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# High Temperature Thermal Devices for Nuclear Process Heat Transfer Applications

Piyush Sabharwall and Eung Soo Kim  
*Idaho National Laboratory  
United States of America*

## 1. Introduction

Thermal device is a broad term used to describe a component whose main objective is to transport thermal energy across a system. The two thermal devices discussed in this chapter are heat exchangers and thermosyphons.

### 1.1 Heat exchangers

A heat exchanger is a component used to transfer heat from one medium to another. The media may be separated by a solid wall, so they never mix, or they may be in direct contact (Kakac and Liu 2002). Heat exchangers generally have no external heat nor work interactions and are typically used in the following applications:

- Space heating
- Refrigeration
- Air conditioning
- Power plants
- Chemical plants
- Petrochemical plants
- Petroleum refineries
- Natural gas processing.

#### 1.1.1 High temperature heat exchangers

The high temperature heat exchanger technology is emerging in many industrial applications such as gas turbines, chemical plants, and nuclear power plants. For this reason, many researchers have been studying the development of heat exchangers that can operate at high temperatures. Unlike typical heat exchangers, high temperature heat exchangers require and exhibit some special characteristics to be operated in severe environments (Sunden 2005):

- The transfer of radiative heat may have a significant role in the heat transfer process.
- Tube diameters and pitch should be larger for lower pressure drop. Adding a fan or blower to work at high temperatures might be prohibitive.

- Fins are generally not used in high temperature units because of suspended dirt particles that will foul and low available pressure drops. The advantage of fins is negligible.
- Ceramic and high-temperature alloys (such as: Alloy 800H, 617, 230) are used for construction materials.
- The thickness and mechanical design of selected materials are mainly governed by thermal stress, but the extent of the materials oxidation, thermal shock bearing capability, and erosion from suspended dirt particles, fouling, and corrosion because of metallic salts, sulfates, etc., also need to be considered.
- Differential expansion is an important factor in high temperature units and should be accounted for by using either expansion bellows or by using bayonet-type units. Floating tube sheets cannot generally be used, because sealing gaskets or packing materials do not work effectively at such high temperatures.
- Heat losses from the outside surface to the environment have to be considered in the mechanical design of the unit and design of the foundation.
- Gases, air, liquid metals, or molten salts are preferred over steam for high temperature heat transfer, because the latter require a very thick shell and tubes to contain its high pressure.

Therefore, the thermal stress during startup, shutdown, and load fluctuations can be significant for high temperature heat exchangers. The heat exchanger must be designed accordingly for reliability and long life. The thermal capacitance should therefore be reduced for high temperature heat exchangers for shorter startup time. High temperature heat exchangers also require costly materials contributing to the high cost of balance of power plant. Heat exchanger costs increase significantly for temperatures above 675°C.

### 1.1.2 Heat exchanger types and classifications

A variety of heat exchanger types with various features are used in industry. This subsection generally explains how to classify and categorize them. According to Kakac and Liu (2002), heat exchangers can be generally classified as follows:

1. Recuperator/Regenerator
  - a. Recuperations
  - b. Regenerations
2. Transfer Process
  - a. Direct contact
  - b. Indirect contact
3. Geometry of Construction
  - a. Tubular heat exchanger
    - i. Double pipe heat exchanger
      - High pressure (in both sides)
    - ii. Shell and Tube heat exchanger
    - iii. Spiral tube type heat exchanger
  - b. Plate heat exchanger
    - i. Gasketed plate heat exchanger
    - ii. Spiral plate heat exchanger
  - c. Extended surface heat exchanger
    - i. Plate-fin heat exchanger

- ii. Tubular-fin heat exchanger (Gas to Liquid)
- 4. Heat Transfer Mechanism
  - a. Single phase convection on both sides
  - b. Single phase convection on one side, two phase convection on other side
  - c. Two phase convection on both sides
- 5. Flow Arrangement
  - a. Parallel flow
  - b. Counter flow
  - c. Cross flow

Table 1 shows the principle features for several types of heat exchangers (Shah and Sekulic 2003). According to this table, shell-and-tube, Bavex (plate heat exchanger), printed-circuit, and Marbond are available for high temperature applications above 700°C.

The **shell and tube heat exchanger** is the most common type found in industry. This exchanger is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes and the other fluid flows across and along the tubes. The major components of this exchanger are tubes (or tube bundles), shell, front-end head, rear-end head, baffles, and tube sheets (Shah and Sekulic 2003). The diameter of the outer shell in a shell and tube heat exchanger is greatly increased, and a bank of tubes rather than a single central tube is used, as shown in Figure 1 (Sherman and Chen 2008). Fluid is distributed to the tubes through a manifold and tube sheet. To increase heat transfer efficiency, further modifications to the flow paths of the outer and inner fluids can be accomplished by adding baffles to the shell to increase fluid contact with the tubes, and by creating multiple flow paths or passes for the fluid flowing through the tubes (Sherman and Chen 2008). These heat exchangers are used for gas-liquid heat transfer applications, primarily when the operating temperature and/or pressure is very high (Shah and Sekulic 2003).

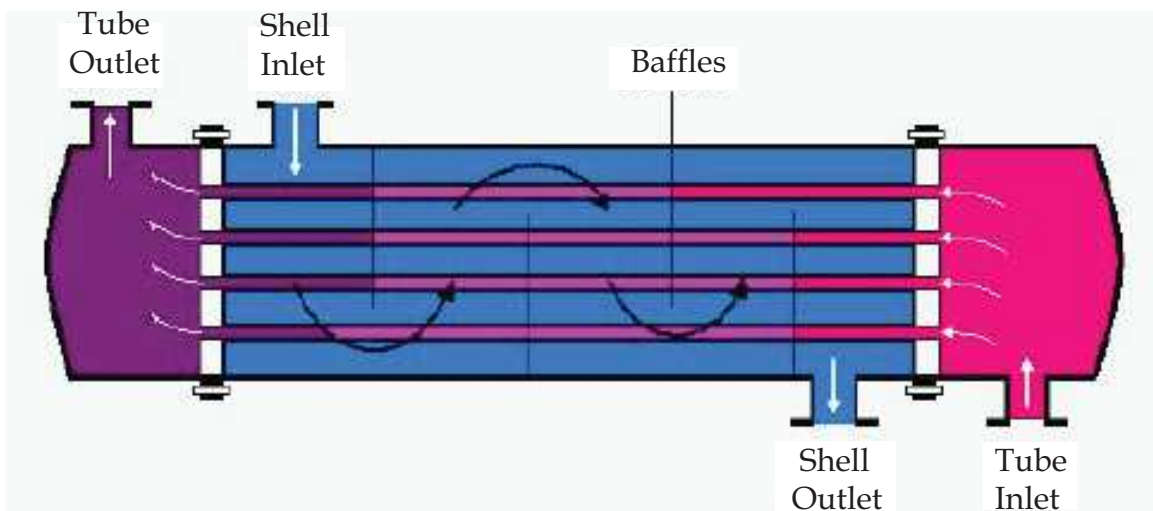


Fig. 1. Shell and tube heat exchanger with baffles (Sherman and Chen 2008)

The **Bevax hybrid welded-plate heat exchanger** is a plate type heat exchanger that deploys metal plates arranged in a stack-wise fashion and sealed with welds as shown in Figure 2. This heat exchanger is reported to be operational at 900°C with pressures to 6 MPa on the

Table 1. Principal features of several types of heat exchangers (Shah and Sekulic 2003)

