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NUMERICAL SIMIULATIONOF THE PERFORMANCE OF HEAT PIPE OPERATING WITH WATER

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ABSTRACT

The study of heat transfer and fluid flow behaviors in the heat pipe is investigated by study of two dimensional steady flow in a horizontal heat pipe in vapor region, wick region and wall region is investigated numerically. The governing differential equations in vapor region, wick region and wall region are solved by using a finite difference method. The numerical results of heat transfer and fluid flow are presented for the Reynolds numbers of (*Re*=4, 10) and the pipe dimension is taken to be (L/r_o =63.3).The results showed that the stream lines band at the wick increases linearly in the evaporator, decreases linearly in the condenser and is constant in the adiabatic region . Also, it can be seen that as the Reynolds number increases, the pressure distributions shift up without considerable change in their shapes. The numerical analysis showed that for the low and moderate Reynolds number, the shear stress becomes zero at a point that is very close to the end of the condenser. Also the study showed the effect of Re on temperature distribution in condenser and evaporator, whereas it has no effect in the adiabatic region. For verification of current model, it is compared with the previous studies, the comparison showed good agreement and match.

Keywords— Vapor region; heat pipe; numerical study.

Nomenclature

Cp	heat capacity at constant pressure (kJ/kg.K)
hfg	latent heat of vaporization (kJ/kg)
k	thermal conductivity (W/m.K)
L	length (m)
P	pressure (N/m2)
Q	heat transfer (W)
R	gas constant (kJ/kg. K)
Re	Reynolds number
r	redial coordinate
rv	Vapor radius (m)
rw	Wick radius (m)
ro	outer radius of pipe (m)
T	temperature (K)
v	radial vapor velocity (m/sec)
u	axial velocity (m/sec)
x	axial coordinate

Greek Symbols

μ	viscosity(kg/m.sec)
ρ	density (kg/m3)
τ	shear stress (N/m2)
μ	viscosity(kg/m.sec)

Subscripts

a	adiabatic
c	condenser
e	evaporator
int	interface
l	liquid
sat	saturated
w	wall
v	vapor

INTRODUCTION

Heat pipes are the most effective passive method of transferring heat available today. Heat pipes can transmit heat at high rates and have a very high thermal conductance. They can transfer heat with low temperature drop and quick response time in a wide range of temperatures. The vapor flow in heat pipes has been investigated by various authors in the past four decades.

Faghri and Thomas,-1989 [1], Rajashree and Sankara-1990 [2], Faghri and M. Buchko- 1991 [3], Tournier and El-Genk-1994 [4], and Chan and Faghri, 1995 [5] have published many techniques, theories and applications of different heat pipe structures.

Zhu and Vafai, 1998 [6], analyzed a three dimensional vapor and liquid flow in an asymmetric flat plate heat pipe. They studied the vapor flow by finite element method using FIDAP code. They also used a non-Darcian model for investigation of liquid flow in porous media.

Zhu and Vafai, 1999 [7] studied the liquid-vapor coupling and non-Darcian transport in the cylindrical heat pipe. They analyzed the flow and pressure distribution in a low-temperature heat pipe.

Kim et al, 2003 [8], used an analytical and experimental investigation on the operational characteristics and thermal optimization of a miniature heat pipe with a grooved wick structure.

A. Nouri- Borujerdi and M. Layeghi 2004 [9],used a numerical method based on the SIMPLE algorithm has been developed for the analysis of vapor flow in a concentric annular heat pipe. The steady-state response of a concentric annular heat pipe to various heat fluxes in the evaporator and condenser sections are studied. The fluid flow and heat transfer in the annular vapor space are simulated using Navier-Stokes equations. The governing equations are solved numerically, using finite volume approach.

Shoeib Mahjoub, and Ali Mahtabroshan 2008 [10], solved the steady incompressible flow in cylindrical coordinates in both vapor region and wick structure. The governing equations in vapor region are continuity, Navier-Stokes and energy equations. These equations have been solved using SIMPLE algorithm. For study of parameters variation on heat pipe operation, a bench mark has been chosen and the effect of changing one parameter has been analyzed when the others have been fixed.

In this paper a numerical model has been used for analysis of vapor flow in vapor region and heat transfer in both wick and wall region of heat pipe. The steady state incompressible flow has been solved in cylindrical coordinates in vapor region. The governing equations have been solved using finite difference

with collocated grid scheme. The objective of this paper was to study the heat transfer and fluid flow behavior of a conventional heat pipe operation.

