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A NEW PROPOSAL FOR THE DESIGN AND MODELING OF PASSIVE SPENT FUEL AUXILIARY POOL COOLING SYSTEM

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Abstract: Spent nuclear fuel at the spent fuel storage in the research reactors must be in the safe temperature conditions. Utilizing heat–pipe as stand–by cooling system is a suitable method to remove heat generated from the fuel at the storage in the reactors, which considered the second grade of the passive systems from four grads scale. Heat–pipes are considered as passive heat transfer appliances which, when well designed and fabricated, have very long live spans when operating with in temperature limits. In This article, a theoretical that new design and completely passive cooling system using gravity assisted closed two–phase heat–pipe for dissipation the heat from 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 section of the heat–pipe to confine the residual heat. Focus on heat–pipe configuration effect, evaporator and condenser lengths ranges (75, 100, 125 and 150m) and outer diameter ranges was (15, 20 and 25 cm) with 3mm thickness at heat load 150 kW and full filing ratio at evaporator section and ambient air temperature at 30˚C, a numerical simulation using a new design of gravity assisted heat–pipe loops was used to investigate the heat–pipe thermal characteristics and performance will analysed. The heat–pipe material is stainless steel and demineralized water was used as the running working fluid. The observation from the results showed that the best performance and thermal characteristics were obtained at a higher heat–pipe length and diameter that the heat pipe could remove 150kW with safe conditions. The computer simulation can then be used to predict the concepts of heat transfer with different inputs

Keywords: safe temperature conditions, passive systems, heat–pipe as stand–by cooling system

INTRODUCTION

Multi–purposes reactors (MPR) after shutdown have residual heat. Accordingly, in case of emergency and after nuclear reactor shut down, heat should be removed to keep fuel temperature within safe limit. Also, MPR have an auxiliary spent fuel tank that should be cooled also to remove the heat generated. In this direction, heat–pipes are suggested to be inactive heat energy transport systems. With appropriate design and made–up, they are considered to be expatriate sealed tubes which have very long lives.

A completely passive cooling system implementing loop heat pipe for cooling and dissipating the nuclear reactor's residual heat is suggest in the current study. The most important parameters that determine if the public will accept nuclear reactors plant are safety features. The only thing which public can imagine is destruction or radioactive hazards on human health whenever they heard about the word nuclear [1].

Safety regulations have been developed, matured, and implemented in the last decades in a very restricted way. Consequently, the accidents risk in nuclear power facilities is declining [2]. There are three serious accidents only at nuclear power plants over 16,210 cumulative reactor of commercial runs in 33 countries: No adverse health or environmental consequences occurred in the Three Mile Island (USA 1979) in which the reactor was completely destroyed.

Significant environmental and healthiness consequences happened in Chernobyl (Ukraine 1986), where the vapor bang and bonfire destruction of the reactor. Since then, the deaths have increased to 56 after 31 people are killed. Fukushima (Japan 2011) where three reactors from a fourth were suffering 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 face from contact to other sources [3].

The design of nuclear power stations are directed to be in safe hands in functioning 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 [4].

Because of the unbalancing between passive and active safety systems, the three previous mentioned nuclear reactors accidents were occurred [5]. Using passive component system and/or the lack of human action step down power failure. Safety will increase in direct proportion with increasing passive safety systems. Thus, passive systems will improve the overall factor of safety for the Nuclear power plant [6]. 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 [7]. Passive systems must obey some conditions to be accepted: Reliability and availability in short terms, long terms, and under retrograde conditions [8].The accident at the Daiichi nuclear plant proposed reducing credence on active systems to reduce human error factor.

Passive systems types are classified to different categories: Grade A, B, C and D Grade B: no signal input, external power source, no moving mechanical parts but moving working fluids.[9] and [10]. 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 pure conduction through solid metal. It is more than 200–500 that of copper [11].

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 evaporator. 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 evaporator 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 [12].

A suggested new design of heat–pipe loop with evaporator and condenser helical configuration with 3 mm 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 figure 1.

The design is focused on removing heat from the spent fuel tank of research reactor to be in safely operated. The model considers convection by air naturally for the condensing section of the heat–pipe loop to compensate the residual heat. A numerical model using unique design of heat–pipe loops were investigated the heat transfer [13].

