Capillary Tube and Shell Heat Exchanger Design for Helium to Liquid Salt Heat Transfer

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Liquid salts have been suggested as the intermediate heat transport fluid for the Next Generation Nuclear Plant, due to their very high volumetric heat capacity when compared to high-pressure helium, and capability to transfer heat at low pressure over long distances with low pumping power. While it is recognized that liquid salts could bring attractive benefits, the use of liquid salts for this application raises two key issues for the design of the NGNP intermediate heat exchanger (IHX). First, the IHX must operate with a large pressure differential, and with acceptable creep deformation rates. Second, the IHX must have acceptable corrosion behavior. Both creep and corrosion can ultimately limit the lifetime of the IHX. This report presents an NGNP IHX design that is optimized to address these two key issues.

Because the IHX is a small fraction of the total capital cost of the plant, it is economically acceptable for the IHX to be replaced periodically, but it is preferable if the IHX has a lifetime of at least several years or longer. This calls for an IHX design that is capable of operating with minimal creep deformation while sustaining a large pressure differential. Previous work at UC Berkeley examined stress distributions in a Heatric type heat exchanger with very small (1-mm) liquid salt channels to reduce the compressive stresses on the channel. But it was found that the non-circular geometry of the channels resulted in stress concentrations that would be expected to cause creep deformation over time, potentially slowing as the deformation resulted in a more circular channel shape. Over time, creep deformation closes the salt channels and increases the salt pressure loss and pumping power.

Likewise, corrosion is a design issue for the IHX. Because the IHX is heated by helium, corrosion on the liquid salt side will result in the dissolution of wall material (particularly chromium) and transport to and deposition in the cold part of the intermediate loop (e.g., in process heat exchangers). Therefore plugging is not expected to be a problem for the IHX, and instead gradual weakening of the IHX due to material removal becomes the primary concern. Minimizing the corrosion rates using nickely plating or cladding has clear potential benefits.

The ideal geometry to sustain compressive stresses for IHX heat transfer surfaces is a cylinder, and here it is proposed that the NGNP IHX use capillary tubes (inside diameter less than 2 mm) for the liquid salt, with sufficiently thick tube walls to keep compressive stresses below the values that would result in significant creep deformation. Figure 1 shows a scaled drawing of a potential, non-optimized capillary tube and shell IHX module configuration.