MATHEMATICAL MODEL AND GOVERNING EQUATIONS

The simplified model and the coordinate system of the heat pipe used in the present study is shown in Fig.1. The heat pipe configuration can be divided into three radial regions, namely, vapor space, wick region and wall region .The working fluid is saturated with wick in liquid phase. The power applied to the heater in evaporator causes the liquid in the wick to vaporize. The vapor flows to the condenser section and releases the heat as it condenses. The released heat is rejected through the wall to the ambient. The condensed working fluid in the wick returns to heater section by the capillary force of the wick structure. To analysis the behavior of flow of fluid and heat through the heat pipe by using continuity, momentum and energy equations as flows:

1. Vapor Space:

The governing equations of the two –dimensional incompressible laminar flow with constant viscosity in cylindrical r-x coordinate, steady state flow and without heat generation are given as follows [2]:

(i)-Conservation of mass

$$\frac{\partial u}{\partial x} + \frac{1}{r}\frac{\partial (rv)}{\partial r} = 0 \tag{1}$$

(ii)-Conservation of momentum

$$\rho u \frac{\partial u}{\partial x} + \rho v \frac{\partial u}{\partial r} = -\frac{\partial p}{\partial x} + \mu \left[\frac{\partial^2 u}{\partial x^2} + \frac{1}{r}\frac{\partial u}{\partial r} + \frac{\partial^2 u}{\partial r^2}\right] \tag{2}$$

$$\rho u \frac{\partial v}{\partial x} + \rho v \frac{\partial v}{\partial r} = -\frac{\partial p}{\partial r} + \mu \left[\frac{\partial^2 v}{\partial x^2} + \frac{1}{r}\frac{\partial v}{\partial r} + \frac{\partial^2 v}{\partial r^2} - \frac{v}{r^2}\right] \tag{3}$$

(iii)-Conservation of Energy

$$\rho cp \left[u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial r}\right] = K \left[\frac{\partial^2 T}{\partial x^2} + \frac{1}{r}\frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial r^2}\right] \tag{4}$$

The boundary conditions of vapor space at both ends of the heat pipe, at which is no slip condition and adiabatic [2] as illustrated in Fig.1.

$$at\ x = 0\ ,\ v(0,r) = 0\ ,\ u(0,r) = 0\ ,\ \frac{\partial T(0,r)}{\partial x} = 0 \tag{5}$$

And

$$at\ x = L\ ,\ v(L,r) = 0u(L,r) = 0,$$

$$\frac{\partial T(L,r)}{\partial x} = 0 \tag{6}$$

At the centerline, the symmetry conditions are applied,

$$at\ r = 0\ ,\ v(x,0) = 0\ ,\ \frac{\partial u}{\partial r} = 0\ ,$$

$$\frac{\partial T(x,0)}{\partial r} = 0 \tag{7}$$

At the vapor –wick interface, the temperature is assumed to be saturated, corresponding to interface pressure during heat pipe operation. For the velocity, the boundary conditions are defined based on the rate of evaporation and condensation of working fluid. The radial and the axial velocities along the vapor –wick interface are given as

At $r = r_v$

$$v(x,r_v) = v_e = -\frac{\dot{Q_e}}{2\pi r_v L_e \rho_v h_{fg}}\ ,\quad v(x,r_v) = v_a = 0,\ v(x,r_v) = v_c = +\frac{\dot{Q_c}}{2\pi r_v L_c \rho_v h_{fg}},\ u = 0 \tag{8}$$

According to the coordinate system shown in Fig. 1, interface velocity $v(x,r_v)$ is negative in evaporator and positive in the condenser while in adiabatic regain is zero.

Thus the interface temperature is calculated using the Clausius –Clapeyron equation [10] given by

$$T(x,r_v) = \frac{1}{\frac{1}{T_{0(sa)}} - \frac{R_g}{h_{fg}} \ln\left(\frac{p_v}{p_{0(sa)}}\right)} \tag{9}$$

Pressure gradient at the wall at a particular time is calculated from the momentum equation and is given by:

$$\frac{\partial P}{\partial x} = \frac{2}{Re}\left[\frac{1}{r}\frac{\partial u}{\partial r} + \frac{\partial^2 u}{\partial r^2}\right] - 2v\frac{\partial u}{\partial r} \tag{10}$$

The shear rate is also calculated from the equation:

$$\tau = \frac{2}{Re}\frac{\partial u}{\partial r} \tag{11}$$

For the numerical analysis, it is convenient to use the governing equations in stream function and vorticity as follows:

$$ru = \frac{\partial \psi}{\partial r} \tag{12}$$

$$rv = -\frac{\partial \psi}{\partial x} \tag{13}$$

$$\omega = r\omega^{*} = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial r} \tag{14}$$

Now, we convert the governing equations and the boundary conditions to non-dimensional form using the following dimensionless quantities:

$$\left.\begin{array}{l} x^{-} = \frac{x}{r_0} \;,\quad r^{-} = \frac{r}{r_o} \;,\quad u^{-} = \frac{u}{U_a} \\[2ex] v^{-} = \frac{v}{U_a} \;,\; \psi^{-} = \frac{\psi}{U_a r_o^2} \;,\quad \omega^{*-} = \frac{\omega^{*} r_o^2}{U_a} \\[2ex] Pr = \frac{cp\,\mu}{k} \;,\; Re = \frac{U_{ar_o}}{\nu} \;,\; p^{-} = \frac{p}{\frac{1}{2}\rho U_a^2} \\[2ex] \theta = \frac{T - T_s}{T_0 - T_s},\; \theta_1 = \frac{T_{Lw} - T_s}{T_0 - T_s}, \theta_2 = \frac{T_w - T_s}{T_0 - T_s} \end{array}\right\} \tag{15}$$

