Study on Failure Characteristics of 20kW Printed Circuit Heat Exchangers for High-Temperature Reactors

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Abstract

As a candidate heat exchanger for high-temperature reactors, the Printed Circuit Heat Exchanger (PCHE) operates in extremely harsh environments with low technology maturity. In this work, a 20kW PCHE was designed and the factors influencing its potential failure were analyzed. This paper proposes a zonal homogenization method for the heat exchange core, simulates the high-temperature and high-pressure operating conditions of the heat exchanger, and conducts numerical simulations of the macroscopic and mesoscopic temperature fields within the core. Finally, through theoretical and preliminary numerical simulation analyses, the potential failure locations, failure modes, and consequences of the PCHE heat exchanger are estimated. This work provides direction for further research on safety assessment of PCHE heat exchangers.

Full Text

Preamble

Failure Analysis of a 20 kW Printed Circuit Heat Exchanger for High Temperature Reactors

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Abstract

As a candidate heat exchanger for high-temperature reactors, the printed circuit heat exchanger (PCHE) operates in extremely demanding environments while exhibiting low technological maturity. In this work, a 20 kW PCHE was designed and analyzed for potential life-limiting factors. A partitioned homogenization method for the heat exchanger core is proposed, and the high-temperature, high-pressure operating environment of the heat exchanger was simulated. Macroscopic and microscopic temperature field numerical simulations of the heat exchanger core were conducted. Finally, through theoretical and preliminary numerical simulation analyses, predictions were made regarding potential failure locations, failure modes, and consequences of the PCHE. This work provides direction for further research on PCHE safety assessment.

Keywords: heat exchanger; homogenization; high temperature; high pressure; failure analysis

1 Introduction

The printed circuit heat exchanger (PCHE) features numerous microchannels within its core, providing high heat transfer area density, while the unetched material surrounding the channels offers natural structural support. Consequently, PCHEs exhibit higher heat transfer efficiency and pressure-bearing capacity compared to conventional shell-and-tube or plate-fin heat exchangers. However, as a candidate heat exchanger for fourth-generation ultra-high temperature gas-cooled reactors, PCHEs face even more severe operating conditions. If structural safety design is inadequate or operation and maintenance are improper, structural failure accidents may occur during early, middle, or late service life, seriously threatening nuclear power system operational safety. Sustained high-temperature and high-pressure conditions introduce new problems and challenges for structural safety design. To enable targeted safety assessment analysis and periodic maintenance inspection, it is necessary to preliminarily predict potential failure modes and hazardous locations during the design phase.

This study designed a 20 kW PCHE and summarized the structural configurations and characteristics of its overall assembly, heat exchanger core, headers, and nozzles. A partitioned homogenization method for the heat exchanger core was proposed. The high-temperature, high-pressure service environment was characterized, and numerical simulations of macroscopic and microscopic temperature fields in the heat exchanger core were performed to predict temperature and temperature gradient distributions in the cold fluid, hot fluid, and solid domains. The impact of sustained high temperatures on structural safety was discussed. Finally, potential failure locations, failure modes, and consequences of the designed 20 kW PCHE were estimated.

2 Structural Design and Homogenization

2.1 Overall PCHE Structure

The PCHE designed in this study consists of one heat exchanger core, four headers, and four nozzles, with overall dimensions of 571 × 373 × 102 mm. The heat exchanger core is fabricated by alternately stacking 25 hot plates and

25 cold plates, with nine cover plates placed on the top and bottom surfaces, followed by diffusion bonding. The cold and hot plates are pre-etched with microchannels using photochemical machining. The top and bottom cover plates primarily serve sealing and structural reinforcement functions. The headers and nozzles, serving as fluid inlets and outlets for the cold and hot sides, are welded to the side walls of the heat exchanger core to form an integrated assembly. [Figure 1: see original paper] shows a schematic diagram of the overall PCHE structure.

The core mass accounts for approximately 88.8% of the total PCHE mass. Although the numerous microchannels reduce core stiffness, the bulk structure exhibits greater overall stiffness than the thick-walled structures of the headers and nozzles.

2.2 Heat Exchanger Core Structure and Homogenization

The cold plates, hot plates, and cover plates each measure 382 mm × 184 mm × 1.5 mm. The cold and hot plates feature semi-circular cross-section channels etched on their surfaces. Key geometric parameters include plate thickness t, channel radius r, and channel spacing d. The designed dimensions are $t = 1.5$ mm, $r = 0.75$ mm, and $d = 2.15$ mm. The flow channel configurations differ between cold and hot plates.

