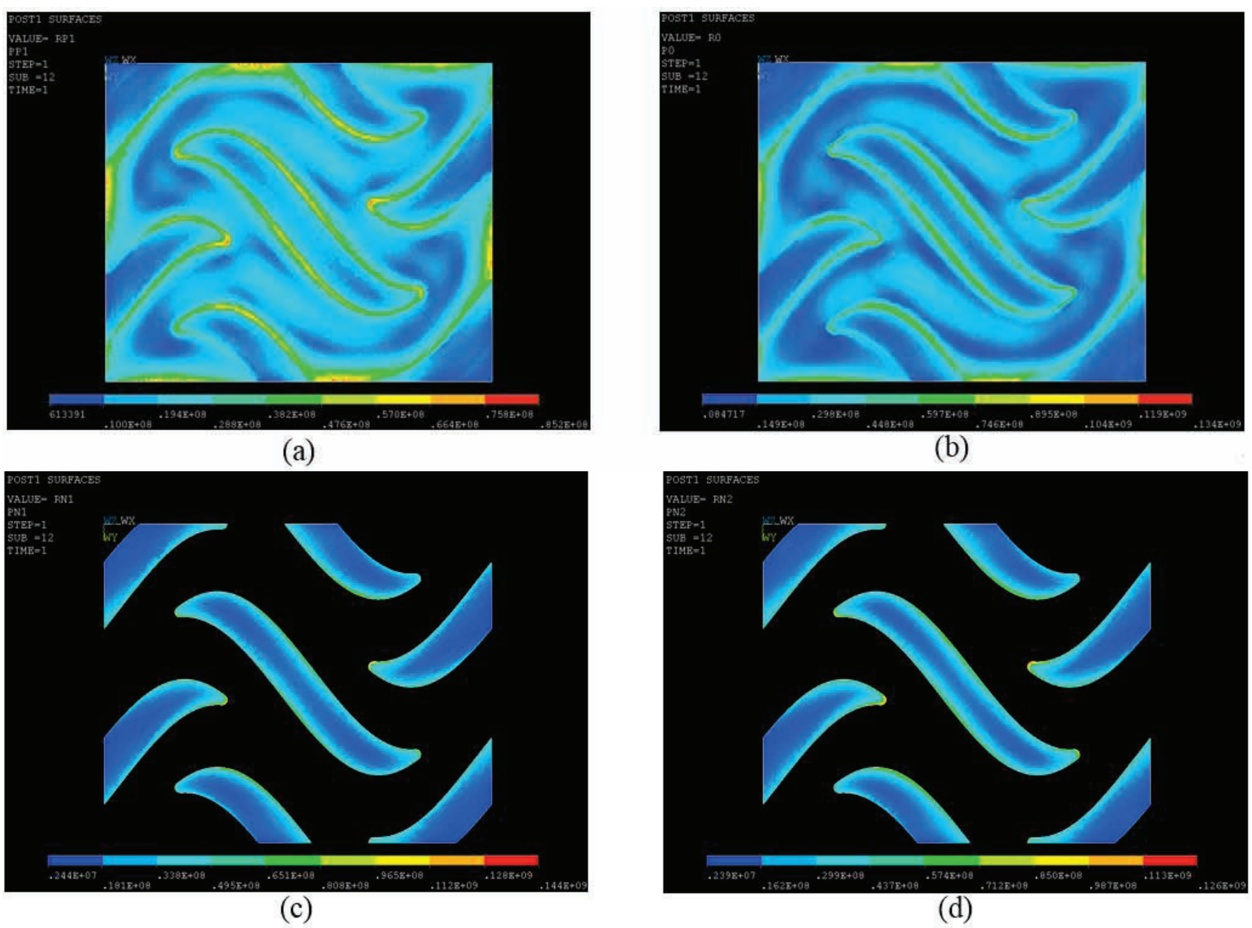
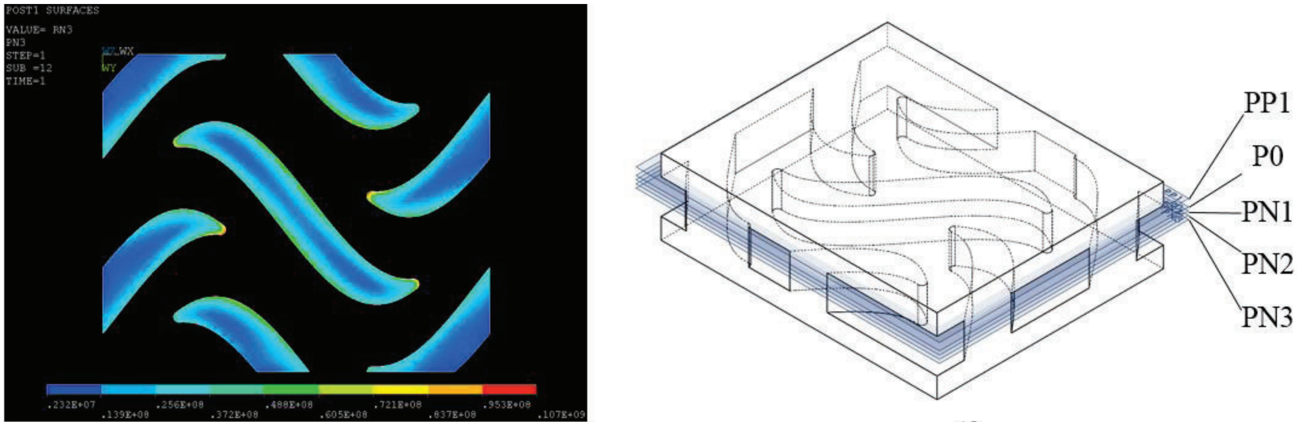


