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

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Abstract

Background: High-temperature reactors (HTRs) represent a significant advancement in fourth-generation nuclear technology, with core outlet temperatures exceeding 900°C. The intermediate heat exchanger (IHX) in HTRs plays a crucial role in transferring heat from the primary circuit to the secondary circuit. However, IHX technology in HTRs remains under development, posing a major challenge to the advancement of HTRs. Microchannel heat exchangers (MCHEs) are emerging as a promising solution for IHX applications due to their compact design and high thermal efficiency. MCHEs operating at temperatures above 750°C are classified as Class A components according to the ASME III-5 standard, requiring rigorous structural safety and integrity assessment. Purpose: The objective of this study is to evaluate the safety performance of high-temperature MCHEs under cyclic conditions. The research focuses on developing a systematic methodology for evaluating stress, strain, fatigue damage, and life prediction, in compliance with the ASME III-5 standard and Code Case N-898. Methods: This study employs a comprehensive finite element modeling approach to analyze the macroscopic and microscopic behavior of a custom-designed MCHE. The model incorporates the effects of argon arc welding and diffusion welding, defining weld zones within the MCHE. The analysis involves establishing an ideal elastic-plastic finite element model to calculate macroscopic and microscopic plastic strain in relation to yield stress. Stress classification lines are established in critical regions such as the shell and nozzles, and stresses are linearized and classified. The study also conducts strain assessment and fatigue damage evaluation according to ASME Code Case N-898, defining allowable strain limits and calculating equivalent plastic strain. Results: The results demonstrate that the investigated MCHE complies with the requirements of ASME III-5 and Code Case N-898. Maximum fatigue damage values are obtained, enabling prediction of the maximum safe service life of the MCHE. Quantitative results include stress and strain calculations in critical regions such as weld zones and the core body. For instance, the study finds that the maximum equivalent plastic strain in the core body of Zone D is 0.00074, below the allowable limit of 0.00110. The fatigue damage values at key points such as Point A on the hot-side inlet manifold and Point B on the cold-side outlet manifold are calculated as 9.3×10^{-8} and 5.1×10^{-7}, respectively, indicating that the MCHE can operate safely for over three years under specified conditions. Conclusion: The study concludes that the safety performance analysis method based on ASME III-5 and Code Case N-898 provides a reliable framework for evaluating MCHE safety. This method effectively accounts for the microchannel structure in the core body, enabling detailed evaluation of stress, strain, and fatigue damage. The research results confirm the safety and reliability of MCHE design for high-temperature reactor applications, with a predicted service life exceeding three years. This study contributes to the development of safer and more efficient heat exchanger technologies for nuclear applications, providing valuable insights for future design and operational strategies.

