Journal of
NUCLEAR
Technology in Applied Science
ISSN 2314-8209 e-ISSN 2314-8217

J. Nucl. Tech. Appl. Sci., Vol. 10, PP. 1 : 10 (2022)

Proposed Modeling of Natural Convection Cooling Heat Pipe

Abdelaziz, M.[1]; Elsayed, M.[2]; Eid, M.A.[2] and Mansour, M.S.[2]

Received: 26/12/2021
Accepted: 26/02/2022
DOI: 10.21608/jntas.2022.112703.1046

E.mail:mabdelazizm36@yahoo.com

ABSTRACT

Heat-pipes and thermosyphons, properly designed and manufactured, are considers as passive heat transfer appliances which have very long live spans when operating with temperature limits. Nuclear spent fuel storage tanks are characterized by a continuous heat emission even after the reactor shutdown. To keep spent fuel temperature within safe limits, this heat must be dissipated in regime of emergency. This paper treats operation of a completely passive cooling system using a wickless gravity assisted two-phase closed heat-pipe loop for dissipation the heat and cooling the nuclear reactor spent fuel pool by running as alternative cooling system to be in safe mode. The model considers natural convection by air for the condenser part of the heat-pipe loop to confine the residual heat. A numerical simulation using a new design of gravity assisted two-phase closed heat-pipe loops was used to investigate the heat-pipe performance and its thermal characteristics. Heat-pipe configuration and the heat load are well known. Focus on atmospheric air temperature effect will be analyzed. The heat-pipe material is stainless steel-AISI316, and the demineralized water was used as the running working fluid. The atmospheric air at ambient temperature was naturally circulated around the heat-pipe condenser for cooling fluid. The results showed that the best thermal performance and characteristics were obtained at a lower atmospheric air temperature and the heat pipe could remove 150kW within safe conditions. The computer simulation refers to a pattern, and a trend line can then be used to predict the concepts of heat transfer with different inputs.

KEYWORDS

Heat pipe, Natural convection, Theoretical modeling, Heat-pipe transient operation, Passive cooling system.

1. Department of Nuclear Safety Research and Radiological Emergencies – Egyptian Atomic Energy Authority, 11787 Cairo, Egypt.
2. Mechanical Engineering Depart. Higher Technological Institute, Tenth of Ramadan city 44629, Egypt

INTRODUCTION

Multi-purposes reactors (MPR) after shutdown still keep residual heat. Accordingly, in case of emergency and after nuclear reactor shut down, the heat should be removed to keep fuel temperature within safe limits. The MPR have an auxiliary spent fuel tank that also should be cooled to remove the heat generated. In this direction, heat-pipes are suggested to be inactive heat energy transport systems. With appropriate design and manufacturing, they are considered to be expatriate sealed tubes which have very long lives. In present study, a completely passive cooling system implementing loop heat pipe for cooling and dissipating the nuclear reactor's residual heat is suggested. The most important aspects affecting the public concerns regarding nuclear reactors operation are their safety features. The public concerns are related to potential nuclear accidents and radioactive hazards on human health by **(International Nuclear Safety Advisory Group. 1999)**. Safety regulations have been developed, matured, and implemented in the last decades leading to very restrictive safety limits. Consequently, the accidents risk in nuclear power facilities is declining by (**Ye *et al*., 2013)**. There were only three serious nuclear accidents, for over 16,210 nuclear power reactors operating in 33 countries. No adverse health or environmental consequences occurred in the Three Mile Island accident (USA 1979), in which the reactor was completely destroyed. Significant environmental and healthiness consequences happened in Chernobyl accident (Ukraine 1986), where the vapor bangs and bonfire leading to destruction of the reactor. Since then, the deaths have increased from 31 people killed to 56. The third one was the Fukushima accident (Japan 2011), where three reactors from fourth units were suffered from the effects of cooling losses because of a huge tsunami. Only the emission due to accidents in Chernobyl and Fukushima leaded to larger hazardous for the community than those faced from contact to other radioactive sources **(Tokyo Electric Power Company, Inc. 2012)**. The design of nuclear power stations are directed to be in safe hands in operating situations and secure in case of any fail or scourge. Possible accidents guide to progressive safety enhancements. Mainly current power reactors use a mixture of intrinsic safety characters and engineered safety systems that can be active or passive **(IAEA Safety Standards)**. Because of the unbalancing between passive and active safety systems, the three previous mentioned nuclear reactors accidents were occurred **(Jackel**, **2015)**. Using passive component system and/or the lack of human action step down power failure. Safety will increase in direct proportion with enhancing and improving the passive safety systems. Thus, passive systems will improve the overall factor of safety for the nuclear power plant **(Ognerubov *et al*., 2014)**. The term Passive Safety is not equivalent with the deep-rooted safety because it remains subject to other faults such as structure, mechanical failure and/or human interference. Regardless of that, we will concentrate on the passive safety, since our work is mainly about the usage of heat-pipe for nuclear safety which is a Passive system from grade B, **(Wu *et al*., 2014)**. Passive systems must obey some conditions to be accepted: Reliability and availability in short terms, long terms, and under retrograde conditions by **(Sailor *et al*., 1987)**. The accident at the Daiichi NPP proposed reducing credence on active systems to reduce human error factor.