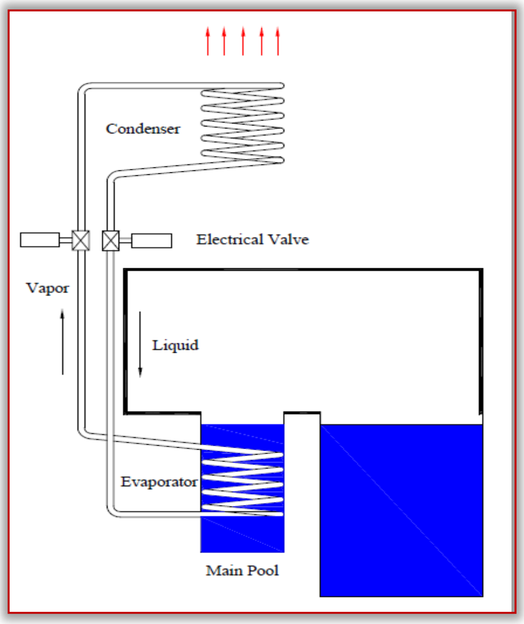

Figure 1. Schematic of Proposed Passive heat–pipe Loops Cooling System

The effects of heat–pipe configuration (evaporator and condenser) were analyzed. Demineralized water was used as the working fluid. The atmospheric air temperature equals (30°C) and circulated around the condensing section as a cooling procedure. Heat input of (Q ≤ 150kW) and working fluid filling ratio (100%) was taken in account. Thermal performance and characteristics was investigated at different heat–pipe lengths (75, 100, 125 and 150m) and diameters (15, 20 and 25cm).

Transient experiments were performed using a water heat pipe having a uniformly heated boiling part and a convectively cooled condenser section [14].

Transient response of the heat pipe to sudden changes of the input electric power to the evaporator heater, at different cooling rates, was investigated [15]. The heat pipe has dimensions according to the followings, 39.3 cm long boiling part, 3.7 cm long adiabatic part, and 17 cm long condensing part. The time constants for the temperature of vapor and the effective throughput power during the start heat–up and shut down cooling transients were estimated depending on the electric power input and the cooling water mass flow rate.

Anhar R. Antariksawan [16] studied heat pipe performance in the reactor cavity cooling system RCCS design. As part of the study, the performance of U–shaped heat pipe is assessed both through an experiment and numerical simulation methodology. A small U–shaped heat pipe with fins in the condensing section is modeled using the RELAP5/SCDAP code.

The result predicted from model illustrates that the heat pipe could remove the received heat in the boiling section to the environment through the condensing part of that heat pipe using the phenomena of boiling and condensation. However, the steady state condition could not be fulfilled until the end of the calculation. The predicted thermal resistance of the heat pipe is approximately 0.012580°C/W. It is concluded that the model could model reasonably the performance of the heat pipe.

MODEL ASSUMPTIONS

- A one–dimensional flow models.
- The heat–pipe loop is in the vertical orientation.
- The vapor superheat is very small; the vapor is taken at the saturated conditions.
- Constant wall material (Stainless–AISI316) properties, such as (density, specific heat and thermal conductivity).
- The kinetic and potential energy components are neglected in the energy balance equations when compared with heat transfer rate.
- The density, thermal conductivity, enthalpy and other properties of saturated liquid are temperature depended.
- The local wall, working fluid and heat transfer coefficient of the evaporator and the condenser are calculated at mean value for both.
- The heat–pipe starts up from initial condition when the power is suddenly on.

EQUATIONS OF THE MODEL

A model describing both thermal and phase flows of the heat–pipe loop has been performed by M. Abdelaziz, et al [14]. This simpler model has been improved on the one hand in order to provide numerical expressions of the different system variables, and, on the other side, to give the expression of the heat–pipe loop response time as function of the various parameters. Such a model can also give a guide to the design of heat pipe loop.

The primary body investigates 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). This model investigates a theoretical study of heat–pipe loop behaviour in transient rule.

In this model, the transient thermal behaviour of the heat–pipe loop has been utilized so as to obtain a mathematical expression of the system response. A program depend on the simulation technique was developed to calculate temperature of the heat–pipe loop in addition to the time required to reach ideal condition. This program can be known as a simple tool for modelling and designing heat–pipe loop in transient rule. The heat balance calculations for each body (wall and fluid) give:

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

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

From equation (1) and by using the finite difference method in certain time step Δt, we obtain T_w.

