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Intermediate Heat Transfer Loop Study for High Temperature Gas-Cooled Reactor

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C. H. Oh
C. Davis
S. Sherman

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INTERMEDIATE HEAT TRANSFER LOOP STUDY FOR HIGH TEMPERATURE GAS-COOLED REACTOR

C.H. Oh[1], C. Davis[1] and S. Sherman[2]

[1] Thermal Fluids & Safety Analysis,
Idaho National Laboratory, U.S.A.
Email: Chang.Oh@inl.gov,

[2] Savannah River National Laboratory, U.S.A.
773-42A, Room 134
Aiken, SC 29808
Email: Steven.Sherman@srnl.doe.gov

ABSTRACT
A number of possible configurations for a system that transfers heat between the nuclear reactor and the hydrogen and/or electrical generation plants were identified. These configurations included both direct and indirect cycles for the production of electricity. Both helium and liquid salts were considered as the working fluid in the intermediate heat transport loop. Methods were developed to perform thermal-hydraulic and cycle-efficiency evaluations of the different configurations and coolants. The thermal-hydraulic evaluations estimated the sizes of various components in the intermediate heat transport loop for the different configurations. This paper also includes a portion of stress analyses performed on pipe configurations.

INTRODUCTION
The Department of Energy and the Idaho National Laboratory are developing a Next Generation Nuclear Plant (NGNP) to serve as a demonstration of state-of-the-art nuclear technology. The purpose of the demonstration is two fold: 1) efficient low cost energy generation and 2) hydrogen production. Although a next generation plant could be developed as a single-purpose facility, early designs are expected to be dual-purpose. While hydrogen production and advanced energy cycles are still in its early stages of development, research towards coupling a high temperature reactor, electrical generation and hydrogen production is under way. Many aspects of the NGNP must be researched and developed in order to make recommendations on the final design of the plant. Parameters such as working conditions, materials, stresses, cycle components, working fluids, coupling of the hydrogen production plant and power conversion unit (PCU) configurations must be understood.

This paper describes various PCU configurations coupled to a High Temperature Steam Electrolysis (HTSE) plant by means of an intermediate heat transport loop (IHTL). The key issues that are addressed in this document are:

1. PCU configuration options
2. Coupling of the HTSE to the reactor
3. Working fluids in the PCU and IHTL
4. Efficiency
5. Component sizing
6. IHTL thermal-hydraulic performance
7. IHTL piping mechanical performance

BASIC ASSUMPTIONS
The NGNP was assumed to produce 600 MW of thermal power with a 900 °C outlet temperature and use helium coolant on the primary side. The nominal rise in fluid temperature across the core was assumed to be 400 °C, based on the point design (MacDonald et al. 2003). However for the reheat option this value was not used and a smaller temperature rise was calculated and applied to take advantage of the cycle.

The IHTL was assumed to be the same for all configurations. The loop was developed by Davis et al. (2005) and consists of piping to the hydrogen process plant, a heat exchanger between the loop and hydrogen process plant called the process heat exchanger (PHX), and a circulator. The IHTL was assumed to receive 50 MW of thermal power (ANLW 2004). Estimations of the required separation distance between the nuclear and hydrogen process plant vary considerably. For example, Sochet et al. (2004) recommended 500 m for the High-Temperature Reactor while Smith et al. (2005) recommended a separation distance of from 60 to 120 m for the NGNP. For this analysis, a nominal value of 90 m was used. The working fluid in the loop is assumed to be helium.

Hydrogen production is achieved by HTSE. The HTSE plant receives the necessary process heat from the IHTL. This heat is transferred to the HTSE plant by means of three process heat exchangers (PHXs). The electrical power needed for the electrolyzer will come from the electrical power produced by the PCU.

The nominal reactor pressure was assumed to be 7 MPa (INEEL 2005). The cycle working pressure was also assumed to be 7 MPa. The pressure drop across the hot-side of the IHX was assumed to be 0.05 MPa. For the HTLHX the nominal cold side pressure drop was taken to be 0.139 MPa (Davis et al 2005). The recuperator was assumed to have a hot side pressure drop of 0.1 MPa. The precoolers and intercoolers in the three-shaft and reheated cycles were assumed to have a 0.05 MPa pressure drop.

The working conditions in that loop are summarized in Table 1. Both helium and liquid salts were considered as working fluids for the IHTL. The liquid salt NaBF4-NaF in molar concentrations of 92% and 8% was used because of its low freezing temperature of 385 °C. The use of this liquid salts can potentially increase the heat transfer and reduce the pumping power; however it also introduces material problems such as compatibility and freezing.