The cold plate channels incorporate flow disturbance structures in the middle region of the straight channels to enhance heat transfer coefficient, as shown in [Figure 2: see original paper]. The hot plate channels form a "Z" -shaped configuration through baffling, as shown in [Figure 3: see original paper]. After diffusion bonding, connection structures form at the channel ribs and edges, providing necessary pressure-bearing support. The minimum plate wall thickness, minimum rib thickness, and channel edge thickness directly affect the pressure-bearing performance of the PCHE core. The channel tips represent local discontinuities that generate stress concentrations.

The prototype PCHE contains 74 microchannels on each hot and cold plate, resulting in 1,850 hot and cold flow channels within the heat exchanger core. This large number of channels creates difficulties for numerical modeling and mechanical analysis. Therefore, micromechanical theory is employed to homogenize the repetitive microchannel structures, treating the heat exchanger core as a homogeneous continuum. Based on the combination characteristics of cold and hot plate channels, the cored region can be divided into seven zones, as shown in [Figure 4: see original paper], neglecting the disturbance structure effects. In the central Zone A, cold and hot channels run parallel. In the left and right end Zones B1 and B2, cold and hot channels cross perpendicularly. In the cold-side inlet and outlet Zones C1 and C2, only cold channels exist. In the hot-side inlet and outlet Zones D1 and D2, only hot channels exist. The remaining core region consists of cover plates and channel edge areas without flow channels.

Based on the periodic arrangement characteristics of channels in each zone, different unit cell structures (representative volume elements) can be established. Figure 5: see original paper shows the unit cell for Zone A; Figure 5: see original paper for Zones B1 and B2; Figure 5: see original paper for Zones C1 and C2; and Figure 5: see original paper for Zones D1 and D2. Without loss of generality, the lower holes in the unit cell are designated as cold channels and the upper holes as hot channels. Using micromechanical homogenization methods, equivalent elastic modulus, equivalent Poisson' s ratio, and equivalent thermal expansion coefficient can be obtained for different zones.

2.3 Header and Nozzle Structures

The header and nozzle structures are shown in [Figure 6: see original paper]. Each header consists of a header end plate and header shell connected by argon arc welding. Both inner and outer surfaces of the header end plate and shell are cylindrical. The header dimensions are 184 mm × 88.8 mm × 44.4 mm with a 6 mm wall thickness. The nozzle outer diameter is 48 mm with a 6 mm thickness. Both headers and nozzles are thick-walled structures. The connections between nozzle and header, header end plate and header shell, and header and core are gross structural discontinuities that induce local primary and secondary stresses.

3 Thermal-Fluid Modeling

3.1 Temperature and Pressure Environment

The design parameters for the 20 kW PCHE are listed in Table 1 . The unit primarily withstands internal pressure and high temperature. The pressure difference and temperature difference between cold and hot side channels significantly affect microchannel stresses. The cold and hot side channels employ counterflow design, with the hot inlet adjacent to the cold outlet (average temperature difference of 80°C) and the hot outlet adjacent to the cold inlet (average temperature difference of 74°C). Temperature gradients along the channels induce longitudinal thermal stresses. The average temperature difference between cold channel inlet and outlet is 241°C, while for hot channels it is 321°C.

3.2 Coupled Heat Transfer Equations for Cold Fluid, Hot Fluid, and Solid

The temperature field distribution in the solid region is obtained by solving heat transfer equations. Neglecting microchannel structures, a homogenization approach establishes coupled heat transfer equations for the heat exchanger core. Since hot flows are separated from each other by channel walls, as are cold flows, fluid heat transfer by radiation and conduction can be neglected while convection dominates. The heat transfer equations for cold and hot flows are:

$$\epsilon_c \rho_c c_c \mathbf{v}_c \cdot \nabla T_c = q'_{c,s} \quad (1)$$

$$\epsilon_h \rho_h c_h \mathbf{v}_h \cdot \nabla T_h = q'_{h,s} \quad (2)$$

where ϵ_c and ϵ_h are the cold and hot channel void fractions, ρ_c and ρ_h are cold and hot fluid densities, c_c and c_h are cold and hot fluid heat capacities, $\mathbf{v}_c$ and $\mathbf{v}_h$ are cold and hot fluid velocity vectors, and $q'_{c,s}$ and $q'_{h,s}$ are the heat exchange power densities (heat exchange power per unit volume) between cold/hot fluid and solid regions.