Using equations (12 through 15) terms and eliminating pressure the governing equations are transformed and substitution the dimensionless, the governing equation of motion assumes the following form [2]:

$$-\omega^{*-} = \frac{1}{r^{-2}}\frac{\partial^2 \psi^{-}}{\partial x^{-2}} + \frac{1}{r^{-2}}\frac{\partial^2 \psi^{-}}{\partial r^{-2}} - \frac{1}{r^{-3}}\frac{\partial \psi^{-}}{\partial r^{-}} \tag{16}$$

$$\frac{1}{r^{-}}\left[\frac{\partial \psi^{-}}{\partial r^{-}}\frac{\partial \omega^{*-}}{\partial x^{-}} - \frac{\partial \psi^{-}}{\partial x^{-}}\frac{\partial \omega^{*-}}{\partial r^{-}}\right] = \frac{1}{Re}\left[\frac{\partial^2 \omega^{*-}}{\partial x^{-2}} + \frac{\partial^2 \omega^{*-}}{\partial r^{-2}} + \frac{3}{r^{-}}\frac{\partial \omega^{*-}}{\partial r^{-}}\right] \tag{17}$$

$$\frac{1}{r^{-}}\left[\frac{\partial \psi^{-}}{\partial r^{-}}\frac{\partial \theta}{\partial x^{-}} - \frac{\partial \psi^{-}}{\partial x^{-}}\frac{\partial \theta}{\partial r^{-}}\right] = \frac{1}{RePr}\left[\frac{\partial^2 \theta}{\partial x^{-2}} + \frac{\partial^2 \theta}{\partial r^{-2}} + \frac{1}{r^{-}}\frac{\partial \theta}{\partial r^{-}}\right] \tag{18}$$

2. Wick Region:

Based on these assumptions, the wick region is modeled as pure conductance with an effective thermal conductivity (K_{eff}). The corresponding governing equation is : [10]

$$k_{eff}\left[\frac{\partial^2 T_{Lw}}{\partial x^2}+\frac{1}{r}\frac{\partial T_{Lw}}{\partial r}+\frac{\partial^2 T_{Lw}}{\partial r^2}\right]=0 \tag{19}$$

Where the effective thermal conductivity, k_{eff} is calculated based on the metal screen wick [10] ,[2] .

$$k_{eff}=\frac{k_L[(k_L+k_s)-(1-\phi)(k_L+k_s)]}{[(k_L+k_s)+(1-\phi)(k_L-k_s)]}$$

(20)

The boundary conditions of wick regain at both ends of the heat pipe are:

$$\frac{\partial T_{Lw}(0,r)}{\partial x}=0 \tag{21}$$

$$\frac{\partial T_{Lw}(L,r)}{\partial x}=0 \tag{22}$$

At the wick and wall interface, where $r=r_w$

$$k_w\frac{\partial T_w}{\partial r}=k_{eff}\frac{\partial T_{lw}}{\partial r} \tag{23}$$

At the vapor –wick interface same equation (9)

3. Wall Region:

The heat transfer through the heat pipe wall is purely by conduction. The corresponding governing equation is [9]

$$k_w\left[\frac{\partial^2 T_w}{\partial x^2}+\frac{1}{r}\frac{\partial T_w}{\partial r}+\frac{\partial^2 T_w}{\partial r^2}\right]=0 \tag{24}$$

The boundary conditions for wall region at both ends of the heat pipe are:

$$at\quad x=0,\quad \frac{\partial T(0,r)}{\partial x}=0 \tag{25}$$

$$at\quad x=L\,,\quad \frac{\partial \boldsymbol{T}(L,r)}{\partial \boldsymbol{x}}=0 \tag{26}$$

At the wick and wall interface $at\quad r=r_w$,

$$k_w \frac{\partial T_w}{\partial r} = k_{eff} \frac{\partial T_{lw}}{\partial r} \quad (27)$$

At the outer wall, a constant heat flux boundary is used.

$at \quad r = r_o \quad x = 0 \, ... L_e$

$$k_w \frac{\partial T_w}{\partial r} = \frac{Q_e}{A} \quad (28)$$

$at \quad r = r_o \quad x = L_e \, ... L_a$

$$k_w \frac{\partial T_w}{\partial r} = 0 \quad (29)$$

$at \quad r = r_o \quad x = L_a \, ... L_c$

$$k_w \frac{\partial T_w}{\partial r} = -\frac{Q_c}{A} \quad (30)$$

The solution procedure of the discretised equations is based on a line-by-line iteration method in the axial and radial directions using FORTRAN program. The sequence of numerical steps based on upwind difference method.

The accuracy of the numerical solution is checked first by summation of the absolute values of the relative errors which should be equal or less than 10^{-4}. Second, the spot value should approach a constant value. The relative error, *RE*, in the numerical procedure is defined as:

$$Err = \sum_{cells} \left| \frac{\emptyset^{n+1} - \emptyset^{n}}{\emptyset^{n+1}} \right| \leq 10^{-4} \quad (31)$$

where superscript *n* refers to the previous iteration and term (ϕ)refers to ($\phi = \phi(u,v,P,\tau,\theta)$).