Table 1. Working conditions in the IHTL.

Parameter	Nominal Value	
	He	$NaBF_4$-NaF
Power, MW	50.4	49.3
Heat Loss, MW	1.79	1.79
Outlet temperature of HTLHX, °C	875.1	875.1
Pressure drop, kPa	139.0	5.0
Pressure, MPa	2	2
Mass Flow, kg/s	27.5	94.8

METHODS

The cycles were modeled and optimized in HYSYS. The efficiency of the power conversion unit was calculated as follows

$$\eta_{PCU} = \frac{\Sigma W_T - \Sigma W_C - \Sigma W_{CIR}}{Q_{th} - Q_{IHTL}}, \tag{1}$$

where ΣW_T is the total turbine workload, ΣW_C is the total compressor workload, W_{CIR} is the circulator workload in the primary and secondary side, Q_{th} is the reactor thermal power and Q_{IHTL} is the heat delivered to the IHTL through the HTLHX. The efficiency of the overall cycle including the HTSE plant was calculated as follows

$$\eta_{overall} = \frac{\Sigma W_T - \Sigma W_C - \Sigma W_{CIR} + \Sigma W_{H2} - Q_{EL} + Q_{H2}}{Q_{th}} \tag{2}$$

where W_{H2} is the workload in the HTSE plant, Q_{EL} is the power supplied to the electrolyzer, Q_{H2} is the lower heating value of the produced hydrogen and Q_{th} is the reactor thermal power.

Once the cycle efficiencies had been calculated the relative sizes of the turbomachinery and heat exchangers were estimated. The actual size of the turbomachinery was not calculated but rather parameters that gave some indication to their relative size. The volume of the heat exchangers was calculated.

To determine the relative sizes of the heat exchanger, the UA values (overall heat transfer coefficient times the heat transfer area) of the heat exchangers were calculated by HYSYS. The U values were calculated, the heat transfer areas were determined, and the heat exchanger volume was calculated. This gives a relative estimation of the heat exchanger sizes for the different configurations.

The IHX, HTLHX, and recuperator were assumed to be printed circuit heat exchangers (PCHE) as designed by Heatric (2005). PCHE are composed of channels chemically etched into plates. The plates are then stacked and diffusion bonded together and headers are attached to form the heat exchanger. For this study Alloy 617 was used as the construction material for the heat exchangers. The thermal conductivity was assumed to be constant over the length of the heat exchangers and was obtained from www.specialmetals.com.

The detail analysis methods are described by Oh et al. (2006) and Davis et al. (2005).

SYSTEM CONFIGURATIONS

Seven plant configurations were evaluated. For convenience, the following nomenclature is used relative to the heat exchangers:

- IHX - The first heat exchanger downstream of the NGNP outlet
- PHX - The heat exchanger that connects the intermediate heat transport loop to the hydrogen production plant
- SHX - The heat exchanger that, if present, is located between the IHX and the PHX, and is referred to as the secondary heat exchanger (SHX).

The seven plant configurations evaluated are illustrated in Figures 1 through 7. The configurations include direct and indirect electrical cycles as shown in Figures 1 – 4 and 5 – 7, respectively. The configurations include both serial and parallel heat exchanger options. In the serial option, which is illustrated in Figures 1, 3, and 5, the IHX or SHX is located upstream of the power conversion unit (PCU). In the serial option, the heat exchanger removes less than 10% of the reactor power and directs it towards the hydrogen production plant. With this configuration, the hydrogen production plant receives the highest possible temperature fluid while the PCU receives a lower temperature fluid. In the parallel heat exchanger option, which is illustrated in Figures 2, 4, 6, and 7, the hottest fluid is divided, with most going towards the PCU and the remainder going towards the hydrogen production plant. This configuration is more complicated, but results in a higher overall efficiency because both the electrical and hydrogen production plants see the maximum possible temperature. The final option uses a SHX as shown in Figures 3, 4, 5, and 6. This option utilizes a third or tertiary coolant loop that provides additional separation between the nuclear and hydrogen plants, which should increase the safety of both plants and may make the nuclear plant easier to license. However, this option requires more capital investment and lowers the overall efficiency of the plant.

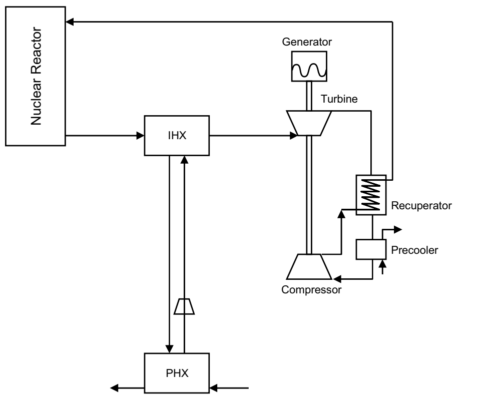


Figure 1. Configuration 1 (direct electrical cycle and a serial IHX).