HX Type	Compactness (m <sup>2</sup> /m <sup>3</sup> )	System Types	Material	Temperature Range (°C) <sup>a</sup>	Maximum Pressure (bar) <sup>b</sup>	Cleaning Method
Shell and Tube	~100	Liquid/Liquid, Gas/Liquid, 2Phase	Stainless steel (s/s), Ti, Inoloy, Hastelloy, graphite, polymer	~ +900	~ 300	Mechanical Chemical
Plate-and-frame (gaskets)	~200	Liquid/Liquid, Gas/Liquid, 2Phase	s/s, Ti, Inoloy, Hastelloy, graphite, polymer	-35 ~ +200	25	Mechanical
Partially welded plate	~200	Liquid/Liquid, Gas/Liquid, 2Phase	s/s, Ti, Inoloy, Hastelloy	-35 ~ +200	25	Mechanical Chemical
Fully welded plate (Alfa Rex)	~200	Liquid/Liquid, Gas/Liquid, 2Phase	s/s, Ti, Ni alloys	-50 ~ + 350	40	Chemical
Brazed plate	~200	Liquid/Liquid, 2Phase	s/s	-195 ~ +220	30	Chemical
Bavex plate	200 ~ 300	Gas/Gas, Liquid/Liquid, 2Phase	s/s, Ni, Cu, Ti, special steels	-200 ~ +900	60	Mechanical Chemical
Platular plate	200	Gas/Gas, Liquid/Liquid, 2Phase	s/s, Hastelloy, Ni alloys	~700	40	Mechanical
Packinox plate	~300	Gas/Gas, Liquid/Liquid, 2Phase	s/s, Ti, Hastelloy, Inconel	-200 ~ +700	300	Mechanical
Spiral	~200	Liquid/Liquid, 2Phase	s/s, Ti, Incoloy, Hastelloy	~400	25	Mechanical
Brazed plate fin	800 ~ 1500	Gas/Gas, Liquid/Liquid, 2Phase	Al, s/s, Ni alloy	~650	90	Chemical
Diffusion bonded plate fin	700 ~ 800	Gas/Gas, Liquid/Liquid, 2Phase	Ti, s/s	~500	>200	Chemical
Printed circuit	200 ~ 5000	Gas/Gas, Liquid/Liquid, 2Phase	Ti, s/s	-200 ~ +900	> 400	Chemical
Polymer (e.g. channel plate)	450	Gas/Liquid	PVDF, PP	~150	6	Water Wash
Plate and shell	–	Liquid/Liquid	s/s, Ti	~350	70	Mechanical Chemical
Marbond	~10,000	Gas/Gas, Liquid/Liquid, 2Phase	s/s, Ni, Ni alloys, Ti	-200 ~ +900	>400	Chemical

- Heat exchanger operational temperature ranges.
- Heat exchanger maximum applicable pressure.
- Capability to connect several independent flow loops in a single heat exchanger.
- Capability to split flow into several paths in the heat exchanger.

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Sindelar 2008). It is reminiscent of a shell and tube arrangement with substantially greater surface area. Plates can be produced up to 0.35 m wide and 16 m long (Fisher and Sindelar 2008). Other variants of the welded plate-type heat exchanger are produced, some of which do not require external shells.

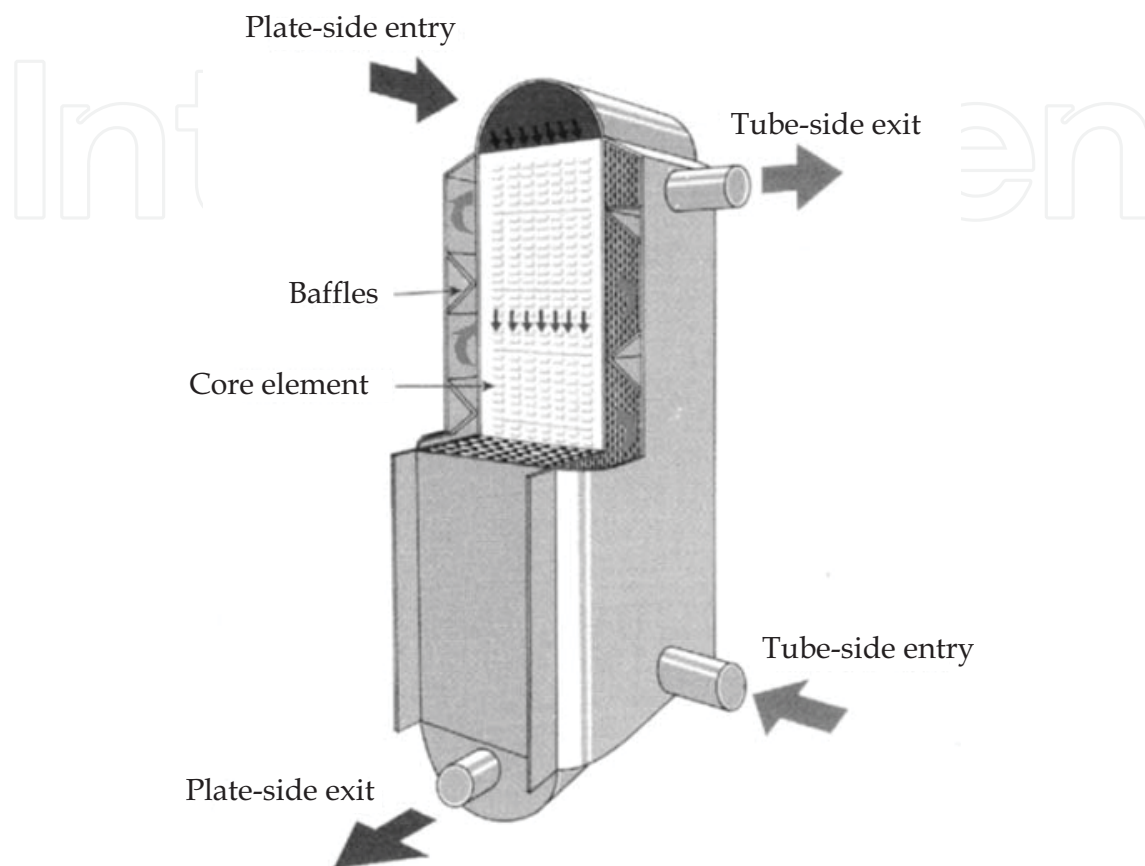


Fig. 2. Bevac welded-plate heat exchanger (Reay 1999)

The **printed circuit heat exchanger (PCHE)** is a relatively new concept that has only been commercially manufactured by Heatric™ since 1985. PCHEs are robust heat exchangers that combine compactness, low pressure drop, high effectiveness, and the ability to operate with a very large pressure differential between hot and cold sides (Heatric™ Homepage 2011). These heat exchangers are especially well suited where compactness is important. The Heatric™ heat exchanger falls within the category of compact heat exchangers because of its high surface area density ( $2,500 \text{ m}^2/\text{m}^3$ ) (Hesselgreaves 2001). As the name implies, PCHEs are manufactured by the same technique used for producing standard printed circuit boards for electronic equipment. In the first step of the manufacturing process, the fluid passages are photochemically etched into the metal plate (See Figure 3). Normally, only one side of each plate is etched-out. The etched-out plates are thereafter joined by diffusion bonding, which is the second step and results in extremely strong all-metal heat exchanger cores. Plates for primary and secondary fluids are stacked alternately and formed into a module. Modules may be used individually or joined with others to achieve the needed energy transfer capacity between fluids. The diffusion bonding process allows grain growth, thereby essentially eliminating the interface at the joints, which in turn gives the parental metal strength. Because of the use of diffusion bonding, the expected lifetime of the heat

exchanger exceeds that of heat exchangers that are based on a brazed structure (Dewson and Thonon 2003).

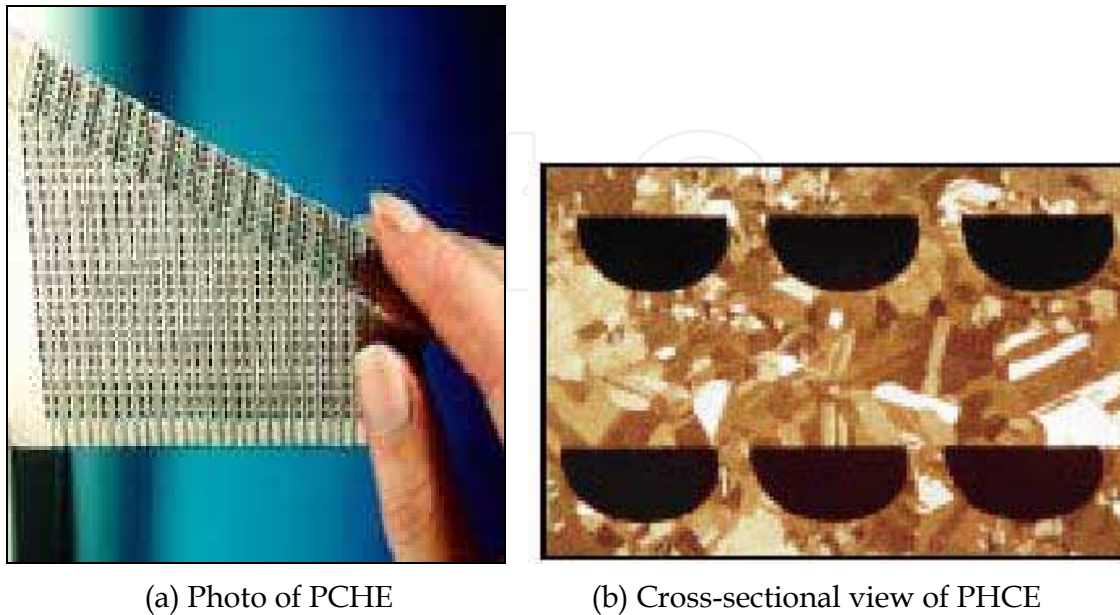


Fig. 3. Printed circuit heat exchanger (Heatric™ Homepage 2011)

The **Marbond heat exchanger** is a type of compact heat exchanger based on a novel combination of photochemical etching and diffusion bonding (Phillips 1996). The internal construction of this heat exchanger comprises a stack of plates that are etched photochemically to form a series of slots as shown in Figure 4. The plates are stacked with high positional tolerance such that series of slots form discrete flow paths. Adjacent flow paths are separated by means of intervening solid plates. Thus, two or more separate flow paths may be formed across a group of plates, enabling different fluid streams. Injecting a secondary reactant into the flow of the primary reactant may be achieved by means of perforations in the solid separator plate that are aligned exactly with the flow paths of the primary reactant. The use of a positive pressure differential between the secondary and primary reactant streams ensures that the secondary reactants flow in the desired direction.

