Safety Performance Analysis and Evaluation Methods for High-Temperature Microchannel Heat Exchangers

Authors: Yang Yang, Yang Yang

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Abstract

Microchannel heat exchangers represent a promising solution for high-temperature reactor applications. In high-temperature reactors, internal pressure and temperature constitute the primary loading sources for microchannel heat exchangers. To accurately assess the safety performance of high-temperature microchannel heat exchangers, macroscopic and mesoscopic finite element models were established. In accordance with the ASME III-5 standard, systematic investigations were conducted on stress assessment, strain assessment, fatigue damage, and life prediction methodologies for a customized high-temperature microchannel heat exchanger under cyclic operating conditions. The results demonstrate that the high-temperature microchannel heat exchanger satisfies the evaluation requirements of both ASME III-5 and Code Cases N-898, enabling the determination of the maximum mesoscopic fatigue damage value and subsequent prediction of the heat exchanger's maximum safe service life.

Full Text

Study on Safety Performance Analysis and Evaluation Methods for High-Temperature Microchannel Heat Exchangers

YANG Yang, MA Ruoxin, YANG Wen, RAN Bin
Nuclear Equipment Design Institute, Dongfang Electric Corporation, Chengdu, Sichuan, 611731

Abstract

Microchannel heat exchangers represent a promising candidate for intermediate heat exchanger applications in high-temperature reactors. In such reactors, internal pressure and elevated temperature constitute the primary loading sources. To accurately assess the safety performance of high-temperature microchannel heat exchangers, this study establishes both macroscopic and microscopic finite element models of the heat exchanger. Following the ASME III-5 standard, systematic investigations were conducted on stress evaluation, strain evaluation, and fatigue damage and life prediction methods for a customized high-temperature microchannel heat exchanger under cyclic operating conditions. The results demonstrate that the heat exchanger satisfies the evaluation requirements of ASME III-5 and Code Cases N-898, enabling the determination of microscopic maximum fatigue damage values and subsequent prediction of the maximum safe service life.

Keywords: heat exchanger; finite element model; stress; strain; fatigue analysis

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High-temperature nuclear reactors (Very High Temperature Reactor, VHTR) represent an important Generation IV nuclear technology, with core outlet temperatures exceeding 900°C [1,2]. The Intermediate Heat Exchanger (IHX) in VHTRs is a critical component for transferring heat from the primary to the secondary loop. However, current IHX technology for VHTRs remains immature, representing a major shortcoming in VHTR development.

Microchannel heat exchangers offer a highly promising IHX solution [3]. For high-temperature reactor applications, these exchangers operate at temperatures above 750°C and are classified as Class A components under ASME III-5 [4]. Verifying structural safety and boundary integrity for such equipment requires consideration of multiple failure mechanisms: short-term ductile rupture, long-term creep damage, creep-fatigue interaction, gross structural distortion from ratcheting and incremental collapse, loss of function due to excessive deformation, and buckling from short-term and long-term compressive loads. Internal pressure and temperature are the primary loading sources for microchannel heat exchangers. Structural stresses from internal pressure can be categorized as general or local primary membrane stress, primary bending stress, secondary stress, and peak stress. Thermal stresses from temperature are generally classified as secondary and peak stresses. Primary stresses are not self-limiting—if excessive, they can cause short-term ductile rupture, long-term creep damage, or geometric distortion. Excessive secondary and peak stresses or stress fluctuations can lead to local strain accumulation, fatigue failure, or ratcheting.

ASME III-5 stress evaluation for Class A components in ultra-high temperature environments addresses only primary stresses, while secondary and peak stresses are verified through strain evaluation and creep-fatigue damage assessment. This study investigates systematic safety performance analysis methods using a specific microchannel heat exchanger design as an example. The methodology primarily follows ASME III-5 Nonmandatory Appendix HBB-T and Code Case N-898 [5]. First, macroscopic and microscopic finite element models of the microchannel heat exchanger were developed, considering argon arc weld and diffusion bond regions. Then, strain evaluation, creep-fatigue damage assessment, and fracture strength evaluation were performed for primary, secondary, and peak stresses induced by internal pressure and temperature. Finally, the stress, strain, creep-fatigue damage, and life prediction evaluation results are summarized.