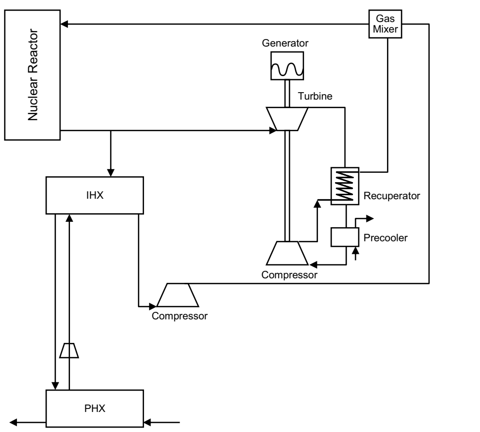


Figure 2. Configuration 2 (direct electrical cycle and a parallel IHX).

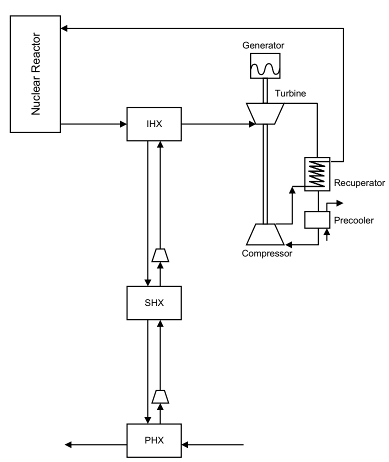


Figure 3. Configuration 3 (direct electrical cycle, serial IHX, and SHX).

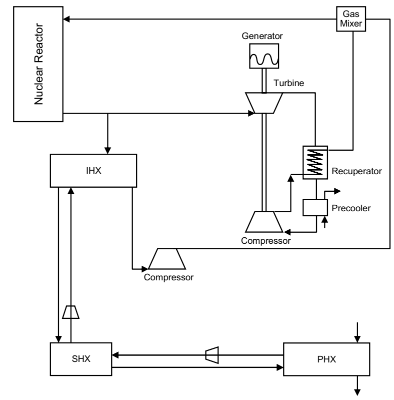


Figure 4. Configuration 4 (direct electrical cycle, parallel IHX, and SHX).

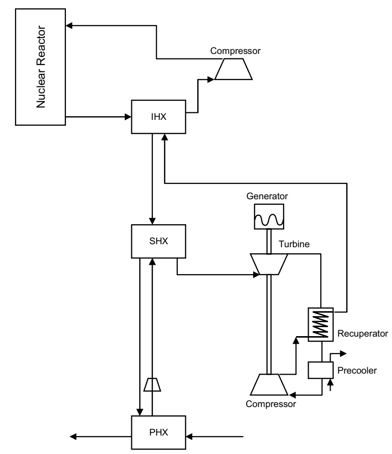


Figure 5. Configuration 5 (indirect electrical cycle and a serial SHX).

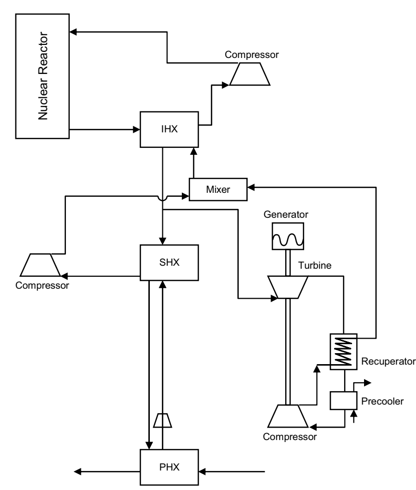


Figure 6. Configuration 6 (indirect electrical cycle and a parallel SHX).

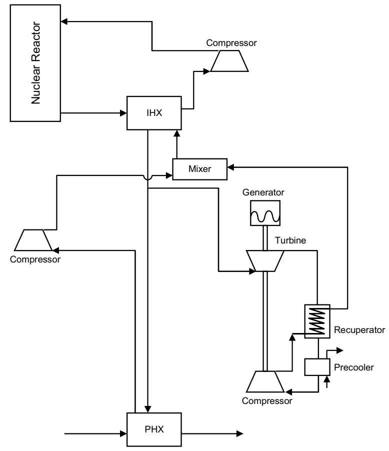


Figure 7. Configuration 7 (indirect electrical cycle and a parallel PHX).