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

The average fluid temperature from equation (2) and by using the finite difference method we obtain T_f

$$T_f^{n+1}(C_f + \Delta t.h_{e.}.S_e + \Delta t.h_{c.}.S_c) - \Delta t.h_{e.}.S_e.T_W^n$$
$$= C_{f.}T_f^n + \Delta t.h_{c.}.S_c.T_{wat} \qquad (4)$$

where:

T_w: is the average evaporator wall temperature.

T_f: is the average fluid temperature.

S_e: is the surface area of the evaporator = π*D*L_e

S_c: is the surface area of the condenser = π*D*L_c

Radial heat flux, q_r in evaporator is given as:

$$q_r = Q_{net}/A_r = Q_{net}/(\pi * di * Le) \qquad (5)$$

Axial heat flux q_{ax} of heat–pipe is given as:

$$q_{ax} = Q_{net}/A_{cs} = Q_{net} / (\pi * di^2/4) \qquad (6)$$

Time constant

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

$$x(t) = x_{ss}\left(1 - e^{\frac{-(t-t_0)}{\tau_h}}\right) \qquad (7)$$

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

Where: X_{ss}: variable Value at steady state.

Average evaporator heat transfer coefficient

The process of the heat transfer in the liquid pool of the evaporator section is generally assumed to be common nucleate boiling whose heat transfer coefficient may be calculated from Forester–Zuber equation [17]:

$$h_e = \frac{0.00122 * \Delta T_{sat}^{0.24} * \Delta P_{sat}^{0.75} * cp^{0.45} * \rho_l^{0.49} * k_l^{0.79}}{\sigma^{0.5} * h_{fg}^{0.24} * \mu_l^{0.29} * \rho_v^{0.24}} \quad (8)$$

Average condenser heat transfer coefficient

Natural convection heat transfer on a external surface depends on the geometry in addition to its location. Moreover it depends on the disparity of temperature on the external surface and the properties of the thermo–physical of the fluid involved [18]. The correlations for the average Nusselt number Nu in natural convection are of the form:

$$Gr_L = \frac{g * \beta * (T_s - T_\infty).L_{cr}^3}{\nu^2} \quad (9)$$

where β is Coefficient of volume expansion and equal (1/T) and T is temperature in Kelvin.

The Characteristic length of the pipe is its diameter D.

$$Nu = \frac{h.L_{cr}}{\nu^2} = C(Gr_L * Pr)^n = C * Ra_L^n \quad (10)$$

The values of the constants C and n depend on the geometry of the surface and the flow rule, which is characterized by the variety of the Rayleigh number.

The value of n is (1/4) for laminar flow and (1/3) for turbulent flow. The value of the constant C is normally less than 1.

Where Ra_D is the Rayleigh number, which is the product of the Grashof and Prandtl numbers:

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

The Empirical correlation for the average Nusselt number for natural convection over horizontal cylinder is expressed as:

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

When the average convection coefficient and Nusselt number are known, the heat transfer rate Q_{conv} by natural convection from a solid surface at a uniform temperature T_s to the surrounding fluid is known by Newton's law of cooling.

A_s is the heat transfer surface area and h is the average heat transfer coefficient on the surface [19].

$$Q_{conv} = h.A_s(T_s - T_\infty) \quad (13)$$

where A is condenser surface area and expressed as:

A_s=π.D.L

$$U = \left(\frac{1}{h_i} + \frac{t}{k} + \frac{1}{h_o}\right)^{-1} \quad (14)$$

PROGRAM DESCRIPTION

According to ranges program illustrated in table 1. A mathematical program is designed to solve the presented model. A computer simulation program based on the method of finite difference (Euler) is developed to estimate temperature of heat–pipe loop as well as the time needed to reach steady state condition, also thermal performance and thermal characteristics. The equations are solved by Engineering Equation Solver program (EES) [20].

Table 1. Ranges program of the tests

Heat–pipe material	Stainless–AISI316	Studied parameters	Heat–pipe configuration
Evaporator filing ratio [%]	100	Condenser and evaporator lengths [m]	75, 100, 125 and 150
Ambient temperature [°C]	30	Heat–pipe diameter. [mm]	150, 200 and 250
Heat load [Kw]	150	Material thick. [mm]	3

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, cooling flow rate as an 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 is activated the program at the saturation temperature of the fluid. This means the transient calculation of the two–phase is started. In this section the mean wall, fluid, evaporator and condenser heat transfer coefficient and cooling flow rate temperature 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

TRENDS AND 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 results involve the change of wall and fluid temperatures during the start–up transient operation. Prior to start–up the loop is initially (t = 0 s), then the power input to the evaporator is sudden started. The program is running for each interval of time Δt defined by 4 second till the loop temperature reach the steady state values.

