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PASSIVE CONTAINMENT COOLING: MODEL EXPERIMENTS

Someshwar Pandey
Department of Mechanical Engineering
Indian Institute of Technology Kanpur
Kanpur 208016 India

Vadiraj A. Hemadri
Department of Mechanical Engineering
Indian Institute of Technology Kanpur
Kanpur 208016 India

Sameer Khandekar*
Department of Mechanical Engineering
Indian Institute of Technology Kanpur
Kanpur 208016 India
samkhan@iitk.ac.in

Pavan Sharma
Reactor Safety Division
Bhabha Atomic Research Center
Trombay
Mumbai 400094 India

ABSTRACT

The feasibility of using a closed cylindrical two-phase thermosyphon and a closed loop Pulsating Heat Pipe (PHP) for containment cooling application is experimentally studied in this work. The reactor containment at the time of accident is presently mimicked by administering steam in the evaporator chamber ~ 1 bar/ 100°C (high pressure tests will be reported later). The condenser chamber is modeled as heat sink containing a stagnant pool of water. The latent heat of condensation of the steam is transferred to the stagnant pool of water through the thermosyphon or the PHP, as per the case. The axial cross-sectional area and the radial surface area for heat transfer are kept identical for both the thermosyphon and the PHP. In this way a direct comparison of the thermal performance of these two devices was possible. We find that while both the devices transferred approximately the same amount of heat (~500W), the average Nusselt number for PHP was higher than the thermosyphon, indicating that it operated at a lesser temperature difference between the wall and the coolant fluid, i.e., PHP had a higher effective thermal conductivity than the thermosyphon.

Keywords: Passive nuclear containment cooling, Thermosyphon, Pulsating heat pipes

INTRODUCTION

Containment cooling systems under accident scenario are an important part of a nuclear reactor. Although many heat transfer systems are available, which meet the present industry requirements, yet cooling technology is being continuously challenged to dissipate heat, preferably passively, at minimum cost, weight and other technological factors related to compatibility with the reactor. This is the prime motivation for development of more efficient containment cooling system designs.

Primarily, the containment cooling systems can be broadly classified into two major types:

a) Active cooling systems
b) Passive cooling systems

Industry and nuclear regulatory bodies prefer passive heat removal systems as compared to active cooling solutions; the latter requiring additional power and active control strategies for coolant circulation. On one side nuclear energy is an indispensible energy source for the emerging economies, recent accidents in Fukushima, Japan pose tough challenges for thermal-hydraulic control and safety operations. Despite the three major nuclear accidents worldwide, number of operating and new nuclear reactors will see a modest growth in the coming years with more emphasis on reliable

and stringent systems. In this scenario, passive systems would be the preferred choice for cooling and heat removal systems as they require less human intervention, thus increasing reliability. Some of the main contemporary concepts available for passive cooling of nuclear containments include, amongst others [Lee et al., 1997]:

a) External spray from an elevated tank
b) External moat for steel containment
c) Suppression pool
d) Internal/external condenser with passive spray
e) Closed loop two-phase thermosyphons

One of the early studies for the use of heat pipes heat exchangers for passive containment cooling was done by Razzaque et al. [1989]. The concept relied on the shutdown heat removal from the reactor vessel by heat rejection to the outside atmosphere, through a system of heat pipe heat exchangers located at the top of the containment structure. Lee et al., [1997] have presented an overview and comparative assessment on passive containment cooling concepts for advanced Pressurized Water Reactors (PWRs). Condensation experiments in the presence of non-condensables (e.g., air and helium) were conducted by Liu et al., [2000] to evaluate the heat removal capacity of a passive cooling unit in a post-accident containment. Experiments on Internal Evaporator Only (IEO) type passive containment cooling system were performed and an empirical correlation for heat transfer was developed. An experimental study on evaporative heat transfer coefficient and application for passive cooling of AP600 steel containment was studied by Kang and Park [2001]. The evaporative heat and mass transfer at the surface of a falling water film with counter-current air flow, was investigated in a vertical duct having a one-side heated plate. With the generated data, a new correlation on evaporative mass transfer coefficient was developed as a function of operating parameters. Conceptual design and analysis of a semi-passive containment cooling system for a large concrete containment has been developed by Byun et al. [2000]. Here, it was concluded that a practical system requiring four IEO loops could be utilized to meet design criteria for severe accident scenarios.