Figure 8 shows the HTSE system. The process water enters on the left. The water is then pumped up to the operating pressure of 5 MPa. The efficiency of the pumps and circulators is assumed to be 75%. This water is then combined with water condensate returned from the hydrogen/water product stream. This stream then enters the low temperature recuperator. The pressure drop through the heat exchangers is assumed to be 20 kPa. From there the steam is further heated by PHX 3. Upon leaving PHX 3 the steam is mixed with hydrogen from the product stream by a recirculator which works to overcome the pressure drops in the system. A mole fraction of 90% water and 10% hydrogen is maintained in this model. This hydrogen helps to maintain reducing conditions at the electrolysis stack to prevent oxidation. The mixed stream then enters the high temperature recuperator which takes advantage of the high temperature outlet from the electrolysis stack. The hydrogen/water stream is then heated to the operating temperature for the electrolysis stack, in this case 827 °C, in PHX 1.

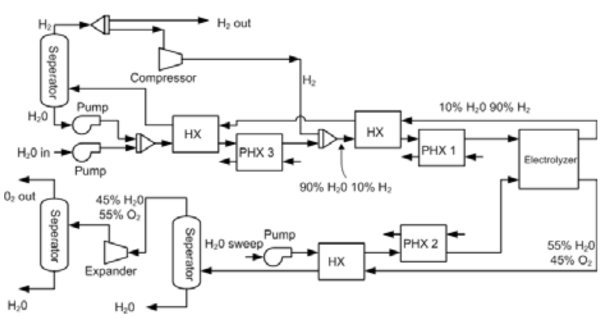


Figure 8. Schematic of HTSE process.

THERMAL HYDRAULIC EVALUATION

Figure 9 shows a snapshot of the HYSYS simulation of Configuration 1. Table 2 summarizes the important parameters in the simulation. The overall cycle efficiency calculated from Equation (2) was 50.6%.

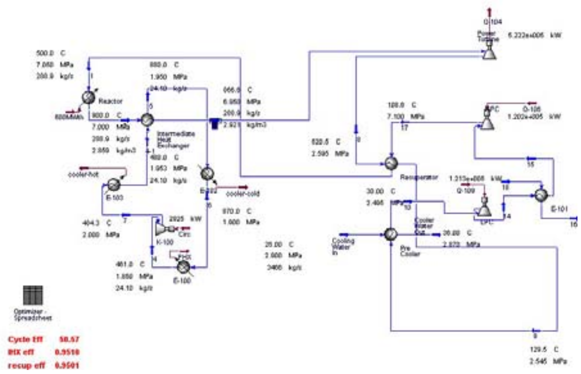


Figure 9. Snapshot of the HYSYS model of Configuration 1.

Configuration 2 utilizes a direct electrical cycle and a parallel IHX. The helium from the reactor is split with 88.9% flowing towards the PCU and the rest towards the IHX. The higher turbine inlet temperature results in a higher efficiency, but the reduced mass flow results in a lower efficiency. The overall cycle efficiency is slightly higher than that of the Configuration 1, which indicates that the effect of temperature was larger than the effect of mass flow. The overall efficiency for Configuration 2 is 50.7 %, which is the highest efficiency among the cycles evaluated.

Configuration 3 is similar to Configuration 1 except that it utilizes a SHX and a short tertiary loop to provide better separation of the nuclear and hydrogen plants. The overall efficiency is 50.3% which is slightly less than that of Configuration 1. Configuration 4 is similar to Configuration 2 except that it also utilizes a SHX and a short tertiary loop. Again, the use of the SHX resulted in only a slight decrease in overall efficiency.

Configurations 5 and 6 utilize an indirect cycle. The efficiencies of Configurations 5 and 6 are 1.1% lower than for Configurations 1 and 2, respectively, which utilized a direct cycle.

Configuration 7 utilizes an indirect cycle with the intermediate heat transport loop and the PCU arranged in parallel. Because a low pressure of 2 MPa was used to limit the stress in the PHX, the PCU also operated at a low pressure. The lower pressure reduced the efficiency of this configuration to 38.6%. This configuration is not considered to be a viable option because of its low efficiency.

Table 2 summarizes the results for all seven configurations with helium coolant.

