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NGNP PROCESS HEAT UTILIZATION: LIQUID METAL PHASE CHANGE HEAT EXCHANGER

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ABSTRACT

One key long-standing issue that must be overcome to fully realize the successful growth of nuclear power is to determine other benefits of nuclear energy apart from meeting the electricity demands. The Next Generation Nuclear Plant (NGNP) will most likely be producing electricity and heat for the production of hydrogen and/or oil retrieval from oil sands and oil shale to help in our national pursuit of energy independence. For nuclear process heat to be utilized, intermediate heat exchange is required to transfer heat from the NGNP to the hydrogen plant or oil recovery field in the most efficient way possible. Development of nuclear reactor - process heat technology has intensified the interest in liquid metals as heat transfer media because of their ideal transport properties. Liquid metal heat exchangers are not new in practical applications. An important rationale for considering liquid metals as the working fluid is because of the higher convective heat transfer coefficient. This explains the interest in liquid metals as coolant for intermediate heat exchange from NGNP. The production of electric power at higher efficiency via the Brayton Cycle, and hydrogen production, requires both heat at higher temperatures and high effectiveness compact heat exchangers to transfer heat to either the power or process cycle. Compact heat exchangers maximize the heat transfer surface area per volume of heat exchanger; this has the benefit of reducing heat exchanger size and heat losses. High temperature IHX design requirements are governed in part by the allowable temperature drop between the outlet of NGNP and inlet of the process heat facility. In order to improve the characteristics of heat transfer, liquid metal phase change heat exchangers may be more effective and efficient. This paper explores the overall heat transfer characteristics and pressure drop of the phase change heat exchanger with Na as the heat exchanger coolant. In order to design a very efficient and effective heat exchanger one must optimize the design such that we have a high heat transfer and a lower pressure drop, but there is always a trade-off between them. Based on NGNP operational parameters, a heat exchanger analysis with the sodium phase change is presented to show that the heat exchanger has the potential for highly effective heat transfer, within a small volume at reasonable cost.

Keywords: intermediate heat exchanger, liquid metal, NGNP, phase change,, process heat application, multi-phase flow dynamics

1. INTRODUCTION

A conceptual schematic of an advanced nuclear reactor (prismatic or pebble bed gas cooled reactor) coupled to a hydrogen production plant is shown in Fig.1. The high temperature gas cooled reactor (HTGR) supplies thermal energy to drive the power cycle and provides the process heat for either high temperature electrolysis (HTE) or for thermochemical sulfur iodine (SI) Process. In this design, the primary helium coolant is heated in the reactor to outlet temperatures in the range from 1100K to 1300K. A portion of the hot helium outlet stream serves as the working fluid in a gas-turbine power cycle, and a separate helium stream flows through a high temperature heat exchanger, providing process heat to hydrogen production plant. The overall efficiency of electrolytic hydrogen production can be improved by conducting the electrolysis of water vapor at high temperature (high temperature electrolysis or HTE), which has lower electrical energy requirements as compared to the conventional alkaline electrolysis. Thermochemical sulfur iodine (SI) cycle uses thermal energy directly, avoiding the thermal to electrical energy conversion, and is potentially more attractive as the process alternatives for hydrogen production.

Hybrid Sulfur is also being considered as a possible hydrogen production process but the required parameters are bounded by the HTE/SI analysis. Both for HTE and the SI process efficient transfer of thermal energy (heat) from the NGNP to the hydrogen production facility is required [1].

