Conceptual Design for a 50 MW(t) Metallic Intermediate Heat Exchanger for the Next Generation Nuclear Plant

Per F. Peterson, H. Zhao, D. Huang, and G. Fukuda
U.C. Berkeley
Report UCBTH-04-001

December 20, 2004

ABSTRACT

This UCB report summarizes preliminary design analysis for the 50 MW(t) intermediate heat exchanger (IHX) for the demonstration of hydrogen production with the Next Generation Nuclear Plant (NGNP). Sulfur-iodine (SI) hydrogen production efficiency is maximized when sulfuric acid decomposition is performed at a low pressure (~1 MPa) compared to the NGNP primary coolant pressure (~7 MPa). For a metallic IHX, however, operation with a large pressure differential may cause unacceptable creep. This report investigates creep-resistant metallic IHX designs using molten salt (MS) as an intermediate fluid, to determine the maximum operating temperature at which creep can be maintained at an acceptable level. Preliminary results indicate that a metallic He-to-MS IHX may be able to operate with a peak helium inlet temperature between 850 and 900°C, while accommodating a ~ 6 MPa pressure differential with acceptable creep. By selecting a metallic IHX design that permits hydrogen production at the optimal SI process pressure, a subsequent upgrade to high-temperature (1000°C) operation using a ceramic IHX then becomes possible.

INTRODUCTION

The Next Generation Nuclear Plant (NGNP) will use a 50-MW(t) intermediate heat transfer loop to demonstrate the direct production of hydrogen by thermochemical processes [1]. The selection of the intermediate heat-transfer fluid, where high-pressure helium and low-pressure molten fluoride salts (MS) are the primary candidates, has been identified as a major design choice for the reactor/process interface [2]. UC Berkeley performed a preliminary design comparison between helium and molten salt [3] that showed the potential for major differences in performance and in the potential sources of technical risk. Here we present a sufficiently detailed design for a metallic helium-to-MS intermediate heat exchanger (IHX) based on the Printed Circuit Heat Exchanger (PCHE) design to show the potential advantages of the MS approach.

A number of factors will affect the efficiency of the NGNP hydrogen production system. These include

1) the peak temperature of the intermediate fluid,

2) the temperature drop from the intermediate fluid to process fluids (the log-mean temperature difference (LMTD) across the process heat exchangers),

3) the degree to which the IHX and process heat exchangers operate in counterflow to achieve high effectiveness while limiting pressure drop,

4) the pressure of the process fluid (for the SI process) and the affect of the process pressure on driving the decomposition reaction toward completion (lower pressure is better), and

5) recirculated power required for the intermediate fluid pumping.

Likewise, a number of factors will affect the capital cost of the intermediate loop system, and the approach to achieving safe and reliable operation.

To meet the NGNP schedule, the first NGNP IHX must be constructed from metallic materials. It is also desirable that the design be capable of being upgraded to use ceramic composite heat exchangers, to allow for future increases in the intermediate loop peak temperature. For peak intermediate loop temperatures above 850°C, desired for efficient hydrogen production, steady-state stresses in the hot portions of a metallic heat exchanger and outlet manifold must be maintained below around 10 to 30 MPa to avoid long-term creep deformation and damage.

For a helium-to-helium IHX, current designs call for the intermediate helium to be at approximately the same pressure as the primary helium (around 7 MPa). This "pressure-balanced" operation is required to limit steady-state stresses in the heat exchanger due to the fluids differential pressure, and to provide sufficiently high helium density to give acceptable (although still large) intermediate-loop pumping power. Because the helium intermediate loop must operate at high pressure, the high-temperature process heat exchangers must either be designed to either (1) operate with a pressure difference between the helium and the sulfuric acid undergoing thermal decomposition, giving the same creep issues as the IHX experiences, or (2) operate with the sulfuric acid at high pressure in pressure balance. However, if the decomposition pressure is increased, the reaction shifts away from the desired degree of completion, reducing the efficiency of hydrogen production.

In the helium-to-MS IHX design presented here, the IHX is designed to accommodate a pressure difference of approximately 6 MPa between the primary helium and the intermediate MS. The sulfuric acid decomposition can then be performed in pressure balance at the optimal process pressure, anywhere from 0.1 to 2.0 MPa. The helium-to-MS IHX system then permits increased hydrogen production efficiency.

The IHX affects cycle efficiency in two other key ways as well, through the log-mean temperature difference (LMTD) and through pumping power. For a helium-to-helium IHX, pumping power is large and creates a strong tradeoff between LMTD and pumping power. While the LMTD can be reduced by increasing the IHX surface area, this increases the pumping power. More subtly, the LMTD also depends on the heat exchanger effectiveness, which depends on the degree to which the heat exchanger operates in counterflow.

For the plate-type PCHE heat exchanger, effectiveness depends on the fraction of the plate area that operates in cross-flow, rather than counterflow, and the uniformity of the flow distribution to the individual flow channels. For the helium-to-helium IHX, the equal volumetric flow rates of the two fluids creates large challenges in achieving high effectiveness. With equal volumetric flows a substantial fraction of the plate area must be devoted to cross-flow. Likewise, to obtain reasonable size, the inlet and outlet manifolds must operate with relatively high fluid velocities, so that the changes in the dynamic pressure from one location to another in the manifold can be substantial compared to the total pressure drop across the heat exchanger, which promotes flow maldistribution among individual flow channels and reduces effectiveness.

Conversely, the helium-to-MS IHX operates with a very large difference between the helium and MS volumetric flow rates. For this reason, the helium flow can occur as almost perfect through-flow across the plates (Fig. 1), allowing the helium flow to be easily distributed uniformly to all flow channels. For the MS, the flow channel configuration can be optimized to minimize the fraction of area devoted to cross flow, and also to give uniform flow distribution between flow channels, allowing the heat exchanger effectiveness to be high.