Figure 7. The Stress Intensity Distribution of the Reference Model at one of the Tips of Fins, Maximum Stress Intensity 146.4 MPa, with Element Number = 595,395.





(e) (f)

Figure 8. Sliced Planes of Stress Intensity Distribution of the Reference Simulation Model (a) Plane PP1; (b) Plane P0; (c) Plane PN1; (d) Plane PN2; (e) Plane PN3; (f) Configuration of the Sliced Planes in the Reference Simulation Model.

Table III. Information of the sliced planes of stress distribution of the reference simulation model

Plane Surface	Offset distance relative to Plane P0, mm	Maximum stress intensity, MPa	Yielding, %	Local Stress > 101 MPa, %
PP1	+0.0437	85.2	0	0
P0	0	134.3	0.320	0.782
PN1	-0.0092	143.5	0.923	1.845
PN2	-0.0196	126.3	0.517	2.114
PN3	-0.0319	107.0	0	0.323

4. SENSITIVITY STUDIES

Based on the reference model, a series of sensitivity studies have been carried out. There are several issues concerning the accurate simulation of the stress. Firstly, it is necessary to find an appropriate meshing scheme for accurate simulation. The meshing refinement needs to be balanced with the computational cost. Furthermore, the reference simulation model needs to be compared with a large-scale models to verify whether or not the effect of the boundary conditions on the local stress concentration simulation is negligible. In this case, a larger rectangular-shape model containing multiple fins has been simulated. Besides, due to the geometric periodicity of the S-shaped fin configuration, it is proposed to study whether a fin-scale simulation is feasible. The performed sensitivity studies may provide some insights on stress simulations for complex surface geometries in compact heat exchangers.

Table IV has summarized the mesh-sensitivity study results for various meshing schemes. As is mentioned before, the mesh sizing for the diffusion-bonded areas and the tips of the fin should be different. The indicator of reasonable meshing scheme is the convergence of the maximum stress intensity. Regarding the tips of fins, the results show that the meshing scheme of case 5 with the 0.015 mm element sizing is the most appropriate because the maximum stress intensity changes little while the element number of the case 6 with 0.01 mm element sizing almost doubles. This indicates the maximum stress intensity converges with mesh sizing 0.015 mm at tips of fins. The same approach can be used to select the most suitable meshing scheme for diffusion-bonded areas. Therefore, case 5 has been selected for the reference simulation model, as discussed in the previous section.