Full Text

Preamble

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

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Abstract

Background: High-Temperature Reactors (HTRs) represent a significant advancement in fourth-generation nuclear technology, with core outlet temperatures exceeding 900°C. The Intermediate Heat Exchanger (IHX) in HTRs plays a crucial role in transferring heat from the primary circuit to the secondary circuit. However, IHX technology for HTRs remains in the developmental stage, posing a major challenge to HTR advancement. Microchannel Heat Exchangers (MCHEs) are emerging as a promising solution for IHX applications due to their compact design and high thermal efficiency. Operating at temperatures above 750°C, MCHEs are classified as Class A components under the ASME III-5 standard, necessitating rigorous structural safety and integrity assessments. **Objective:** This study aims to evaluate the safety performance of high-temperature MCHEs under cyclic conditions, focusing on developing a systematic methodology for stress assessment, strain assessment, and fatigue damage and life prediction in accordance with ASME III-5 and Code Case N-898. **Methods:** The study employs a comprehensive finite element modeling approach to analyze both macroscopic and microscopic behavior of a custom-designed MCHE. The models incorporate the effects of argon arc welding and diffusion welding, defining weld regions within the MCHE. The analysis involves establishing ideal elastoplastic finite element models to calculate the relationship between macroscopic and microscopic plastic strain and yield stress. Stress classification lines are established in critical regions such as the shell and nozzles, with stresses linearized and categorized accordingly. The study also conducts strain assessment and fatigue damage evaluation based on ASME Code Case N-898, defining allowable strain limits and calculating equivalent plastic strain. **Results:** The results demonstrate that the studied MCHE complies with ASME III-5 and Code Case N-898 requirements. Maximum fatigue damage values were obtained, enabling prediction of the MCHE' s maximum safe service life. Quantitative results include stress and strain calculations for critical regions such as weld areas and the core body. For instance, the maximum equivalent plastic strain in the D-region core body was found to be 0.00074, below the allowable limit of 0.00110. Fatigue damage values at key points such as point A in the hot-side inlet manifold and point B in the cold-side outlet manifold were calculated as 9.3×10^{-8} and 5.1×10^{-7}, respectively, indicating that the MCHE can operate safely for over three years under specified conditions. **Conclusion:** The study concludes that the safety performance analysis methodology based on ASME III-5 and Code Case N-898 provides a robust framework for evaluating MCHE safety. This approach effectively accounts for microchannel structures within the core body, enabling detailed assessment of stress, strain, and fatigue damage. The findings confirm that the MCHE design is safe and reliable for high-temperature reactor applications, with a predicted service life exceeding three years. This research contributes to the development of safer and more efficient heat exchanger technologies for nuclear applications, providing valuable insights for future design and operational strategies.

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

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Introduction

High-temperature nuclear reactors (Very High Temperature Reactors, VHTRs) represent an important reactor type among Generation IV nuclear technologies, 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 circuit to the secondary circuit [3]. However, IHX technology for VHTRs remains immature, representing a significant shortcoming in VHTR development. Microchannel heat exchangers offer a highly promising IHX solution [4]. Microchannel heat exchangers for high-temperature reactors operate at temperatures above 750°C and are classified as Class A components under ASME III-5 [5]. Therefore, verifying the structural safety and boundary integrity of such equipment requires simultaneous consideration of multiple failure mechanisms, including short-term ductile rupture, long-term creep rupture, creep-fatigue interaction damage, gross structural distortion caused by ratcheting and incremental collapse, loss of function due to excessive deformation, and buckling under short-term and long-term compressive loads [6-8].

Internal pressure and temperature represent the primary loading sources for microchannel heat exchangers. Structural stresses induced by internal pressure can be categorized as general and local primary membrane stresses, primary bending stresses, secondary stresses, and peak stresses [9]. Thermal stresses induced by temperature generally belong to secondary and peak stresses [10]. Primary stresses are not self-limiting; if excessive, they can cause short-term ductile rupture, long-term creep rupture, or geometric distortion of the overall structure. Conversely, excessive secondary and peak stresses or stress variations can lead to excessive local strain, fatigue failure, or ratcheting [11].

ASME III-5 addresses stress assessment for Class A components in ultra-high temperature environments only for primary stresses, while secondary and peak stresses are evaluated through strain assessment and creep-fatigue damage assessment. This study investigates a systematic safety performance analysis methodology 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 [12]. The approach involves: (1) establishing macroscopic and microscopic finite element models of the microchannel heat exchanger that incorporate argon arc weld and diffusion weld regions; (2) conducting corresponding strain assessment, creep-fatigue damage assessment, and fracture strength assessment studies for primary, secondary, and peak stresses caused by internal pressure and temperature; and (3) summarizing the stress assessment, strain assessment, creep-fatigue damage assessment, and life prediction results for the microchannel heat exchanger.