An important advantage to using molten salt is that the MS flow channels can be much smaller than equivalent helium flow channels would need to be, as shown in Fig. 2, while still maintaining very low pumping power. When the heat exchanger is immersed in the helium environment, the low-pressure channels are placed into compression. Because the roots between the MS channels are very thick (particularly compared to the roots between the helium channels), the maximum stresses can be kept to within 150% to 300% of the pressure difference between the helium and molten salt, that is, to around 10 to 30 MPa. This is the range of stresses where very low creep rates can be achieved in advanced metals (Haynes 214, etc.) at temperatures above 850°C.

In PCHE type heat exchangers, the flow channels are sufficiently small that laminar flow is expected. Under laminar flow, the Nusselt number $Nu = hD_h/k$ is a constant and is independent of the Reynolds number, where h is the heat transfer coefficient, D_h the hydraulic diameter of the flow channel, and k the thermal conductivity of the fluid. Because the thermal conductivity of molten salts is approximately 3.5 times larger than the conductivity of high-temperature helium, the heat transfer coefficient is also 3.5 times larger. Therefore, for the same heat flux the temperature drop from the channel wall to MS is 3.5 times smaller than for helium.

When a MS flow channel is made smaller, the heat transfer coefficient h increases inversely with decreasing D_h, while the heat transfer area (the channel perimeter) for heat transfer decreases linearly with D_h. Therefore, for a given total heat transfer rate, the temperature drop from the channel surface to the MS does not depend on the size of the channel. Thus the heat transfer performance of the heat exchanger is almost independent of the size of the MS channel. Instead, their optimal size is determined by the balance between reducing the peak compressive stresses by making the channels small, and decreasing the pressure drop, pumping power, and the blockage effects of corrosion and fouling, by making the MS channels larger.

Detailed calculations show that in a He-to-MS IHX more than 70% of the total temperature drop from helium to MS occurs between the helium and the helium channel surface. Detailed stress analysis shows that when a He-to-MS IHX is immersed in the helium environment, stresses between the helium channels are very small. This suggests that more complex helium flow channel geometries can be readily used to enhance the helium-side heat transfer coefficient, further reducing the IHX LMTD and size. UC Berkeley will be investigating the effect of helium-side heat transfer enhancement during the next phase of its current NERI project.

This report provides detailed mechanical stress analysis showing the effects of using smaller MS channels, provides thermal analysis and pressure loss information, and discusses the design of MS outlet and inlet headers for PCHE-type IHX designs. Candidate IHX materials are discussed, including the status of their ASME code certification, corrosion resistance with candidate molten fluoride salts, and their high-temperature thermal creep properties.

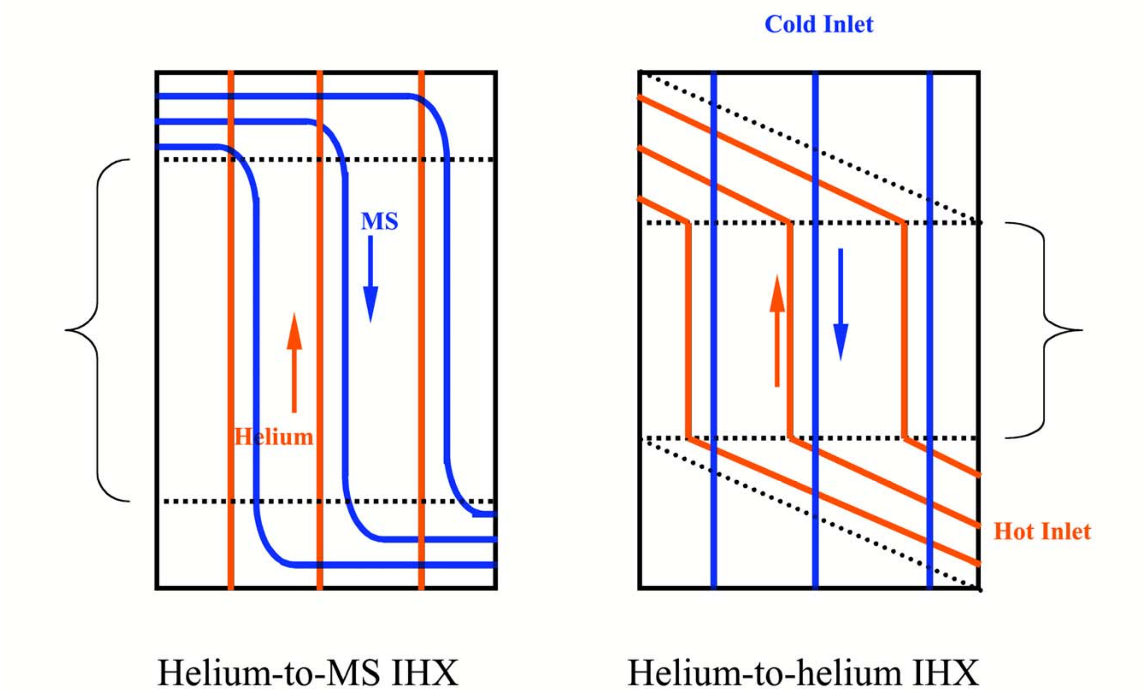

Fig. 1 Illustration of the flow paths through a helium-to-MS IHX, compared to a helium-to-helium IHX (note that the blue and red fluids are in different layers).