Table IV. Information for different meshing schemes

Case	Type of model	Mesh sizing at diffusion-bonded areas, mm	Mesh sizing at tips of fins, mm	Element number	Maximum stress intensities, MPa
1	Single-fin	0.10	0.05	161,742	107.4
2	Single-fin	0.10	0.03	245,127	123.8
3	Single-fin	0.10	0.025	382,557	133.4
4	Single-fin	0.10	0.02	391,874	133.8
5	Single-fin	0.10	0.015	595,395	146.4
6	Single-fin	0.10	0.01	1,385,467	149.6
7	Single-fin	0.08	0.025	418,177	138.9
8	Single-fin	0.08	0.015	702,232	147.0
9	Single-fin	0.07	0.025	455,047	138.7
10	Single-fin	0.06	0.015	888,674	145.5
11	Multiple-fin	0.08	0.025	1,019,119	131.6
12	Multiple-fin	0.10	0.015	1,742,121	149.5

The comparison of the reference simulation model with the large-scale model has been made. The major concern about the reference model is that it is uncertain whether stress fields near the tips and the diffusion-bonded areas of fins are affected by the assumed fixed support boundary conditions at four sides of the model. During the stress analysis of the large-scale model, it is assumed that the S-shaped fin at the center of the large-scale model will be the least affected by the specified boundary conditions, since it is relatively far away from them. Quantitatively, the effect of the boundary conditions on the tips of the fin can be evaluated by comparing the maximum stress intensity near the tips. The information of case 5 and 12 from Table IV shows an agreement on the maximum stress intensity for two models. It should be noted that there is still lack of effective tools to compare the stress distribution of two models from quantitative point of view because of the complicated surface geometry of S-shaped fins. An alternative way for the comparison is to analyze the yielded portion of the entire model. It is found that there has been 2.83% of elements in the large-scale model that yield, while only 2.14% of the reference model yield. Additionally, 4.04% of elements in the large-scale model have local stress beyond 101 MPa, whereas in the reference model it is 3.16%. This can be explained by observing the stress field near the tips of the fin shown in Fig. 9. It can be found that in the reference simulation model the stress field at the bottom surface of the upper plate near the tip of the fin is distorted compared with the similar location in the large-scale model. This implies that the stress field in the reference model is affected by specified boundary conditions in the simulation, especially when it comes to the portion of elements at tips of the S-shaped fin that have large local stresses. However, with little deviation of the maximum stress intensity and distortion of the stress filed at certain locations, the reference model is still capable of simulating the stress filed of S-shaped fins in the preliminary stress analysis, considering the extremely saved computational cost.

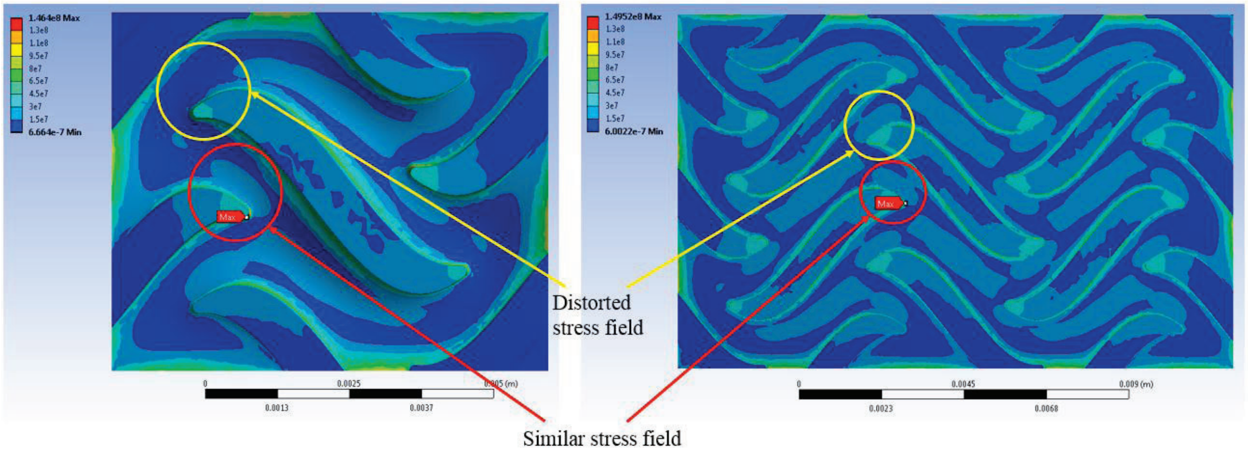


Figure 9. The Comparison of Global Stress Intensity Distribution between the Reference Model (Left) and the Multiple-fin Large-scale Model (Right).