A cylindrical heat pipe with water as working fluid is selected, in which the length of the evaporator is the same as the length of the condenser and a comparatively long adiabatic section is considered. The pipe dimension is taken to be (L/r_o), increment in space coordinates are $\Delta x = \frac{L/r_o}{m-1}$ and $\Delta r = \frac{r_o}{n-1}$. The computations are done for a mesh (m=101×n=21) and for Reynolds number 4 and 10, and the Prandtl number taken as Pr=0.00101. In the present analysis, the axial conduction along the heat pipe wall is neglected. The evaporator is maintained at constant temperature over its entire length and the condenser is cooled uniformly and is also kept at constant temperature.

Checking For Grid Independency

The domain was discretized with structural homogenous meshes. The governing equations in vapor region, wick region and wall region have been solved with various numbers of meshes and as shown in Fig. 2, where 15251nodes were sufficient to achieve results that were independent to mesh structure.

Results and Discussion:

The temperature of the outer wall surface of the heat pipe through its length is in Fig.3. As the thermal conductivity is increased the outer surface temperature is decreased. It is due to the high heat transfer rate and decreasing of thermal resistance of the heat pipe system. The relation between the thermal resistance and the thermal conduction of the heat pipe is shown in Fig.4. The relation is reciprocal.

The effective thermal conductivity of the wick is decreased as the porosity of it is increased. The temperature difference between the condenser and evaporator region increased slightly. This is shown in Fig.5. The effect of porosity of the thermal resistance is shown in Fig.6.

The heat load on the evaporator will affect on the temperature along the heat pipe. Fig.7 shows that as the heat load on the evaporator is increased the temperature difference between the evaporator and condenser is increased. It is also shown that the thermal resistance is changed with the change of heat load (heat transmitted in the heat pipe). It is decreased slightly as in Fig.8.

For the same amount of charge of working fluid (here water) in the heat pipe, the temperature distribution through the pipe is decreased as the diameter of the heat pipe increased. That is because the heat transfer surface area increased so the heat transfer with the outer medium increased as shown in Fig.9. Also it is shown from Fig.10 that as the radius of the heat pipe is increased the thermal resistance decreased.

The length of the heat pipe effect on the temperature distribution along the heat pipe is shown in Fig.11 it is indicated the temperature difference between evaporator and condenser is still constant. The thermal resistance of the heat pipe also still constant. This is shown in Fig.12. Fig.11 also indicates that the adiabatic region length increased with increasing of overall heat pipe length. It means that the distance

between the heat source and heat sink doesn't affect the temperature difference and the amount of heat transfer. This is the major difference between the heat pipe and the solid pipe in transferring of heat.

The flow in the heat pipe depends on the heat flow between the two ends of it (heat load), the flow of vapor from evaporator to condenser. The velocity component in the axial direction increased in the evaporator as the vapor is generated and then it will be constant at the mid-region section of the pipe (adiabatic region) because the quantity of vapor be full the section and volume of the vapor is constant so the velocity is also constant (the flow is fully developed. At the end of the adiabatic section and beginning of the condenser, the liquid will face the liquid which prevent its movement and decelerate its velocity as shown in Fig.13. This figure shows the effect of Reynolds number on the axial velocity ratio to the average velocity (u/U_{av}). The evaporating will be fast as heat load is increased and the velocity of vapor moving increases and so Reynolds number.

The radial component of velocity in the evaporator is increased in the direction towered the center line. This is due to diffusion of the vapor from around surfaces to the center. This component becomes zero at the adiabatic region. this means that the velocity in the adiabatic region is parallel to axial line of the heat pipe. At condenser region the flow diffuses to the outer surfaces far away from the center line of the pipe and the radial component of the velocity will increase because the vapor will condense on the pipe surfaces. The ratio of radial component to average velocity (v/U_{av}) along the heat pipe is shown for two values of Reynolds numbers in Fig.14.

Fig.15 shows the pressure distribution along the heat pipe space for various Reynolds numbers. In general the pressure change is very small because of the very slow velocity

It can be seen that as the Reynolds number increases, the pressure distributions shift up without considerable change in their overall profile in the pipe. As the Reynolds number increases the pressure in the condenser section is more recovered. The pressure distribution in the adiabatic section is a straight line similar to Poiseuille flow results, while the profiles in the evaporator and condenser section demonstrate the effects of pressure head absorbed or created by evaporation or condensation.

The shear rate as shown in Fig.16 with distribution in the heat pipe. The increasing in the Reynolds numbers cause to shift the shear rate up without considerable change in its profile. Shear rate increases in vapor region in the negative direction until it reaches its maximum value at the beginning of

the adiabatic region and remains constant through this region. The shear rate decreases in the condenser region until it reaches zero value. This difference in shear rate depends on the pressure variations in the heat pipe.