Passive systems types are classified in different categories: Grade A, B, C and D, where Grade B means: no signal input, external power source, no moving mechanical parts but moving working fluids by **Throm (1991)** and **Advisory Group (1994)**. Gravity assisted two-phase, closed heat pipe loops are sealed inactive two-phase heat transfer devices that make use of the highly efficient evaporator and condensation thermal transfer procedure to attain upper limit thermal conductivity among a heat source and a heat sink. The quantity of heat that these devices can carry is several orders higher of magnitude than the pure conduction through solid metal, being 200-500 higher comparatively with that of copper by **(Noie, 2005)**.

The two-phase closed heat-pipe loop may be inclined with small angle or vertically oriented, with a liquid sink at the bottom. At operation, heat-pipe loop transfer heat through the evaporator from an external source to the liquid think. Consequently, a part of the fluid evaporates. Towards the condenser section, the vapor driven by pressure difference force between the condenser and evaporator flows through the adiabatic length. In connection with the condenser part, the steam is condensed into fluid leaving its latent heat of evaporation to the heat pool in the condenser segment. The liquid returns from the condenser to the evaporator internally due to force of gravity. Thus, the hydraulic cycle of the working fluid completed by equilibrated heat transfer by **Reay *et al*. (2013)**.

A heat-pipe loop with evaporator and condenser lengths each 100m helical configuration with an outer diameter of 150 mm and 3 mm in thickness, as a passive cooling system for a nuclear research reactor main pool and nuclear spent fuel storage pool is proposed to be used to remove this decay heat, as presented in “Figure 1”. The design is focused on removing heat from the spent fuel tank of the research reactor to be in safely operated. The model considers convection by air naturally for the condenser part of the heat-pipe loop to compensate the residual heat. A numerical simulation model using unique design of heat-pipe loops was used to investigate the heat transfer by **Alizadehdakhel *et al*.** (**2010)**. The effects of atmospheric air temperatures were analyzed. Demineralized water was used as the working fluid. The atmospheric air was (15-20-25-30-35-40 ^{o}C) and circulated around the condenser as a cooling procedure. The main parameters considered were, as follows: Heat input the effect of $Q \leq 150$kW, 100% working fluid filling ratio, and the heat-pipe loop configuration with 100m length and 150mm outer diameter. As it was expected, the results show that the best thermal performance was obtained at a low atmospheric air temperature.

THEORETICAL MODEL

Model assumptions

1. Flow model is one-dimensional.
2. Vertical orientation heat-pipe loop is considered.
3. Saturated vapor state is considered due to a very small vapor superheat.
4. Energy balance equations neglect the potential and kinetic energies compared with the heat transfer rate.
5. The thermal conductivity, density and other physical properties for saturated liquid are temperature depended.
6. Working fluid, wall temperatures and coefficient heat transfer of the heat-pipe sections are at mean value.
7. The passive loop starts up when the power is suddenly on from initial condition.

Equations of the model

A model describing both phase flow and thermal of the passive gravity assisted heat-pipe loop has been performed by **Shouman *et al*., (2013)**. This simpler model has been improved in order to provide numerical expressions of the different system variables, and, to give the expression of the passive heat-pipe loop response time as function of the various parameters. A model can also give a guide to the design the current passive heat-pipe loop. The primary body investigates the section of the evaporator wall. It is worth to mention that the primary body can be classified as a thermally thin body investigated by a temperature (T_w). The second body was connected to the whole working fluid which contacts with mutually evaporator wall and working fluid. The working fluid was classified as saturated temperature (T_f), as shown in “Figure 1”.