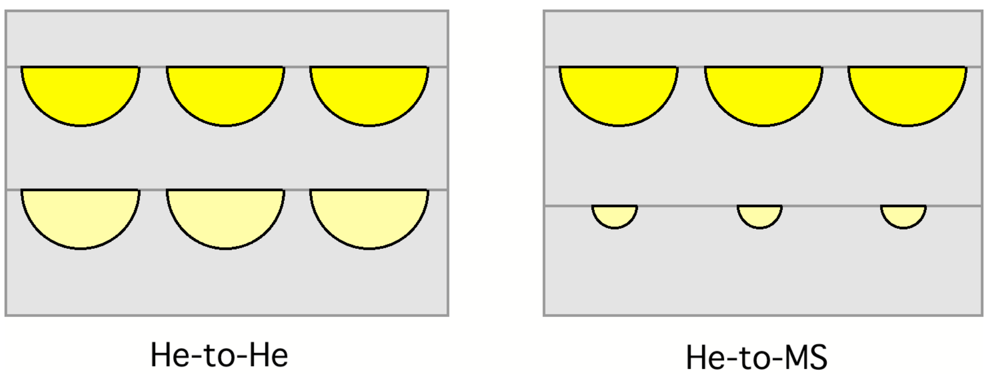

Fig. 2 Schematic of He-to-He and He-to-MS PCHE IHX flow channels.

MECHANICAL STRESS ANALYSIS

For the metallic PCHE-type heat exchangers for heat transfer from high-pressure helium (7 MPa) to intermediate pressure molten salt (1 MPa), the heat exchangers will be immersed in the helium environment, so the material will be loaded dominantly in compression. As shown in Fig. 2, the hydraulic diameter of the molten salt channels can be reduced by a factor potentially ranging from 2 to 4, while leaving the helium channels with their original size. In addition, because heat transfer coefficient for the molten salt is much higher than that of the helium, the MS channels can also be spaced with a larger pitch than the helium channels. Thus, the amount of solid material between each of the

MS channels is increased significantly, and the maximum stresses are reduced to values somewhat larger than the pressure difference between the helium and salt.

Detailed local stress analysis for different metals and geometries was performed using Pro/MECHANICA (Pro/M, an integral part of Pro/Engineer Software). The ratio of He channel diameter to the MS channel diameter was selected as 3:1. Haynes 214 and 800H are compared in this study. The following table shows the mechanical properties of these metals at 900ºC:

Table 1. High-temperature metal mechanical properties at 900ºC

	Haynes 214	800H
Density, kg/m^3	8200	7940
Young's modulus, GPa	151	133
Poisson's coefficient	0.298	0.41
Average Initial Stress to produce 0.5% creep over 10,000 hrs, MPa	20	18

Figures 3 and 4 show compressive stresses for each material. For the stress analysis for the Haynes 214 He-to-MS (HL-Haynes 214) IHX, the maximum compressive stress was about 8 to 8.5 MPa through most of the middle section between two MS channels and is around 12 MPa near the corners of MS channels. Because finite element analysis (FEA) doesn't give definite stresses at divergent corners the exact stress at the corners can not be predicted using Pro/M. Other more advanced FEA software may be used to obtain more precise solution for the stress concentration points. Similar results were found for Alloy 800H. For comparison, a He-to-He IHX with pressure balance at 7MPa is shown in Fig. 5. As expected, because it is in pressure balance stresses are small.

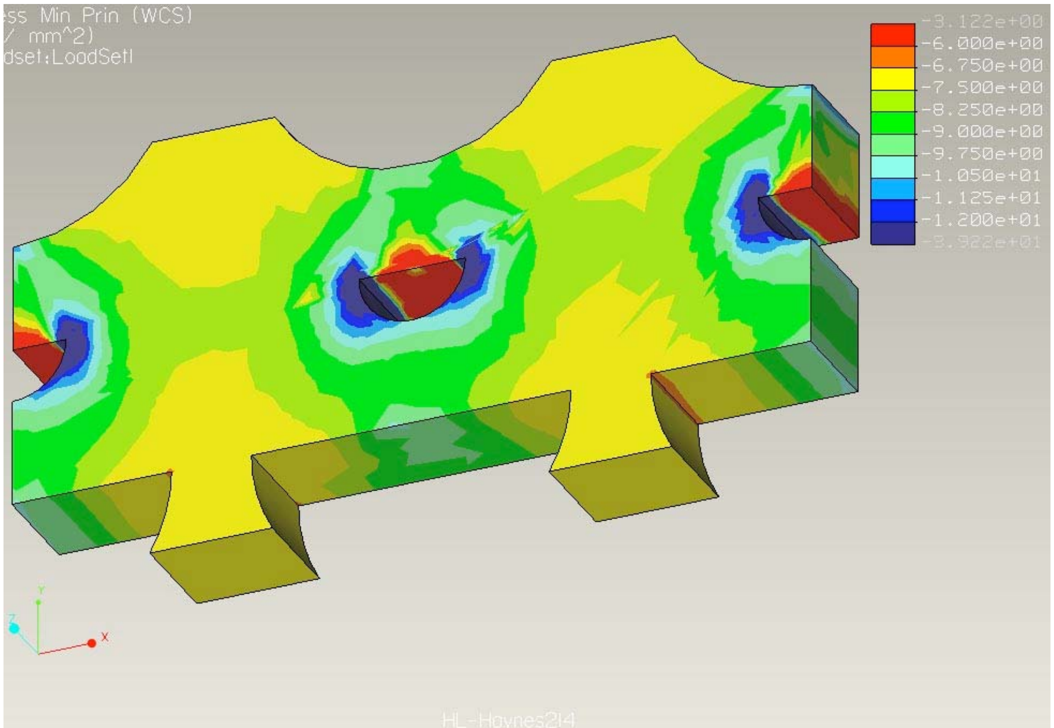

Fig. 3 Local FEM stress analysis result (MPa) for HE-to-MS Haynes 214 PCH IHX for He at 7 MPa and MS at 1 MPa.