MODEL VERIFICATION

For the verification of current model, the results of temperature distributions along the heat pipe for various values of heat load have been compared with other results as shown in Fig. 17. The results have good agreement with that of Mahjoub[10]. Figure.18 shows stream function comparison with study of Faghri[2]. This demonstrates that the present numerical analysis is in a good agreement of a comparison for predicting the stream function distribution for Reynolds number (Re=4).

For more confidence of the theoretical results, the temperature distribution along the heat pipe is compared with experimental result of heat pipe studied with two different heat input as shown in fig. 19. The comparison gave a good agreement with small deviations between the theoretical and experimental results [11].

CONCLUSION

Steady state two-dimensional heat transfer and flow equations in vapor region, wick region and wall region are simulated using finite difference approach model with persuasive accuracy. The governing equations have been solved using upwind difference with collocated grid scheme. This model has been verified with available numerical data and has shown good agreement. For design of a heat pipe for a special purpose it is possible to study parameter variation and their effect on system behavior that as shown by this approach. It is concluded that thermal resistance of a conventional cylindrical heat pipe, grows with increasing wick porosity, and decreases with increasing of wall thermal conductivity and heat pipe radius.

The qualitative behavior of heat and mass transfer in a heat pipe can be effectively studied using the present analysis. Due to the small pressure drop along the heat pipe at low and moderate Reynolds numbers, the present analysis predicts very small vapor temperature drop along the heat pipe.

The numerical analysis have shown that for the low and moderate Reynolds number, the shear stress becomes zero at a point very close to the end of the condenser. The results are compared with the

available numerical data which is done in the literature and have shown a good agreement. Also the results of this study have been compared with experimental study results and gave a good match.

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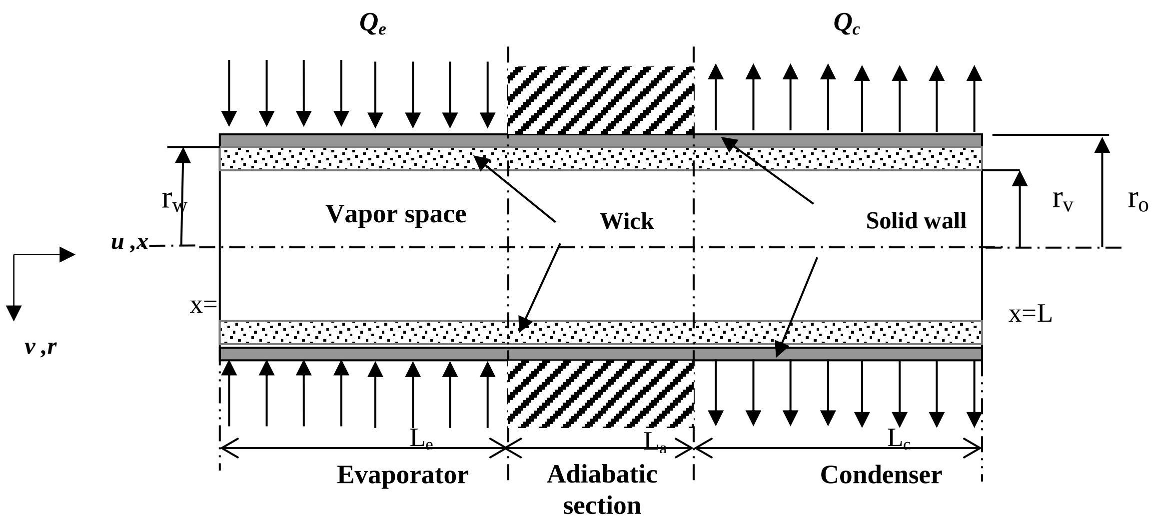