Figure 2 and figure 3 show the average heat–pipe evaporator wall and working fluid temperatures at different evaporator and condenser lengths ranging (75, 100, 125, and 150m) and at different heat–pipe diameters ranging (15, 20 and 25 cm). It was noticed that the best conditions are at evaporator and condenser lengths at 150 m and at 25cm heat–pipe outer diameter.

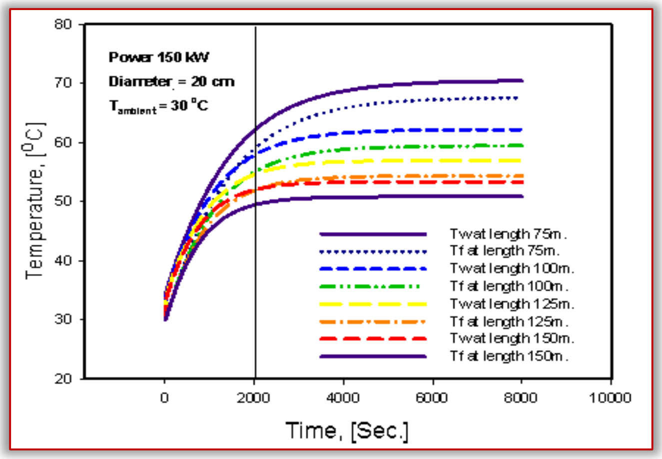

Figure 2. Average wall and fluid temperature at different evaporator lengths.

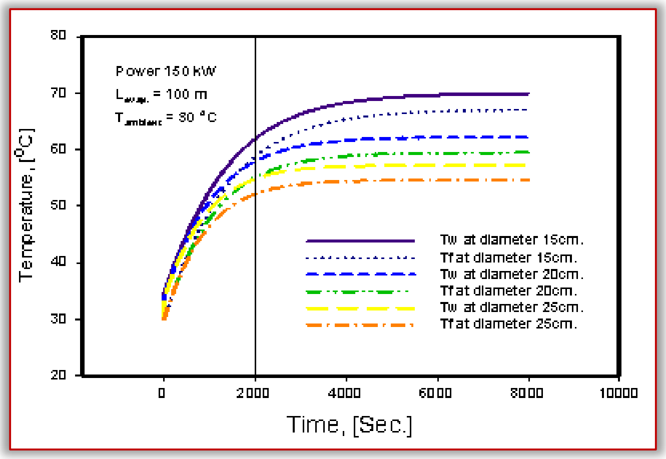

Figure 3. Average wall and fluid temperature at different heat–pipe diameter.

In the current theoretical tests, shown in the above figures, for the loop the time required to reach the steady state operation is about 2000 seconds. The figures indicate the increment of average wall and fluid temperatures with time. The phenomenon is divided into two regions. In the first region, the vapour density is too low to support continuum flow.

Heat gained by the heat–pipe 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 vapour 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.) the majority heat energy is absorbed as latent heat in working fluid thus, increasing the generated vapour. The vapour temperature is enough to sustain continuum flow. Finally, as the steady state is approached, the rate of temperature increase slows down. This is due to the reduction in temperature difference between vapour and working fluid.

Heat–pipe performance and thermal characteristics

At certain value 150kW of thermal load, Figure 4 and figure 5 illustrates the theoretical average evaporator heat transfer confidents. The estimation is carried out in the case of different heat–pipe lengths ranged from 75 to 150m and diameters ranged from 15 to 25cm. It was shown that the evaporator and condenser heat transfer confidents are reverse proportional with heat–pipe lengths and diameters.

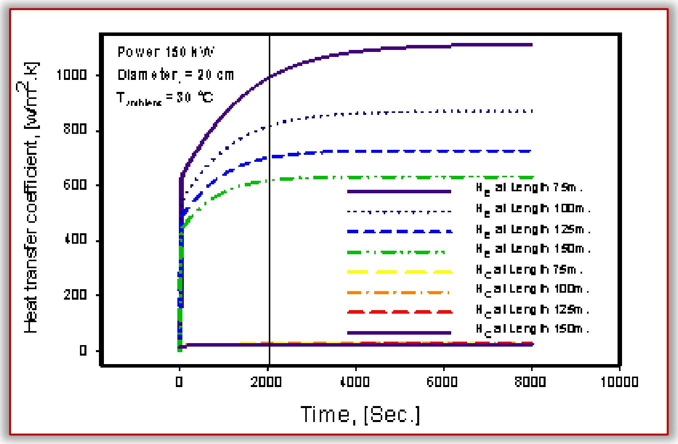

Figure 4.Average evaporator and condenser heat–transfer coefficients at different heat–pipe lengths.