The heat exchange power densities are expressed as:

$$q'_{c,s} = h_c A_c (T_s - T_c) \quad (3)$$

$$q'_{h,s} = h_h A_h (T_h - T_s) \quad (4)$$

where A_c and A_h are the heat transfer area densities per unit core volume for cold and hot channels, h_c and h_h are the convective heat transfer coefficients between cold/hot fluid and solid, and T_s is the macroscopic temperature (homogenized temperature) of the solid domain. The void characteristics of each core zone are listed in Table 2 . The average convective heat transfer coefficient for helium on the hot side is taken as 1,400 W/(m^2 · °C), and for air on the cold side as 200 W/(m^2 · °C).

Solid heat transfer considers primarily thermal conduction:

$$\rho_s c_s \frac{\partial T_s}{\partial t} = \nabla \cdot (k_s \nabla T_s) - q'_{c,s} - q'_{h,s} \quad (5)$$

where ρ_s, c_s, and k_s are the density, heat capacity, and effective thermal conductivity of the solid domain, respectively.

3.3 Finite Element Model and Boundary Conditions

A mapped solid mesh using hexahedral elements and a few prism elements was generated for the entire microchannel heat exchanger structure, as shown in Figure 4.1, with a total of 61,017 elements. The core interior was simplified into seven different homogenized zones, as shown in Figure 4.2. Continuity between the header and nozzle, between different homogenized zones within the core, and between the core interior and outer shell was achieved through shared element nodes. The mesh nodes of the headers and core are independent but coupled through constraint equations to ensure continuity at connection surfaces. Three element layers were used through the thickness of headers, nozzles, and core shell, employing quadratic reduced integration elements.

Assuming uniform flow distribution among all cold and hot channels, velocity vectors can be determined based on total flow rate, porosity, and channel orientation in different zones, as shown in Figure 4.3. The total flow rates for cold and hot fluids are listed in Table 3 .

Boundary conditions were set as follows: 1. Inlet surfaces of the core for cold and hot sides were defined as inflow boundaries with specified upstream temperature and pressure. 2. Outlet surfaces of the core for cold and hot sides were defined as outflow boundaries. 3. The outer surface of the heat exchanger was set as a convective heat transfer boundary with ambient air at 80°C. 4. Inner surfaces of inlet headers and nozzles for cold and hot sides were set as convective boundaries with fluid temperatures equal to core inlet temperatures. 5. Inner surfaces of outlet headers and nozzles for cold and hot sides were set as convective boundaries with fluid temperatures equal to average core outlet temperatures.

The convective heat transfer coefficients used in the model are listed in Table 4 .

3.4 Macroscopic and Microscopic Temperature Field Analysis

Assuming uniform flow distribution among channels, velocity vectors were determined from the flow rates in Table 1 and channel orientations in different zones. Using the finite element method, the coupled heat transfer equations (1)-(5) were solved to obtain temperature fields for cold fluid, hot fluid, and the solid macroscopic temperature field within the core. Figures 10 and 11 show the isosurface distributions and gradient directions for cold-side and hot-side fluid temperatures, respectively. The cold fluid is gradually heated along the hot channel direction, while the hot fluid is gradually cooled along the cold channel direction. Consequently, fluid temperatures at the cold and hot outlets are non-uniform across the cross-section, with maximum temperature differences reaching 100°C.

Due to the high average convective heat transfer coefficient of helium, the macroscopic temperature distribution in the solid core resembles that of the hot fluid, meaning the solid macroscopic temperature gradient direction is approximately along the hot channel direction. This macroscopic temperature gradient induces macroscopic thermal stress. Figure 12 [Figure 12: see original paper] shows the macroscopic temperature gradient magnitude distribution in the solid core, revealing a large gradient of approximately 2.7°C/mm near the hot-side inlet interior. This region will experience significant macroscopic thermal stress. Thermal stress is a secondary stress with self-limiting characteristics, but repeated cycling can cause fatigue and ratcheting effects. Therefore, the hot-side inlet interior represents one of the critical locations for fatigue and ratcheting failure.

The microscopic temperature field in the solid core correlates with local temperature differences between cold and hot channels. Figure 13 [Figure 13: see original paper] presents the temperature difference distribution between cold and hot flows within the heat exchanger core. Large temperature differences up to 210°C occur near the hot-side inlet interior, explaining the large macroscopic temperature gradient in this region, as greater local temperature differences yield higher local heat transfer rates. Additionally, large local temperature differences between cold and hot flows induce significant microscopic thermal stress, further indicating high failure probability in this region.