In the present study, two passive devices, a two-phase closed thermosyphon and a Pulsating Heat Pipe (PHP), are tested in view of potential applications in containment cooling and allied systems. Although these two systems have been extensively tested for various applications, ranging from electronics thermal management to heat exchangers and reboilers, there is no data available in the literature on the performance of these two devices under the type of boundary conditions which may be encountered during nuclear containment heat removal. In the series of tests reported here, the devices have been subjected to condensing steam at one atmosphere (in the evaporator section) and a stagnant pool of water (in the condenser section). The performance of these two devices has been compared under such boundary conditions in a scale-down model. This study forms the basis for doing more extensive experimental investigations for further exploring two-phase passive heat transfer systems for thermal management of nuclear reactors.

EXPERIMENTAL SETUP/ PROCEDURE

We envisage the use of the thermosyphon and PHP as a passive heat removal system during a steam leak accident in the reactor containment. Figure 1 shows the experimental setup constructed to simulate this scenario. There are two main chambers in the set up, the heat transfer coupling between the two being achieved by the passive two-phase heat transfer device under scrutiny. The lower chamber (evaporator chamber) represents the reactor containment. Steam at atmospheric conditions is generated in a boiler and sent to this lower chamber. This simulates the steam leakage in the containment zone. At the time the steam injection commences, this chamber is in communication with the atmosphere through the bottom valve. Thus, the steam condensation in this lower chamber cannot be termed as a ‘single-component phase-change system’; a certain amount of air is always present during the process. The upper chamber (condenser chamber) is always filled with stagnant pool of coolant water at room temperature. This represents the water tank for dumping the heat released in the reactor containment, represented by the lower evaporator

chamber of the setup, as described earlier. This upper chamber is connected to a constant temperature bath which can replenish/ circulate cooling water whenever needed. The thermosyphon or the PHP, as the case may be, is placed such that, one half of the device is inside the lower evaporator chamber (exposed to steam at one atmosphere) and the other half is located in the upper condenser section directly in contact with the stagnant pool of water. Thus, in this design, the steam continuously loses its latent heat to the stagnant water pool in the upper chamber, the two-phase passive device being the medium of heat transfer. For benchmarking and comparison between the two devices, the axial as well as the radial heat transfer area of both the devices have been made identical.