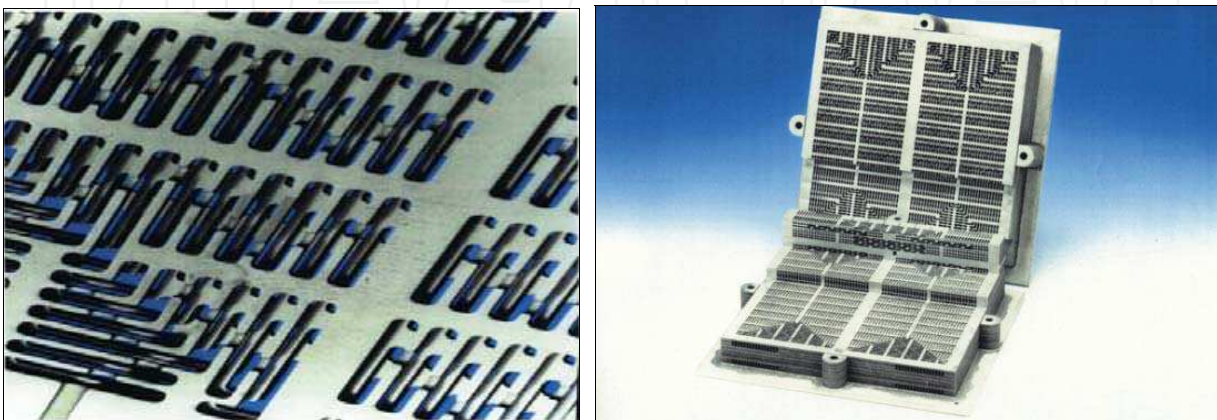


Fig. 4. Marbond heat exchanger (Phillips 1996)

### 1.1.3 Heat exchanger fluid types and comparisons

A variety of heat transfer fluids are available for high temperature heat exchangers including gases, liquid metals, molten salts, etc. The following lists some general characteristics required for the heat transfer fluid:

- High heat transfer performance to achieve high efficiency and economics
- Low pumping power to improve economics through less stringent pump requirements
- Low coolant volume for better economics
- Low structural materials volume for better economics
- Low heat loss for higher efficiency
- Low temperature drop for higher efficiency.

Characteristics of heat transfer fluids have been extensively investigated by Kim, Sabharwall, and Anderson (2011) for high temperature applications based on the following Figures-of-Merit (FOMs):

- $FOM_{ht}$  represents the heat transfer performance of the coolant. It measures the heat transfer rate per unit pumping power for a given geometry.
- $FOM_p$  represents the pumping power of the coolant. It measures the pumping power required to transport the same energy for a given geometry.
- $FOM_{cv}$  represents the volume of the coolant. It measures the coolant volume required for transferring heat with the same heat and pumping power.
- $FOM_{ccv}$  represents the volume of the structural materials. It measures the volume of the coolant structural materials required for transferring heat with the same heat duty and pumping power under given operating conditions (T and P).
- $FOM_{hl}$  represents the heat loss of the coolant. It measures the heat loss of the coolant when it is transported the same distance with the same heat duty and pumping power.
- $FOM_{dt}$  represents the temperature drop in the coolant while transferring thermal energy with a given heat duty and pumping power.

Table 2 shows the comparisons of the thermal-hydraulic characteristics of the various coolants based on the estimated FOMs (Kim, Sabharwall, and Anderson 2011). In this estimation, the water at 25°C and 0.1 MPa was selected to be the reference coolant. The following summarizes the results:

- Higher  $FOM_{ht}$  is preferred for better heat transfer performance. According to the comparisons, sodium shows the highest value (=19.05) and argon has the lowest value (0.05). Overall,  $FOM_{ht}$  is the highest in liquid metal followed by liquid water, molten salt, and gases, respectively.
- Lower  $FOM_p$  is preferred for better efficiency and economics. According to the comparisons, liquid water has the lowest value (=1.0) and argon has the highest value (=72592). Overall,  $FOM_p$  is the lowest in molten salt followed by liquid metals and gases, respectively.
- Lower  $FOM_{cv}$  is preferred because it requires less coolant volume for providing the same amount of heat transfer performance under the same pumping power. According to the comparisons, the liquid water has the lowest value (=1.0) and argon has the highest (=101.44). Overall,  $FOM_{cv}$  is the lowest in molten salt followed by liquid metals and gases, respectively.
- Lower  $FOM_{ccv}$  is preferred because it requires less structural material volume for both heat transfer pipes and components. Overall, the same result was obtained as the  $FOM_{cv}$ . The  $FOM_{ccv}$  is the lowest in molten salt followed by liquid metals and gases, respectively.



	Coolant	FOM <sub>ht</sub>	FOM <sub>p</sub>	FOM <sub>cv</sub>	FOM <sub>ccv</sub>	FOM <sub>hl</sub>	FOM <sub>dl</sub>
Ref.	Water (25°C, 1 atm)*	1.00	1.00	1.00	1.00	1.00	1.00
Gas (700°C, 7 MPa)	He	0.12	25407.41	67.74	4741.80	0.40	0.40
	Air	0.07	40096.15	80.10	5607.14	0.26	0.26
	CO <sub>2</sub>	0.11	11390.17	47.19	3303.46	0.32	0.32
	H <sub>2</sub> O (Steam)	0.11	10012.63	45.10	3157.12	0.32	0.32
	Ar	0.05	72592.09	101.44	7100.53	0.20	0.20
Molten Salt (700°C)	LiF-NaF-KF	0.80	2.87	1.57	1.57	0.92	0.92
	NaF-ZrF <sub>4</sub>	0.45	5.02	1.98	1.98	0.56	0.56
	KF-ZrF <sub>4</sub>	0.38	8.69	2.49	2.49	0.51	0.51
	LiF-NaF-ZrF <sub>4</sub>	0.40	5.36	2.05	2.05	0.50	0.50
	LiCl-KCl	0.55	14.99	3.07	3.07	0.76	0.76
	LiCl-RbCl	0.47	23.03	3.66	3.66	0.70	0.70
	NaCl-MgCl <sub>2</sub>	0.58	16.26	3.18	3.18	0.81	0.81
	KCl-MgCl <sub>2</sub>	0.50	14.30	3.02	3.02	0.70	0.70
	NaF-NaBF <sub>4</sub>	0.71	5.66	2.04	2.04	0.88	0.88
	KF-KBF <sub>4</sub>	0.64	8.98	2.47	2.47	0.84	0.84
	RbF-RbF <sub>4</sub>	0.54	14.61	3.01	3.01	0.75	0.75
Liquid Metal (700°C)	Sodium	19.05	33.62	4.19	4.19	28.91	28.91
	Lead	6.05	111.64	6.90	6.90	10.82	10.82
	Bismuth	6.61	100.69	6.60	6.60	11.66	11.66
	Lead-Bismuth	4.86	142.94	7.65	7.65	8.95	8.95

Table 2. Principal features of several types of heat exchangers (Shah and Sekulic 2003)

Fluid	h [W/m <sup>2</sup> K]
Gases (natural convection)	3–25
Engine Oil (natural convection)	30–60
Flowing liquids (nonmetal)	100–10,000
Flowing liquid metal	5000–25,000
Boiling heat transfer:	
Water, pressure < 5 bars, dT < 25K	5000–10,000
Water, pressure 5–100, dT = 20K	4000–15,000
Film boiling	300–400
Condensing heat transfer:	
Film condensation on horizontal tubes	9000–25,000
Film condensation on vertical surface	4000–11,000
Dropwise condensation	60,000–120,000

Table 3. Order of magnitude of heat transfer coefficient (Kakac and Liu 2002)

- Lower FOM<sub>hl</sub> is preferred because it requires less insulation for preventing heat loss. According to the comparisons, argon has the lowest value (0.2), and sodium has the highest (28.9). Overall, the FOM<sub>hl</sub> is the lowest in gases followed by molten salt and liquid metal, respectively.