Fig.1: Heat pipe model and coordinate system

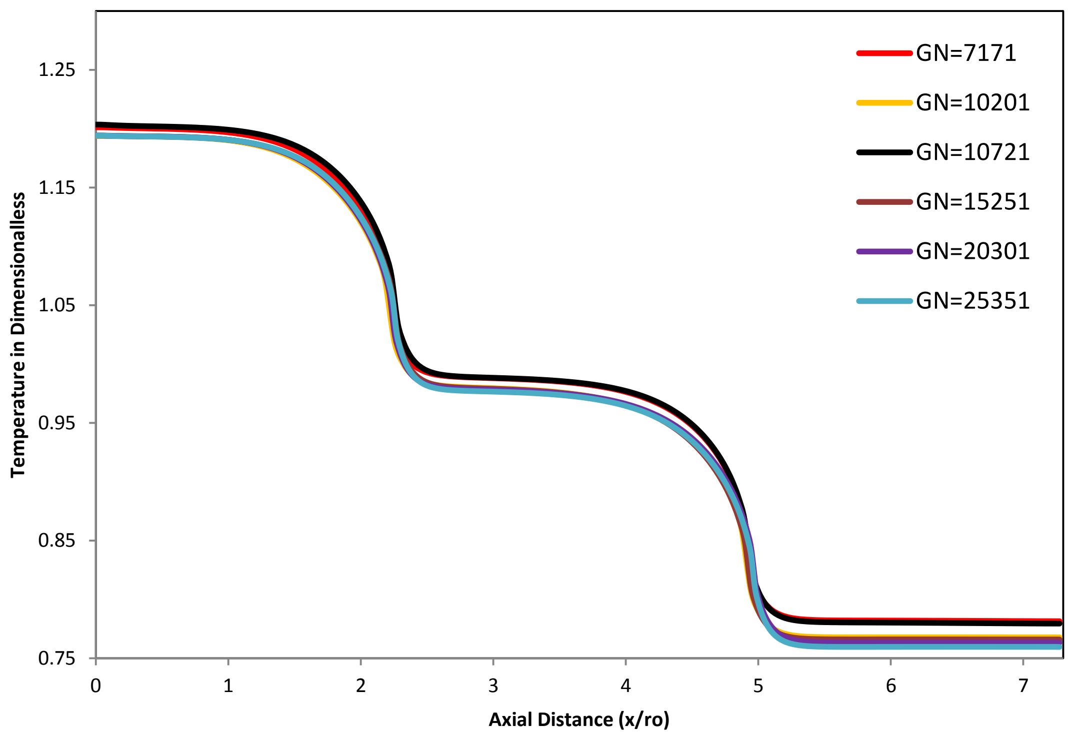

Fig. 2: Checking for grid independency

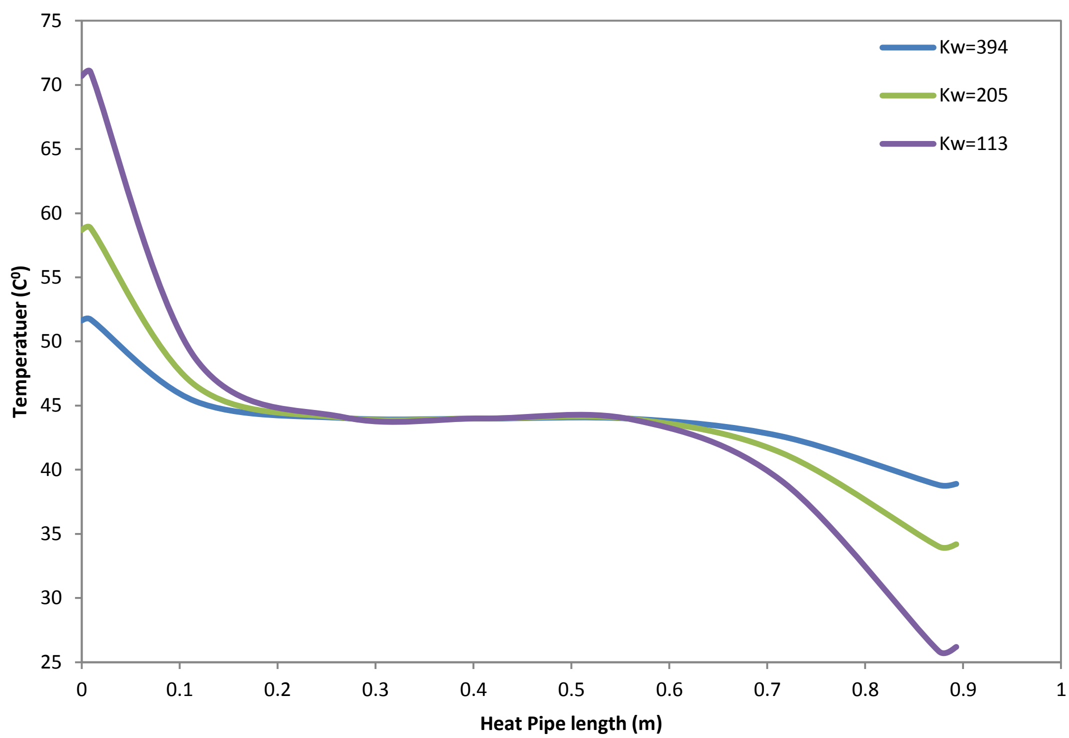

Fig. 3. Wall temperature distribution along the heat pipe for different thermal conductivity