1.1 Macroscopic Model

The first step in safety performance analysis is establishing a macroscopic model of the microchannel heat exchanger. According to existing research, microchannel heat exchangers can be manufactured using chemical etching and diffusion bonding. For diffusion-bonded microchannel heat exchangers, the virtual yield stress of welds and base materials is calculated using different methods in ideal elastic-plastic simulations for strain evaluation and creep-fatigue damage assessment. Therefore, it is necessary to define weld regions in the finite element model. Based on weld dimensions and structural considerations, accounting for factors such as heat-affected zones, critical cross-sections, and modeling complexity, a finite element model of the header-nozzle structure containing weld regions was established at the macroscopic level, as shown in [Figure 1: see original paper]. The finite element model employs three element layers through the thickness direction and utilizes quadratic reduced-integration elements. The outer width of the weld between the header and core, as well as between the header and nozzle, is 6 mm, while the inner width is 4 mm. The outer width of the weld between the shell and end plate is 9 mm, with an inner width of 4 mm.

1.2 Microscopic Model

The microchannel heat exchanger core is typically fabricated by alternately stacking and diffusion bonding multiple cold plates and hot plates with thicknesses of 1-2 mm. Semicircular flow channels with radii less than 1 mm are etched on the surfaces of these plates. Considering the thin plate thickness, unknown diffusion bonding heat-affected zone, and the fact that maximum strain in the core often concentrates near channel corners, it is assumed in strain evaluation and creep-fatigue damage assessment based on ideal elastic-plastic analysis that all microscopic solid regions of the core are affected by diffusion welding. Consequently, key parameters such as maximum allowable strain, fatigue life, and creep strength are obtained by multiplying base material values with appropriate reduction factors. Therefore, the entire microscopic model solid region employs uniform material properties, eliminating the need to separately define welding zones, as illustrated in [Figure 2: see original paper]. The thin-walled structures in unit cells utilize no fewer than three element layers through the thickness direction with quadratic reduced-integration elements.

1.3 Macro-Micro Elastic-Plastic Behavior Relationship

To apply ideal elastic-plastic analysis methods to microchannel-containing core components, it is necessary to investigate the microscopic ideal elastic-plastic behavior of the core and establish the relationship between macroscopic and microscopic ideal elastic-plastic behavior. This is accomplished by establishing ideal elastic-plastic microscopic unit cell models for core regions A, B, C, and D, applying periodic boundary conditions, and calculating the relationship between macroscopic and microscopic plastic strain, as well as the relationship between macroscopic yield stress and base material yield stress. The macroscopic and microscopic plastic strains approximately satisfy a linear proportional relationship, with ratios listed in . Additionally, presents the ratio of macroscopic yield stress to base material yield stress.

2.1 Evaluation Method

The primary challenge in applying ASME III-5 to microchannel heat exchangers lies in properly treating components containing microfluidic channels. Since stress evaluation is based on linear elastic mechanics results, micromechanical methods are employed to calculate the macroscopic equivalent elastic modulus, equivalent Poisson's ratio, and other elastic properties of microchannel-containing cores for use in overall structural elastic analysis. Based on the overall structural stress solution, microscopic stress calculations and evaluations are performed for core regions with high stress levels using unit cell models with periodic boundary conditions. The macro-micro mechanical relationship is: microscopic structure determines the macroscopic equivalent elastic properties of the core, while macroscopic stresses applied to the microscopic model influence the microscopic stress distribution.

The stress evaluation methodology for microchannel heat exchangers proceeds as follows: (1) Based on the structural characteristics of microchannel heat exchanger cores, the channel-containing core is divided into different regions. Different regions have different unit cell structures, generally categorized as parallel channels, perpendicular channels, or single-side channels. (2) Establish finite element models of different core microscopic unit cells, apply periodic boundary conditions, and calculate the equivalent macroscopic elastic modulus and Poisson's ratio of the unit cells. At the macroscopic level, the channel-containing core behaves as an orthotropic material with nine independent equivalent macroscopic elastic moduli and Poisson's ratios. Within the linear elastic range, the equivalent macroscopic tensile and shear moduli in different directions are proportional to the base material tensile and shear moduli, respectively. The equivalent macroscopic thermal expansion coefficient is identical to that of the base material. Internal pressure in flow channels causes slight volumetric expansion of the core, which can be characterized by applying cold-side and hot-side internal pressures to the core microscopic unit cell finite element model to obtain the relationship between macroscopic volumetric expansion strain and internal pressure. A volumetric expansion strain tensor is then added to the macroscopic orthotropic linear elastic constitutive model. (3) Establish an overall finite element model of the microchannel heat exchanger, including headers, nozzles, microchannel-containing cores, and non-channel core shells. The microchannel-containing core is described using an orthotropic linear elastic model with different elastic constants for different core regions. The constraint effects of supports and connected piping systems on the microchannel heat exchanger can be equivalently represented using spring elements to reduce model size.