- Lower FOMdt is preferred because more thermal energy can be transferred long distances without much of a temperature drop. Same values were obtained for the FOMdt as obtained from FOMhl.

In the heat exchanger design, the heat transfer coefficient is a very important parameter because it determines overall heat exchanger sizes and performance. Table 3 lists some coolant types and the ranges of their heat transfer coefficients (Kakac and Liu 2002). As can be seen, water exhibits the highest heat transfer coefficient in the drop-wise condensation, and gases exhibit the lowest in the natural circulation.

#### 1.1.4 Heat exchanger materials and comparisons

Material selection is one of the most important things in the high temperature application. There are four main categories of high temperature materials: high temperature nickel-based alloy, high temperature ferritic steels and advanced carbon silicon carbide composite, and ceramics (Sunden 2005).

Ohadi and Buckley (2001) extensively reviewed materials for the high temperature applications. High temperature nickel-based material has good potential for helium and molten salts up to 750°C. High temperature ferrite steels shows good performance under fusion and fission neutron irradiation to around 750°C. Advanced carbon and silicon carbide composite has excellent mechanical strength at temperatures exceeding 1000°C. It is currently used for high temperature rocket nozzles to eliminate the need for nozzle cooling and for thermal protection of the space shuttle nose and wing leading edges. Many options are available that trade fabrication flexibility and cost, neutron irradiation performance, and coolant compatibility. Table 4 compares the properties of most commonly used high temperature materials (Ohadi and Buckley 2001). It includes nickel-based alloy, ceramic materials, and carbon and SiC composites. Figure 5 shows the specific strength versus temperature for various composite materials.

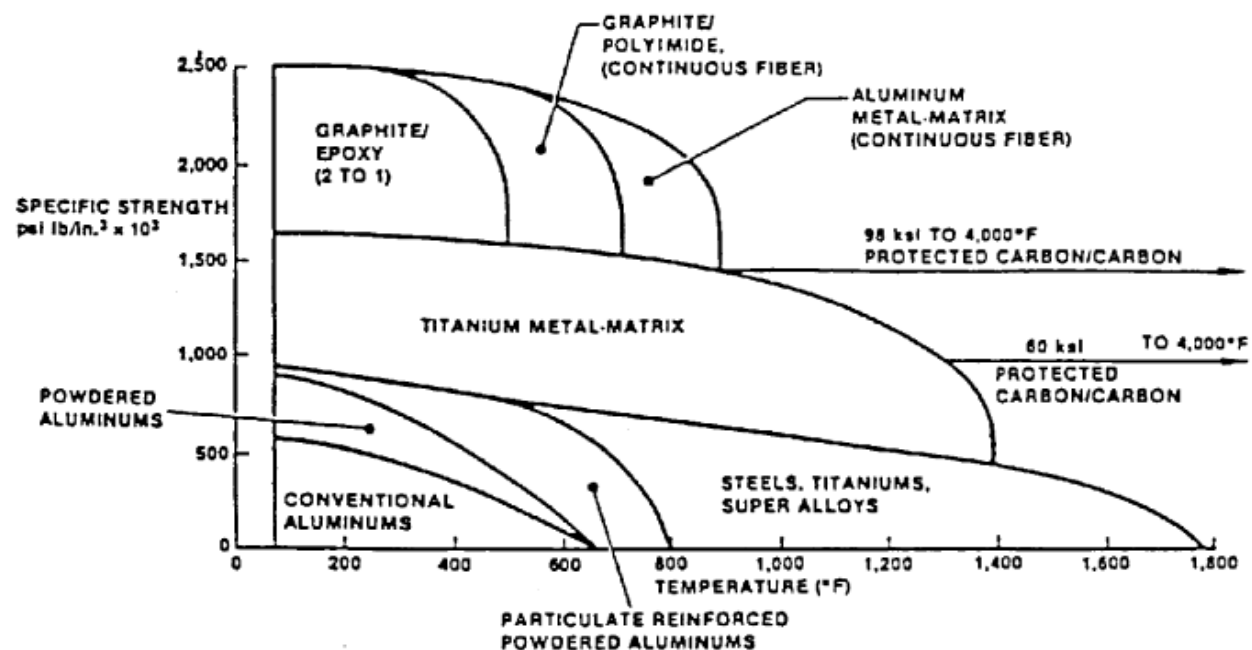


Fig. 5. Specific strength vs. temperature (Brent 1989)

High temp. material/fabrication technology	Metallic Ni alloys (Inconel 718)	Ceramics oxides of Al, Si, Sr, Ti, Y, Be, Zr, B and SiN, AiN, B4C, BN, WC94/C06	Carbon-carbon composite	Carbon fiber-SiC composite
Temperature range	1200 - 1250 °C	1500 - 2500 °C	3300 °C (inert environment) 1400 - 1650 °C (with SiC layer)	1400 - 1650 °C
Density	8.19 g/cm <sup>3</sup>	1.8 - 14.95 g/cm <sup>3</sup>	2.25 g/cm <sup>3</sup>	1.7 - 2.2 g/cm <sup>3</sup>
Hardness	250 - 410 (Brinell)	400 - 3000 kgf/mm <sup>2</sup> (V)	0.5 - 1.0 (Mohs)	2400 - 3500 (V)
Elongation	< 15%	N/A	N/A	-
Tensile strength	800 - 1300 MPa	48 - 2000 MPa	33 (Bulk Mod.)	1400 - 4500 MPa
Tensile modulus	50 GPa	140 - 600 GPa	4.8 GPa	140 - 720 GPa
Strength of HE	Strength - adequate, but limited due to creep and thermal exp	Strength - not adequate, low mechanical parameters for stress. Good thermal and electrical parameters	Strength - poor, oxidation starts at 300 °C	Highest due to carbon fiber and SiC
Electrical conductivity	125 μΩ cm	2E-06 - 1E+18 Ω cm	1275 μΩ cm	1275 μΩ cm
Thermal conductivity	11.2 W/m K	0.05 - 300 W/m K	80 - 240 W/m K	1200 W/m K
Thermal expansion	13E-06 K <sup>-1</sup>	0.54 - 10E-06 K <sup>-1</sup>	0.6 - 4.3E-06 K <sup>-1</sup>	-
Comments	Metallic expansion joints are the weak link	Often very expensive fabrication cost for conventional applications. Technology proprietary for the most part. Technologically hard to produce	Life-time is low even protected by SiC (adhesion is poor)	Comparatively less expensive, successful proprietary fabrication technologies available.

Table 4. Selected properties of most commonly used high-temperature materials and fabrication technologies (Ohadi and Buckley 2001)

Dewson and Li (2005) carried out a material selection study of very high temperature reactor (VHTR) intermediate heat exchangers (IHXs). They selected and compared the following eight candidate materials based on ASME VIII (Boiler and Pressure Vessel Code): Alloy 617, Alloy 556, Alloy 800H, Alloy 880HT, Alloy 330, Alloy 230, Alloy HX, and 253MA.

Alloys	UNS No	Tmax (°C)	S898°C (MPa)	UTS (MPa)	0.2%PS (MPa)	EI (%)	Nominal compositions (wt%)
617	N06617	982	12.4	655	240	30	52Ni-22Cr-13Co-9Mo-1.2Al
556	R30556	898	11.0	690	310	40	21Ni-30Fe-22Cr-18Co-3Mo-3W-0.3Al
800HT	N08811	898	6.3	450	170	30	33Ni-42Fe-21Cr
800H	N08810	898	5.9	450	170	30	33Ni-42Fe-21Cr
330	N08330	898	3.3	483	207	30	Fe-35Ni-19Cr-1.25Si
230	N06230	898	10.3	760	310	40	57Ni-22Cr-14W-2Mo-0.3Al-0.05La
HX	N06002	898	8.3	655	240	35	47Ni-22Cr-9Mo-18Fe
253MA	S30815	898	4.9	600	310	40	Fe-21Cr-11Ni-0.2N

Table 5. Candidate materials for VHTR IHXs (Dewson and Li 2005)

Table 5 lists the allowable design stress ( $S$ ) at  $898^{\circ}\text{C}$ , minimum required mechanical properties (ultimate tensile stress [UTS]), 0.2% proof stress (PS), and elongation (El) at room temperature, together with the nominal compositions of the alloys. They extensively compared the mechanical properties, physical properties, and corrosion resistance for the candidate materials, and finally concluded that Alloy 617 and 230 are the most suitable materials for an IHX.