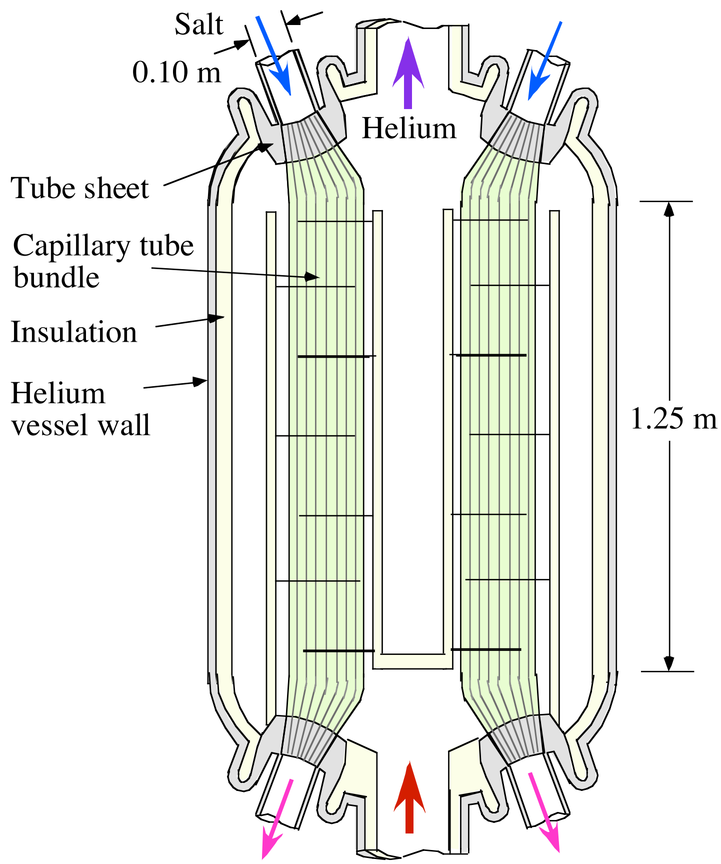

Fig. 1 Capillary tube and shell heat exchanger showing the proposed tube-bundle geometry formed by diffusion bonding of multiple bundles of ~2,500 3.0-mm diameter tubes with hexagonally tapered ends to form inlet and outlet tube sheets (Fig. 2).

For liquid salts the use of small diameter tubes is acceptable, because a relatively large pressure drop can be accepted since the low volumetric flow rate of salt still gives a reasonable pumping power.

A capillary tube and shell heat exchanger design could have several potential advantages:

1) Excellent response to thermal stresses and thermal transients, due to the flexibility of the tubes.

2) Compact geometry, with similar power density as a Heatric heat exchanger.

3) An optimal geometry to resist high temperature creep while operating with high pressure differential.

4) A geometry where creep-related failure is graceful, because it involves gradual closing of the flow channels and increase in the salt pressure drop, rather than rupture or other rapid failure, and it can be monitored online by measuring the IHX pressure drop.

5) Potential capability to fabricate with an internal cladding material, or to apply nickel plating, to the internal surface of the tubes for corrosion resistance with liquid salt.

6) Excellent geometry for experimental validation of corrosion and creep resistance under prototypical chemical and thermal conditions by testing single capillary tubes or small bundles of capillary tubes.

7) A geometry that allows physical access to the outsides of the heat exchanger tubes, allowing in-service inspection of heat transfer surfaces that is not possible for diffusion bonded plate type (Heatric) heat exchangers.

8) A geometry that allows a greater flow area for helium than a corresponding Heatric type heat exchanger, allowing the potential for reduced pressure drop and helium recirculating power (a substantial source of irreversibility).

It is possible that the tubes can be nickel plated on their interiors before the tubes are fabricated into a heat exchanger by running a nickel anode wire down the center with an electrolyte solution and running current from the tube to the nickel wire. Corrosion studies at the University of Wisconsin have found nickel plating to provide an effective method for corrosion control. Also, it is possible that the tubes can be co-extruded with an internal cladding material with high corrosion resistance such as nickel or Hastelloy N. Either the plating or the cladding method would help to decouple the question of corrosion from the question of adequate creep strength for the heat exchanger.

The major technical issue for the capillary tube and shell heat exchanger is the design of the tube sheet, because it must have adequate strength and creep resistance while withstanding the required pressure differential, and it must have appropriate thermal stress response during thermal transients. Several potential solutions exist, but here it is proposed that the ends of the capillary tubes be machined with a long hexagonal taper, so that a large number of capillary tubes can be bundled together and compressed into a tapered collar assembly, and diffusion welded into this configuration. Figure 2 illustrates a typical cross section of a tube machined in this manner. Normally this approach to fabricating a tube sheet could be problematic due to the crevices created on the shell side that could facilitate corrosion, but in this case the shell side fluid is dry helium. Figure 1 shows a schematic illustration of a multiple-bundle capillary tube and shell heat exchanger. A variety of tube bundle geometries are possible, and they can be optimized to provide for the relative thermal expansion of the tubes and the shell, for drainage, fabrication cost, in-service inspection, etc.