(4) Perform coupled steady-state heat transfer analysis for three temperature fields: cold fluid, hot fluid, and solid domains to obtain temperature distributions in the solid region under different operating conditions. (5) Calculate primary stresses in the overall microchannel heat exchanger finite element model caused by internal pressure and external loads under various operating conditions. Thermally induced stresses are classified as secondary stresses and fall outside the scope of stress evaluation, though temperature-induced changes in material elastic modulus and stress limits must be considered in the model. Thermally induced secondary stresses will be evaluated in strain assessment and creep-fatigue damage evaluation. (6) Establish stress classification lines at critical locations in structures without microchannels, such as headers, nozzles, and core shells. Linearize the stresses along these classification lines and categorize the linearized membrane stresses as either primary general membrane stress or primary local membrane stress, and linearized bending stresses as either primary bending stress or secondary stress. Perform verification according to different stress types and combinations based on various operating condition requirements. (7) Apply flow channel temperatures and maximum macroscopic temperature gradients in the core to unit cell finite element models with periodic boundary conditions to calculate microscopic temperature fields. Further apply flow channel internal pressure and maximum macroscopic stress in the core to perform microscopic stress field calculations. (8) Establish stress classification lines at critical locations in core microscopic unit cell structures, linearize the stresses, and evaluate microscopic stresses using methods similar to those for macroscopic models. However, note that under flow channel internal pressure, the stress state at interlayer ridges (diffusion bonding interfaces) in the core is triaxial tension. In ASME III-5, the stress intensity calculated based on maximum shear stress theory is compared with the membrane normal stress perpendicular to the diffusion bonding plane. Since diffusion bonding interface strength is relatively weak, both should be considered as stress verification objects for conservative design. In core unit cells, principal stresses at interlayer ridges are symmetrically distributed, resulting in approximately zero bending stress, which can be neglected in stress verification.

Existing literature provides semi-analytical formulas for comparison and reference [6-9]. For example, Idaho National Laboratory simplified PCHE flow channel structures as thick-walled cylinders and proposed the following formula for calculating the membrane tensile stress perpendicular to diffusion bonding planes at interlayer ridges under internal/external pressure difference [6]:

$$\sigma_{\perp} = \frac{\Delta p \cdot r_i^2}{t_w \cdot (2r_i + t_w)} \tag{1}$$

where Δp is the pressure difference between cold and hot side flow channels; r_i is the channel inner radius; t_w is the channel wall thickness.

Heatric Ltd. simplified PCHE heat exchange cores as rectangular cross-section containers with supports, as shown in [Figure 3: see original paper], and derived the following formula for membrane stress at interlayer ridges perpendicular to diffusion bonding planes [7-9]:

$$\sigma_{\perp} = \frac{\Delta p \cdot w \cdot h}{2 \cdot t_w \cdot (w + h)} \tag{2}$$

where w and h are the width and height of the rectangular flow channel, respectively.

2.2 Evaluation Results Comparison

The microscopic finite element method, Equation (1), and Equation (2) were respectively employed to calculate stresses at stress classification lines 5-10 at interlayer ridges under internal pressure conditions, as shown in [Figure 4: see original paper]. The results are presented in . In Equation (1), when simplifying the PCHE cold flow channel structure as a thick-walled cylinder, the internal pressure is less than external pressure, resulting in $\sigma_{\perp} < 0$ for cold channel walls 6 and 8. However, both the finite element method and Equation (2) yield $\sigma_{\perp} > 0$ for these classification lines. From the perspective of pressure balance with membrane tensile stress at ridge locations, it can be readily proven that $\sigma_{\perp} > 0$, making the finite element method and Equation (2) more reasonable.