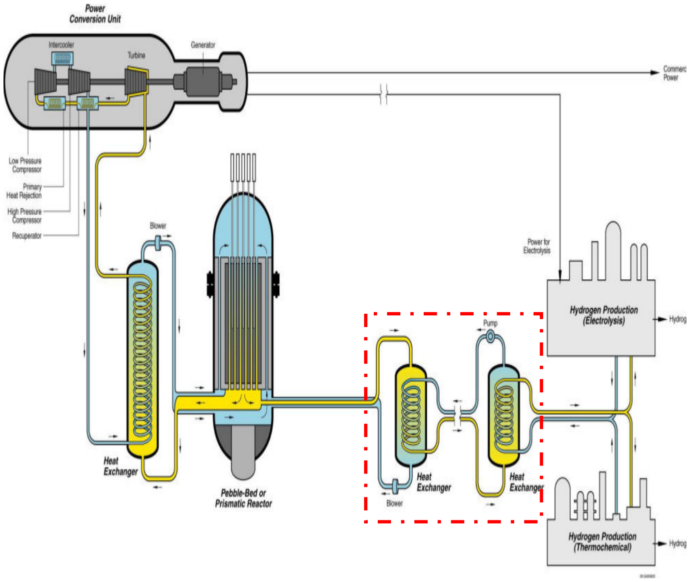

Fig.1 NGNP Conceptual Schematic of power generation and hydrogen production modified from [1]

Liquid metals are attractive as the secondary heat transfer medium for transferring the thermal energy of NGNP to a process plant. Liquid metals have high thermal conductivities, very low Prandtl numbers and typically have a very high heat transfer coefficient. Further, they are elementary substances that do not decompose, and behave as Newtonian fluids. The normal boiling points of these metals are also high, and hence the liquid metal systems can be operated at near atmospheric pressures. Research related to the vaporization of liquid metals has been quite intensive [3], technological interests derives mainly from the potential use of liquid metals as working fluids in the power plants for space application. The development of nuclear reactor technology has intensified the interest in liquid metals because of their ideal transport properties. Some acceptable equations have been developed but only limited information is available for boiling sodium. The objective of this work is to analyze the heat transfer and pressure drop characteristics of a heat exchanger utilizing boiling sodium as the secondary heat transfer medium. Overall heat transfer and pressure drop are determined and detailed analytical solution is presented for the phase-change heat exchanger. Sodium is selected as the heat transfer medium as it is among the most promising media for transporting heat from the NGNP to the hydrogen plant [2].

2. MATHEMATICAL MODEL DEVELOPMENT

The important difference between liquid metals and nonmetallic fluids in turbulent flows is that temperature distributions in the latter are relatively insensitive to local changes at the duct walls and thus to boundary conditions and to duct shape. Liquid metal temperature distributions however can be quite sensitive to boundary conditions [3].

The counter current heat exchanger is used in the analysis with liquid metal (Na) flowing through one side and He gas flowing through the other, as can be seen from Figure 2. Na side of the pipe is divided in three sections, exhibiting different flow regime. L_1 section refers to section where Na is in the vapor phase, L_2 section refers to section where Na is in two-phase regime and L_3 section refers to section where Na is in liquid phase, as can be seen from Figure 2. The process heat available from NGNP is 50MW. The key assumptions in this analysis are as follows:

(1) Flow is steady i.e. fluid properties are independent of time

(2) Fluid (Na) density is treated constant in the single phase regime but does vary in the two phase regime with varying quality

(3) Friction factor is considered as constant with passage flow length for single phase but does vary for two phase regime

(4)Body force is only caused by gravity, but is neglected because of horizontal position

(5)There are no energy sinks or source along a streamline; flow stream mechanical energy dissipation is idealized as zero

(6) Inlet temperature of He is assumed to be 1300K (i.e. outlet temperature from NGNP)

(7) Inlet temperature for Na is assumed to be 700K (i.e. because of one of the reaction for SI cycle takes place at this temperature)

(8) Outlet temperature for Na is assumed to 1223K (for higher efficiency of H_2 production)

(9) Properties of He are kept constant throughout the analysis, but the properties of Na do vary depending on the section