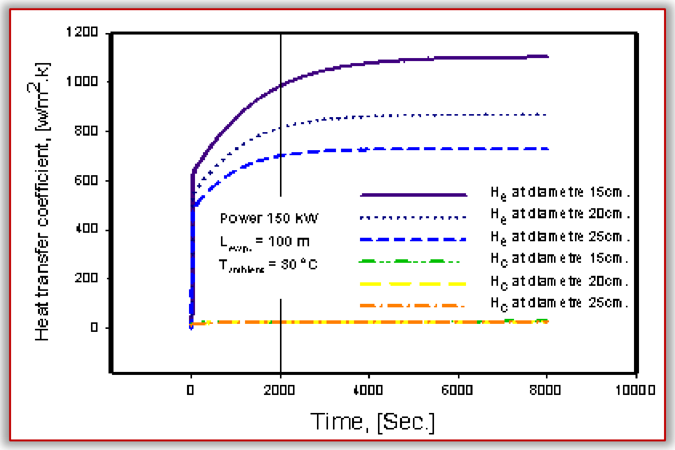

Figure 5. Average evaporator and condenser heat transfer Coefficients at different heat–pipe diameters

In 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 figure 7 estimated by the mathematical model Equation (14), show that there little change at 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 heat–pipe configuration change .

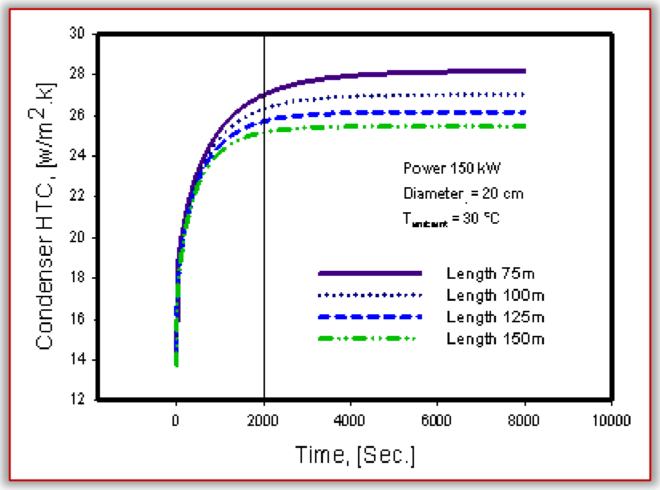

Figure 6. Average condenser heat transfer coefficient at different heat–pipe lengths

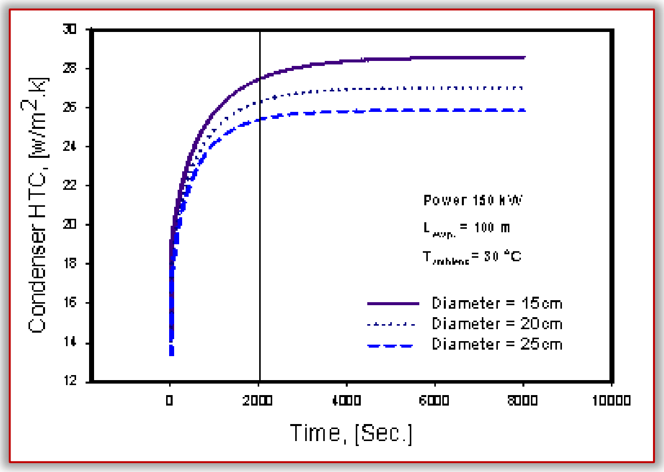

Figure 7. Average condenser heat transfer coefficient at different heat–pipe diameters

Also, the power output from condenser section respectively 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 8 and 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 power–output remain constant.

From the previous figures it was shown that the time constant needed to reach steady–state is directional proportional with heat–pipe lengths and diameters due to increasing in thermal inertia.

From the results the heat–pipe loop as passive cooling system could remove heat load 150 kW with approximately safe condition especially at heat–pipe configuration at evaporator and condenser length 150m and heat–pipe diameter at 25cm.