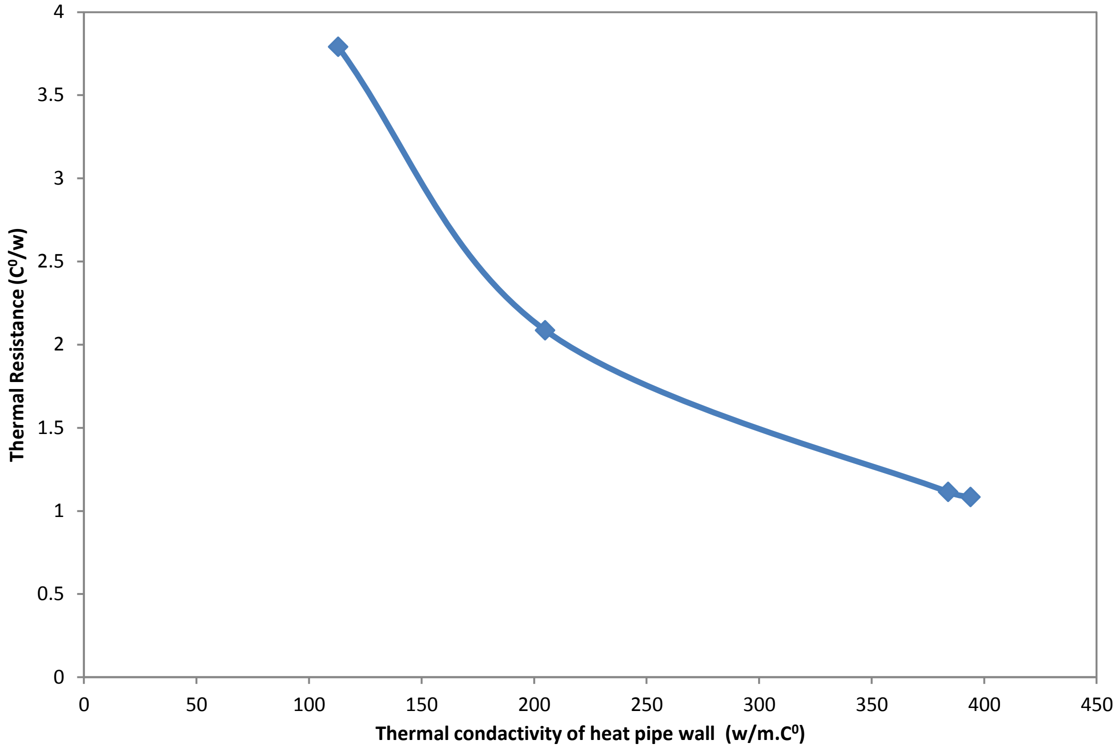

Fig. 4. Thermal resistance of heat pipe as function of wall thermal conductivity

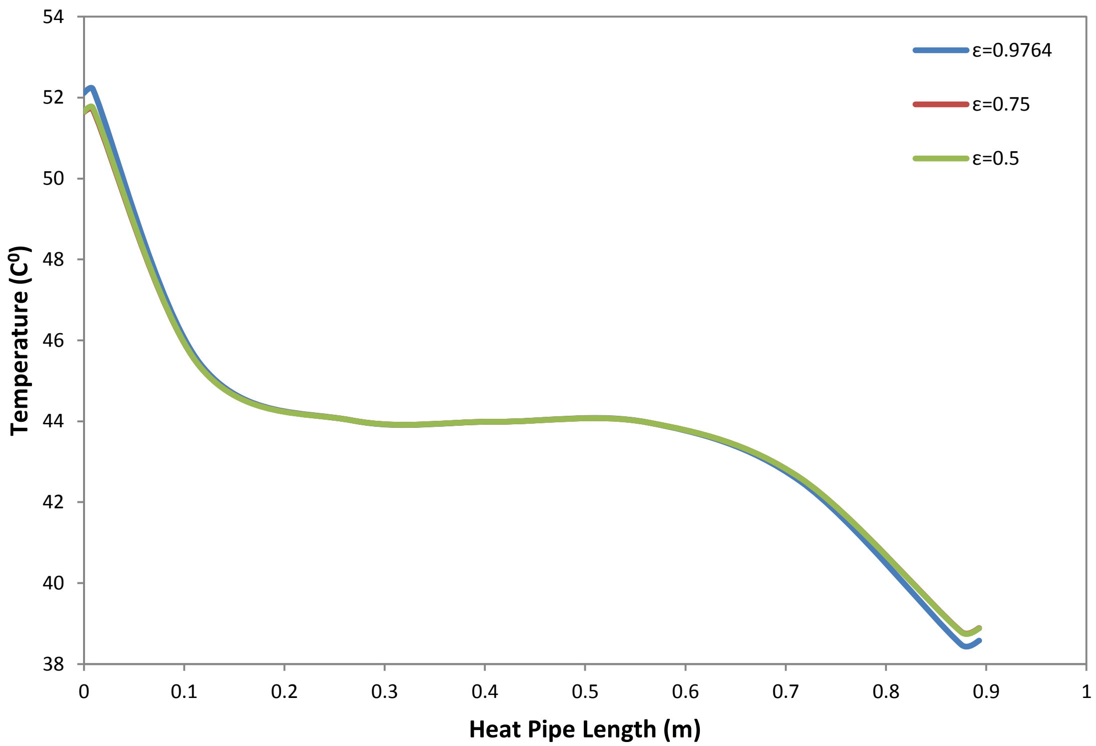

Fig. 5 Heat pipe wall temperature with variation of wick porosity