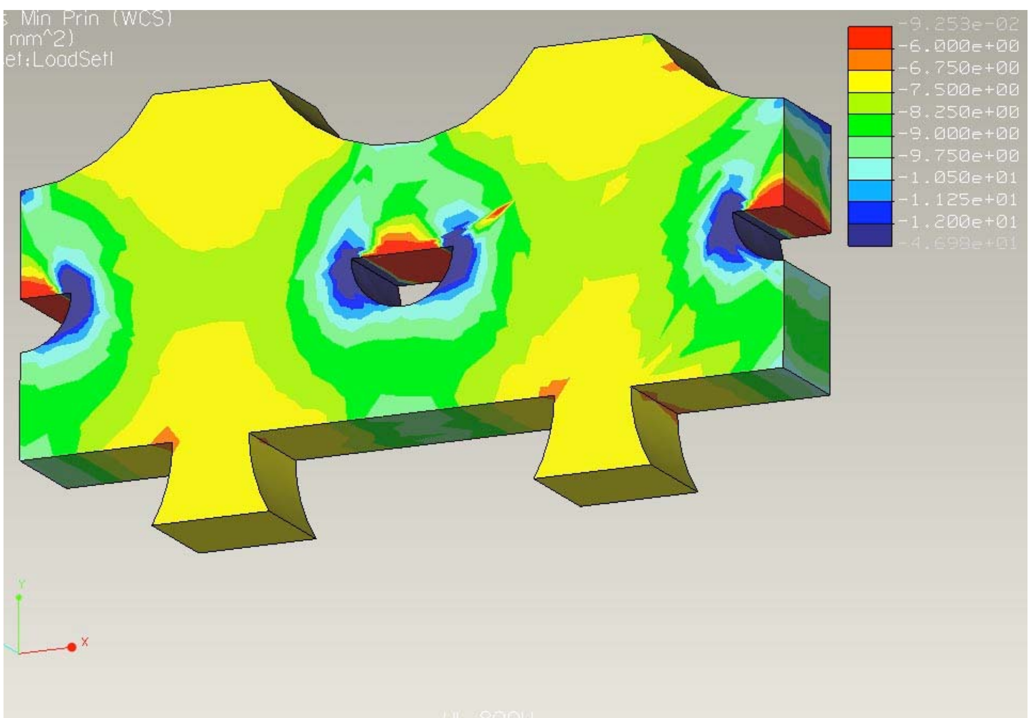

Fig. 4 Local FEM stress analysis result (MPa) for HE-to-MS Alloy 800H PCH IHX for He at 7 MPa and MS at 1 MPa.

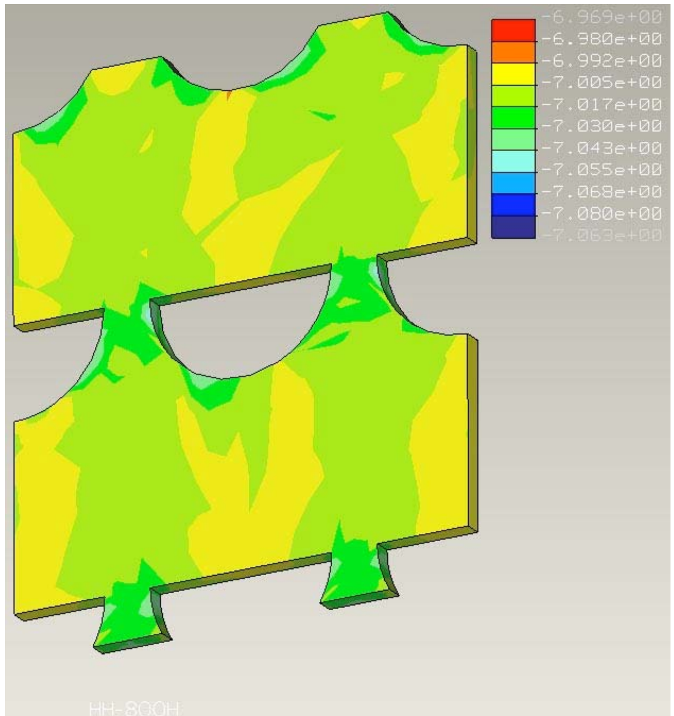

Fig. 5 Local FEM stress analysis result for an He-to-He 800H PCH IHX operated in pressure balance, with primary He at 7 MPa and intermediate He at 7 MPa.

THERMAL AND MECHANICAL DESIGN

Preliminary thermal designs have been carried out. The analysis assumed that the primary side inlet temperature is 900ºC and outlet temperature is 635ºC, and secondary side inlet temperature is 615ºC and outlet temperature is 880ºC. With these temperatures, the LMTD for the counter flow IHX is 20ºC and the thermal effectiveness for a perfect counter flow IHX is 0.93. Table 2 compares design results for a He-to-MS PCHE IHX with 7 MPa helium and 1 MPa MS, and a He-to-He PCHE IHX with both sides at 7 MPa. Both of the heat exchangers use Haynes 214 and have the same inlet and outlet temperatures. Both of designs operate in laminar flow and consume similar total pumping power. The helium side half circular diameter is 3 mm. When a plate type heat exchanger operates in laminar region, the total core volume and mass don't change with frontal area or pumping power. From the table, a He-to-He PCHE IHX will use 90% more volume and mass than a He-to-MS PCHE IHX. As discussed in the introduction section, a gas-to-gas HX needs much more volume and mass for inlet and outlet manifolds than a gas-to-liquid HX. Therefore, the total volume and mass for a He-to-He PCHE IHX is at least two times greater than a He-to-MS PCHE IHX.