### 1.1.5 General heat exchanger design methodology

Once the process requirements are given, the high temperature heat exchanger is designed using the same methods used for the typical heat exchangers. This subsection summarizes the basic logical structure of the process heat exchanger design procedure described by Bell (2004).

Figure 6 shows the logical process for heat exchanger design. The fundamental goal of any such process is to find the optimal design among the infinite set of designs that could satisfy the thermal-hydraulic and mechanical requirements. Because of large number of qualitative factors in the design process, optimal needs are to be considered broadly. Generally, the design process aims at the least costly (which usually means the smallest) heat exchangers that meet the required thermal duty within the allowed pressure drops and satisfy the mechanical requirements.

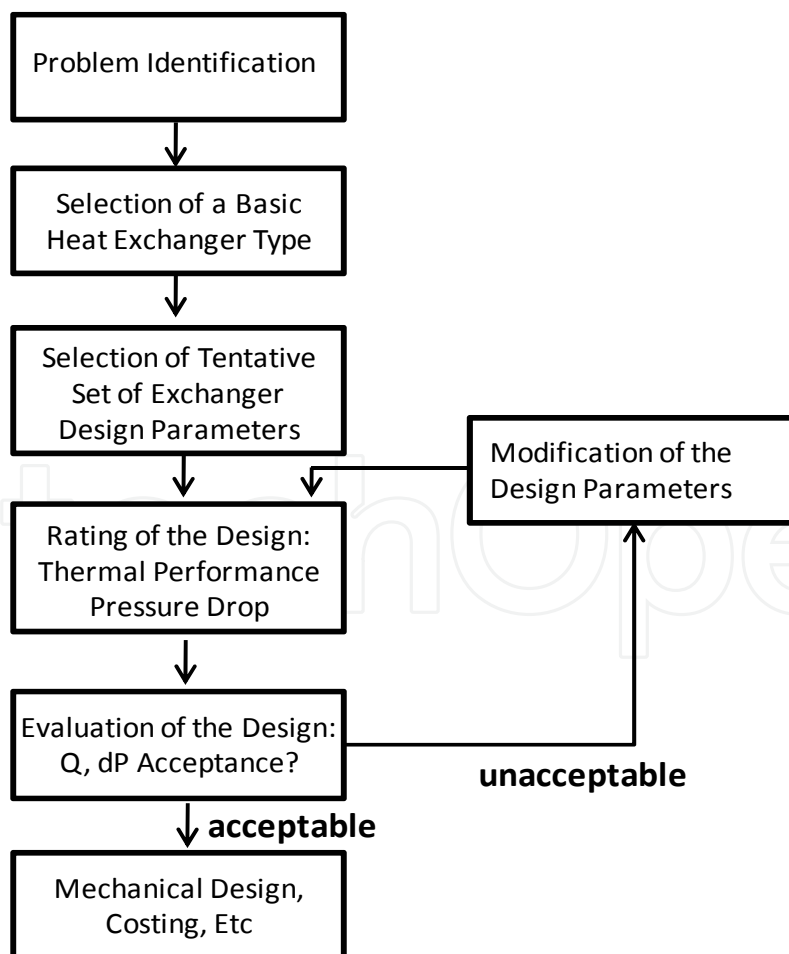


Fig. 6. Basic logical structure of the process heat exchanger design procedure (Bell 2004)

First, a basic heat exchanger type is selected based on the given operational requirements and problems identified. The basic criteria for heat exchanger selection are as follows:

- It must satisfy the process specifications (performance) for temperature and pressure
- It must withstand the service conditions of the plant environment (reliability) for temperature and pressure
- It must be maintainable for cleaning or replacement of a special component
- It should be cost effective (installed operating and maintenance costs)
- It must meet the site requirements or limitations for diameter, length, weight, and tube configurations, and lifting and servicing capability or inventory considerations.

Second, main heat exchanger design parameters are selected based on the followings main design factors:

- Heat duty
- Materials
- Coolant
- Pressure drop
- Pressure level
- Fouling
- Manufacturing techniques
- Cost
- Corrosion control
- Cleaning (with ease).

Third, rating and evaluation of the heat exchanger designs are iteratively conducted to thermally optimize the design. The main purpose of this is to find all possible configurations to meet the process requirements. Generally, the optimum heat exchanger is determined based on the cost.

Once the optimal thermal design is provided, a mechanical design is developed based on the following factors:

- Plate, tube, shell, and header thickness and arrangements
- Corrosion resistance
- Manifold
- Location of pressure and temperature measuring device
- Thermal stress analysis under steady and transient
- Flow vibrations
- Level of velocity to eliminate fouling and erosion
- Maintenance.

Detailed cost analyses are also conducted at this stage including both capital and operational costs. The capital cost includes materials, manufacturing, testing, shipment, and installation. The operational cost includes pumping power, repair, and cleaning.

### 1.1.6 Heat exchanger cost analyses

The cost of a heat exchanger is an important factor for heat exchanger design and selection. Generally, manufacturers have their own methods for cost estimation. This section introduces a simple heat exchanger costing methodology based on empirical cost data of the ESDU (1994) for various feasible heat exchanger types (Shah and Sekulic 2003). Detailed cost will depend on the operating conditions and materials used. The decision variable in this method is cost of a heat exchanger per unit of its thermal size defined by

$$C_{UA} = \frac{\text{cost}}{\frac{q}{\Delta T_m}} \quad (1)$$

Where:

*cost* = cost of the heat exchanger

*q* = thermal duty

$\Delta T_m$  = mean temperature.

Table 6 shows a selection of the cost data represented by the values of  $C_{UA}$  (Shah and Sekulic 2003). This table is prepared for an application between gas as a hot fluid at a medium pressure of about 2 MPa and cold fluid as treated water. An extensive set of  $C_{UA}$  data for various heat exchangers can be found in ESDU (1994).

$q/\Delta T_m$	$C_{UA}$ (\$/(W/K))				Welded Plate	
	Shell-and-Tube U = 484 (W/m <sup>2</sup> K)	Double Tube U = 484 (W/m <sup>2</sup> K)	Printed Circuit U = 1621 (W/m <sup>2</sup> K)	Plate-Fin U = 491 (W/m <sup>2</sup> K)	U (W/m <sup>2</sup> K)	$C_{UA}$ (\$/(W K))
10 <sup>3</sup>	3.98	2.5	12	-	349	4.9
5x10 <sup>3</sup>	1.00	0.75	2.4	3.1	1187	1.22
3x10 <sup>4</sup>	0.29	0.31	0.6	0.513	1068	0.42
10 <sup>5</sup>	0.17	0.31	0.42	0.210	1112	0.28
10 <sup>6</sup>	0.106	0.31	0.28	0.115	1173	0.22

\* Original cost data in ESDU are approximated to the US dollar value in 2000.

Table 6. Cost data  $C_{UA}$  vs.  $UA$  for various heat exchanger types (Shah and Sekulic (2003), ESDU (1994))

## 2. Thermal energy transfer for process heat application

Recent technological developments in next generation nuclear reactors have created renewed interest in nuclear process heat for industrial applications. The Next Generation Nuclear Plant (NGNP) will most likely produce electricity and process heat for hydrogen production. Process heat is not restricted to hydrogen production, but is also envisioned for various other technologies such as the extraction of iron ore, coal gasification, and enhanced oil recovery. To utilize process heat, a thermal device is needed to transfer the thermal energy from NGNP to the hydrogen plant in the most efficient way possible. There are several options to transferring multi-megawatt thermal power over such a distance. One option is simply to produce only electricity, transfer it by wire to the hydrogen plant, and then reconvert the electric energy to heat via Joule or induction heating. Electrical transport, however, suffers energy losses of 60 to 70% because of the thermal-to-electric conversion inherent in the Brayton cycle. A second option is to transport thermal energy via a single-phase forced convection loop where a fluid is mechanically pumped between heat

exchangers at the nuclear and hydrogen plants. High temperatures, however, present unique challenges for materials and pumping. Single phase, low pressure helium is an attractive option for NGNP, but is not suitable for a single-purpose facility dictated to hydrogen production because low pressure helium requires higher pumping power and makes the process very inefficient. A third option is two-phase heat transfer utilizing a high-temperature thermosyphon. Heat transport occurs via evaporation and condensation, and the heat transport fluid is recirculated by gravitational force. Thermosyphons have the ability to transport heat at high rates over appreciable distances, virtually isothermally, and without any requirement for external pumping devices.