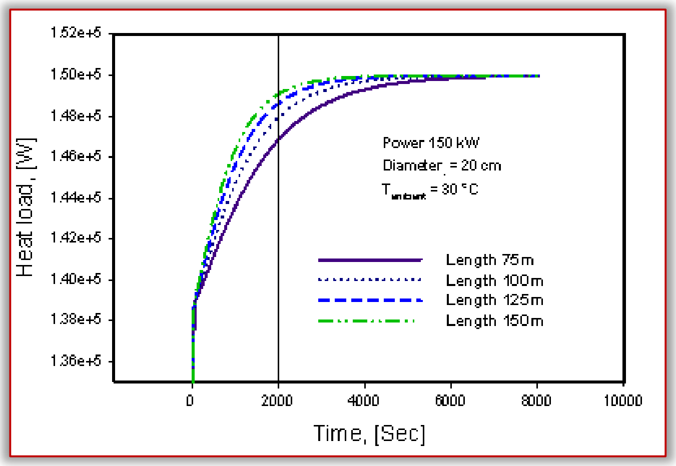

Figure 8. output load from condense section at different heat–pipe lengths

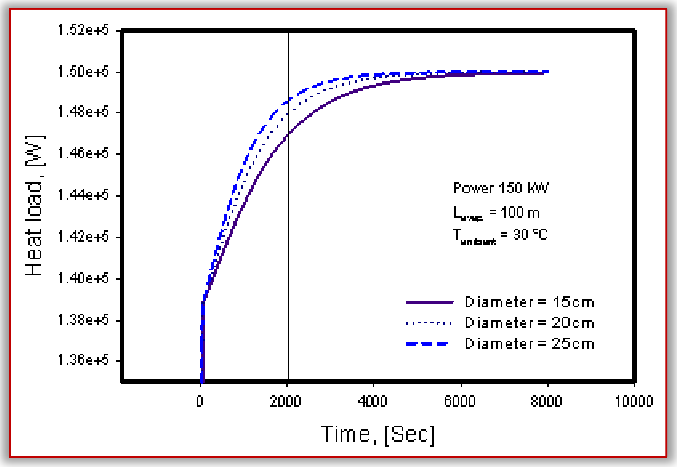

Figure 9. output load from condense section at different heat–pipe diameters

CONCLUSION

A theoretical network model has been proposed to predict the transient response of a heat–pipe loop. A numerical simulation using special design of Stainless–AISI316 heat–pipe loops of passive spent nuclear fuel cooling system used to investigate the thermal performance and characteristics at constant heat–load 150 kW and 30°C ambient temperatures.

Demineralized water was used as the working fluid. The atmospheric air was circulated around the condenser as a cooling system. The effects of heat–pipe configuration (length and diameter) were analysed. The results show that the best thermal characteristic was obtained at higher heat–pipe length and diameter.

The numerical simulation refers to a pattern, and a trend line can be used to predict the phenomena of heat transfer with different inputs. It was observed that heat pipe's transient response was found to be depending primarily upon heat–pipe configuration. Decreasing at the heat–pipe length and diameter causes a reduction in time constants.

Also it was found that the evaporator heat transfer coefficient was increased with heat–pipe configuration decreased, while the condenser heat transfer coefficients were found slightly changed. From the previous figures the heat–pipe able to remove heat load at 150 kW.

NOMENCLATURE

Ac	[m^2]	Cross section area
cp	[J/kg. K]	Specific heat
D	[m]	Diameter
g	[m/s^2]	Acceleration gravity
h	[W/m^2 .K]	Heat transfer coefficient
h_{fg}	J/kg	Latent heat of vaporization
k	[W/mK]	Thermal conductivity
p	[N/m^2]	Pressure
S	[m^2]	Surface area
T	[K]	Temperature
t	[s]	Time
Gr		Grashof number
Nu		Nusselt number
Pr		Prandtl number
Re		Reynolds number
Ra		rayleigh number
X		Any variable
Special characters		
β	[–]	Coefficient of volume expansion
μ	[N.s/m^2]	Dynamic viscosity
ρ	[Kg/m^3]	Density
τ	[sec]	Time constant
ν	[m^2/s]	Kinematic viscosity
Subscripts		
c		Condenser
cr		Characteristic length
cs		Condenser cross sectional area
e		Evaporator
eq		Equivalent
f		Filling ratio
ss		Steady state
L		Liquid
M		Mean
0		Initial
sat		Saturation
v		Vapour
wat		Water
s		Surface
∞		ambient

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