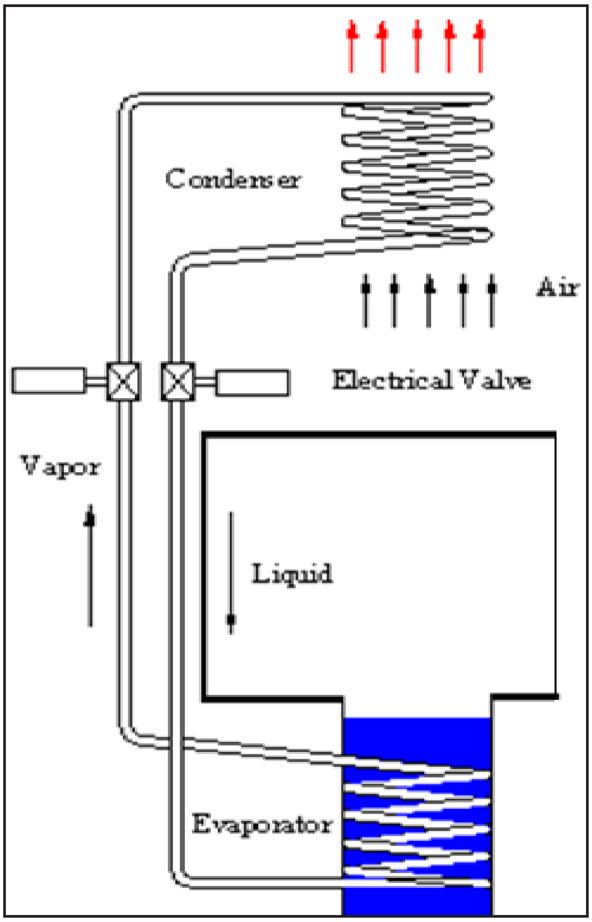

Fig. (1): Schematic of proposed passive heat-pipe loops cooling system.

This model was investigates as a theoretical study of heat-pipe loop behavior in transient regime. In this model, from the transient to steady-state, the thermal behavior and thermal characteristics of the heat-pipe loop has been utilized to obtain a mathematical expression of the system response and performance.

A program depend on the simulation technique was developed to calculate the mean temperature of the heat-pipe loop in addition to the time required to reach ideal condition. This program could be used as a simple tool for modeling and designing heat-pipe loop in steady-state and transient regime. The heat balance calculations for each body (wall and fluid) give:

$$C_w \frac{\partial T_w}{\partial t} = Q_s - h_e.S_e.(T_w - T_f) \qquad (1)$$

$$C_f \frac{\partial T_f}{\partial t} = h_e - h_e.S_e.(T_w - T_f) - h_c.S_c(T_f - T_{wat}) \qquad (2)$$

From “Eq. (1)”and “Eq. (2)”, by using the finite difference (Euler method) in certain time step Δt, we obtain Tw and Tf

$$T_w^{n+1}(C_w + h_e.S_e.\Delta t) - .T_f^{n+1}(h_e.S_e.\Delta t) = C_w.T_w^n + \Delta t.Q_s \qquad (3)$$

$$T_f^{n+1}(C_f + \Delta t.h_e.S_e + \Delta t.h_c.S_c) - \Delta t.h_e.S_e.T_w^n = C_f.T_f^n + \Delta t.h_c.S_c.T_{wat} \qquad (4)$$

Radial heat flux, qr in evaporator section is given as:

qr = Qnet /Ar = Qnet/ (π * di * Le) (5)

Axial heat flux qax of heat-pipe is given as:

qax = Qnet / Ac.s = Qnet / (π * di2 / 4) (6)

Time constant

From observations of various operating variables of the heat-pipe, can be noticed that many nonlinear physical phenomena occur inside it (phase change, counter current flows . . . etc.), and the values of variables can be well fitted by a simple “one exponential law”:

$$X(t) = X_s\left(1 - e^{\frac{-(t-t_0)}{\tau_h}}\right) \qquad (7)$$

(τh) is the time constant corresponding to the variation of the variable X in the heat-up phase.

Where: Xss: variable Value at steady state.

Average evaporator heat transfer coefficient, he

The heat transfer process in the pool of the heat pipe evaporator section is generally assumed to be nucleate boiling, and the heat transfer coefficient may be calculated from Forester–Zuber by **Whalley (2002)**

$$h_e = \frac{0.00122 * \Delta T_{sat}^{0.24} * \Delta P_{sat}^{0.75} * CP_l^{0.454} * \rho_l^{0.45} * K_l^{0.79}}{\sigma^{0.5} * h_{fg}^{0.24} * \mu_l^{0.29} * \rho_g^{0.24}} \quad (8)$$