A reference (non-optimized) design has been developed to demonstrate viability. Capillary tubes with D_i = 1-mm inside diameter, D_o = 3-mm outside diameter tubes that have a heated length of approximately 1.8 m can be selected. The capillary tubes are loaded in compression, with an average compressive stress of $\sigma = D_o/(D_o-D_i)\Delta P = 1.5\ \Delta P$, which for a ΔP = 7 MPa pressure differential gives an average compressive stress in the tubes of 10.5 MPa that is well within the acceptable creep limited stress for available high temperature alloys such as Alloy 617, as shown in Fig. 3. If a co-extruded cladding is used, a somewhat thicker wall could be required.

A 50-MW IHX constructed using these capillary tubes and operating with a 25°C temperature differential between the helium and the liquid salt, would have design parameters as shown in Table 1 and a total mass of 30,000 kg. This can be compared to a Heatric helium to salt IHX design, which would have total mass of 43,000 kg (including inlet/outlet manifolds). Likewise a He to He IHX would weigh 44,000 kg, but would be required to operate in pressure balance. The liquid salt side pumping power is 430 kW, a very small recirculated power.

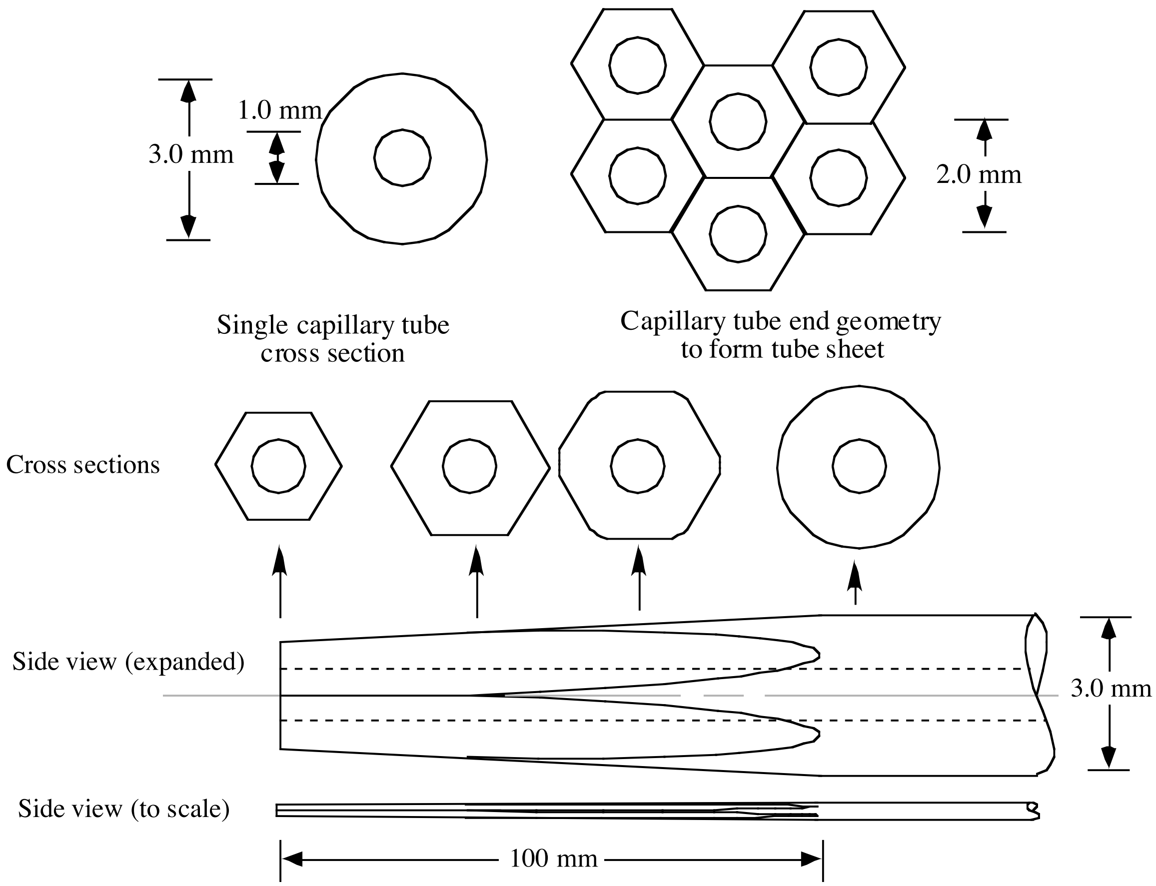

Fig. 2 Cross sections and side view of the hexagonal taper on the end of a capillary tube.