	Conf-1	Conf-2	Conf-3	Conf-4
PCU configuration	Direct	Direct	Direct	Direct
IHX	Serial	Parallel	Serial	Parallel
SHX	N/A	N/A		
Turbine inlet	866.6 °C	900 °C	866.6 °C	900 °C
	288.9 kg/s	256.8 kg/s	288.9 kg/s	256.8 kg/s
HPC outlet	108.6 °C	119.8 °C	108.6 °C	119.8 °C
	7.1 MPa	7.1 MPa	7.1 MPa	7.1 MPa
Flow rate to IHX (cold side)	24.1 kg/s He	27.5 kg/s He	32.1 kg/s He	27.5 kg/s He
Flow rate to SHX	N/A	N/A	24.38 kg/s He	26.5 kg/s He
Pressure ratio	2.85	3.23	2.83	3.23
Overall cycle efficiency	50.6%	50.7%	50.3%	50.6%
	Conf-5	**Conf-6**	**Conf-7**	
PCU configuration	Indirect	Indirect	Indirect	
IHX				
SHX	Serial	Parallel	N/A	
Turbine inlet	853.7 °C	886.3 °C	880.4 °C	
	292. kg/s	260.1 kg/s	270. kg/s	
HPC outlet	110.3 °C	121.2 °C	144.7 °C	
	7.1 MPa	7.1 MPa	2.0 MPa	
Flow rate to IHX (cold side)	292. kg/s He	292.2 kg/s He	22. kg/s He	
Flow rate to SHX	24.1 kg/s He	27.5 kg/s He	22. kg/s He	
Pressure ratio	2.90	3.29	4.10	
Overall cycle efficiency	49.5%	49.6%	38.6%	

Using Configuration 1, the helium in the intermediate heat transport loop was replaced with Flinak in the HYSYS model to investigate the sensitivity of the calculated results to the working fluid. The results are compared in Table 3. The pumping power in the intermediate heat transport loop was significantly smaller with Flinak as the working fluid. However, the difference in cycle efficiency was relatively small (0.2%) because the pumping power was small compared to the reactor power even with helium as the working fluid.

Table 3. The effect of working fluid on the overall efficiency for Configuration 1.

	Helium	Flinak
Reactor power	600 MW-thermal	600 MW-thermal
Configuration	Direct and serial IHX	Direct and serial IHX
Reactor inlet	500 °C	500 °C
	7.05 MPa	7.05 MPa
Reactor outlet	900 °C	900 °C
	7.0 MPa	7.0 MPa
Helium mass flow to PCU	288.9 kg/s	288.9 kg/s
Turbine inlet	866.6 °C	866.6 °C
	6.95 MPa	6.95 MPa
HPC outlet	108.6 °C	108.6 °C
	7.1 MPa	7.1 MPa
Flow rate of intermediate loop	24.1 kg/s He	133 kg/s Flinak
Pressure ratio	2.85	2.85
Pump power	3.2 MW	47.9 kW
Cycle efficiency	50.6%	50.8%

The diameters and insulation thicknesses of the hot leg of the intermediate loop are compared in Figures 10 and 11 for each configuration. The variations in hot leg diameter and insulation thickness were generally small with helium as the working fluid and were primarily due to differences in the assumed flow rates. The figures also show that the diameter and insulation thickness were much smaller when the helium coolant was replaced by liquid salt.

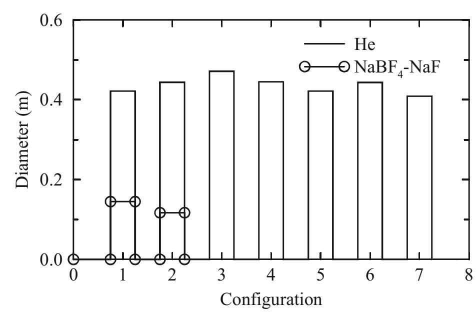


Figure 10. A comparison of hot leg diameters in the various configurations of the intermediate heat transport loop.

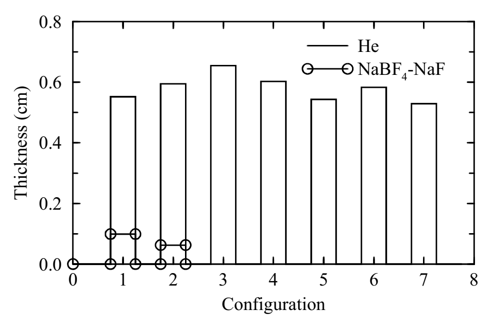


Figure 11. A comparison of insulation thicknesses in the hot leg for the various configurations of the intermediate heat transport loop.

The relative sizes of the configurations are illustrated in Figure 12, which is based on the calculated metal volume in the intermediate heat transport loop, and, if present, the SHX and the tertiary loop. The volume includes that of the fluid in the compact heat exchangers, but does not include that of equipment in the PCU, such as the recuperator and the precooler, but does include the IHX.