(10) Neglecting Axial conduction contribution

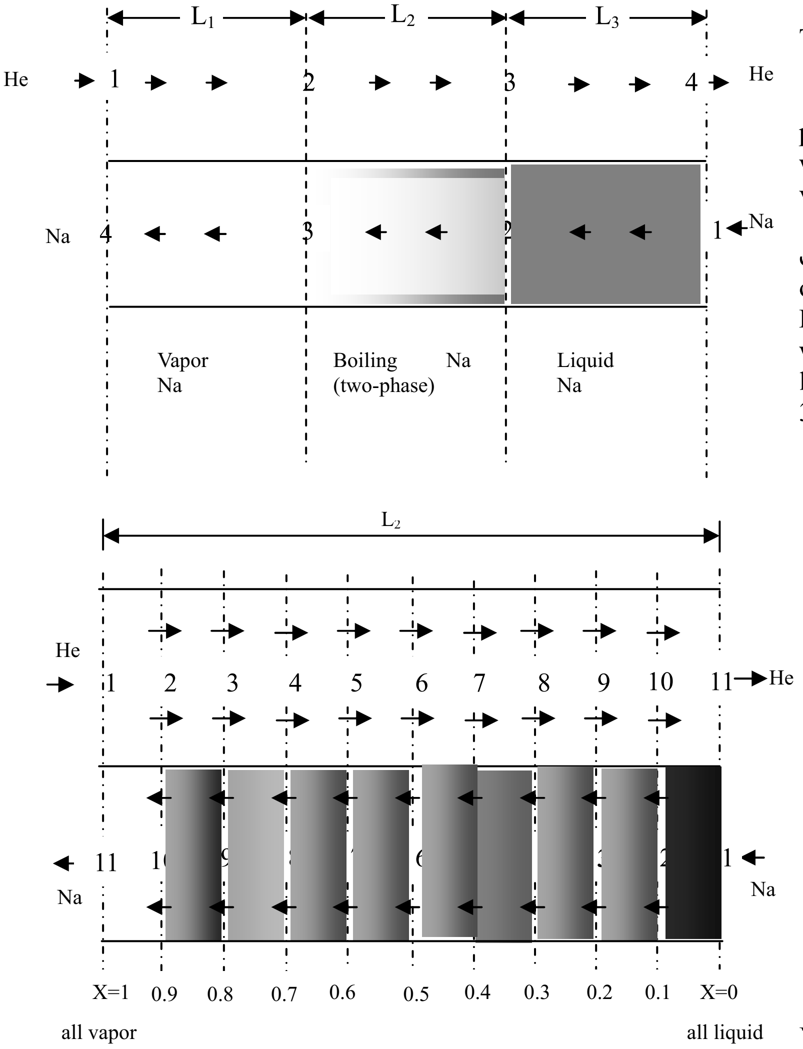

Fig. 2 Schematic of the Phase Change Heat Exchanger

In this analysis the boiling section (L_2) is divided into various sub-sections, such that quality in each section varies between completely liquid state to complete vapor state. The assumed geometry is a simple straight tube, where the primary side working fluid is He and the secondary side working fluid is Na which does goes through phase change. MATLAB code is written for the development and better understanding of phase change heat exchangers using Na as coolant. In this code for two-phase regime the macro-convective mechanism associated with over-all flow and micro-convective mechanism associated with bubble growth in the liquid film is taken into account. For single phase regime, based on the fluid temperature and properties overall heat transfer coefficient is determined and based on the Reynolds number value appropriate fluid friction factor value is determined, which is important since it determines the pressure drop for a given flow rate and geometry.

The choked flow condition is a fundamental limit on the axial vapor flow in a heat exchanger. Having the vapor velocity equal to sonic will maximize the heat transfer but will lead to propagation of a shock wave in the system, so in the analysis we assumed the vapor velocity of He equal to half its sonic velocity. The outlet temperature of He duct was varied such that we obtain positive value for LMTD for He and correspondingly diameter of He duct was obtained such that we operate at half the sonic velocity of He (i.e 882.5 m/sec). Fixing the value of mass flow rate, the diameter was varied till we obtained a value where the effective heat transfer area is low and so is the pressure drop, which can be seen from Figure 3.