This is a non-optimal design, and many things can potentially be done to improve the heat transfer and reduce the surface area, such as using fins on the helium side to increase heat transfer.

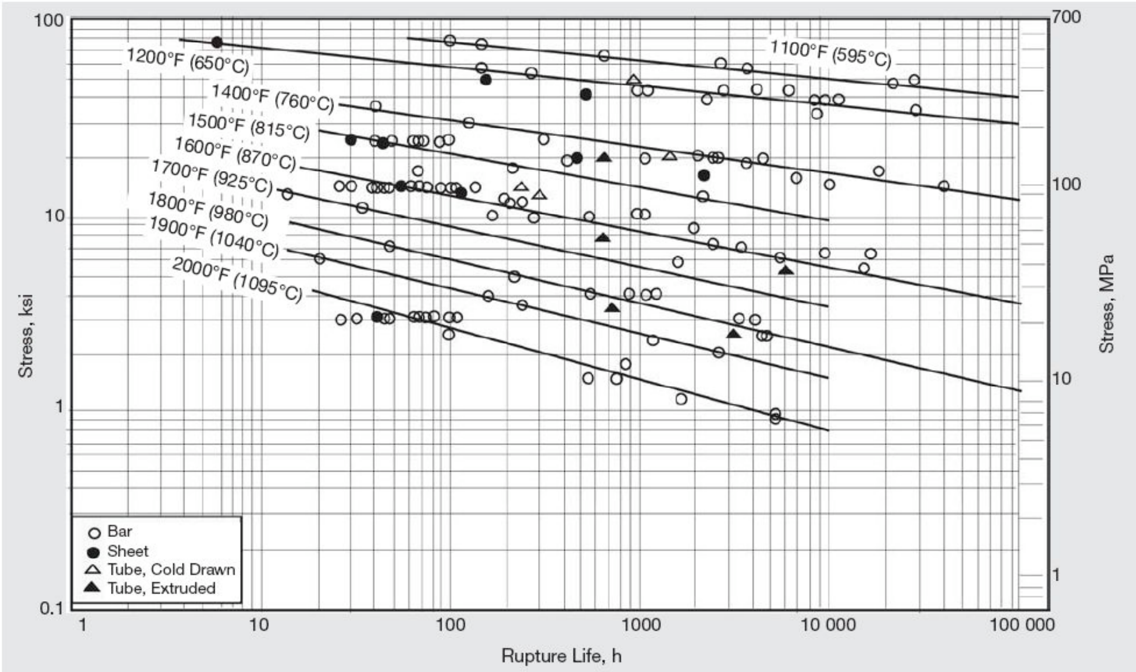

Fig. 3 Rupture life for Alloy 617, as a function of temperature [Special Metals, 2007].

Table 1 Non-optimized reference design parameters

Total thermal power	50 MW(t)
Flinak inlet/outlet temperatures	565°C / 925°C
Flinak total mass flow rate	73.718 kg/sec
Helium inlet/outlet temperatures	950°C / 590°C
Capillary tube inner/outer diameters	1.0 mm / 3.0 mm
Heated/total tube length	1.8 m / 2.0 m
Total number of tubes	259,000
Number of tubes in one 10-cm diameter tube sheet bundle	2,500
Total number of tube bundles	104
Flinak pressure drop	810 kPa
Flinak pumping power	430 kW

References

Special Metals, 2007, Alloy 617 Data Sheet, www.specialmetals.com.