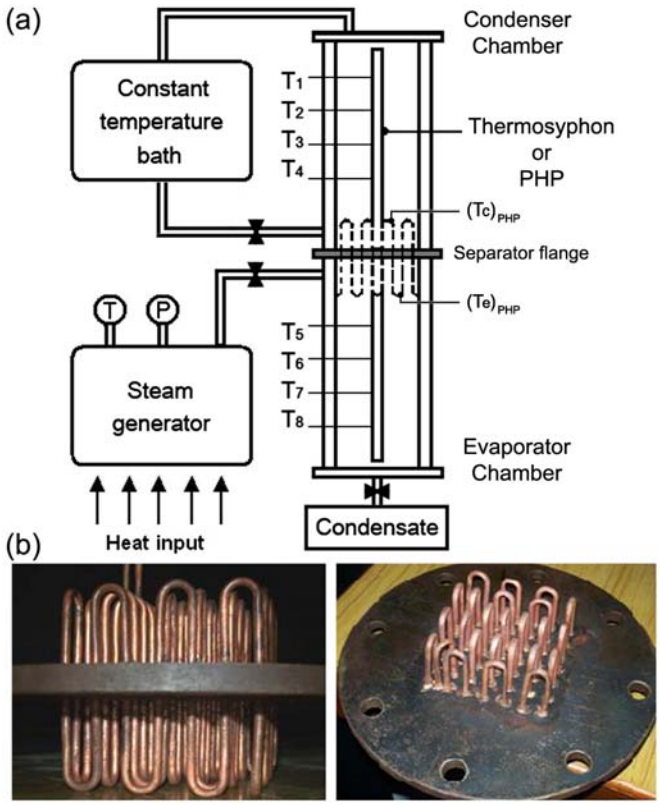

Figure 1: (a) Schematic of the experimental setup and, (b) PHP module fitted with the separator plate.

The evaporator and condenser chambers were made from a 550 mm long mild steel cylindrical hollow pipe of ID 150 mm and OD 162 mm, respectively. This hollow pipe was divided into two sections to form the evaporator and the condenser zones, as shown in Fig. 1. The separator flange, which physically divides these two chambers, is integrated with the thermosyphon or the PHP, as the case may be. The entire exposed surface of the setup is properly insulated by 40 mm thick foam insulation. From energy balance, the heat losses were found to be lower than about 12% of the net heat input.

The thermosyphon is made of copper tube 1000 mm long (Lc=Le=500 mmID = 23.4 mm; OD = 25.4 mm). This tube is inserted into the separator flange and welded to it at the mid section. The closed loop PHP is also made across the separator flange by inserting U turns of copper tube (ID = 3.4 mm; OD = 5.0 mm) and eventually welding them together to form 24 turns each in the evaporator and the condenser section. The length of one PHP U-turn (52.7 mm) and number of turns (24 x 2 = 48) of the PHP tube is chosen in such a manner that the radial as well as the axial heat transfer area of both the devices is exactly the same.

The working fluid tested in the thermosyphon/PHP was distilled, deionized and degassed water at 50% filling ratio for both the devices, respectively. To benchmark the data and to assess the contribution of the thermosyphon/PHP action in the net heat transfer, dry tests under internal vacuum conditions inside the devices (with no working fluid) were performed prior to actual tests.

The time response of the thermosyphon tube in terms of the eight thermocouples (T_1-T_8), four each in the condenser and the evaporator section, located as shown in Fig. 1, was recorded by a PC based DAQ for the entire duration of experiment.

In all experiments, initially the temperature of condenser section (upper chamber) was maintained with the help of a cryostat. The evaporator section steam supply was then started. Once a steady steam-supply condition was achieved and the entire evaporator section reached a steady uniform temperature, the coolant water circulation in the upper chamber was stopped. After this point, heat transfer to the stagnant pool of water commenced through the passive two-phase device, increasing its temperature. The experiment was stopped when the coolant pool temperature in the condenser zone reached about 75°C.

All experiments with thermosyphon as well as PHP can be divided into three distinct zones, indicated on the results as Z-1, Z-2 and Z-3. The attributes of these three zones is given hereunder:

- Zone Z-1: Temperature measurement and data logging starts in this zone. It is a precursor to the actual heat transfer experiment. We only record the temperature at all the eight thermocouples for some period of time before starting the steam injection into the evaporator chamber. In this period the upper chamber is being continuously circulated with coolant water at a fixed temperature of 25°C. After a while when the entire setup is isothermal, we enter Zone Z-2.
- Zone Z-2: At this stage we open the steam injection valve to the bottom evaporator section because of which the temperature of the evaporator section starts increasing (T_5-T_8). In this zone, the temperature of the cooling water in the condenser section is continued to be maintained at a fixed value by circulating it with the help of the constant temperature bath. We continue the experiment in this stage until all the thermocouples (T_1 to T_8) come to a quasi steady state. Once this happens, we stop the circulation of coolant in the upper chamber and now the cooling is achieved only by the stagnant pool. We now enter the last zone of the experiment.
- Zone Z-3: At the end of Zone 2, the coolant circulation from the constant temperature bath is stopped so as to allow the water in the upper chamber to start getting heated up by the heat coming from the condensation of the steam in the lower chamber, through either the thermosyphon or the PHP, as the case may be. The temperature of the thermocouples located in the upper chamber (T_1-T_4) then starts to rise.