A fin-scale model is also established to be simulated for stress analysis. Figure 10 shows the simulation results for different boundary conditions imposed on the sides of the model. The fin-scale model was originally proposed for the stress assessment due to its small scale to save computational cost. Theoretically, this model can be imposed with linear periodic symmetry to address the inherent periodicity of S-shaped fins. Unfortunately, the non-linear simulation always failed with linear periodic boundary condition at either direction. With alternative boundary conditions, such as fixed support and free support, the results shows that the stress fields are distorted largely and thus are strongly affected by boundary conditions compared with the reference model. Therefore, it is no longer suitable for S-shaped fin stress analysis.

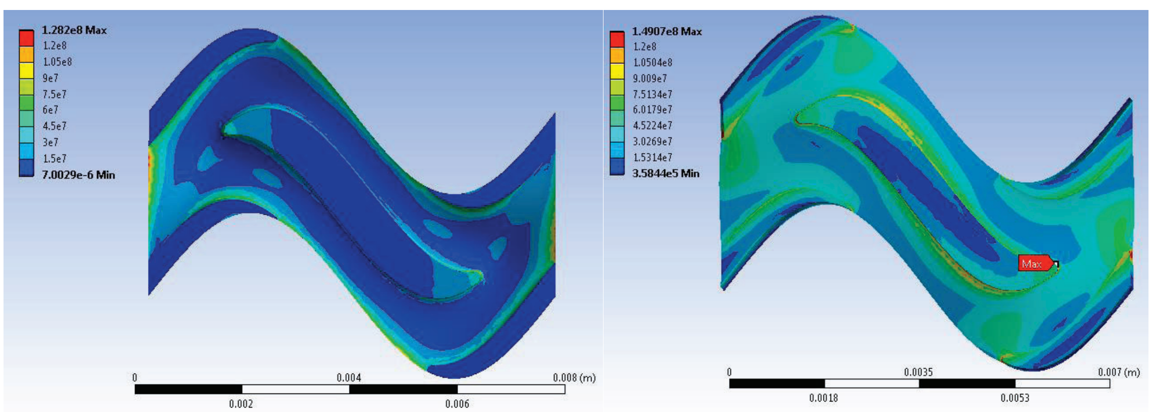


Figure 10. The Stress Intensity Distribution of the Fin-scale Model with Fixed Support (Left) and Free Support (Right) Boundary Conditions.

5. STRESS ANALYSIS AND DISCUSSION

Using ASME standards, it is possible to evaluate the structural integrity of S-shaped fin PCHEs with the obtained stress simulation results. In the ASME BPVC section III [12], specific rules are described for construction of nuclear facility components. As is also discussed in Lee et al. [10], these rules are applicable to PCHEs, where channels are treated as pressure-containing vessels. Considering the Design loading and Level A service loadings, we can compare the local stress levels obtained from the simulation results with specified design criteria in ASME code. For design limit, the general membrane stress intensity, derived from P_m, shall not exceed S_0, the maximum allowable stress intensity:

$$P_m \leq S_0 \ , \tag{1}$$

and the combined primary membrane stress plus bending stress intensity, shall not exceed 1.5 S_0:

$$\left(P_L + P_b\right) \leq 1.5 S_0 \ , \tag{2}$$

where P_L and P_b are the local primary-membrane stress and the bending stress, respectively. For service limits of Level A service loadings, the general primary membrane stress intensity P_m, shall not exceed S_{mt}:

$$P_m \leq S_{mt} \ , \tag{3}$$

where the S_{mt} values are the lower of two stress intensity values, S_m, which is the lowest stress intensity value at a given temperature among the time-independent strength quantities, and S_t, which is a temperature and time-dependent stress intensity limit. The combined primary membrane plus bending stress intensities for Level A service loadings, shall satisfy the following limits with:

$$P_L + P_b \leq K S_m \ , \tag{4}$$

$$P_L + P_b / K_t \leq S_t \ . \tag{5}$$

The factor K_t accounts for the reduction in extreme fiber bending stress due to the effect of creep, given by:

$$K_t = \left(K + 1\right) / 2 \ . \tag{6}$$

The factor K is the section factor for the cross section being considered. For conservative estimation, it is assumed that K is set to be 1. The specified stress intensities limits are listed in Table V. Then it is possible to use local stress (P_L+P_b) for stress assessment in compliance with the rules specified above.