Figure 12 shows that the smallest size was obtained with Configuration 1. The components that contained the most metal in Configuration 1 were the hot leg, the PHX shell, and the PHX tubes. The metal volumes for Configurations 3 and 4, which contained a SHX, were 40 to 60% greater than for Configurations 1 and 2, which did not contain a SHX. In addition to its volume, the presence of the SHX required the other heat exchangers to be larger to achieve the necessary effectiveness. The total metal volumes for the configurations with an indirect electrical cycle were significantly larger than for those with a direct cycle. For example, the total volume for Configuration 5, which utilized an indirect cycle, was five times greater than for Configuration 1, which utilized a direct cycle. The large increase in volume was a consequence of the IHX being designed to remove 600 MW of thermal power for the indirect cycles versus 50 MW for the direct cycles. However, the costs of the indirect and direct options will not differ by a factor of five because of the relatively large components, such as the recuperator and precooler, that are present in both options but are not included in Figure 12.

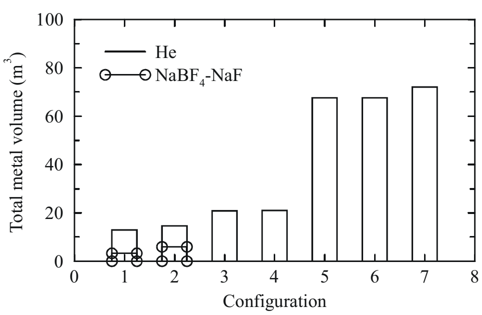


Figure 12. Relative sizes of the intermediate heat transport loop in each configuration.

STRESS ANALYSIS

Simple stress calculations were performed for three configurations of the IHTL piping. The configurations included the parallel and concentric arrangements described in Figure 13 and a jacketed arrangement, in which a pressurized pipe was placed outside of the hot leg.

The stress evaluations were based on the case with helium working fluid, 50-MW of loop power, and a separation distance of 90 m. The evaluations used thin-walled approximations in the metal and neglected the strength of the insulation. The assumed conditions and geometry are summarized in Table 4. In the jacketed configuration, the pressure in the jacket was half of that of the hot leg. The thickness-to-diameter ratios of the hot leg and the jacket were half of the value used for the legs in the parallel configuration.

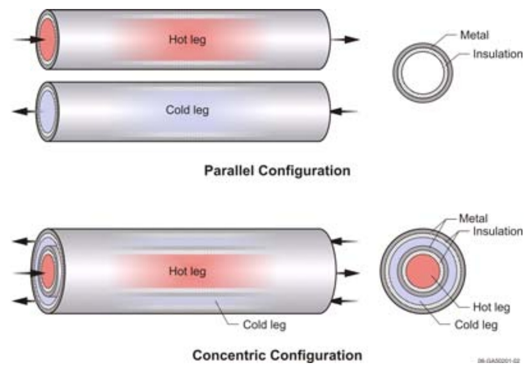


Figure 13. Parallel and concentric piping configurations for the IHTL.

Table 4. Stress analysis parameters.

Parameter	Value
Hot / cold leg pressure, MPa	6.95 / 7.0
Metal temperature, °C	355
Metal material	Carbon steel
Parallel configuration:	
Hot / cold leg inner diameter, m	0.330 / 0.304
Hot / cold leg insulation thickness, m	0.0125 / 0.00305
Metal thickness-to-inner-diameter ratio	0.06
Concentric configuration:	
Hot / cold leg inner diameter, m	0.3300 / 0.5186
Hot / cold leg insulation thickness, m	0.00920 / 0.00305
Hot leg metal thickness-to-inner-diameter ratio	0.02
Cold leg metal thickness-to-inner-diameter ratio	0.06
Jacketed configuration:	
Jacket pressure, MPa	3.45
Hot leg inner diameter, m	0.330
Hot leg insulation thickness, m	0.0125
Hot leg metal thickness-to-inner-diameter ratio	0.03
Jacket inner diameter, m	0.400
Jacket metal thickness-to-inner-diameter ratio	0.03

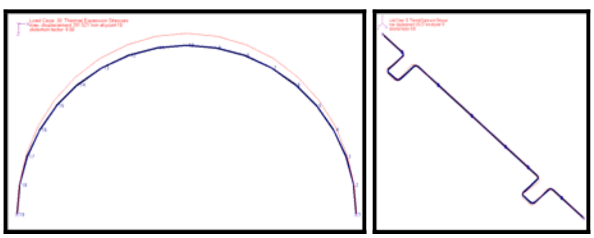
The hoop stress calculated for each configuration was then compared to 132 MPa, the allowable value for Class C carbon steel seamless pipe at 355 °C (ASME 2002). Ratios of the allowable stress to the calculated stress are presented in Table 5. In each case, the allowable stress was slightly more than two times the calculated stress. The difference between the allowable and calculated values provides some margin for increases in stress at elbows and fittings and reductions in strength at welds. The stress evaluation indicates that the design appears feasible if the insulation is effective in reducing the temperature of the metal to 355 °C.