Heat pipes and thermosyphons have the ability to transport very large quantities of heat over relatively long distances with small temperature losses. The applications of heat pipes and thermosyphons require heat sources for heating and heat sinks for cooling. The development of the heat pipe and thermosyphon was originally directed towards space applications. However, the recent emphasis on energy conservation has promoted the use of heat pipes and thermosyphons as components in terrestrial heat recovery units and solar energy systems. Thermosyphons have less thermal resistance, wider operating limits (the integrity of the wick material might not hold in heat pipes at very high temperatures), and lower fabrication costs than capillary heat pipes, which makes a thermosyphon a better heat recovery thermal device. Perhaps the most important aspect of thermosyphon technology is that it can easily be turned off when required, whereas a heat pipe cannot be turned off. This safety feature makes the licensing of NGNP process heat transfer systems comparatively easier. This section describes the thermosyphon system and the potential benefits of using it in order to transfer process heat from the nuclear plant to the hydrogen production plant.

## 2.1 Thermosyphon design

Considerable effort has been invested in thermosyphon and heat pipe development, resulting in broad applications. One significant advantage of heat transfer by thermosyphon is the characteristic of nearly isothermal phase change heat transport, which makes the thermosyphon an ideal candidate for applications where the temperature gradient is limited and high delivery temperatures are required, as in the case of thermochemical hydrogen production (Sabharwall and Gunnerson 2009; Sabharwall 2009). The nature of isothermal heat transport results in an extremely high thermal conductance (defined as the heat transfer rate per unit temperature difference). A schematic diagram of a thermosyphon system is shown in Figure 7.

The controllable thermosyphon, conceptually illustrated in Figure 7, is a wickless heat pipe with a separate liquid return line, which is an intriguing option to traditional pumped fluid heat transfer. Thermosyphons rely on convection to transport thermal energy inside pipes and high-temperature heat exchangers for the evaporation and condensation end processes. Ideally, no pumping power is required in contrast to single-phase gas or liquid loops that require compressors or pumps, both of which are problematic at very high temperatures.

Heat is transported by saturated or superheated vapor expanded from an evaporative heat exchanger, through a long pipe, to a condensation heat exchanger. Liquid condensate returns to the evaporator assisted by gravity through a separate liquid return line with a

liquid return control valve. When the thermosyphon is started by applying power (process heat from NGNP) to the evaporator, the working fluid is evaporated and the latent heat of vaporization is transported (~isothermally) along the thermosyphon to the condenser region. Expansion joints are added near the condenser section and at the inlet to the condensate return line in order to accommodate the thermal expansion of the thermosyphon piping at higher temperature. The condensate returns to the evaporator region through a liquid bypass line containing a liquid storage reservoir and a control check valve as shown in Figure 7. The storage reservoir and part of the liquid lines may incorporate electric resistance heating if necessary in order to melt the working fluid and restart the thermosyphon after a long shutdown period. Liquid from the storage reservoir passes into the thermosyphon system evaporator through a control valve which, as needed, plays a role in controlling the rate of heat transfer and shutting off or isolating the thermosyphon. In order for the thermosyphon system to be shut and completely disabled from heat transfer, the control valve is closed wherein all the working fluid is collected in the liquid storage reservoir and the condensing-evaporating cycle is terminated. When it is desired to resume the thermosyphon action, the control valve is opened to again allow the liquid to flow into the evaporator region of the system. The heat input governs the rate of evaporation and the subsequent rate of heat transfer. The rate of thermal energy exchange can be regulated over a spectrum of conditions from “off” to “fully on,” hence the term *controllable thermosyphon*.

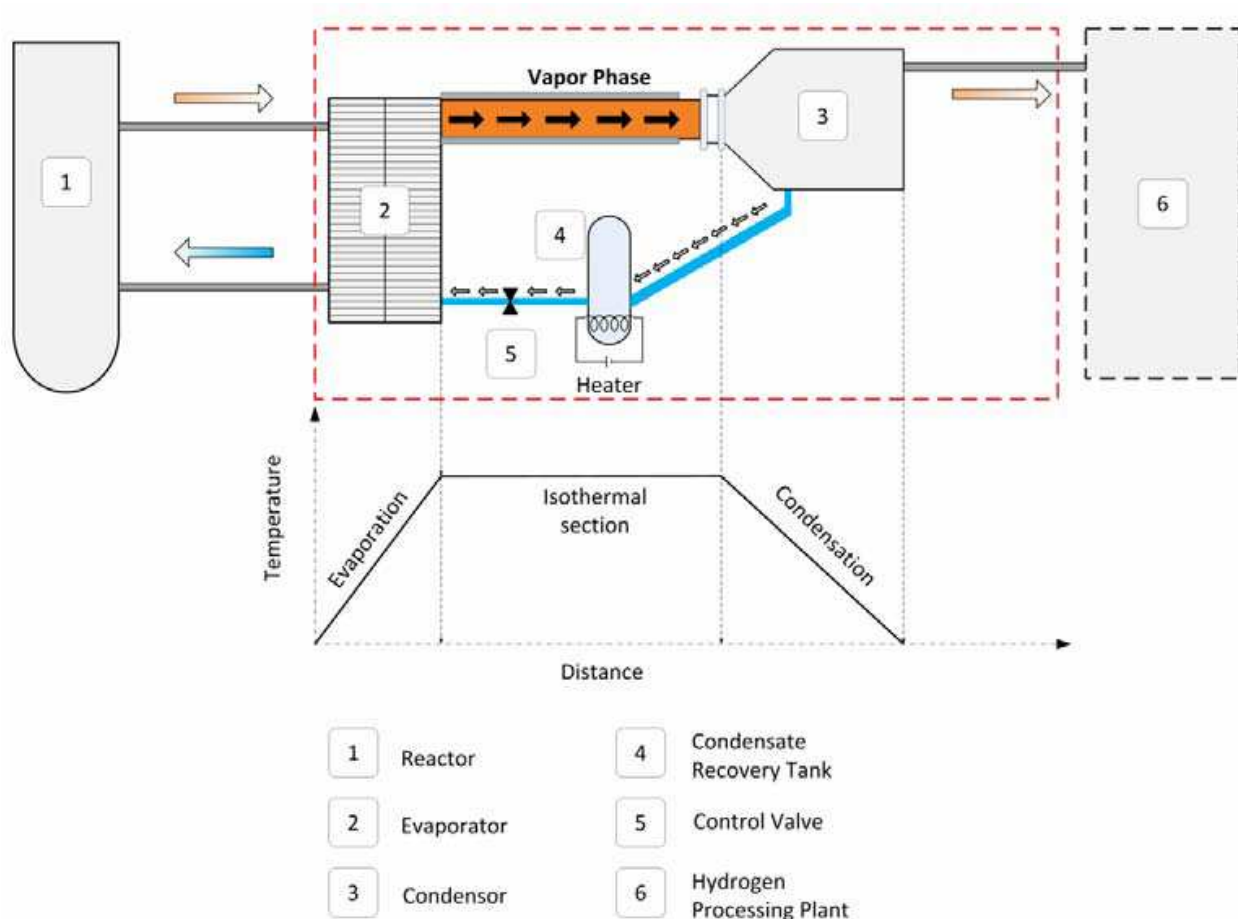


Fig. 7. Schematic of a simple controllable thermosyphon



Depending on the temperature and pressure of operation, favorable working fluids can be identified. Alkaline metals, for example, may be suited for process heat transfer because they have the characteristics of:

- High boiling temperature
- Availability and cost effectiveness
- Good heat transfer properties (latent and specific heat are both high)
- Typically good chemical compatibility (except Li).

Working fluids more suitable than alkali metals may exist, such as molten salts (Sohal et al. 2010). Corrosive behavior at high temperatures or lack of high temperature thermodynamic properties, especially for superheated vapors, rule out fundamental analysis of many possible thermosyphon working fluids.