Table V. The specified stress intensities limits in ASME BPVC [12]

Temp., ^{o}C	S_0, MPa	S_{mt}, 10^5 hr, MPa	S_{mt}, 3×10^5 hr, MPa	S_t, 10^5 hr, MPa	S_t, 3×10^5 hr, MPa
500	107	106	106	131	125
525	101	105	105	118	108
550	88	101	87	101	87
575	77	79	67	79	67

First of all, according to the design limit (2), the maximum allowable local stress is 132 MPa. Only 1.38% of total elements in the reference model fail to satisfy this criteria, and all of them occur at tips of S-shaped fins. Secondly, according to the service time limit, except 3.16% of total elements beyond 101 MPa, 10^5 hours of service time is allowed for S-shaped fin PCHEs, which is approximately 11.5 years. This implies that the S-shaped fin body can easily satisfy the design limit and service limit, with only small portion of the fin at fin tips being possible to limit the design and reduce service time. Therefore, the mechanical integrity concerning the S-shaped fin bodies and diffusion-bonded areas can be maintained in compliance with ASME standards.

As is mentioned before, the reference model is different from the S-shaped fins in the real situation. Thus, it is still challenging to state that the simulation can represent the real stress situation of S-shaped fins. One of major concerns is the difference between the selected simulation model and the real S-shaped fin channels. Firstly, as shown in Fig. 11(e), the reference simulation model does not include any fillets that connect the fin body to the bottom plate. As is known, in the photo-chemical etching technique, as the etchant attacks the exposed plate surface area to form desired channels, a characteristic roundness will occur with rounded interior corners, also known as fillets in S-shaped fin channels [13]. These fillets can mediate the stress concentration that could occur at bottom plates. Hence, the current simulation results tend to be conservative in the absence of fillets. Furthermore, the roundness of tips of the S-shaped fins in the reference model is set to be 0.1 mm according to the actual shape obtained by etching in TIT experiment. Figure 12 shows the particular round effect of the etching process of S-shaped fins that has been investigated before [11]. However, the dimension of the tips' roundness varies because it is strongly dependent upon the manufacturing conditions. It is still unclear how the roundness dimension may affect the mitigation of stress concentration at tips of fins. Moreover, the diffusion-bonded areas that connect the fin body with the upper plate have sharp 90º angle in the reference model, as shown in Fig. 11(c). However, this is not necessarily the case since in reality the diffusion-bonding process leads to physical roundness at the sharp tips [10]. Using elastic model for simulation, such sharp tips at the diffusion-bonded areas may result in diverging stresses. The similar situation for zigzag channel tips has been discussed in the literature [10]. Fortunately, in this study the plasticity of SS 316 at sharp tips have been well captured in the simulation. The plastic deformation, or the yielding of some portion of the fin tips, accommodates the excessive strain energy that caused by the sharp tips. In other words, the current simulation is more conservative than that in real situation where sharp tips are unintentionally rounded during diffusion-bonding. Therefore, the actual stress situation of the manufactured S-shaped fins may be less harsh than the simulated models thanks to the nature of the chemical etching and diffusion bonding technique.