Table 5. Stress analysis results.

Configuration	Allowable stress/ calculated stress
Parallel configuration:	
Hot leg	2.16
Cold leg	2.14
Concentric configuration:	
Cold leg	2.14
Jacketed configuration:	

Parallel Configuration

The thermal expansion characteristics of the hot leg in the parallel configuration were analyzed using PipeStress (DST Computer Services 2004) software. The hot leg was assumed to be anchored at both ends. Calculations were performed for two materials; Inconel 617 and carbon steel (SA-106 Grade B). The operating temperature was assumed to be either 355 °C, which represents a case in which the internal insulation is effective in reducing the temperature of the metal, or 850 °C, which represents a case without internal insulation. This pipeline was modeled in either an expansion loop or semi-circular configuration (See Figure 14). The lengths of the expansion loops were adjusted so the maximum calculated stress is about equal to the maximum allowable stress.



(a) semi-circular configuration (b) expansion loop

Figure 14 Pipe line configurations.

Four cases were evaluated using different materials, configurations, and operating temperatures. Results are summarized in Table 5.

Table 5. Results of the thermal expansion calculations

Case	Material	Configuration	Temperature (°C)	Pipe Length (m)	Max. Code Stress Ratio	Max. Deformation (mm)
1	Inconel	Semi-circular	355	141.4	0.188	386.7
2	Inconel	Exp. loop	355	114	1.000	131.1
3	Carbon steel	Semi-circular	355	141.4	0.309	391.5
4	Carbon steel	Exp. loop	355	122	0.953	129.2

Inconel 617 has a much higher allowable temperature range than carbon steel. Consequently, the high temperature cases were not performed for carbon steel. The results with carbon steel and Inconel 617 are reasonably comparable for the lower temperature cases. Because of its lower cost and similar performance, carbon steel is preferred over Inconel 617 for the IHTL if the internal insulation can keep the metal operating temperatures sufficiently low.

The lengths of the IHTL legs must be increased over the required separation distance between the nuclear and hydrogen plants to accommodate the stresses associated with thermal expansion. For the cases with carbon steel and an operating temperature of 355 °C, the increase in length varies from 35 to 60%.

The semi-circular configuration has a lower maximum stress than the expansion loop configuration.

Concentric Configuration

The concentric pipe configuration was modeled with the hot and cold leg piping lines overlaid (coincident nodes). The relative displacement between the two legs was limited to the free space available in the cold leg (calculated as half the difference between the inner diameter of the cold leg and the outer diameter of the hot leg, 78.5 mm). With the only load considered for this model being thermal expansion, no spacers were needed between the two legs to keep the pipes from colliding (the calculated relative displacement between the two pipes was less than the allowable value).

Figure 15 shows the deformation and the maximum stress locations. In Figure 15, the purple line shows the exaggerated displacement of the hot and cold legs (distortion factor of 32) and the black line shows the undeformed pipes.

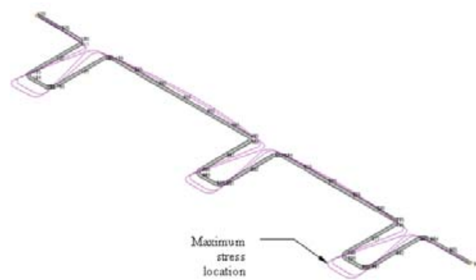


Figure 15. Deformation plot for concentric configuration with Inconel and maximum stress location.

Jacketed Configuration

The piping lines in the jacketed configuration were modeled similarly to the concentric configuration with the two lines overlaid. Similar to the concentric configuration, the results for the two cases modeled below also showed that no spacers were needed between the hot leg and jacket to limit their relative displacement.

Since it was noted that geometry and operating conditions of the legs can be assumed to be the same as in the parallel configuration, the cold leg geometry was not modeled since the results would be the same for this case as those found for the parallel configuration (if a parallel arrangement is desired, the cold leg can use the same layout shown below for the jacketed hot leg since this latter leg has the less conservative parameters). The hot leg behaves differently in this configuration (compared to the parallel configuration) due to the presence of the jacket (this larger pipe needed larger expansion loops in order to meet the stress criteria).