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, virtual yield stresses for welds and base materials are calculated using different methods in ideal elastoplastic simulation for strain assessment and creep-fatigue damage assessment. Therefore, it is necessary to define weld regions in the finite element model. Based on weld dimensions and structure, considering factors such as heat-affected zones, critical cross-sections, and modeling complexity, a finite element model of the nozzle-pipe box connection containing weld regions was established at the macroscopic level, as shown in [Figure 1: see original paper]. The finite element model uses three element layers through the thickness direction with quadratic reduced integration elements. The outer width of welds between the pipe box and core body, and between pipe box and nozzle, is 6 mm, while the inner width is 4 mm. The outer width of welds between the shell and end plate is 9 mm, and the inner width is 4 mm.

1.2 Microscopic Model

The core body of a microchannel heat exchanger is typically manufactured by alternately stacking and diffusion bonding multiple cold and hot plates with thicknesses of 1-2 mm. Semicircular flow channels with radii less than 1 mm are formed on the plate surfaces through etching. Considering the thin plate thickness, unknown diffusion bonding heat-affected zones, and the fact that maximum strains in the core often concentrate near channel corners, it is assumed in strain assessment and creep-fatigue damage assessment based on ideal elastoplastic analysis that all solid regions in the microscopic core are affected by diffusion welding. Their maximum allowable strain, fatigue life, and creep strength properties are obtained by multiplying base material properties by corresponding reduction factors. Consequently, the entire microscopic model solid region employs uniform material properties without explicitly defining weld regions, as shown in [Figure 2: see original paper]. The thin-walled structures in unit cells use no fewer than three element layers through the thickness direction with quadratic reduced integration elements.

1.3 Macro-Micro Elastoplastic Behavior Relationship

To apply ideal elastoplastic analysis methods to core components containing microchannels, microscopic ideal elastoplastic behavior analysis must be performed to establish the relationship between macroscopic and microscopic elastoplastic behavior of the core. This is achieved by establishing ideal elastoplastic microscopic unit cell models for A, B, C, and D regions of the core, 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. 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 Stress Assessment Methodology

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

The microchannel heat exchanger stress assessment methodology consists of the following steps:

(1) Based on the flow channel structure characteristics of the microchannel heat exchanger core, divide the flow channel-containing core into different regions. Different regions have different unit cell structures, generally categorized as parallel flow channels, perpendicular flow channels, and single-sided flow 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 flow channel-containing core behaves as an orthotropic material with a total of nine independent equivalent macroscopic elastic moduli and Poisson' s ratios. Within the linear elastic range, equivalent macroscopic tensile and shear moduli in different directions are proportional to base material tensile and shear moduli, respectively. The equivalent macroscopic coefficient of thermal expansion is identical to that of the base material. Internal pressure in flow channels causes slight volumetric expansion of the core, which can be obtained by applying cold-side and hot-side internal pressures in the core microscopic unit cell finite element model to derive 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 pipe boxes, nozzles, microchannel-containing cores, and core outer shells without flow channels. The microchannel-containing core is described using an orthotropic linear elastic model with different elastic constants for different core regions. Constraints from supports and connected piping systems on the microchannel heat exchanger can be equivalently represented using spring elements to reduce finite element model size.

(4) Perform coupled steady-state heat transfer analysis for three temperature fields (cold fluid, hot fluid, and solid) to obtain solid domain temperature distributions under different operating conditions.

(5) Calculate primary stresses caused by internal pressure and external loads for the overall microchannel heat exchanger finite element model under different operating conditions. Thermally induced stresses are classified as secondary stresses and are not within the scope of stress assessment, but 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 fatigue-creep damage assessment.

(6) Establish stress classification lines at critical locations without microchan-

nels, such as pipe boxes, nozzles, and core outer shells. Linearize stresses along these classification lines and categorize the linearized membrane stresses as either general primary membrane stress or local primary membrane stress, and linearized bending stresses as either primary bending stress or secondary stress, based on their characteristics. Perform verification for different types of stresses and combinations according to stress assessment requirements for different operating conditions.