Table 2. Thermal design results for a 50 MW(t) He-to-MS PCHE IHX and He-to-He PCHE IHX

	He-to-MS PCHE IHX	He-to-He PCHE IHX
Core volume, m^3	11	21
Frontal area, m^2	8.5	16
Core flow length, m	1.3	1.3
Total pumping power, W	$9.6x10^4$	$9.7x10^4$
Hot side counter flow region pressure loss, Pa	$6.7x10^3$	$3.8x10^3$
Cold side counter flow region pressure loss, Pa	$1.2x10^5$	$3.6x10^3$
Total core weight, metric tons	79	131
Core thermal density, MW/m^3	4.4	2.4

According to thermal design results, detailed preliminary mechanical designs have been developed for the NGNP IHX. These designs are not optimized, but still appear to be attractive. For the primary helium side, inlet/outlet manifolds are not needed because the IHX will be immersed in the primary helium environment. So the main task is to design a high effectiveness MS plate that minimizes the fraction of area used for cross flow. The design criteria for designing the flow channels on the MS side are:

- Gas bubbles can be reliably purged and the liquid can be reliably drained.
- The MS inlet and outlet distribution manifolds must provide uniform mass flows to the core region of the plate, to provide for high effectiveness. The distribution manifold must do this with a reasonable pressure drop, to avoid excessively high MS pressures and pumping power (although total pumping power will typically be dominated by the helium pumping).
- Cross-flow reduces the effectiveness of the HX. Because regions with cross-flow operate with larger temperature differences than counterflow regions, they reduce the temperature difference available in the counterflow region. Thus it is important to minimize the fraction of the total plate area that operates with cross-flow, which means minimizing the area of the plate occupied by the distribution flow channels.
- The HX must be able to sustain transient thermal stresses that would occur if there were a sudden change in the inlet temperature of helium or MS. Special design attention must be paid to the manifold holes through the plates, where the plate material must be significantly thicker.

Figure 6 shows a preliminary design for a MS plate. If all the inlet and outlet distribution channels have same hydraulic diameter as the core heat transfer region, the molten salt side flow maldistribution in the width direction will be more than 16% of average speed for a HX module with an average salt speed of 0.5 m/s in the core heat transfer region and a core heat transfer area of 1.3 m length by 1 m width. The reason for this large maldistribution is the large change of MS properties, especially viscosity, that occurs from inlet to outlet. For example, for flinak, the viscosity at the inlet (615ºC) is $2.1\text{x}10^{-6}$ m^2/s and at the outlet (880ºC) is $7.9\text{x}10^{-7}$ m^2/s, almost 3 times smaller. This results in a large difference in Reynolds numbers for inlet and outlet distribution areas. Because for laminar flow the friction factor is proportional to the inverse of Reynolds number, the difference in friction factors for inlet and outlet areas are large. The difference in friction factors for the similar inlet and outlet distribution channels with similar inlet and outlet pressures results in asymmetric velocity distribution between parallel channels. Fig. 7 shows the pressure loss distributions relative to the core pressure loss for inlet area, outlet area, and the total. The pressure losses in the inlet and outlet distribution area dominate the total pressure loss, since flow speeds are highest here. The maximum inlet pipe dynamic pressure is very small (0.3 kPa) relative to the total pressure loss through HX (min. 0.5 MPa and max. 0.9 MPa). Therefore, effects of flow acceleration on flow distribution are expected to be small.