Average condenser jacket heat transfer coefficient

The heat-transfer for natural convection on the condenser surface depends on: the condenser geometry; its location; the temperature difference on the external surface, and the properties of the thermo-physical of the fluid involved **(Cengel, 2002)**. The average Nusselt number (Nu) correlation is treated as natural convection by using the following relation:

$$Gr_L = \frac{g.\beta.(T_s - T_\infty)L_c^3}{\gamma^2} \quad (9)$$

The Characteristic length of the pipe is its diameter D.

$$Nu = \frac{h..D}{k} = C(Gr_D.\Pr)^n = C.Ra._D^n \quad (10)$$

The constant values of n and C depend on the flow and surface configuration, which is characterized by the variety of the Rayleigh number. For laminar flow the n value is (1/4), and (1/3) for turbulent flow, respectively. The constant C value is less than (1). RaD is the Rayleigh number **(Anjakar *et al.*, 2012)** and we have:

$$Ra_D = Gr_L \Pr = \frac{g.\beta.(T_s - T_\infty)}{\gamma^2} pr \quad (11)$$

The Empirical correlation for the average (Nu) for natural convection is expressed as:

$$Nu = \left[0.6 + \frac{0.387 Ra_D^{1/6}}{\left[1 + (0.559/\Pr)^{9/16}\right]^{8/27}}\right]^2 \quad (12)$$

Where Nusselt is used to calculate convective heat transfer coefficient as below:

$$Nu = \frac{h.D}{k} \quad (13)$$

The heat transfer rate Q_{conv} by natural convection from a uniform solid surface at a uniform temperature Ts to the surrounding fluid is known by Newton's law of cooling, where As is the heat transfer surface area, and h is the average heat transfer coefficient on the surface when the (Nu) and average convection coefficient are known.

$$Q_{conv} = hA_s(T_s - T_\infty) \quad (14)$$

$$A_s = \pi.D.L \quad (15)$$

$$U = \left(\frac{1}{h_i} + \frac{t}{k_p} + \frac{1}{h_o}\right)^{(-1)} \quad (16)$$

Program description

In this part, the proposed mathematical model will be simulated. A simulation program was developed to estimate the mean temperature of heat-pipe loop, performance and thermal characteristics as well as the time to reach steady state condition. The equations of the model are solved by Engineering Equation Solver program (EES) **(Klein, 2001)**. The developed program comprises three main sections.

The first section contains the initial conditions such as the ambient temperature, the basic heat-pipe loop dimensions, material and configuration, and evaporator heat input rate. As input data, these parameters are combined to calculate the physical properties of the working fluid and heat-pipe for each section.

The second section of program activates the program at the saturation temperature of the fluid. This means the transient calculation of the two-phase is started. In this section, the average heat-pipe mean wall, working fluid temperature and heat-pipe characteristics for start-up and steady process are calculated.

In the third section, replacement of the initial conditions is achieved by the new calculated data. The program progresses until a steady-state is reached. "Table (1)." represents the ranges program of the tests.

Table (1) : *Ranges program of the tests.*

Heat-pipe material	Studied parameter	Heat load [Kw]	Ambient temperature [°C]	filing ratio [%]	Condenser length L_c. [m]	Heat-pipe outer diameter. [mm]
Stainless-AISI316	Ambient temperatures	150	15-20-25-30-35-40	100%	100	150

DISCUSSION OF THE ANALYTICAL MODELING RESULTS

Transient start-up operation

Transient test operation is performed at start-up operation and steady-state for loop for nuclear fuel storage tank. These measurements involve the change of wall and fluid temperatures during the start-up transient operation. Prior to start-up, the loop is initially set at (t = 0 s), then the power input to the evaporator is sudden started. The program is running for each interval of time Δt till the loop temperature reach the steady state values. "Figure 2." shows the average heat-pipe evaporator wall and working fluid temperatures.

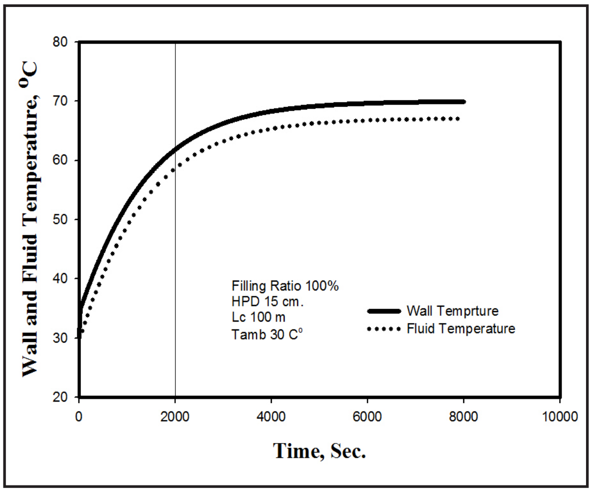

Fig. (2): Wall and fluid temperature at 150 kW.