(7) Apply flow channel temperatures and maximum macroscopic temperature gradients in the core to the core unit cell finite element model with periodic boundary conditions to perform microscopic temperature field calculations. Subsequently 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 the core microscopic unit cell structure, linearize the stresses, and perform assessment verification for microscopic stresses using methods similar to those for the macroscopic model. However, it should be noted 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 the strength of diffusion bonding interfaces 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 to thick-walled cylindrical structures and used the following simplified formula to calculate the membrane tensile stress perpendicular to the diffusion bonding plane at interlayer ridges under inner and outer pressure differences [13]:

$$\sigma_{film} = \frac{P_{diff} \cdot d}{2t_{rib}} \tag{1}$$

where P_{diff} is the pressure difference between cold and hot side flow channels, d is the flow channel diameter, and t_{rib} is the inter-rib thickness, i.e., the number of inter-ribs per unit length multiplied by rib thickness.

Heatric Corporation simplified PCHE heat exchanger cores to rectangular cross-section vessels with supports, deriving the following formula for membrane stress perpendicular to the diffusion bonding plane at interlayer ridges [14-16]:

$$\sigma_{film} = \frac{P_{diff} \cdot w}{2t_{rib}} \tag{2}$$

where w is the flow channel width.

2.2 Stress Assessment Results Comparison

The microscopic finite element method, Equation (1), and Equation (2) were respectively used to calculate stresses at stress classification lines 5-10 at interlayer ridges under internal pressure conditions, as shown in [Figure 3: see original paper], with results compared in . In Equation (1), after simplifying PHCE cold flow channel structures to thick-walled cylinders, the internal pressure is less than external pressure, i.e., $P_{diff} < 0$, resulting in calculated stresses at cold channel inter-rib locations 6 and 8 being $\sigma_{film} < 0$. However, both the finite element method and Equation (2) calculate $\sigma_{film} > 0$ at locations 6 and 8. From the perspective of force balance between flow channel pressure and membrane tensile stress at ridges, it can be easily proven that $\sigma_{film} > 0$ at locations 6 and 8, making the finite element method and Equation (2) more reasonable.

Comparing the finite element method with Equation (2) for membrane principal stress calculations, except for the cold channel inter-rib 8 stress classification line in the B-region core with perpendicular cold/hot flows, the calculation errors at other flow channel inter-rib stress classification lines do not exceed 15%, with Equation (2) yielding larger stress values. However, at the 8 stress classification line, the finite element method calculates larger stress values. Therefore, when designing PCHE flow channel dimensions, Heatric Corporation' s stress calculation formula can be used initially to determine preliminary designs, followed by detailed strength verification of flow channel inter-ribs using the microscopic finite element method.

3.1 Strain Assessment Methodology

According to ASME Code Case N-898, for ultra-high temperature Class A components made of high-temperature nickel-based Alloy 617, strain assessment can be performed using the ideal elastoplastic finite element method through the following steps:

(1) Define t_{design} as the total service time at temperatures exceeding 427°C under A, B, and C-level operating conditions.

(2) Select target inelastic strain ε_{target} (where average strain limit $\varepsilon_{avg} = \varepsilon_{target}$ and local strain limit $\varepsilon_{local} = 1.5\varepsilon_{target}$). In Equations (3) and (4), a, b, and c represent base material, argon arc weld, and thin-plate diffusion weld, respectively.

(3) Define temperature-dependent virtual yield strength $S_y(T)$ at each point in the structure, obtained from transient heat transfer analysis, with values varying spatially and temporally.

(4) Employ an ideal elastoplastic model to describe the high-temperature nickel-based alloy used in the microchannel heat exchanger, with yield stress being the temperature-dependent virtual yield strength $S_y(T)$. Perform ideal elastoplastic 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 equivalent plastic strain ε_{eq} at structural points using the finite element results at the end of the combined cycle according to Equation (5):

$$\varepsilon_{eq} = \sqrt{\frac{2}{3}\varepsilon_{ij}^{p}\varepsilon_{ij}^{p}} \tag{5}$$