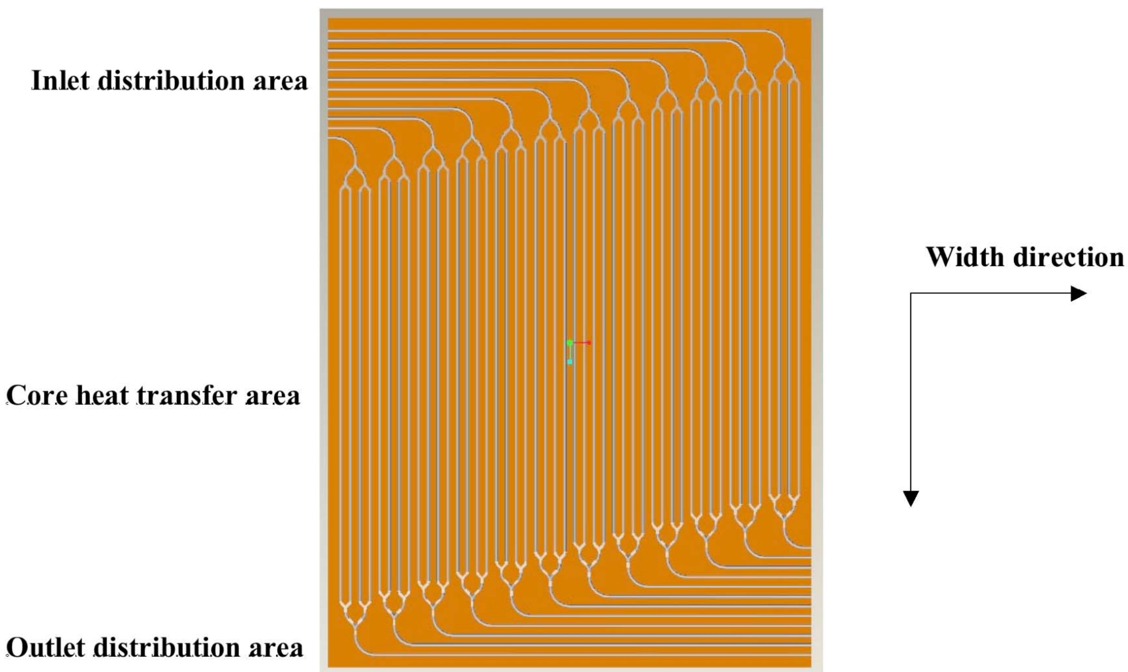

Fig. 6 MS plate design

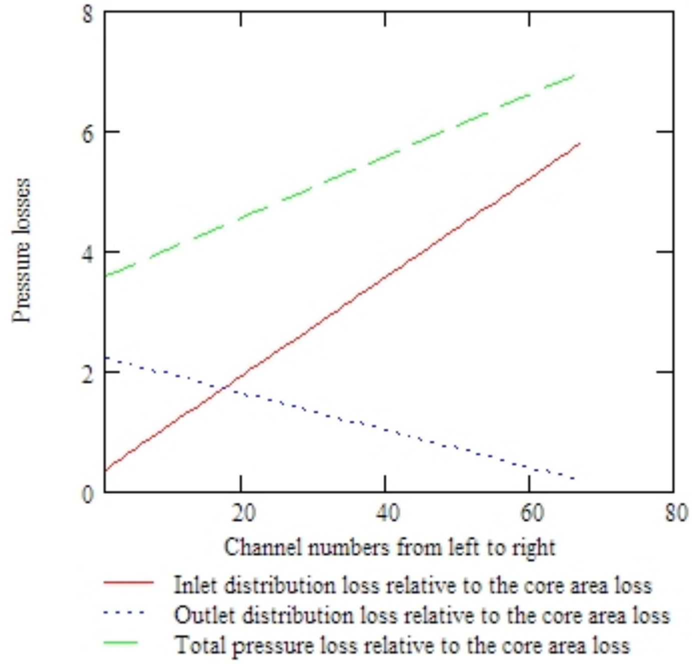

Fig. 7 Pressure loss distributions for parallel channels for inlet area, outlet area and the total for a MS plate with constant channel dimensions, assuming equal flow speed in each channel.

Detailed calculations show that the pressure loss from the MS inlet manifold to the outlet manifold is mainly due to the first layer inlet and outlet distribution channels(first layer for the inlet distribution channels means the channels starting from the inlet connection to the first forks). MS velocities in these distribution channels are approximate 4 times greater than the velocity in the core area. Also the pressure loss in the first layer channels is mainly due to viscous stresses (more than 90% of the total pressure loss in these layers), not from the form losses such as 90° turns, forks, and 135° turns. For the outlet side, which operates at high temperature, the same small channel dimension as in the core area must be used in order to maintain spacing between the channels to limit the stress below the low creep stress. For the inlet distribution channels, the available creep strength is much higher than the hot side, so larger channels will not cause problems with creep deformation. Thus it is possible to increase the inlet distribution channel dimensions to adjust for the property change effects. For simplicity, only the first layer inlet distribution channels would use variable dimensions. Scaling analysis shows that the pressure loss is proportional to the inverse of the 4th power of speed. Therefore we could change the channel hydraulic diameters from left to right in the following way to obtain uniform pressure losses for all the parallel channels:

$$d_i(j) := d_h \cdot \left[1 + \left(\frac{x_1(j)}{x_1(n_1)} \right)^{\frac{1}{4}} \cdot c_1 \right] \qquad (1)$$

$$c_1 := 0.265$$

where d_h is the hydraulic diameter for the core heat transfer area, $x_1(j)$ width distance measured from the left edge for the jth channel, n_1 the total number of channels for the first distribution layer, and c_1 a constant determined by pressure distribution balance. In this case, the largest channel diameter is only 26.5% more than the diameter for core channels. Fig. 8 shows the calculated pressure loss distributions for this inlet design. The inlet pressure loss distribution is well balanced by the outlet pressure loss distribution. The total pressure losses for the channels are very uniform. The maximum flow maldistribution is within 2% of the average value. Another benefit using larger variable inlet distribution channels is that the total pressure loss of the MS through the IHX is much reduced, since the higher viscosity of the MS at the inlet makes the inlet losses the largest. Uniform molten salt flow distributions in both width direction and height direction will be expected for the new design. This type of distribution design has a relative high salt pressure loss. However, due to the small volumetric flow rate of molten salt, the total pumping power for molten salt is acceptable and has a negligible effect on the net efficiency for hydrogen production.

The MS properties change effects may enhance the self-cleaning ability of the MS flow channels and reduce flow maldistribution. When a channel is partly blocked by a gas bubble or particles, the MS flow rate in this channel will decrease and the MS will reach higher temperatures in this channel. With the increase of MS temperature, viscosity will decrease and the MS flow rate will increase in this channel. Higher speed MS may flush out the blocking bubbles or particles in this channel, and will compensate partially for the effects of blockage that can not be cleared.

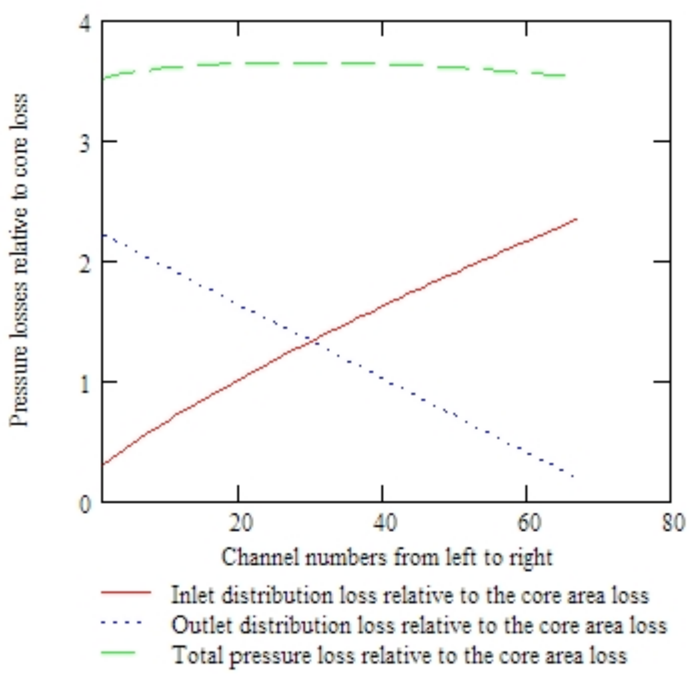

Fig. 8 Pressure loss distributions for parallel channels for inlet area, outlet area and the total for a MS plate with variable inlet distribution channel dimensions

Figure 9 shows the design of the helium side plate. The helium side plate does not need any flow distribution channels because the IHX will be immersed in the helium environment. Therefore, heat transfer enhancement features such as fins or wavy flow channels can be used to enhance helium side heat transfer, which dominates in the total thermal resistance.