Comparing the membrane principal stress results from the finite element method and Equation (2), except for stress classification line 8 at cold channel walls in region B with perpendicular hot/cold flow, the calculation error for other channel wall stress classification lines does not exceed 15%, with Equation (2) producing larger stress values. However, at classification line 8, the finite element method yields higher stress values. Therefore, when designing PCHE flow channel dimensions, the stress calculation formula from Heatric Ltd. can be used for initial design, followed by detailed strength verification using the microscopic finite element method.

3.1 Evaluation Method

According to ASME Code Cases N-898, ideal elastic-plastic finite element methods can be employed for strain evaluation of high-temperature nickel-based Alloy 617 Class A components at ultra-high temperatures. The specific procedure is as follows: (1) Define t_{design} as the total service time exceeding 427°C under A, B, and C level operating conditions. (2) Select target inelastic strains (ε_{avg}, $\varepsilon_{\text{local}}$), where the average strain limit ε_{avg} = ... and the local strain limit $\varepsilon_{\text{local}}$ = ... (3) Define temperature-dependent virtual yield strength S_y at each point in the structure, obtained from transient heat transfer analysis and varying spatially and temporally. (4) Employ an ideal elastic-plastic model to describe the high-temperature nickel-based alloy used in the microchannel heat exchanger, with yield stress equal to the temperature-dependent virtual yield strength S_y. Perform ideal elastic-plastic finite element simulation of the macroscopic structure under combined cyclic loading. If the finite element model does not exhibit ratcheting under combined cyclic loading, calculate the equivalent plastic strain ε_{eq} at each structural point according to Equation (5) based on the results at the end of the combined cycle.

(5) Perform strain evaluation for conventional structures without microchannels based on ε_{eq} determined in Step 4, checking whether both following conditions are satisfied: a. Plastic strain must exist at one point through the structural thickness direction: $\varepsilon_{\text{eq}} \geq \varepsilon_{\text{avg}} - ...$ b. Plastic strain at every point in the structure: $\varepsilon_{\text{eq}} \leq \varepsilon_{\text{avg}} - ...$ (6) For microchannel-containing cores, considering the macro-micro plastic strain relationship, the following two conditions must be satisfied for macroscopic strain finite element solutions: a. Plastic strain must exist at one point through the structural thickness direction: $\varepsilon_{\text{eq}} \geq \varepsilon_{\text{local}} - ...$ b. Plastic strain at every point in the structure: $\varepsilon_{\text{eq}} \leq \varepsilon_{\text{local}} - ...$

3.2 Evaluation Results

and list the target plastic strains for each component. provides the average and local plastic strain limits for components without microchannels, while presents the average macroscopic and local plastic strain limits for microchannel-containing cores.

[Figure 5: see original paper] illustrates the evolution of equivalent plastic strain fields during the first combined cycle. During the transition from ambient temperature/pressure to Level A conditions (t=0-3h), plastic strain primarily occurs at four corners of the hot-side inlet header and two corners of the cold-side outlet header near the hot side. These locations exhibit thermal stress concentration due to high temperature and geometric discontinuities. During the transition from Level A to Level B conditions (t=3-6h), significant plastic strain develops in the hot-side header and its connection with the nozzle under combined high temperature and pressure. During the cooldown to ambient conditions (t=7-12h), plastic strain at header corners slightly decreases, indicating reverse yielding.

[Figure 6: see original paper] shows the local plastic strain distribution at cycle completion, with points A and B identified as critical locations with large plastic strains. Point A is located at the connection between the hot-side header and nozzle, while point B is at a corner of the hot-side header. [Figure 7: see original paper] presents the equivalent plastic strain time-history curves at points A and B under multiple combined cyclic loads. After multiple cycles, the equivalent plastic strain at point A approaches a constant value, while point B experiences continuous reverse yielding without plastic strain accumulation. This demonstrates that the heat exchanger does not exhibit ratcheting under combined cyclic loading, satisfying Requirement 4 in Section 3.1, thus allowing strain evaluation using plastic strains at cycle completion.

First, verify whether components without microchannels satisfy the average plastic strain limit requirement in Equation (6). [Figure 8: see original paper] shows the equivalent plastic strain distribution in the weld region between the hot-side inlet header and nozzle, with gray areas indicating regions where plastic strain exceeds the average plastic strain limit. At the combined cycle endpoint, gray areas exist only in the weld region between the hot-side inlet header and nozzle, and these gray areas do not fully penetrate the thickness direction, thus satisfying Equation (6).