(5) For strain assessment of ordinary structures without microchannels based on ε_{eq} determined in Step 4, verify that both following conditions are satisfied:

a. Plastic strain must exist at one point through the structural thickness direction: $\varepsilon_{eq} \leq \varepsilon_{avg}$

b. Plastic strain at every point in the structure: $\varepsilon_{eq} \leq \varepsilon_{local}$

(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: $\varepsilon_{eq}^{macro} \leq \varepsilon_{avg}$

b. Plastic strain at every point in the structure: $\varepsilon_{eq}^{macro} \leq \varepsilon_{local}$

3.2 Strain Assessment Results

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

[Figure 4: see original paper] illustrates the evolution of equivalent plastic strain fields during the first combined cycle. From ambient temperature/pressure to Condition A (t=0-3h), plastic strain primarily occurs at four corners of the hot-side inlet pipe box and two corners of the cold-side outlet pipe box near the hot side. These locations exhibit thermal stress concentration due to high temperatures and geometric discontinuities. During transition from Condition A to Condition B (t=3-6h), as hot-side fluid temperature and pressure continuously increase, significant plastic strain develops in the hot-side pipe box and its connection with the nozzle under combined high temperature and pressure. During cooldown to ambient conditions (t=7-12h), plastic strain at pipe box corners slightly decreases, indicating reverse yielding.

[Figure 5: see original paper] shows the local plastic strain distribution at the end of the cycle, with points A and B being key locations of high plastic strain. Point A is at the connection between the hot-side pipe box and nozzle, while point B is at a corner of the hot-side pipe box. [Figure 6: 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 but with no further increase in plastic strain. This demonstrates that the heat exchanger does not ultimately exhibit ratcheting under combined cyclic loading, satisfying the requirement in Section 3.1, Step 4, allowing plastic strain calculations at cycle endpoints for strain assessment.

First, verify that components without microchannels satisfy the average plastic strain limit requirement. [Figure 7: see original paper] shows the equivalent plastic strain distribution in the weld region between the hot-side inlet pipe box and nozzle, where gray areas indicate regions exceeding the average plastic strain limit. At the combined cycle endpoint, only the weld region between the hot-side inlet pipe box and nozzle contains gray areas, which do not fully penetrate the thickness direction, thus satisfying the requirement.

Second, verify that microchannel-containing cores satisfy the average plastic strain limit requirement. [Figure 8: see original paper] shows the macroscopic equivalent plastic strain distribution in D-region core at the cycle endpoint, where gray areas exceed the average plastic strain limit. At the combined cycle endpoint, only the hot-side inlet D-region core contains gray areas, which do not fully penetrate the thickness direction, thus satisfying the average plastic strain limit requirement.

Finally, verify that all components satisfy the local plastic strain limit requirements in Equations (7) and (9). provides maximum equivalent plastic strain values and limits for various components at the strain assessment combined cycle endpoint, confirming that all components meet strain assessment requirements.

4.1 Creep-Fatigue Damage Assessment Methodology

The safe service life of microchannel heat exchangers should not be less than their design life, meaning creep damage and fatigue damage during service must not exceed specified limits. According to ASME III-5 HBB-T-1400 and ASME Code Case N-898, creep damage and fatigue damage at any point in the equipment under A, B, and C-level operating loads must satisfy:

$$D_c + D_f \leq D_{limit} \tag{10}$$

where D_c is creep damage, D_f is fatigue damage, and the total creep-fatigue damage must be limited within the envelope defined in ASME Code Case N-898, Figure HBB-T-1420-2.