The evaporator and condenser heat transfer coefficient versus time at power 150kW and 100% filling ratio was shown in "Figure 3."

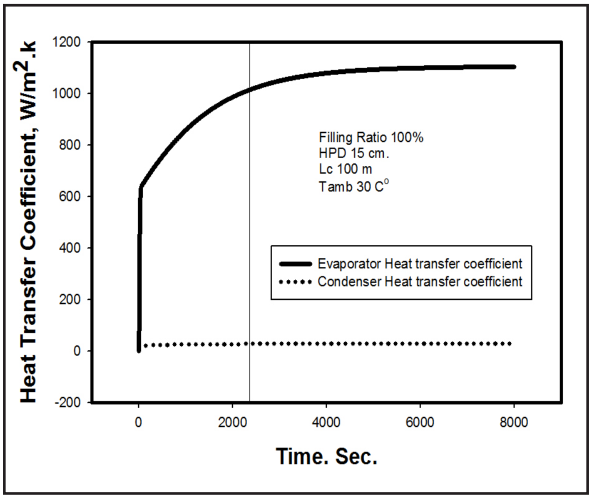

Fig. (3): Average evaporator and condenser heat transfer coefficient at 150 kW.

In the current theoretical tests, shown in the mentioned figures, for the loop the time required to reach the steady state operation is about 2000 seconds. "Figure 2." indicates the increment of average wall and fluid temperatures with time. The phenomenon is divided into two regions. In the first region, the vapor density is too low to support a continuous flow. Heat gained by the heat-pipe loop evaporator is absorbed solely as sensible heating, resulting in temperature rise. As a result, the temperature gradient of evaporator section is considered relatively high in the first interval of heating (t = 0-2000 sec.). While the rest of the heat energy forms some vapor which flows from the evaporator and condenses on the beginning section of the condenser section causing its surface temperature to rise. In this period of time, the response of condenser section is lower than the evaporator section. At the time range (t >2000 sec.), majority of the heat energy is absorbed as latent heat in working fluid increasing thus the generated vapor. The vapor temperature is high enough to sustain the continuous flow. Finally, as the steady state is approached, the rate of temperature increase slows down. This is because of the reduction in temperature difference between vapor and working fluid.

The pool temperature decay at different ambient temperatures

The effect of heat load when sudden heat generation is stopped (imagining Case) was investigated for heat load 150 kW and ambient temperature ranged from 15 to 40°C. "Figure 4" shows the temperature gradient decay along the time inside the nuclear fuel storage tank using distilled water as a working fluid. From this figure it can be noticed that the storage pool temperature readings decayed along approximately two hours. It is clear that the temperature distribution curves tend to have different trend depending on the ambient temperature. For the storage pool which having the heat-pipe loop evaporator, the lowest temperature distribution decay occurs at 15°C, and the highest temperature distribution decay occurs at 40°C.

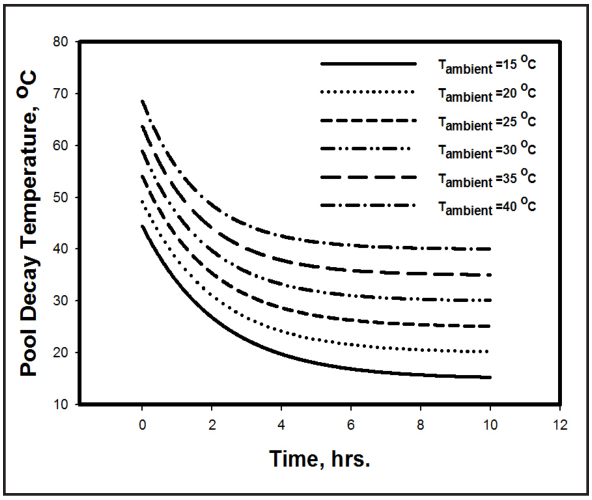

Fig. (4): The spent fuel storage tank temperature decay at different ambient temperature.