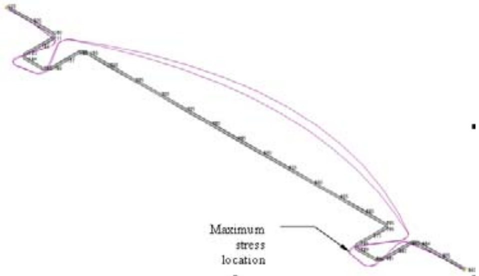


Figure 16. Deformation plot for jacketed configuration with Inconel and maximum stress location.

Table 6 summarizes the results from one case with the concentric configuration and two cases with the jacketed configuration considered in this study.

Table 6. Summary of results from concentric configuration and jacketed configuration.

Configuration	Case	Material	Pipe Length (m)	Max. Code Stress Ratio	Max. Deformation (mm)
Concentric	1, Exp. Loop	Inconel	162	0.895	106.7
Jacketed	1, Exp. Loop	Inconel	118	0.854	325.0
Jacketed	2, Exp. Loop	Carbon Steel	130	0.971	226.8

In summary, the carbon steel and the Inconel are reasonably comparable for the operating temperature and loads (pressure and thermal expansion only). Inconel offers some advantages over carbon steel in that it has better strength at higher temperatures and better corrosion resistance. The semi-circular configuration typically has the lower stresses than the expansion loop, but requires more pipe length. Fabrication and installation costs may be higher for the semi-circular compared to the expansion loop, but it is more efficient at higher temperatures.

The concentric configuration requires the greatest amount of pipe due to the higher operating temperature of the hot leg metal. The carbon steel and the Inconel are reasonably comparable for the jacketed configuration. The concentric configuration also has disadvantages in terms of installation and inspection.

This pipe stress analysis did not account for any dynamic loads such as earthquake or wind loads, deadweight or supports. Furthermore, spacers may need to be added when the analysis accounts for these additional loads in order to prevent the pipe walls from colliding. All of these factors will need to be considered in the final design.

SUMMARY AND CONCLUSIONS

Seven possible configurations for a system that transfers heat between the nuclear reactor and the hydrogen and/or electrical generation plants were identified. These configurations included both direct and indirect cycles for the production of electricity. Both helium and liquid salts were considered as the working fluid in the intermediate heat transport loop.

The addition of a tertiary loop will provide more separation between the nuclear and hydrogen plants, which should make both plants easier to operate and license. However, the addition of a tertiary loop also increases the overall cost of the plant.

The use of a liquid salt as a coolant results in several advantages compared to helium. First, the liquid salt allows the efficient transport of heat at relatively low pressure, which mitigates the problems associated with creep rupture. Second, the use of a liquid salt reduces the total metal volume of the intermediate heat transport loop by 60 to 70% compared with low-pressure helium. Third, the thickness of the insulation required to obtain a given heat loss is reduced by about 80%. All of these factors have the potential to reduce the cost of the NGNP.

The use of a liquid salt also results in several disadvantages compared to helium. First, the material compatibility issues associated with the liquid salts are not as well known as they are for helium. Second, the use of a liquid salt requires additional components that are not required with helium coolant. These additional components, which include an auxiliary heating system and a surge tank to accommodate the contraction and expansion of the liquid salt during transients, will tend to increase the capital cost of the system. Third, a liquid salt has the potential to freeze during transients.

Engineering analyses were performed for several configurations of the intermediate heat transport loop that transfers heat from the nuclear reactor to the hydrogen production plant. The analyses evaluated parallel and concentric piping arrangements and two different working fluids, including helium and a liquid salt.

The differences between the parallel, jacketed, and concentric configurations were relatively small. The parallel configuration was the cheapest, with the jacketed and concentric configurations 20 to 40% more expensive. The carbon steel and the Inconel are reasonably comparable for the operating temperature and loads (pressure and thermal expansion only) we investigated. Inconel offers some advantages over carbon steel in that it has better strength at higher temperatures and better corrosion resistance. The semi-circular configuration typically has the lower stresses than the expansion loop, but requires more pipe length. Fabrication and installation costs may be higher for the semi-circular compared to the expansion loop, but it is more efficient at higher temperatures.

The concentric configuration requires the greatest amount of pipe due to the higher operating temperature of the hot leg metal. The carbon steel and the Inconel are reasonably comparable for the jacketed configuration. The concentric configuration also has disadvantages in terms of installation and inspection.

The mechanical design of the IHTL piping appears feasible if the internal insulation can keep the temperatures sufficiently low. The piping lengths must be increased by 35 – 60% to accommodate the stresses associated with thermal expansion.

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