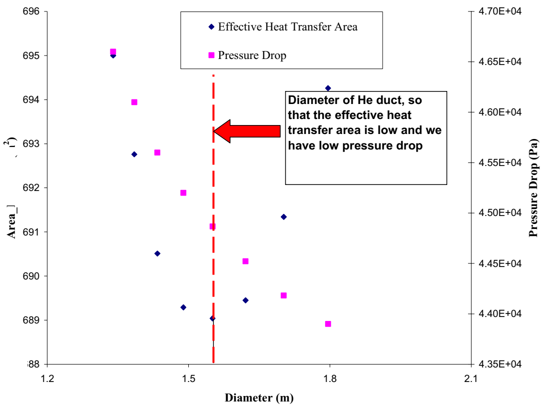

Fig. 3 Determination of He Diameter based on Heat Transfer Characteristics

In order to determine the heat duty for each individual section we have (from Figure 2):

$$Q_{L_1,He} = \dot{m}_{He}\, Cp_{He} \left(T_{He,1} - T_{He,2}\right) \tag{1}$$

$$Q_{L_2,He} = \dot{m}_{He}\, Cp_{He} \left(T_{He,2} - T_{He,3}\right) \tag{2}$$

$$Q_{L_3,He} = \dot{m}_{He}\, Cp_{He} \left(T_{He,3} - T_{He,4}\right) \tag{3}$$

$$L_{He} = L_{Na} \tag{4}$$

(i.e. the total length for both primary and secondary sides are equal)

The well known relationship between pressure drop and friction factor is usually expressed as:

$$\Delta p_{He} = \frac{4 f \rho_{He} \left(L_{He}\right)\left(V_{He}\right)^2}{2 D_{h,He}} \tag{5}$$

The heat transfer parameters for He duct can be seen from

Table 1 Heat Transfer and Geometric Parameters for He Duct

Ti (K)	To (K)	Diameter (m)	mdot_He (kg/sec)	Q_L$_1$ (W)	Q_L$_2$ (W)	Q_L$_3$ (W)	dp (kPa)	h_He (W/m^{2}K)
1300	1182	1.551	81.59	1.42E+06	4.34E+07	5.14E+06	41.56	354.74

2.1 Single Phase Heat Transfer for Straight Tube

For liquid metals the value of Prandtl number is normally less than 0.1, ranging down to less than 0.001. The high thermal conductivity of these fluids accounts for the very low Pr number. Also, the very high thermal conductivity accounts for the unusual heat-transfer characteristics of liquid metals. An empirical equation that corresponds fairly well to the available experimental data for heat transfer inside turbulent flow inside tubes for liquid metals which is utilized for both vapor and liquid phases [5] is:

$$Nu_{Na_v} = 6.3 + 0.0167 \left(Re_{Na_v, D_h}\right)^{0.85} \left(Pr_{Na_v}\right)^{0.93} \tag{6}$$

$$U_{L_1} = \frac{1}{\left(\frac{1}{h_{He}}\right) + \left(\frac{1}{h_{Na_v}}\right)} \tag{7}$$

Using equation 7, we obtain the overall heat transfer coefficient for liquid and vapor phase.

For Pressure drop calculation for Na_{vapor}, we have

$$\Delta T_{1,Na_v} = T_{He,1} - T_{Na,4} \tag{8}$$

$$\Delta T_{2,Na_v} = T_{He,2} - T_{Na,3} \tag{9}$$

$$\Delta T_{m,Na_v} = \frac{\left(\Delta T_{1,Na_v} - \Delta T_{2,Na_v}\right)}{\ln\left(\Delta T_{1,Na_v} - \Delta T_{2,Na_v}\right)} \tag{10}$$

$$L_{Na_v} = \frac{Q}{\left(U_{L_1}\right)\left(2 \pi R_{Na_v}\right)\left(\Delta T_{m,Na_v}\right)} \tag{11}$$

$$\Delta p_{L_1} = \frac{4 f \rho_{Na_v} \left(L_{Na_v}\right)\left(V_{Na_v}\right)^2}{2 D_{h,Na_v}} \tag{12}$$