## 2.2 Thermosyphon startup

The charging of a thermosyphon requires a transfer station wherein molten working fluid under an inert environment or vacuum is transferred to the evacuated thermosyphon pipe(s). Pure fluids without condensable gases are required for proper thermosyphon operation. Otherwise, the impurities, which are more volatile than the fluid itself, will be driven to the condenser section of the thermosyphon and less volatile impurities will be collected in the evaporator causing hot spots and reducing heat transfer. Noncondensable gases will accumulate within the condenser. Although conceptually simple, the startup of a large, high temperature thermosyphon is difficult to accurately predict.

Figure 8 describes the procedure commonly practiced for filling up the thermosyphon. The liquid sodium valve is opened till 20%-by-volume limit is reached for filling up the thermosyphon. The 20%-by-volume is a good approximation for the coolant as described by Gunnerson and Sanderlin (1994).

For a thermosyphon to startup efficiently and effectively, the working fluid has to initially be in molten state. The liquid reservoir, as shown in Figure 8, has provisions for external heating. During normal startup for both heat pipes and thermosyphons, the temperature of the evaporator section increases by a few degrees until the thermal front reaches the end of the condenser as described by Reay and Kew (2006). At this point, the condenser temperature will increase until the pipe structure becomes almost isothermal.

## 2.3 Comparison of thermosyphon with convective loop

Alkaline metal thermosyphons and alkaline metal forced convective loops can both deliver comparable rates of heat transfer through a given size pipe. This can be demonstrated by considering the ideal rate of convective heat transport through a pipe without losses, modeled in terms of enthalpy as:

$$Q'' = \Delta h \rho V = \frac{\dot{m}}{A} \Delta h \quad (2)$$

where:

A -Cross-sectional flow area

$\Delta h$  -Specific enthalpy change of the transport fluid

$\dot{m}$  -Mass flow rate

- $Q''$  -Rate of thermal energy transport per unit flow area  
 $V$  -Average flow velocity  
 $\rho$  -Density of the fluid  
 $\Delta$  -Difference.

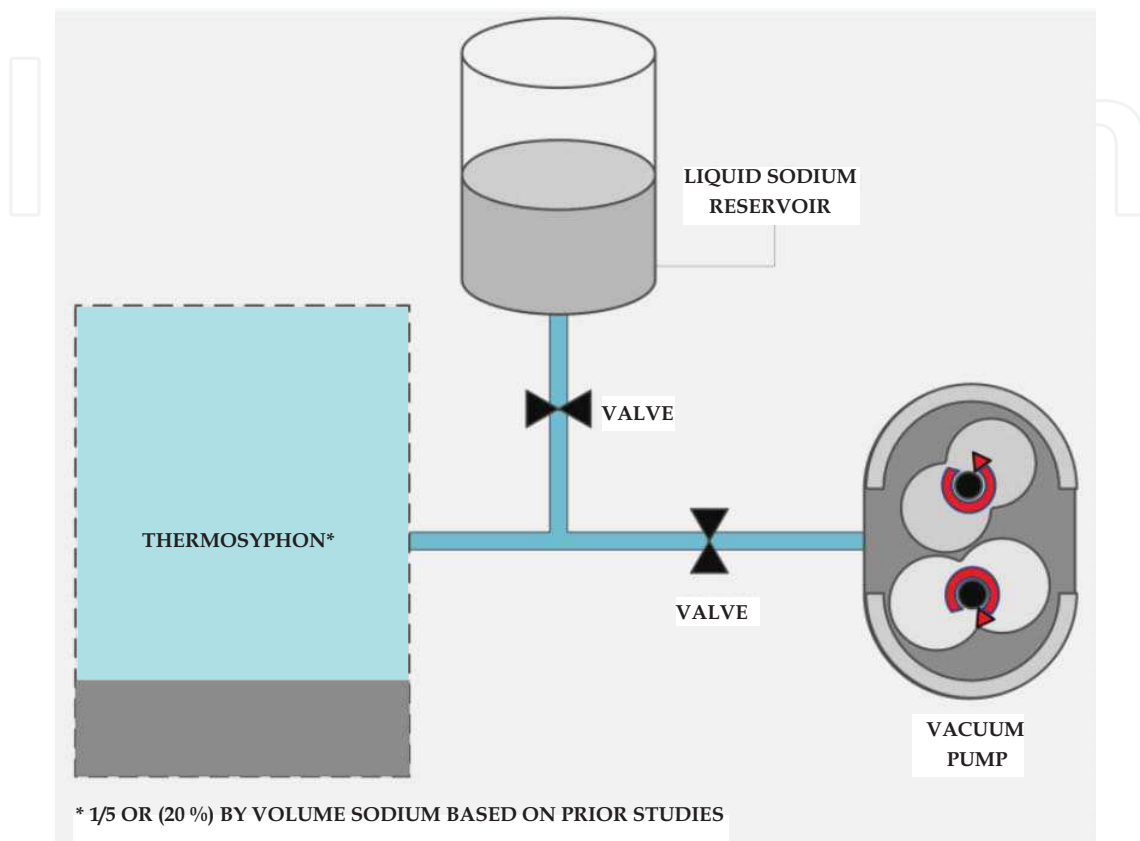


Fig. 8. Na filling procedure for a thermosyphon

Two-phase heat transfer by a thermosyphon has the advantage of high enthalpy transport when compared with single-phase forced convection. Vapor-phase velocities within a thermosyphon can also be much greater than single-phase liquid velocities within a forced convective loop.

Figure 9 exemplifies the enthalpy enhancement in heat transfer afforded by a two-phase thermosyphon versus a single-phase convective loop with sodium as the working fluid as shown by Sabharwall et al. 2009. The specific enthalpy ( $\Delta h$ ) of saturated liquid and vapor, relative to the solid at 298.15 K, is illustrated as a function of temperature. Assuming heat transfer from a high temperature gas-cooled reactor to an industrial facility at 1223 K, the maximum single-phase heat transfer is given by the enthalpy gain from points A to B in Figure 9, or approximately 1,190 kJ for each kilogram of sodium. Compared with two-phase heat transfer from points A to B to C, where the enthalpy gain is approximately 3,864 kJ per kilogram with no vapor superheat, over three times more heat per kilogram of sodium is needed than with the single-phase. The saturation pressure of sodium at 1223 K is only 0.188 MPa, thus minimizing pressure and stress forces. Vapor flow through a pipe is limited by compressible choke flow when the vapor reaches its sonic velocity. The sonic velocity for sodium vapor is approximately 737 m/s at 1223 K as given by Bystrov et al. (1990).

The limiting heat transfer rate for an ideal sodium thermosyphon operating at 1223 K can therefore be estimated as

$$Q'' = h_{fg} \rho_V V_S = (3,864) \text{ kJ/kg} (0.47) \text{ kg/m}^3 (737) \text{ m/s} = 1338 \text{ MW/m}^2$$

where:

$h_{fg}$  -Latent heat & vaporization

Similarly, single-phase liquid sodium could transport the same rate of thermal energy with an average flow velocity of about 2.2 m/s, well within the capabilities of advanced liquid metal pumps. This simple analysis (Sabharwall 2009) for sodium as the working fluid theoretically illustrates that both a thermosyphon and a forced convective loop can deliver comparable rates of heat transfer through comparable diameter pipes. The thermosyphon, however, has the luxury of controllable heat transfer without the need for high temperature pumping and can deliver the heat at the same approximate temperature as the source. The enthalpy gain for sodium that can be achieved by two-phase heat transfer versus a single-phase is about 3.7 times greater (Sabharwall 2009).