Using these plate designs, we completed initial design of IHX modules and vessel assemblies. Figure 10 shows the preliminary design for an IHX module. The total length is about 2 m and total width is about 1 m. The height dimension is decided by fabrication considerations and the IHX pressure vessel assembly design. Figure 11 shows a preliminary vessel assembly design. The pressure vessel dimensions are 3 meters diameter and 6 to 7 meters length. Active cooling is used to cool the pressure vessel wall. The cold high-pressure helium is in pressure balance with the hot IHX inlet helium entering through the hot-duct distribution plenum. The MS cold inlet end has curved turns to accomodate thermal expansion and mechanical vibration. The MS hot outlet end is fixed to the vessel wall to reduce stresses in the hot outlet manifold. The hot MS pipe penetrates the vessel walls through a special hot gas gasket and flange design shown in Fig. 12. In this design, the high-pressure gasket and flange operate at same low temperature as the steel pressure vessel. Outside the IHX vessel, all high-temperature gaskets operate at low pressure.

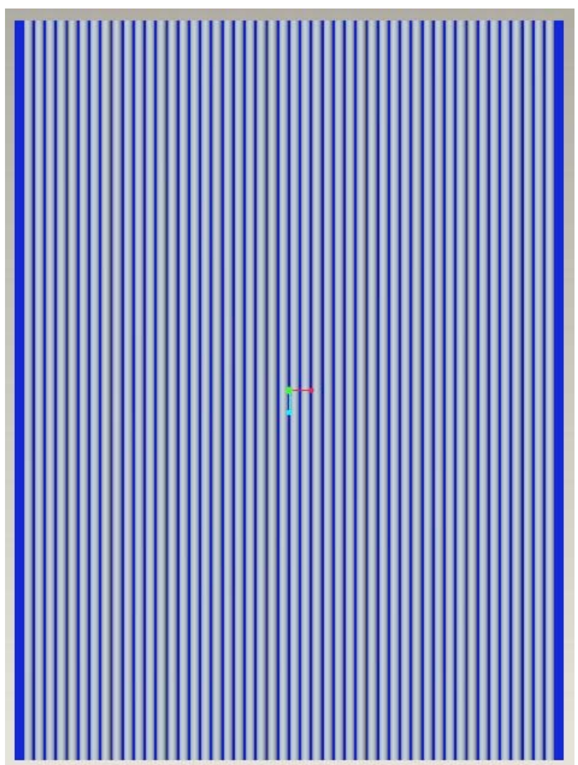

Fig. 9 Helium side plate design

Fig. 10 IHX module design. (Note: In the next design, the thickness of the outlet manifold wall will be increased to reduce stresses and limit creep.)

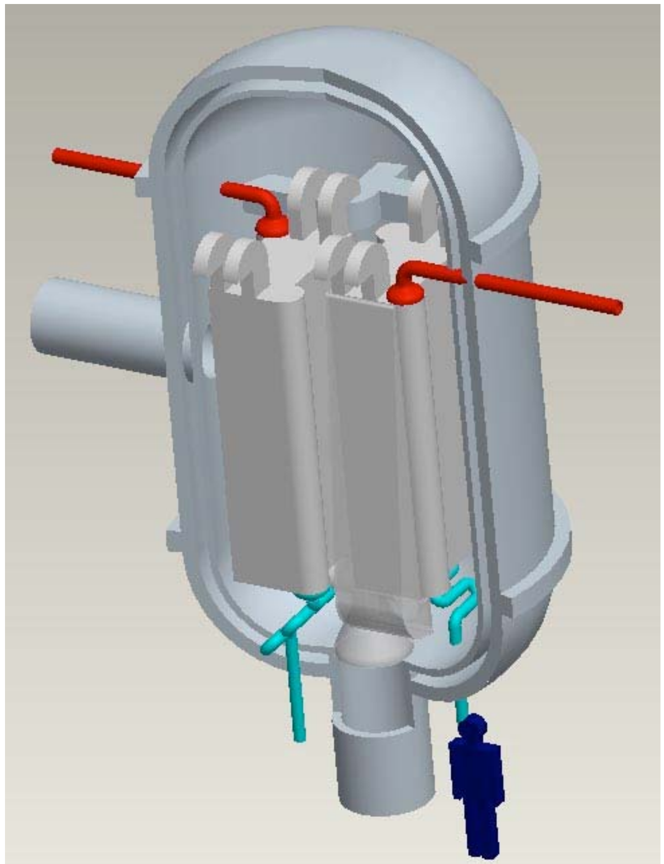

Fig. 11 Preliminary IHX vessel assembly – vertical design. Hot helium enters at the bottom of the vessel though an insulated hot duct, and cooled helium exits from the side of the vessel.