Similarly, for liquid section we have,

$$\Delta T_{1,Na_L} = T_{He,3} - T_{Na,2} \tag{13}$$

$$\Delta T_{2,Na_L} = T_{He,4} - T_{Na,1} \tag{14}$$

$$\Delta T_{m,Na_L} = \frac{\left(\Delta T_{1,Na_L} - \Delta T_{2,Na_L}\right)}{\ln\left(\Delta T_{1,Na_L} - \Delta T_{2,Na_L}\right)} \tag{15}$$

$$L_{Na_L} = \frac{Q}{\left(U_{L_3}\right)\left(2 \pi R_{Na_L}\right)\left(\Delta T_{m,Na_L}\right)} \tag{16}$$

$$\Delta p_{L_3} = \frac{4 f \rho_{Na_L} \left(L_{Na_L}\right)\left(V_{Na_L}\right)^2}{2 D_{h,Na_L}} \tag{17}$$

2.2 Convective Boiling Heat Transfer Characteristics for Two-Phase

There have been a number of experimental studies in convective boiling heat transfer, reporting data on systems with water, organic fluids, and even liquid metals. It is postulated that there are two basic mechanisms which takes part in the heat-transfer process for boiling of saturated fluids with flow. These are a) macroconvective mechanism of heat transfer, and b) microconvective mechanism of heat transfer. It is further postulated that these two mechanisms are additive in their contributions to total heat transfer, which was first suggested in 1952 [5]. In the case of ordinary fluids, i.e., not liquid metals, the Pr number of the liquid and of the vapor are normally of the same magnitude. Based on the proposed methodology [6] for calculation of convective boiling heat transfer to liquid metals, a method for calculating boiling coefficients is developed in MATLAB. Two-dimensionless functions, s and F, are introduced to account for the suppression of bubble growth due to flow and for the increase in convective turbulence resulting from two-phase flow. These two correlating functions were determined empirically from available data and are presented graphically in Figure 4 and Figure 5.

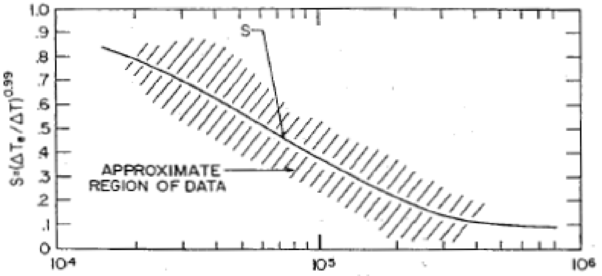

Fig. 4 Suppression Factor (s) [6]

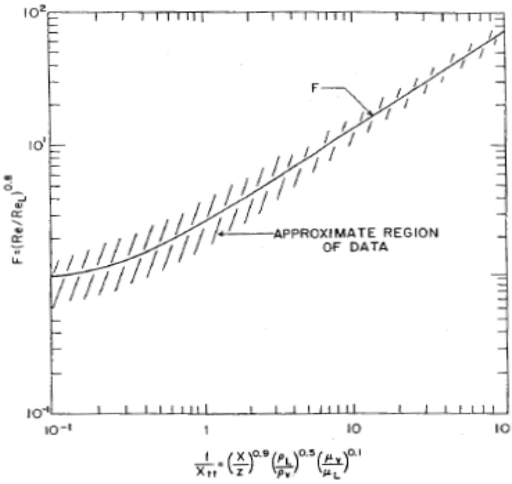

Fig. 5 Reynolds number factor (F) [6]

Chen's proposed model for convective boiling heat transfer is based on the following two postulates [6]:

(1) There are two mechanisms which contribute to total heat transfer, i.e., the macro-convective mechanism associated with over-all flow and the micro-convective mechanism associated with bubble growth in the annular liquid film. Bubble growth is a very complex phenomenon depending on various influences and forces. The heat from the heating surface proceeds to the boundary layer, superheats the layer, and is finally consumed as the latent heat of vaporization for the phase change.