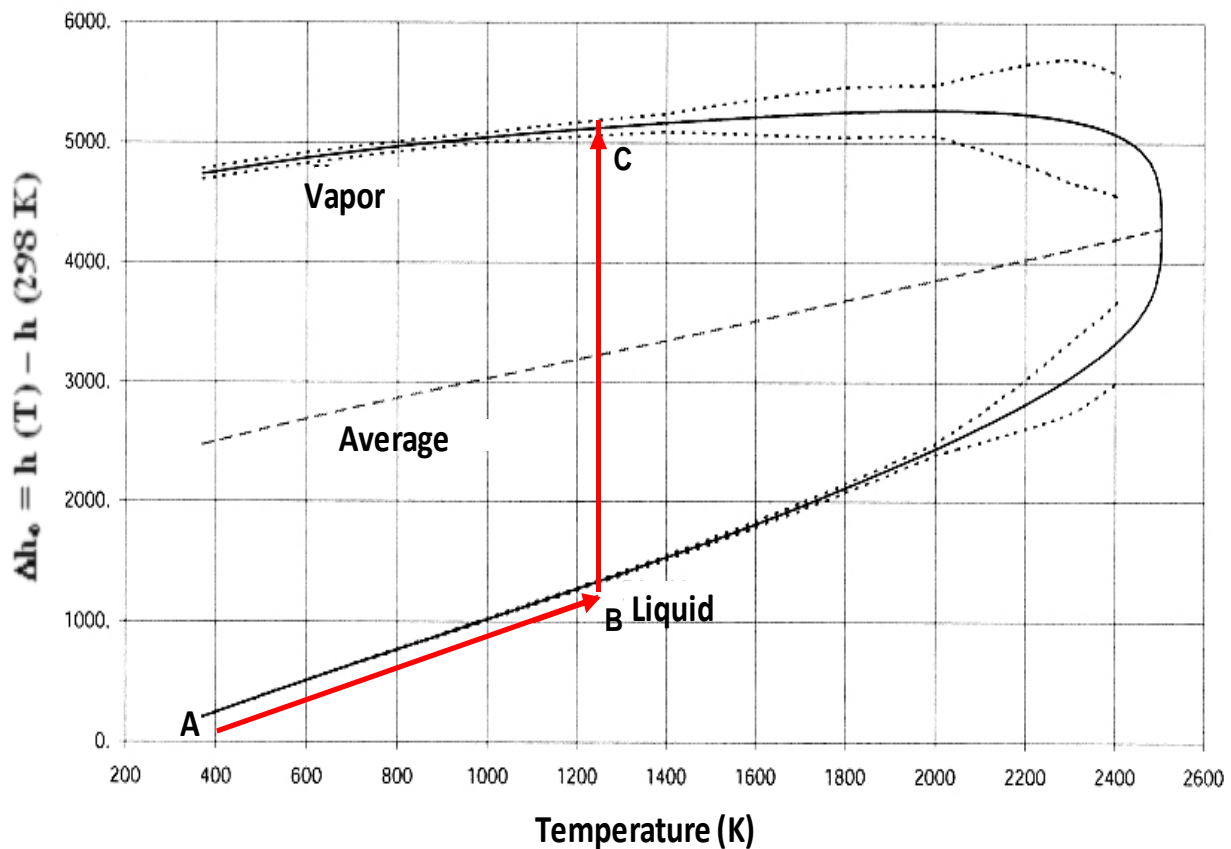


Fig. 9. Enthalpy for saturated sodium: liquid and vapor (thermodynamic data from Fink and Leibowitz [1995]; Gunnerson, Sabharwall, and Sherman [2007])

#### 2.4 Heat transport limitations

Depending on the operational conditions, the heat transport may be limited by one of the following (Sabharwall et al. 2009), as described below:

#### **2.4.1 Sonic limit (choking) of vapor flow**

After continuum flow is established, the evaporator-condenser pressure difference accelerates the vapor until it reaches a maximum velocity at the evaporator exit. The maximum vapor velocity that can exist at the evaporator exit corresponds to sonic velocity, or Mach 1. This choked flow condition is a fundamental limit on the axial vapor flow in a thermosyphon.

#### **2.4.2 Viscous limit**

At startup for liquid metals, the vapor pressure difference between the evaporator and the condenser is zero or very small. In such cases, the viscous forces may be larger than the vapor pressure gradients, thus preventing vapor flow.

### **2.5 Flow instabilities**

Instabilities are common to both forced and natural circulation systems; the latter is more unstable than forced circulation systems. Instabilities can cause operational problems in process heat transfer. Thus, it is important to classify mechanisms that can lead to unstable operational behavior of the thermosyphon. The following processes can lead to an unstable behavior for the thermosyphon.

#### **2.5.1 Surging (chugging) and geysering instability**

Surging and geysering occur mainly because of liquid superheat. Surging occurs when boiling is initiated in the evaporator, but because of nonuniformity in the temperature at the wall and bulk fluid temperature, the vapor being generated becomes trapped, eventually resulting in vapor expulsion as described by Bergles et al. (1981). Geysering is a similar phenomenon that occurs when the heat flux is sufficiently high and boiling is initiated at the bottom. In low pressure systems, this results in a sudden increase in vapor generation because of the reduction in hydrostatic head, usually causing an expulsion of vapor.

#### **2.5.2 Thermosyphon evaporator instability**

If the evaporator section of the thermosyphon system is not sufficiently long for vapor superheat, instability can occur such that the fluid at the outlet of the evaporator experiences a static pressure decrease, leading to the onset of fluid condensation within the thermosyphon. Slight vapor superheat from the evaporator should reduce this concern.

#### **2.5.3 Fluid superheating (alkaline metals)**

Alkaline metals have relatively high boiling temperatures at atmospheric pressure. If the heater surface does not have enough active nucleation sites, boiling may not occur near saturation temperature but rather require significant superheat. At high superheat temperatures a vapor burst expulsion can be expected upon phase change, which could lead to structural damage and flow excursions.

### **3. Summary**

Thermal device is a component whose main objective is to transport thermal energy across a system. In this chapter, two key thermal devices for high temperature heat transfer applications were introduced and discussed in detail; (1) heat exchanger and

(2) thermosyphon. A heat exchanger is a key component in the thermal systems used for transferring heat from one medium to another. Especially, the high temperature heat exchanger technology is emerging in many industrial applications such as gas turbines, chemical plants, and nuclear power plants with increasing demands. Unlike typical heat exchangers, high temperature heat exchangers require and exhibit some special characteristics to be operated in severe environments.

In this chapter, shell-and-tube, Bevax, PCHE, and Marbond heat exchangers are recommended to be the possible option for high temperature applications.

Working fluids and structural materials are the important design parameters that should be carefully considered for high temperature heat exchangers. A variety of heat transfer fluids are available for high temperature heat exchangers including gases, liquid metals, molten salts, etc. General characteristics required for heat transfer coolant are (1) high heat transfer performance, (2) low pumping power, (3) low coolant volume, (4) low heat loss, and (5) low temperature drop. In this chapter, six figures-of-merit (FOMs) for heat transfer coolants were introduced for evaluating important coolant characteristics. Based on these FOMs, various candidate coolants were compared extensively. Material selection is one of the most important things in the high temperature application. There are four main categories of high temperature materials, high temperature nickel-based alloy, high temperature ferritic steels and advanced carbon silicon carbide composite, and ceramics. Basic design methods and cost analysis for the heat exchangers are also briefly mentioned. Another thermal device are heat pipes and thermosyphons, which have the ability to transport very large quantities of heat over relatively long distances with small temperature losses. In this chapter, the thermosyphon system and the potential benefits of using it were described in the heat transfer perspective from the nuclear plant to the hydrogen production plant (also applicable to any other process application industry). One significant advantage of heat transfer by thermosyphon is the characteristic of nearly isothermal heat transport, which makes the thermosyphon an ideal candidate for applications where the temperature gradient is limited and high delivery temperatures are required, as in the case of thermochemical hydrogen production.

Depending on the temperature and pressure of operation, favorable working fluids can be identified. Alkaline metals, for example, may be suited for process heat transfer because of high boiling temperature, availability and cost effectiveness, good heat transfer properties (latent and specific heat are both high), and typically good chemical compatibility (except for Li).

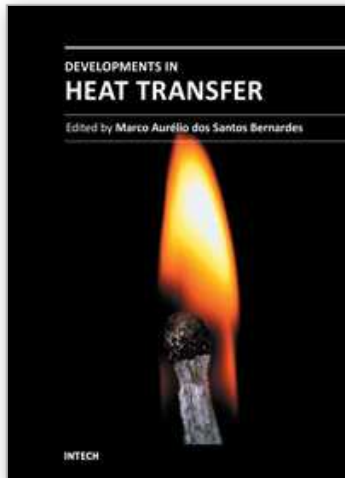
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This book comprises heat transfer fundamental concepts and modes (specifically conduction, convection and radiation), bioheat, entransy theory development, micro heat transfer, high temperature applications, turbulent shear flows, mass transfer, heat pipes, design optimization, medical therapies, fiber-optics, heat transfer in surfactant solutions, landmine detection, heat exchangers, radiant floor, packed bed thermal storage systems, inverse space marching method, heat transfer in short slot ducts, freezing and drying mechanisms, variable property effects in heat transfer, heat transfer in electronics and process industries, fission-track thermochronology, combustion, heat transfer in liquid metal flows, human comfort in underground mining, heat transfer on electrical discharge machining and mixing convection. The experimental and theoretical investigations, assessment and enhancement techniques illustrated here aspire to be useful for many researchers, scientists, engineers and graduate students.

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Phone: +86-21-62489820  
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