RESULTS AND DISCUSSION

Typical temporal variation of thermocouples for the thermosyphon operating in dry state and filled with water are shown in the Fig. 2 (a), (b), respectively. There is no 'steady-state' in the system. So far as the steam is injected in the lower chamber, the thermosyphon/PHP continues to operate and the cooling water stored in the upper chamber keeps getting heated up. The experiment is stopped when the water in the upper chamber is about 75ºC.

In case of dry test of the thermosyphon, Fig. 2 (a), heat transfer from evaporator to condenser is due to conduction effect of copper tube only. In Zone-3, the evaporator temperature quickly comes to about 100°C and the condenser zone temperature continues to rise at a slow rate, the heat being transferred by diffusion through the tube and then through natural convection to the coolant pool. The net heat transfer rate in Zone-3 was about 242 W with a standard deviation of less than 5 W in five tests.

When the thermosyphon is operated with the working fluid, Fig. 2 (b), the temperature of the evaporator section reaches a maximum value and suddenly decreases, as seen in Zone 2. This is because, as the thermosyphon becomes active (onset of nucleate boiling), the temperature of thermocouple T_5 and T_6 go down suddenly while the temperature of T_2 and T_3 increase, thereby decreasing the thermal resistance drastically. As the heat transfer continues through the thermosyphon, (Zone-3), the temperatures of the evaporator as well as the condenser sections continuously increase.

In a series of eight experimental runs, the heat transfer rate of the water filled thermosyphon was found to be about 495 W with a standard deviation of 21 W.

Quite naturally, the surface temperature difference between the evaporator and condenser section was higher in case of the dry-test, its 'effective thermal conductivity' being less than that of filled thermosyphon.

Experiments similar to those detailed for the thermosyphon were performed, after replacing the thermosyphon with the PHP. Figure 3 (a), (b) show the sample temporal response of the thermocouples for the dry test and the filled PHP, respectively. In case of the PHP, only two thermocouples were used to obtain the temperature with one each in the condenser and evaporator section, located at the extremities of the respective U-turns of the evaporator and the condenser section respectively, as shown in Fig. 1.

During the dry-test, the PHP transferred about 260 W (four tests; Standard deviation ~3.5 W) while the water filled PHP transferred about 485 W (eight tests; Standard deviation of ~ 7.0 W).

It is clear that both the devices, with identical radial and axial cross section areas, provide nearly the same heat throughput under the applied boundary conditions.