(2) These two mechanisms interact with each other, i.e., the presence of vapor strongly influences macro-convective heat transfer, and conversely, the presence of flow modifies bubble motion and consequently effects micro-convective heat transfer.

$$h_{mic} = 0.00122 \frac{K_L^{0.79} C_{pL}^{0.45} \rho_L^{0.49} \Delta T^{0.24} \Delta P^{0.75} g_c^{0.25}}{\sigma^{0.5} \mu_L^{0.29} h_{fg}^{0.24} \rho_v^{0.24}} \times s \tag{18}$$

$$s = \left(\frac{\Delta T_e}{\Delta T} \right)^{0.99} \tag{19}$$

where ΔT_e is the effective superheat for bubble growth in the annular liquid film and s, called the suppression function, approaches unity at zero flow rate and zero at infinite flow rate. It was postulated, subject to experimental verification, that in all ranges of flow s can be represented as a function of the local two-phase Re number, as shown in Figure 4. In extending this theory to the case of boiling liquid metals, it is assumed that the micro-convective heat transfer can still be described by equation (18). This assumption is based on indications that pool-boiling theory [5], from which equation (18) was derived, is applicable for liquid metals as well as for ordinary fluids.

$$h_{mac} = \left[\delta + 0.024 \left(F\, Re_L^{0.8} \right) \left(\beta\, Pr_L \right)^{\alpha} \right] \left(\frac{\gamma K_L}{D} \right) \tag{20}$$

The values of the five parameters α,β,γ,δ, F are listed in Table 2.

Table 2. Parameters in Macro-Convective Equation [6]

Symbol	Liquid Metal	Vapor	Two-Phase
δ	7	0	$7Z^{\varepsilon}$
γ	1	K_v / K_L	$Z^{\varepsilon} + (K_v / K_L)(1- Z^{\varepsilon})$
β	1	Pr_v / Pr_L	$Z^{\varepsilon} + (Pr_v / Pr_L)(1- Z^{\varepsilon})$
α	0.8	0.4	$0.4\,(1- Z^{\varepsilon})$

Total convective boiling heat transfer is then obtained as the sum of the two interacting contributions,

$$h = h_{mic} + h_{mac} \tag{21}$$

The determination of the pressure drop is very salient for heat exchanger design because the fluid needs to be pumped through the exchanger, the pumping power is proportional to the exchanger pressure drop and also the heat transfer rate can be influenced significantly by the saturation temperature change for a evaporating/condensing fluid if there is a large pressure drop associated with the flow. This is because saturation temperature changes with changes in saturation pressure and in turn affects the temperature potential for heat transfer. The core pressure drop is determined separately on each side (i.e. for He and Na).

$$-\frac{dp}{dz} = \left(\frac{dp}{dz} \right)_{accleartion} + \left(\frac{dp}{dz} \right)_{friction} + \left(\frac{dp}{dz} \right)_{gravity} \tag{22}$$

$$\left(\frac{dp}{dz}\right)_{accleration} = \frac{d}{dz}\left(\frac{G_m^2}{\rho_m}\right)$$

(23)

$$\left(\frac{dp}{dz}\right)_{friction} = \frac{1}{A_z}\int_{P_z}\tau_w\, dP_z = \frac{\overline{\tau_w}\; P_z}{A_z} = \frac{f_{TP}}{D_e}\left[\frac{G_m^2}{2\,\rho_m}\right]$$