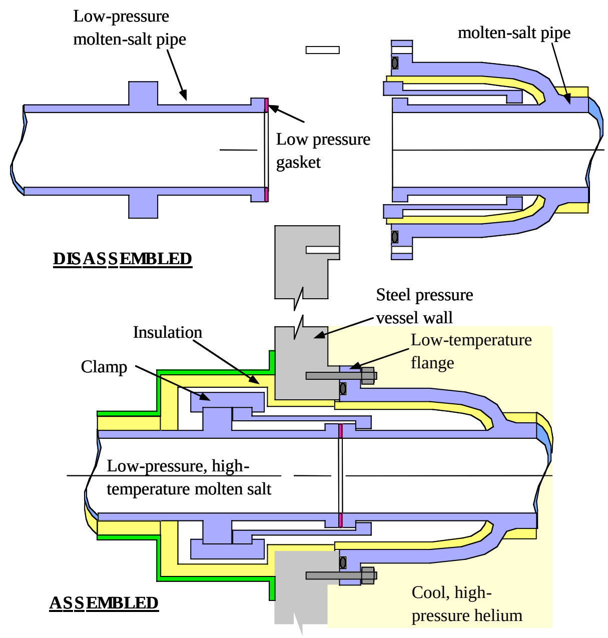

Fig. 12 High temperature gas gasket and flange design

CANDIDATE METALLIC IHX MATERIALS

Two primary issues must be considered in selecting high-temperature metallic materials for the NGNP IHX: corrosion and high-temperature mechanical properties (strength, creep, and fabricability). Clean helium does not have the potential to corrode intermediate loop materials, although contaminants can make helium corrosive. Likewise, clean molten salts exhibit very low corrosion rates if their fluorine potential is controlled to be low, and if the container material is composed of elements which are thermodynamically stable compared to the salt. Figure 12 shows an Ellingham diagram for several container and molten salt materials. Higher free energy (smaller value with a negative sign) implies higher solubility and thus more rapid corrosion. In general, carbon-based container materials will show very low corrosion rates with high-temperature molten salts. For nickel alloys, chromium is typically the most soluble constituent, and corrosion can proceed by dissolution of chromium in hot parts of the loop, with saturation-induced deposition in cold parts of the loop.

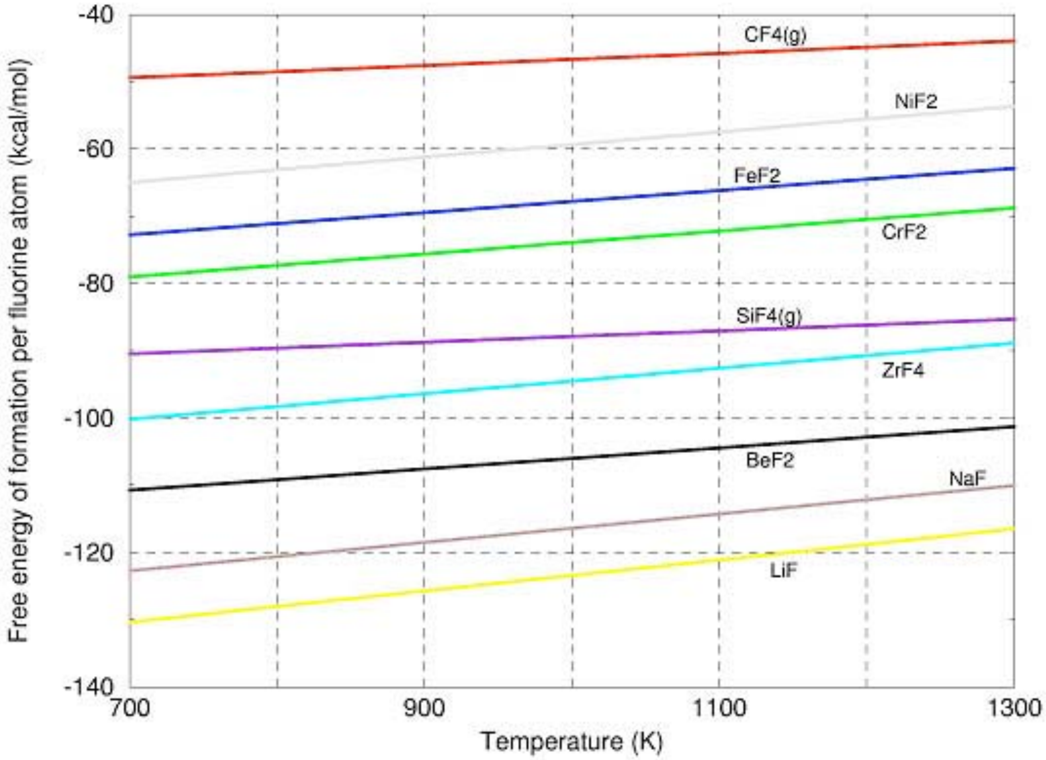

Fig. 13 Ellingham diagram for some container (e.g., carbon, nickel, iron, chromium, silicon) and solvent (e.g., lithium, sodium, beryllium, zirconium) fluorides (per fluorine atom) materials.

There are several high temperature metals which can be used for temperatures up to 900ºC. Examples are Haynes 214 and Alloy 800 H or HT, shown in Table 1. Based on the published data for Haynes 214 creep properties (http://www.haynesintl.com/214H3008C/214crp.htm), Haynes 214 was listed as one of the candidate high-temperature materials for heat exchangers for AHTR that would have good corrosion resistance, but which is not ASME code certified.

The other candidate appears to be Alloy 800 H or HT, which has both ASME Section VIII and III code certification and can work up to 1000°C. Here the primary issue is the corrosion resistance with salt (poor-fair); however, for the intermediate loop salt we can keep the salt highly reducing. ASME Code qualification is a major issue for practical application, so if the corrosion can be managed, 800H looks like it would be a good candidate for the IHX.

The majority of Heatric HXs are currently constructed from 300 series stainless steel, however various other materials are compatible with the chemical etching and diffusion bonding process and have been qualified for use in PCHE manufacture (http://www.PCHE.com/materials.html). Qualified materials include:

- 22 chrome duplex stainless steel

- Copper
- Titanium
- Nickel and nickel alloys
- Stainless steel 300 series.

REFERENCES

1. "Next Generation Nuclear Plant – High-Level Functions and Requirements," Idaho National Engineering Laboratory, INEEL/EXT-03-01163 (2003).
2. "Reactor/Process Interface Heat Exchanger and Intermediate Loop Technical Issues," Argonne National Laboratory, ANL W7500-0002-ES-00, Rev. 0, Sept. 2004
3. P.F. Peterson, "Comparison of Molten Salt and High-Pressure Helium for the NGNP Intermediate Heat Transfer Fluid," U.C. Berkeley Department of Nuclear Engineering Report UCBTH-03-004, Dec. 5, 2003.