The maximum creep damage value under combined cyclic loading is:

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

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

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

$$\varepsilon_{equiv} = \frac{\Delta\varepsilon_{eq}}{2(1+\nu)} \tag{12}$$

where $\Delta\varepsilon_{eq}$ is the equivalent strain amplitude and $\nu = 0.3$ is Poisson's ratio. For microchannel-containing cores, the equivalent strain amplitude should use the microscopic equivalent strain amplitude, obtained by dividing the macroscopic equivalent strain amplitude from Equation (12) by the ratio from to obtain the maximum microscopic equivalent strain amplitude. The fatigue damage value at key points under cyclic loading is then calculated as:

$$D_f = \sum_{i=1}^{n} \frac{n_i}{N_i} \tag{13}$$

where n_i is the number of cycles for load type i, and N_i is the maximum allowable number of cycles for load type i determined from ASME Code Case N-898, Figure HBB-T-1420-1F, with temperature taken as the maximum temperature at key points during the cycle. For microchannel-containing cores, the maximum allowable cycles should be determined using microscopic equivalent strain amplitude.

4.2 Creep-Fatigue Damage Assessment Results

Microchannel heat exchangers for high-temperature reactors experience few start-stop cycles during service, making their contribution to structural fatigue damage negligible. Therefore, only the effects of A-level and B-level condition switching are considered in the assessment. This study examines a microchannel heat exchanger with a three-year design life. Taking time $t' = 4.6$ years = 40,296 hours, elastic safety analysis and fatigue calculations were performed during cyclic switching between A-level and B-level conditions. [Figure 9: see original paper] shows the temperature and pressure variation curves of hot-side inlet fluid during A-level and B-level condition cycling. Cold-side inlet fluid temperature and pressure remain constant. Under each condition, cold-side and hot-side inlet temperatures and pressures are maintained for sufficient duration to allow transient temperature fields to stabilize.

After analyzing equivalent plastic strain in the microchannel heat exchanger following 15 cycles of A-level and B-level condition switching, three critical creep-fatigue damage failure points were identified: point A in the weld region connecting the hot-side inlet pipe box and nozzle; point B at the corner of the cold-side outlet pipe box; and point C at the strain concentration in the hot-side inlet D-region 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 of the microchannel heat exchanger during the three-year service period does not exceed $D_c = 0.65$. Equivalent strain amplitudes at key points were calculated using Equation (12), and allowable cycles N_i were determined from ASME Code Case N-898, Figure HBB-T-1420-1F. Since logarithmic fatigue life approximately satisfies a linear relationship, allowable cycles beyond those provided in the fatigue curve of Figure HBB-T-1420-1F can be obtained through linear extrapolation of $\log(N)$ versus $\log(\varepsilon_{equiv})$, with results shown in .

provides damage values at key points in the creep-fatigue damage assessment. [Figure 10: see original paper] shows the creep-fatigue damage assessment diagram for the microchannel heat exchanger, with all three key points located within the creep-fatigue damage envelope. Therefore, when considering only fatigue events from A-level and B-level condition switching, the microchannel heat exchanger' s creep-fatigue damage values satisfy assessment requirements during the three-year service period, indicating a safe service life exceeding three years. Further estimation based on point C yields a maximum safe service life of:

$$t_{safe} = t_{design} + \frac{0.776 - 0.65}{2} \approx 3.58 \text{ years}$$

Conclusion

This study proposes a safety performance analysis and evaluation methodology for microchannel heat exchangers that considers core microscopic structures, based on ASME III-5 and ASME Code Case N-898. The methodology investigates safety evaluation approaches from multiple aspects including stress, strain, creep-fatigue damage, and life prediction. The main conclusions are:

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

(2) For microchannel structures in heat exchanger cores, Heatric Corporation' s stress calculation formulas can be used to determine initial designs dur-

ing PCHE flow channel dimension design, followed by detailed strength verification of flow channel inter-ribs using the microscopic finite element method.

(3) In strain assessment, verification confirms that the microchannel heat exchanger satisfies ASME III-5 and Code Case N-898 strain assessment requirements under the combined cycle of ambient temperature/pressure → Condition A → Condition B → Condition A → ambient temperature/pressure.

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

Author Contributions Statement

YANG Yang was responsible for conceptualization, methodology, detailed analysis, and manuscript drafting. MA Ruoxin participated in the computational analysis process, reviewed the analysis procedures, and verified the accuracy of results. RAN Bin participated in relevant computational analysis processes.

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