(24)

$$\left(\frac{dp}{dz}\right)_{gravity} = \rho_m\, g\cos\theta \qquad (25)$$

In this analysis we have used homogeneous equilibrium model (HEM)[7] i.e. the two velocities are uniform within the area, and that the two phases are in thermodynamic equilibrium. Thus the total pressure drop when the gas compressibility is accounted for is obtained, also and rearranging to get:

$$-\left(\frac{dp}{dz}\right)_{HEM} = \frac{\frac{f_{TP}}{D_e}\left(\frac{G_m^2}{2\,\rho_m}\right) + G_m^2\, v_{fg}\,\frac{dx}{dz} + \rho_m\, g\cos\theta}{\left(1+G_m^2\, x\,\frac{\partial v_g}{\partial v_p}\right)}$$

(26)

In our case the heat exchanger is horizontally located such that the gravitational effect can be neglected from equation (26), such that we have:

$$-\left(\frac{dp}{dz}\right)_{HEM} = \frac{\frac{f_{TP}}{D_e}\left(\frac{G_m^2}{2\,\rho_m}\right) + G_m^2\, v_{fg}\,\frac{dx}{dz}}{\left(1+G_m^2\, x\,\frac{\partial v_g}{\partial v_p}\right)}$$

(27)

3. RESULTS AND DISCUSSION

As, mentioned earlier about the sonic velocity limitation, at which one can achieve maximum heat transfer in a heat exchanger. In our mathematical model we fixed the velocity of the vapor to go at a constant rate of half the sonic velocity for both He (882.5 m/sec) and Na vapor (365.6 m/sec) in L_2 section.

The Table 3 and 4 shown below, shows the heat transfer parameters which were obtained for both single and two-phase. The diameter of the Na duct for both liquid and vapor phase were selected by fixing the mass flow rate, which was evaluated in previous section, and operating at half the sonic velocity for vapor and at 3m/sec for liquid phase. For the boiling Na section (L_2) we varied our diameter such that we could incorporate the change in quality as the diameter varied from liquid to vapor phase, which can be seen from Table 4.

Table 3 Single Phase Parameters (operating at Vs /2)

Section	Phase	Ti (K)	To (K)	Diameter (m)	Velocity (m/sec)	mdot_Na (kg/sec)	h_Na (W/m^2K)	U (W/m^2K)
L_1	Single Phase Vapor	1156	1223	0.265	365.6	8.806	632.403	227.26
L_3	Single Phase Liquid	700	1156	0.069	3	8.806	1.53E+04	346.7

Table 4 Two-Phase Boiling Na Parameters (operating at Vs /2 & Q_L_2)

Quality	Diameter (m)	h_Na_Boiling (W/m^2K)	U (W/m^2K)
0.05	0.0868	2.21E+07	354.734
0.15	0.104	2.19E+07	354.734
0.25	0.122	2.07E+07	354.733
0.35	0.14	1.93E+07	354.733
0.45	0.158	1.80E+07	354.732
0.55	0.175	1.67E+07	354.732
0.65	0.193	1.54E+07	354.731
0.75	0.211	1.43E+07	354.731
0.85	0.229	1.33E+07	354.730
0.95	0.247	1.24E+07	354.729

The heat transfer coefficient of Na is very high, when in the boiling regime, and is a couple of magnitude higher than what one would expect in a free convection. In this counter current arrangement the heat transfer parameter is solely governed by He, which can be seen from the following expression.

$$\frac{1}{U} = \frac{1}{h_{He}} + \frac{1}{h_{Na}} \qquad (28)$$

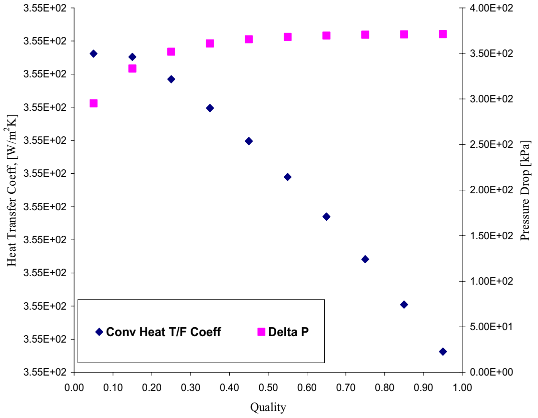

Fig. 6 Overall Heat Transfer Characteristics at Vs/2

From Figure 6, we can observe the pressure drop for the entire system is very high, which requires the need of carrying out the analysis such that we have a lower pressure drop and similar heat transfer coefficient, if not better.