The cooling process is done according to Lumped Capacitance Method, it is assumed that the water in the pool is at temperature Ti, and is cooled naturally to lower temperature T_∞. This reduction can be explained by the convection heat transfer at the liquid-solid interface. The equation of heat is a differential equation governing the spatial temperature distribution due to transient decay. Instead, the transient temperature response is determined by formulating an overall the balance of energy on the heat pipe condenser evaporator portion. This balance must relate the heat loss rate of the surface to change rate of the internal energy in the auxiliary pool tank. For this case, the conservation requirement becomes:

$$E_{in} - E_{out} = 0 \tag{17}$$

Even though energy generation may be occurring in the medium, the process would not affect the energy balance at the control surface. Moreover, for both transient and steady-state conditions, three heat transfer terms are shown for the control surface. On a unit area basis, the pre-mention terms are conduction between the middle and the control surface convection from the external surface to a fluid and net radiation exchange from the surface to the surroundings.

The balance of energy takes the form:

$$q_{cond} - q_{conv} - q_{rad} = 0 \tag{18}$$

The radiation term is neglected, so the equation is reduced to:

$$\rho.(V)\frac{dt}{dT} = U.A.(T - T_\infty) \tag{19}$$

Heat-pipe performance and thermal characteristics

At certain value 150kW of thermal load, "Figure 5 and 6" illustrates the theoretical mean temperatures predicted by the mathematical model for average evaporator wall and fluid temperatures as predicted from "Eq. (3)," and "Eq. (4),".

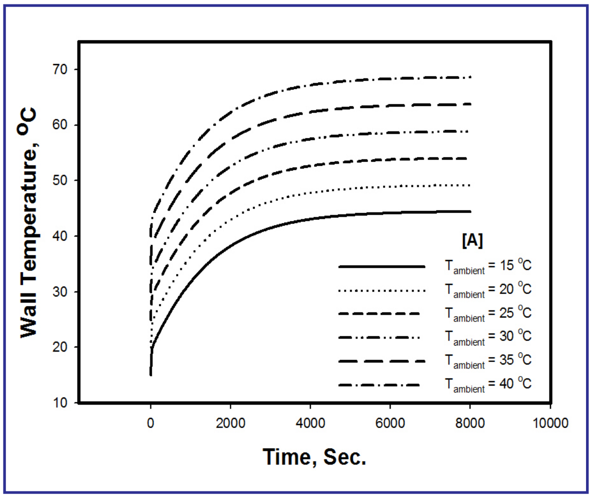

Fig. (5): Wall temperature of heat-pipe at different ambient temperature.

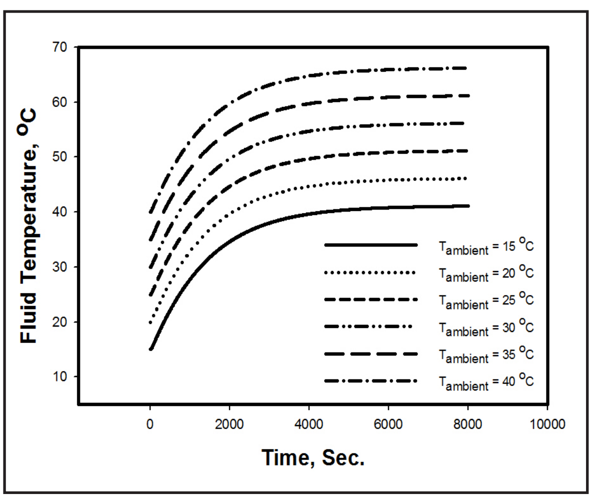

Fig. (6): Fluid temperature at different ambient temperature.

As result of increasing of the ambient temperature, the rate of change temperature with time increased until the steady state is reached and increases as the ambient temperatures become higher. The intensity of heat transfers within the loop is analyzed through the determination of the average heat transfer coefficients (he) and (hc) in evaporator and condenser that estimated for the start-up and steady-state conditions by means of the mathematical model / "Eq. (8)," for evaporator and "Eq. (14)," for condenser, respectively. The calculations are carried out for different thermo-physical properties of working fluid including two subsequent processes, heat-up transient and steady-state, as shown in "Fig. 5-C and D". The estimation is carried out in the case of different ambient temperature ranged from 15 to 40 °C and high values of filling ratio 100%, meaning that the evaporator is fully filled. At the condenser section, a global heat transfer coefficient (hc) has been considered, which combines conduction through the wall and convection (external side of the wall-air) (the cooling section) "Figure 6 and 7" estimated by the mathematical model "Eq. (14),"

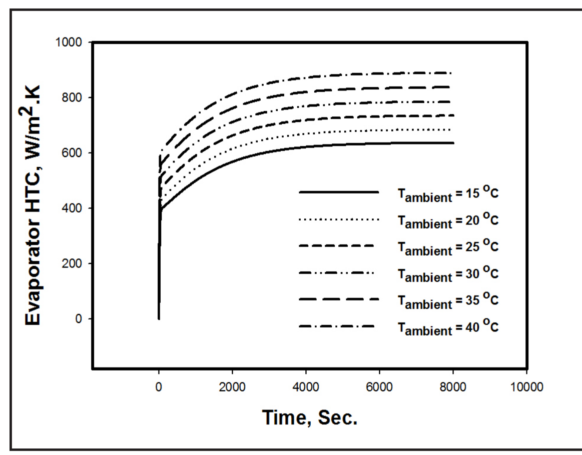

Fig. (6): Average evaporator heat transfer coefficient at different ambient temperature.