4 Effects of High-Temperature Environment on Structural Failure

During operation, the solid domain macroscopic temperature ranges from 742°C to 440°C, while microscopic temperatures range from 750°C to 355°C. High-temperature conditions affect structural failure behavior through two primary mechanisms: degradation of material mechanical properties and thermal stress generation through thermal expansion. At high temperatures, Alloy 617 exhibits strong temperature sensitivity and pronounced time-dependent behavior. Creep becomes significant above 580°C. With a hot-side design temperature of 800°C, the prototype is classified as a Class A high-temperature service component according to ASME Boiler and Pressure Vessel Code Section III Division 5, requiring evaluation of four strength-related failure modes: (1) ductile rupture under short-term loading; (2) creep rupture under long-term loading; (3) creep-fatigue interaction damage; and (4) gross structural distortion due to plastic incremental collapse and ratcheting. Additionally, two deformation-related modes must be considered: (5) deformation causing equipment damage or functional loss; and (6) structural instability under short-term and long-term compressive loading. Modes (2) and (3) are directly related to high-temperature environment, while modes (1), (4), (5), and (6) exhibit pronounced temperature sensitivity under high-temperature conditions.

To characterize mechanical property differences between diffusion-bonded joints and base material for failure analysis, tensile, shear, creep, and fracture toughness tests were conducted on Alloy 617, yielding comprehensive experimental data. For example, Figures 14 [Figure 14: see original paper] and 15 [Figure 15: see original paper] show strain-time curves for base metal and diffusion-bonded specimens at 850°C under various stress levels, demonstrating significantly reduced creep life for welded specimens compared to base metal.

However, the number of diffusion bonding mechanical property tests was limited, precluding probabilistic determination of a strength reduction factor. ASME Section VIII Division 1 Mandatory Appendix 42 (2015–2022 editions) recommended a weld joint efficiency factor of 0.7 for non-nuclear microchannel heat exchangers, though this content was removed in the 2023 edition. Since ASME VIII-1 applicability is limited to temperatures below the creep regime, this factor addresses static strength properties such as S_m. Comparison of measured diffusion-bonded S_m values with ASME standard base metal values is shown in Table 5 , revealing that diffusion-bonded S_m substantially exceeds 0.7 times the base metal standard value.

For creep strength limits S_t, referencing ASME arc welding practices, the diffusion-bonded S_t factor was extended to the minimum value of $0.8 \times S_t$, with the ASME VIII-1 Appendix 42 recommended 0.7 weld joint efficiency factor applied to creep rupture strength, i.e., the ratio of weld metal to base metal creep rupture strength taken as 0.7. At 850°C, measured diffusion-bonded S_t values and 0.7 times base metal ASME standard values are compared in Table 5 . Since the measured S_t exceeds 0.7 times the standard value, using the 0.7 factor yields conservative results. For diffusion-bonded creep strength limits, the minimum value of $S_t = 0.7S_{t,\text{base}}$ is adopted, where both diffusion-bonded and base metal S_t incorporate the 0.7 weld joint efficiency factor.

5 Potential Failure Locations, Modes, and Consequences in PCHE

Comprehensive analysis identifies the following potential failure locations and modes in PCHEs, with estimated consequences:

1. **Interlayer failure of heat exchanger core** due to insufficient diffusion bonding strength, causing gross ductile rupture, creep rupture, or plastic distortion. Consequence: core burst.
2. **Local failure at channel edges** due to microscopic stress concentration or residual crack-like defects from diffusion bonding, causing local ductile rupture, creep rupture, or fatigue fracture. Consequence: core leakage.
3. **Excessive plastic and creep deformation** at the thinnest plate wall location at channel arc centers, caused by combined primary stress and large microscopic temperature gradients, or fatigue failure under cyclic thermal stress. Consequence: significant channel deformation or cross-contamination leakage. This failure is most likely near the hot-side inlet interior where cold-hot temperature differences are large.
4. **Failure at connection welds** between headers and core, headers and nozzles, and header end plates and shells, due to insufficient weld strength or large local primary/secondary stresses, causing ductile rupture, creep rupture, or fatigue failure. Consequence: leakage along weld surfaces.

6 Conclusion

This study systematically investigated PCHE structural configurations, operating environments, and theoretical/numerical simulation aspects. For structural configuration, component characteristics were summarized, and a partitioned homogenization method was proposed for the complex heat exchanger core, providing a solution to modeling difficulties posed by numerous flow channels. For operating environments, high-temperature and high-pressure service conditions were characterized, and a fluid-solid coupled heat transfer numerical model based on homogenization theory was developed to predict temperature distributions in cold fluid, hot fluid, and solid domains, providing essential temperature parameters for PCHE safety assessment. Temperature field predictions revealed non-uniform temperature variations among channels on the same side, with temperature differences reaching 100°C at the cold and hot outlets. Finally, theoretical and preliminary numerical simulation analyses identified potential failure locations, modes, and consequences, guiding future PCHE safety assessment research.

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