4. CONCLUSIONS

The high heat transfer rates are mainly because of high thermal conductivities of liquid metals as compared with other fluids. Liquid metals, behave as Newtonian fluids, have very high thermal diffusivity and very low kinematic viscosity. Thus the Prandtl number is of low order, 10^{-2}. Also, liquid metals remain in liquid state at higher temperature compared to conventional fluids like water and various organic coolants; this also makes the design of a compact heat exchanger more possible. The key advantages of using liquid metal as the coolant is that they do not require operation at high pressure in order to obtain high temperatures and usually, the melting temperatures are low enough such that they can be used as coolants in thermal devices, such as compact heat exchange systems. As a result, their use usually involves pressure which is small compared to thermodynamic critical pressure of the fluid. These advantages, especially the high heat transfer are compelling enough to maintain research interest in the usage of liquid metal as the coolant for the intermediate heat exchanger in the Next Generation Nuclear Plant (NGNP). The net pressure drop obtained for the entire system came out to be very high, which certainly would need further analysis such that the design can be optimized so that we incur lower pressure drop and consequently do not pay penalty on the heat transfer.

NOMENCLATURE

A	cross-sectional flow area [m^2]
C_P	heat capacity [kJ/kg K; equation 18: Btu/lb deg F]
deltaP	pressure drop [Pa]
D_h	hydraulic diameter [m]
dp	pressure drop [Pa]
dz	location [m]
f	friction factor
F	Reynolds number factor $(Re/Re_L)^{0.8}$
F_{DA}	Dengler and Addoms correction factor
g_c	gravitational constant
G	mass flux [Kg / m^2 sec]
h	convective heat transfer coefficient [W/m^{2}K; equation 18: Btu/hr ft^2 deg F]
h_{fg}	latent heat of vaporization [kJ/kg; equation 18: Btu/lb]
K	thermal Conductivity [W/m K; equation 18: Btu/hr ft deg F]
L	length [m]
Nu	Nusselt number
ΔP	difference in vapor pressure corresponding to ΔT [equation 18: psf]
Pr	Prandtl number
Q	process heat from NGNP [watts]
R	radius of the duct [m]
Re	Reynolds number
s	suppression function, $(\Delta T_e / \Delta T)^{0.99}$
T	temperature [K]
ΔT	superheat, $T-T_s$ [K ; equation 18,19: deg R]
U	overall heat transfer coefficient [W/m^{2}K]
v	specific volume [m^3/kg]
V	average flow velocity [m/sec]
Vs	sonic velocity [m/sec]
x	quality, weight fraction of vapor
X_{tt}	Martinelli Parameter, $\left(\frac{Z}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_v}\right)^{0.1}$
Z	weight fraction of liquid, shown in figure 5
z	location [m]

Greek Symbols

ρ	density of the fluid [kg/m^3; equation 18: pcf]
σ	surface tension [equation 18: lb/ft]
μ	dynamic viscosity [newton.sec/m^2; equation 18: lb/ft hr]
τ	shear stress [N/m^2]
θ	angle [radians]
φ_{fo}	friction factor multiplier
α	parameters defined in Table 2
β	parameters defined in Table 2
γ	parameters defined in Table 2
δ	parameters defined in Table 2

Subscripts

e	equivalent
f, L	liquid
g,v	gas/vapor
He	helium
HEM	homogeneous equilibrium model
i	inlet
m	mixture
Na	sodium
o	outlet
TP	two phase
w	wetted
1,2,3	location as can be seen from Figure 2

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