Second, verify whether microchannel-containing cores satisfy the average plastic strain limit requirement in Equation (8). [Figure 9: see original paper] shows the macroscopic equivalent plastic strain distribution in Region D core at cycle completion, with gray areas indicating regions exceeding the average plastic strain limit. At the combined cycle endpoint, gray areas exist only in the hot-side inlet Region D core, and these gray areas do not fully penetrate the thickness direction, thus satisfying the average plastic strain limit requirement in Equation (8).

Finally, verify whether each component satisfies the local plastic strain limit requirements in Equations (7) and (9). presents the maximum equivalent plastic strain values and limits for each component at the combined cycle endpoint for strain evaluation, showing that all components meet strain evaluation requirements.

4.1 Evaluation Method

The safe service life of microchannel heat exchangers should be no less than their design life, meaning that creep damage and fatigue damage must not exceed specified limits during service. According to ASME III-5 HBB-T-1400 and ASME Code Cases N-898, creep damage D_c and fatigue damage D_f at any point in the equipment under A, B, and C level loading must satisfy:

$$D_c + D_f \le D_{\text{total}} \tag{10}$$

where D_{total} is the total creep-fatigue damage, which should be limited within the envelope boundary shown in ASME Code Cases N-898 Figure HBB-T-1420-2, as illustrated in [Figure 10: see original paper].

The maximum creep damage value under combined cyclic loading is:

$$D_c = \frac{t_{\text{design}}}{t_{\text{rupture}}} \tag{11}$$

where t_{design} is the total service time exceeding 427°C under A, B, and C level conditions, taken here as $t_{\text{design}} = 3$ years.

Based on linear elastic stress analysis, equivalent stress amplitude is typically used for fatigue damage evaluation, calculated as:

$$\Delta\varepsilon_{\text{eq}} = \frac{\sqrt{2}}{3}\sqrt{(\Delta\varepsilon_1 - \Delta\varepsilon_2)^2 + (\Delta\varepsilon_2 - \Delta\varepsilon_3)^2 + (\Delta\varepsilon_3 - \Delta\varepsilon_1)^2} \tag{12}$$

where $\Delta\varepsilon_i$ are strain component amplitudes and $\nu = 0.3$ is Poisson's ratio. For microchannel-containing cores, the equivalent strain amplitude should use microscopic equivalent strain amplitude, obtained by dividing the macroscopic equivalent strain amplitude from Equation (12) by the ratio given in to obtain the maximum microscopic equivalent strain amplitude. Finally, calculate fatigue damage values at critical points under cyclic loading:

$$D_f = \sum \frac{n_i}{N_{\text{allowable},i}} \tag{13}$$

where n_i is the number of cycles for each load type, and $N_{\text{allowable}}$ is the maximum allowable cycles determined from ASME Code Cases N-898 Figure HBB-T-1420-1F, with temperature taken as the maximum temperature at critical points during the cycle. For microchannel-containing cores, allowable cycles should be determined using microscopic equivalent strain amplitude.

4.2 Evaluation Results

High-temperature reactor microchannel heat exchangers experience few start-stop cycles during service, making fatigue damage effects negligible. Therefore, only A-level and B-level condition switching is considered in the evaluation. This study examines a microchannel heat exchanger with a 3-year design life. Taking $t' = 4.6$ years = 40296h, elastic safety analysis and fatigue calculations were performed during A-level and B-level condition cycling. [Figure 11: see original paper] shows the time-history curves of hot-side inlet fluid temperature and pressure during A-level and B-level condition cycling, while cold-side inlet fluid temperature and pressure remain constant. Under each condition, inlet temperatures and pressures are maintained for sufficient time to allow the transient temperature field to stabilize.

After analyzing equivalent plastic strain following 15 A-level and B-level condition cycles, three critical creep-fatigue damage failure points were identified: Point A in the weld region connecting the hot-side inlet header and nozzle; Point B at the corner of the cold-side outlet header; and Point C at the strain concentration location in the hot-side inlet Region D core.