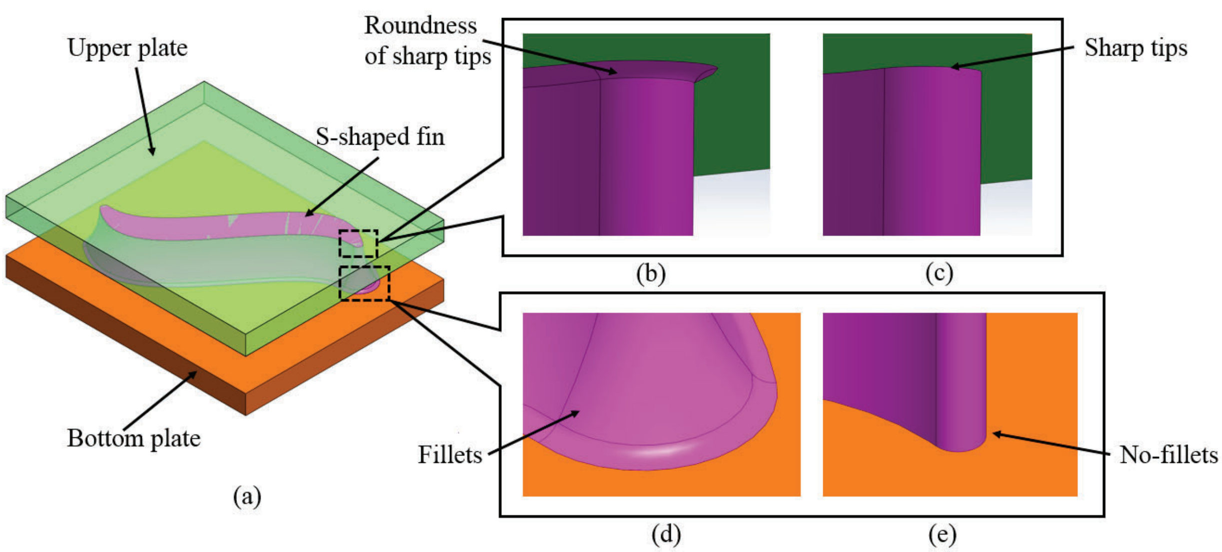


Figure 11. Surface Geometrical Difference between the Actual S-shaped Fin and Reference Model (a) Actual S-shaped Fin; (b) Roundness of Sharp Tips; (c) Sharp Tips in the Reference Model; (d) Fillets at the Bottom Plate; (e) Reference Model with No Fillets.

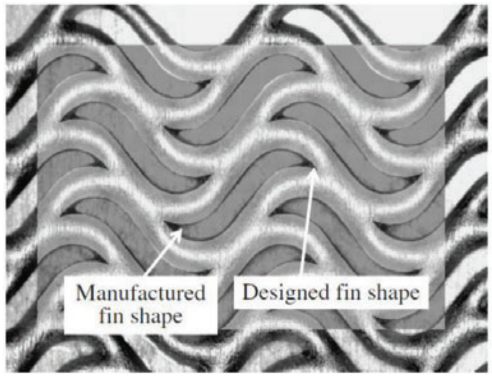


Figure 12. A Plane View of the S-shaped Fin PCHE Surface and the Rounding Effect in Manufacturing Process [11].

The stress distribution of S-shaped fin channels is inherently difficult to be analytically modeled. Accordingly, the unique geometry and high-pressure application of the S-shaped fins make it particularly challenging to be referenced to the existing codes such as ASME code. The ASME BPVC section VIII Division 2 describes the design procedures and criteria for the design of pressure equipment based on design-by-rules methodologies [14]. The diffusion-bonded PCHE channels are generally considered to be pressure-containing vessels. It is possible to follow relevant rules in the BPVC for the mechanical design of straight or zigzag semi-circular channels for intermediate-pressure applications. Unfortunately, when it comes to S-shaped fins and high-pressure applications, the design-by-rules methods are no longer suitable due to the complexity of channel configurations and the elastic-plastic behavior of the material under high pressure. An alternative tool recommended by BPVC is the design-by-analysis method. Although it seems to be powerful tool for the mechanical design of the S-shaped fin channels, it requires the stress linearization and classification of the numerical results based on an accurate modeling to satisfy the design criteria. Particularly, concerning S-shaped fin channels, this task will be arduous and challenging. The numerical simulation of stress analysis for S-shaped fin channels in this paper is a preliminary step in the design-by-analysis method in compliance to BPVC code, and further studies are needed in the future.

It should be noted that the reference temperature 550 ºC does not reflect the highest temperature that some spots of the prototypic PCHE may experience according to the operating conditions, and thermal stresses are not considered in the current stress analysis. These factors may further limit the design and service time. It is also possible that the working fluid s-CO_2 is likely to affect the actual life time of PCHEs. It is then suggested that either should the base structural material of PCHEs change to alloys with higher mechanical strength such as Alloy 617, or the design should be modified and optimized to assure structural integrity and extend service time at prototypic operating conditions.