It shows that a small change is noticed in the condenser heat transfer coefficient during all processes of the operation for the two processes, heat-up transient and steady-state; on the other hand, evaporator heat transfer coefficient is highly changed with the ambient temperature change as shown in "Figure 8".

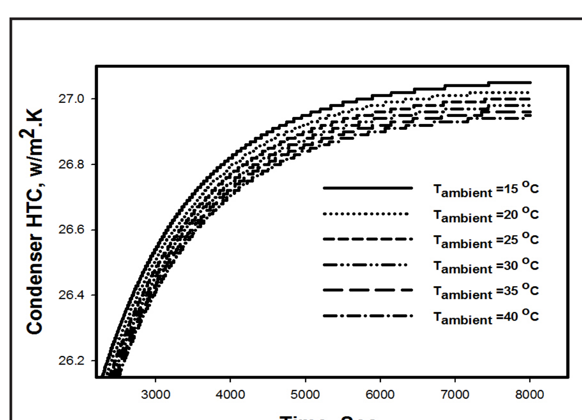

Fig. (7): Average condenser heat transfer coefficient at different ambient temperature.

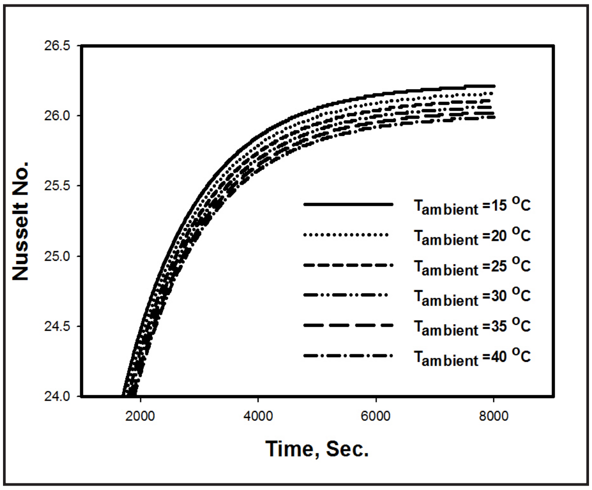

Fig. (8): Average Nusselt No.at different ambient temperature.

Also, the power output from condenser section versus time is observed. The loop is initially at room temperature, and then the power input to the evaporator is increased suddenly from zero to the full power. It was shown from "Figure 9" that the temperatures of each section and the output power increase rapidly at the beginning of operation due to the increase of heat flow from object to another, with time. But as a result of reduction of the temperature driving forces, the rate of change temperature with time decreases until steady state condition is reached. At steady-state, the wall and fluid temperatures, condenser and evaporator heat transfer coefficients and the power-output remain constant.

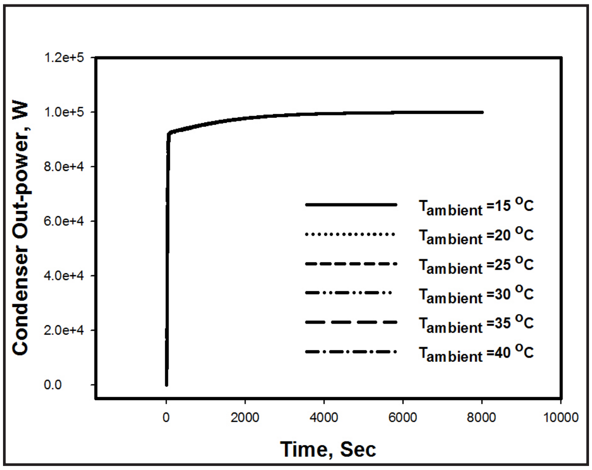

Fig. (9): Average condenser output power at different ambient temperature.