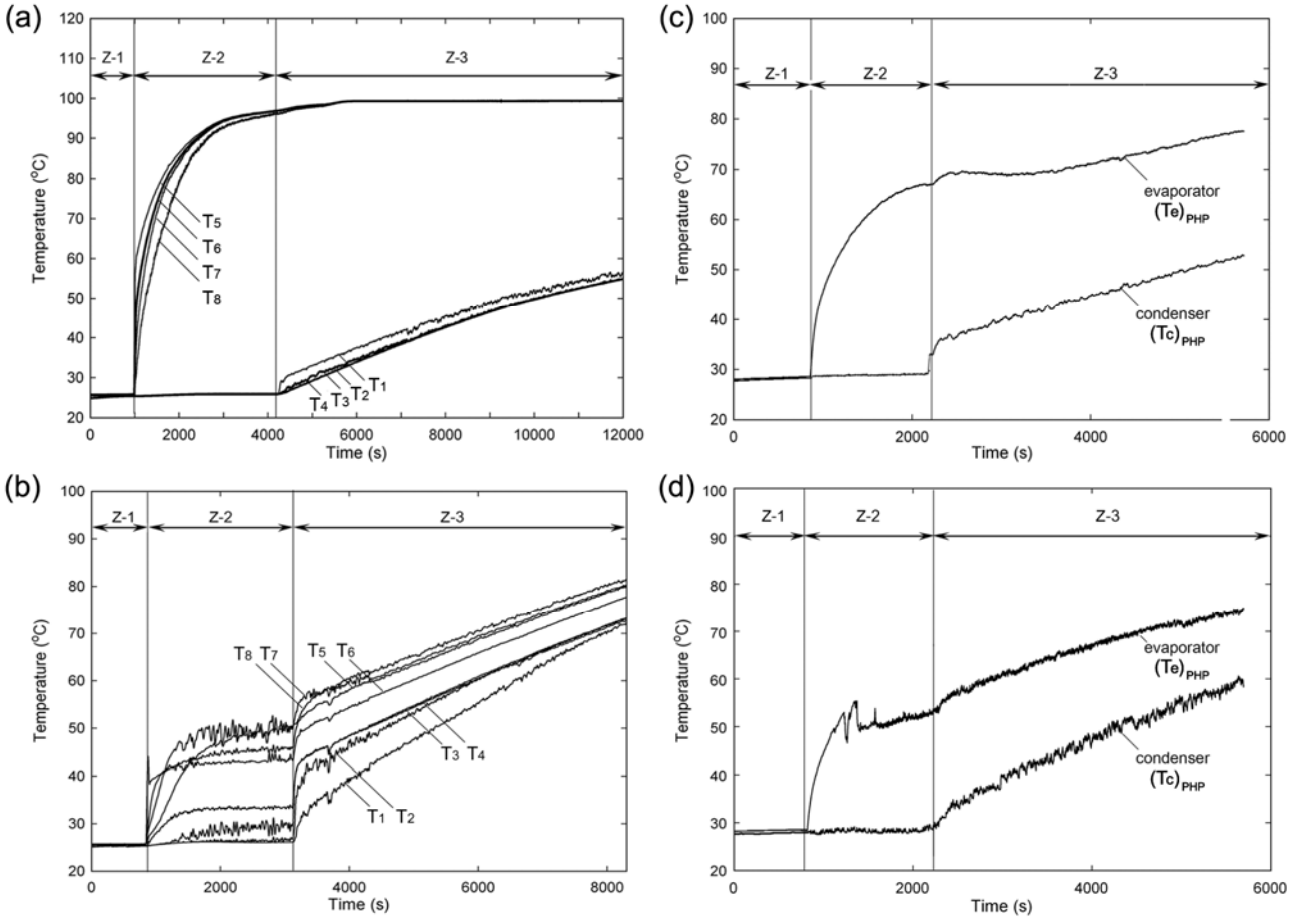

Figure 2: Temporal evolution of various temperatures (a) for dry thermosyphon (b) for thermosyphon operated with water (c) for dry PHP (b) for PHP operated with water

Calculation of average Nusselt number

As the Grashof number in the condenser section is always higher than ~10^{10} (except at very early times, when the heating just commences and the temperature difference $(\overline{T}_{sur-c} - \overline{T}_{l-b})$ is small) and for water the Prandtl number is higher than unity, therefore the Rayleigh number in the condenser is always higher than 10^9, indicating a turbulent flow. As a first estimation, we make use of correlation,

$$\overline{Nu}_c = 0.1(Gr_c \cdot Pr_l)^{1/3} \quad (1)$$

to estimate the average Nusselt number in the condenser. This correlation is applicable as the following condition is generally met by the thermosyphon system,

$$(D / L_c) \geq 35/(Gr_c)^{0.25} \quad (2)$$

Simultaneously, as the heat throughput and the average condenser and fluid temperatures are known, $\overline{Nu}_c$ can be calculated from the experimental data, both for the thermosyphon and the PHP, respectively as,