6. CONCLUSIONS

In this study, four PCHE surface geometries were introduced, and the mechanical concern for S-shaped fin channels was addressed. Mechanical stress induced by original pressure loading in a reference S-shaped fin model was simulated using ANSYS-Mechanical. It was found that the excessive stress concentrations occur at tips of S-shaped fins when imposed with high pressure differential loading. Small portion of the fin yields while the rest of the fin body remains low stress level. A sensitivity study was performed to select appropriate meshing schemes and to investigate the effect of boundary conditions on the reference model. It showed that the stress field is affected by boundary conditions but still acceptable. The design and service limit prescribed in the ASME BPVC code was used to evaluate local stresses of the S-shaped fin, and the reference model was allowed for 11.5 years of service. However, due to differences between the model and real situation, it is suggested to either change base material to alloys with higher mechanical strength or to improve the design to assure structural performance in the future.

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REFERENCES

1. V. Dostal, P. Hejzlar and M.J. Driscoll, "The Supercritical Carbon Dioxide Power Cycle: Comparison to Other Advanced Power Cycles", *Nuclear Technology*, **154**, pp. 283-301 (2006).
2. P. Hejzlar, V. Dostal, M.J. Driscoll, P. Dumza, G. Poullennec and N. Alpy, "Assessment of Gas Cooled Fast Reactor with Indirect Supercritical CO_2 Cycle", *Nuclear Engineering and Technology*, **38**(2), pp. 109-118 (2006).
3. "Heatric Company Website," http://www.heatric.com (2015).
4. S.K. Mylavarapu, "Design, Fabrication, Performance Testing, and Modeling of Diffusion Bonded Compact Heat Exchangers in a High-Temperature Helium Test Facility", *Doctoral dissertation*, The Ohio State University (2011).
5. K. Gezelius, "Design of Compact Intermediate Heat Exchangers for Gas Cooled Fast Reactors," *Master thesis*, Massachusetts Institute of Technology (2004).
6. S.H. Yoon, H.C. No and G.B. Kang, "Assessment of Straight, Zigzag, S-shape, and Airfoil PCHEs for Intermediate Heat Exchangers of HTGRs and SFRs", *Nuclear Engineering and Design*, **270**, pp. 334-343 (2014).
7. N. Bartel, M. Chen, V.P. Utgikar, X. Sun, I.H. Kim, R.N. Christensen, and P. Sabharwall, "Comparative Analysis of Compact Heat Exchangers for Application as the Intermediate Heat Exchanger for Advanced Nuclear Reactors," *Annals of Nuclear Engineering*, 81, pp. 143-149 (2015).
8. T.L. Ngo, Y. Kato, K. Nikitin and T. Ishizuka, "Heat Transfer and Pressure Drop Correlations of Microchannel Heat Exchanger with S-shaped and Zigzag Fins for Carbon Dioxide Cycles," *Experimental Thermal and Fluid Science*, **32**, pp. 560-570 (2007).
9. D.E. Kim, M.H. Kim, J.E. Cha and S.O. Kim, "Numerical Investigation on Thermal-hydraulic Performance of New Printed Circuit Heat Exchanger Model", *Nuclear Engineering and Design*, **238**, pp. 2369-3276 (2008).
10. Y. Lee and J.I. Lee, "Structural Assessment of Intermediate Printed Circuit Heat Exchanger for Sodium-Cooled Fast Reactor with Supercritical CO_2 Cycle", *Annals of Nuclear Energy*, **73**, pp. 84-95 (2014).
11. N. Tsuzuki, Y. Kato, K. Nikitin and T. Ishizuka, "Advanced Microchannel Heat Exchanger with S-shaped Fins", *Nuclear Science and Technology*, **46**, pp. 403-412 (2009).
12. ASME, *ASME Boiler and Pressure Vessel Code*, section III Division 1, (2004).
13. M.D. Carlson, "Measurement and Analysis of the Thermal and Hydraulic Performance of Several Printed Circuit Heat Exchanger Channel Geometries", *Master thesis*, University of Wisconsin—Madison (2012).
14. ASME, *ASME Boiler and Pressure Vessel Code*, section VIII Division 2 (2004).