CONCLUSION

A numerical simulation using special design of Stainless-AISI316 heat-pipe loops with outer diameter 150mm, 3mm thickness and 100m length, working with pure water at heat load 150 kW, evaporator filling ratio 100% and different ambient temperature ranged by 15-20-25-30-35 and 40 °C were used to investigate the thermal performance and characteristics wall and fluid temperatures, coefficients of heat transfer and time constant.

The effects of atmospheric air temperature that circulated around the condenser as cooling were analyzed. The results reveal that the best thermal performance was obtained at lower ambient temperature, the evaporator heat transfer coefficient were increased as the ambient temperature becomes higher, while the condenser heat transfer coefficients were found slightly increasing. The current design of the heat pipe passive cooling system was able to remove 150 kW within safe conditions.

In the future plan the proposed model and its simulation can be modified by using other heat-pipe materials to dissipate more heat from nuclear spent fuel in emergency regime to attain the temperature within safe limits,

REFERENCES

- **Alizadehdakhel, A.; Rahimi, M. and Alsairafi, A. (2010):** CFD modeling of flow and heat transfer in a thermosyphon, *Int. Commun. Heat Mass Transf.*, 37: 312.

- **Advisory Group meeting held in Julich, Germany (1994):** Technical feasibility and reliability of passive safety systems for nuclear power plants. *Proceedings of an Advisory Group meeting held in Julien, Germany.* https://www-pub.iaea.org/MTCD/Publications/PDF/te_920_web.pdf.

- **Jackel, B.S. (2015):** Status of the spent fuel in the reactor buildings of Fukushima Daiichi. *Nucl. Eng*., 283: 2.
- **Ye, C.; Zheng, M.; Wang, M.; Zhang, R. and Xiong Z. (2013):** The design and simulation of a new spent fuel pool passive cooling system. *Ann. Nucl. Energy*, 58:124.
- **Reay, D.; McGlen, R. and Kew, P. (2013):** Theory, Design and Applications, Butterworth-Heinemann, Oxford, UK.
- **Throm, E.D. (1991):** Beyond design basis accidents in spent-fuel pools dgeneric Issue 82. *Nucl. Eng*., 126: 333.
- **International Nuclear Safety Advisory Group (1999):** Reported the safety principles for nuclear power plants 75-*INSAG-3 rev.1, INSAG-12*.
- **IAEA Safety Standards:** for protecting people and the environment Specific Safety Guide International Atomic Energy Agencyviennaisbn, *Design of Electrical Power Systems for Nuclear Power Plants* 978–92 –0–109314–1ISSN 1020–525XNo. SSG-34.
- **Shouman, L.A.; Abd El-Aziz M. and Alyan, A. (2013):** Theoretical Steady-state Forced Flow Boiling Evaporator for Double-tube Thermosyphon (FFDTT). *Int. J. Sci. & Eng. Res.*. 4(6): 2646.
- **Whalley, P.B (1989):** Boiling Condensation and Gas–Liquid Flow, Department of Engineering Science, University of Oxford, Heat transfer by Yunus A. Cengel, 2nd eddition, NewYork: Mc Graw-Hill.
- **Anjakar, P.G. and Yarasu, D.R.B. (2012):** Experimental analysis of condenser length effect on the performance of thermosyphon. *Int. J. Emerg. Technol. Adv. Eng*., 2: 494.
- **Klein, S.A. and Alvardo, F.L. (2001):** EES-Engineering Equation Solver; user's Manual for Microsoft Windows Operating Systems, Version 6.160, F-Chart Software, Madison, WI.
- **Noie, S.H. (2005):** Heat transfer characteristics of a two-phase closed thermosyphon. *Appl. Therm. Eng*., 25:495
- **Tokyo Electric Power Company, Inc. (2012):** Fukushima Nuclear Accident Analysis Report.
- **Sailor, V.L; Perkins, K.R.; Weeks, J.R. and Connell, H.R. (1987):** Severe accidents in spent fuel pools in support of generic safety, Issue 82, Brookhaven National Lab., Upton, NY (USA), Nuclear Regulatory Commission, Washington, DC (USA). Division of Reactor and Plant Systems.
- **Ognerubov, V. and Kaliatka, A.; Vileiniskis, V. (2014):** Features of modeling of processes in spent fuel pools using various system codes, *Ann. Nucl. Energy,* 72: 497.
- **Wu, X.; Li, W.; Zhang, Y.; and Tian, G. (2014):** Analysis of the loss of pool cooling accident in a PWR spent fuel pool with MAAP5. *Ann. Nucl. Energy,* 72: 198.
- **Cengel, Y.A. (2002):** Heat Transfer: A Practical Approach by Hardcover. 2nd edition. Mcgraw-Hill (Tx).