$$\dot{Q} = h_c \cdot A_{sur-c} \cdot (\overline{T}_{sur-c} - \overline{T}_{l-b}) \quad (3)$$

$$\overline{Nu}_c = h \cdot L_c / k_l \quad (4)$$

The comparison of theoretical (Eq. 1: applicable in the case of thermosyphon only), experimental (Eq. 4: applicable in both the cases, $L_{c\text{-}TS}$ = 500 mm and $L_{c\text{-}PHP}$ = 52.5 mm) is shown in Fig. 3. The results indicate that the $\overline{Nu}_c$ increased rapidly in the early part of the transient when the free convective buoyancy layer flow accelerated. After a while when the boundary layer has developed, $\overline{Nu}_c$ asymptotically stabilizes. $\overline{Nu}_{c-PHP}$ is nearly double than $\overline{Nu}_{c-TS}$, as the PHP operates at a much lower wall temperatures in the condenser area. In case of PHP, heat transfer takes place through multiple U-turns, which in turn disrupts boundary layer growth, thus increasing $\overline{Nu}_c$.

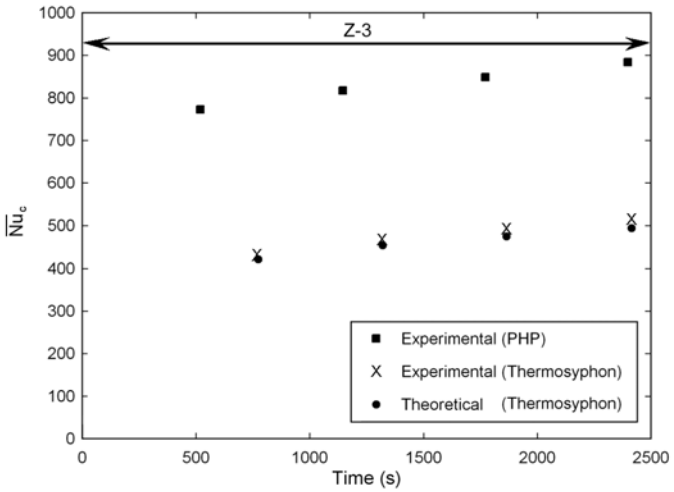

Figure 3: Variation of $\overline{Nu_c}$ with time

SUMMARY AND CONCLUSIONS

We have explored the possibility of using a thermosyphon and a PHP as a potential passive heat removal system from nuclear reactor containments. Both devices had the same radial and axial heat transfer area. Similar type of boundary conditions, as might occur during a steam leakage scenario, was mimicked. The following are the major conclusions of the study:

- Heat transfer during the dry tests for both the devices show considerably low heat transfer rate than the filled devices.
- The net rate of heat transfer by thermosyphon and PHP is comparable with each other.
- The driving temperature difference for heat transfer in the condenser area, $(\overline{T}_{sur-c} - \overline{T}_{l-b})$, is much smaller in case of the PHP, thus increasing the local heat transfer coefficient, as compared to the thermosyphon. In the latter case, a thick boundary layer develops around the vertical cylinder, increasing the average wall temperature for providing the heat throughput.
- Both thermosyphon and PHP are cost effective, passive and reliable systems which can be considered for the purpose of nuclear containment cooling and allied applications.

Further tests at higher pressure/temperature (up to 2.5 bars) of steam, which are closer to the real-time accident scenario, are underway and will be reported at a later date.

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NOMENCLATURE

A	area (m^2)
D	diameter (m)
Gr_c	Grashof number
h	heat transfer coefficient (W/m^2k)
k	thermal conductivity (W/mk)
L	length (m)
$\overline{Nu}$	average condenser Nusselt number
Pr	Prandtl number
$\dot{Q}$	heat throughput (W)
Ra	Rayleigh number
$\overline{T}$	average temperature ($^\circ$C or K)

Subscripts

b	bulk
c	condenser section
f	fluid
l	liquid
sur	surface

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