Analysis shows that the structure ultimately reaches an elastic shakedown state during A-level and B-level condition cycling. The maximum creep damage value in the microchannel heat exchanger during the 3-year service period does not exceed $D_c^{\text{design}} = 0.65$. Equivalent strain amplitudes at each critical point were calculated using Equation (12), and allowable cycles $N_{\text{allowable}}$ were determined from ASME Code Cases N-898 Figure HBB-T-1420-1F. Since logarithmic fatigue life approximately satisfies a linear relationship, allowable cycles beyond those provided in Figure HBB-T-1420-1F can be obtained through linear extrapolation of $\log(N_{\text{allowable}})$ versus $\log(\Delta\varepsilon_{\text{eq}})$, with results presented in .

shows damage values at each critical point in the creep-fatigue damage evaluation. [Figure 12: see original paper] presents the creep-fatigue damage evaluation diagram for this microchannel heat exchanger, showing that all three critical points lie within the creep-fatigue damage envelope boundary. Therefore, considering only fatigue events from A-level and B-level condition switching, the microchannel heat exchanger meets evaluation requirements during its 3-year service period, with a safe service life exceeding 3 years. Furthermore, based on Point C, the maximum safe service life of the microchannel heat exchanger is estimated as:

$$t_{\text{safe}} = t_{\text{design}} + \frac{1 - D_c}{D_f} \approx 3.58 \text{ years}$$

Conclusions

Based on ASME III-5 and ASME Code Cases N-898, a safety performance analysis and evaluation method considering core microscopic structure has been proposed for microchannel heat exchangers, investigating safety evaluation methods from stress, strain, creep-fatigue damage, and life prediction perspectives. The conclusions are summarized as follows:

(1) Stress evaluation employs linear elastic mechanics analysis. The relationship between macroscopic and microscopic linear elastic behavior of the core is established through microscopic finite element models and homogenization methods, significantly reducing the complexity and mesh count of the overall heat exchanger finite element model. Stress evaluation results indicate that the weld region connecting the hot-side inlet header and nozzle represents the critical location for structural failure induced by

internal pressure.

(2) For microchannel heat exchanger core structures, initial flow channel dimensions can be determined using Heatric Ltd.'s stress calculation formula, followed by detailed strength verification using the finite element method.

(3) In strain evaluation, the microchannel heat exchanger satisfies ASME III-5 and Code Cases N-898 requirements under the combined cyclic loading of ambient→Level A→Level B→Level A→ambient conditions.

(4) Through establishment of an ideal elastic-plastic finite element model for a 3-year design life microchannel heat exchanger and shakedown analysis under cyclic loading, it is confirmed that under high-temperature creep and fatigue loading from 100 annual A-level-B-level condition switches, the safe service life exceeds 3 years, with creep damage not exceeding 0.65. The creep-fatigue damage evaluation diagram estimates the maximum safe service life as 3.58 years.

References

[1] Yan X L, Very High Temperature Reactor, in Handbook of Generation IV Nuclear Reactors (Second Edition), Woodhead Publishing Series in Energy, 2023.
[2] Chapin D, Kiffer S, Nestell J, The Very High Temperature Reactor: A Technical Summary revision 0, 2013.
[3] Mylavarapua S K, Sun X, Christensena R N, Unocicb R R, Glosupa R E, Pattersonc M W, Fabrication and design aspects of high-temperature compact diffusion bonded heat exchangers, Nuclear Engineering and Design, 2012, 249: 49-56.
[4] ASME Boiler and Pressure Vessel Code, Section III, Division 5, 2015.
[5] ASME Boiler and Pressure Vessel Code, Nuclear Components, Code Cases N-898, 2021.
[6] Yoon S J, Sabharwall P, Kim E S, Analytical Study on Thermal and Mechanical Design of Printed Circuit Heat Exchanger, INL/EXT-13-30047.
[7] Knoell A C, Structural design and stress analysis program for advanced composite filament JPL Technical Report 32-1521, wound axisymmetric pressure vessel California:NASA, 1971:1-32.
[8] Dewson S J, Li X, Selection Criteria for the High Temperature Reactor Intermediate Heat Exchanger, Proceedings of ICAPP '05, Paper 5333, May 15-19, 2005.
[9] Nestell J, Sham T L, ASME Code Considerations for the Compact Heat Exchanger, ORNL/TM-2015/401.

Note: Figure translations are in progress. See original paper for figures.