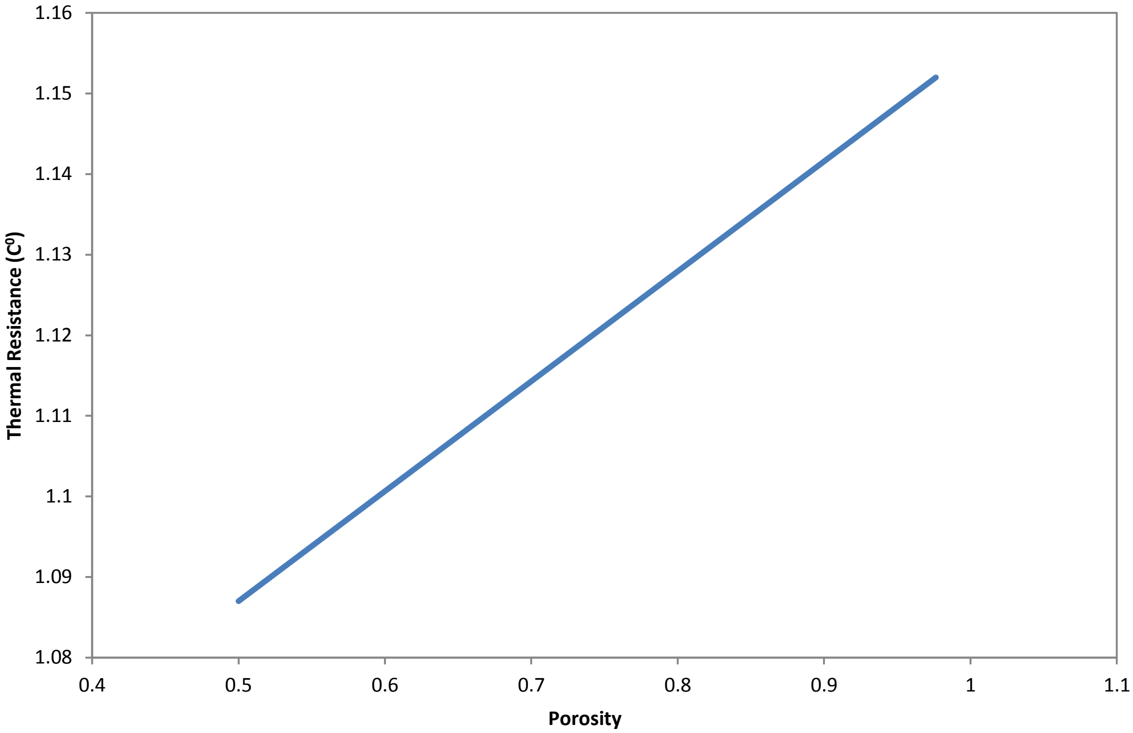

Fig. 6 Thermal resistance of heat pipe with variation of wick porosity

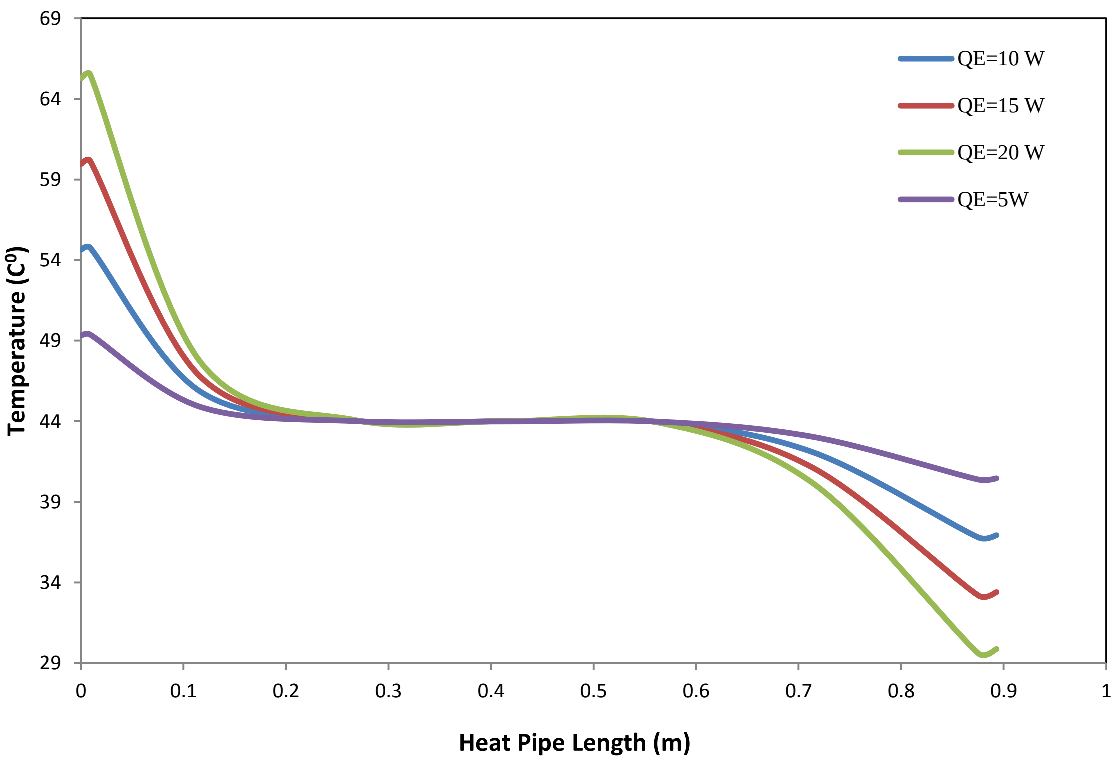

Fig. 7 Heat pipe wall temperature with variation of transmitting heat power

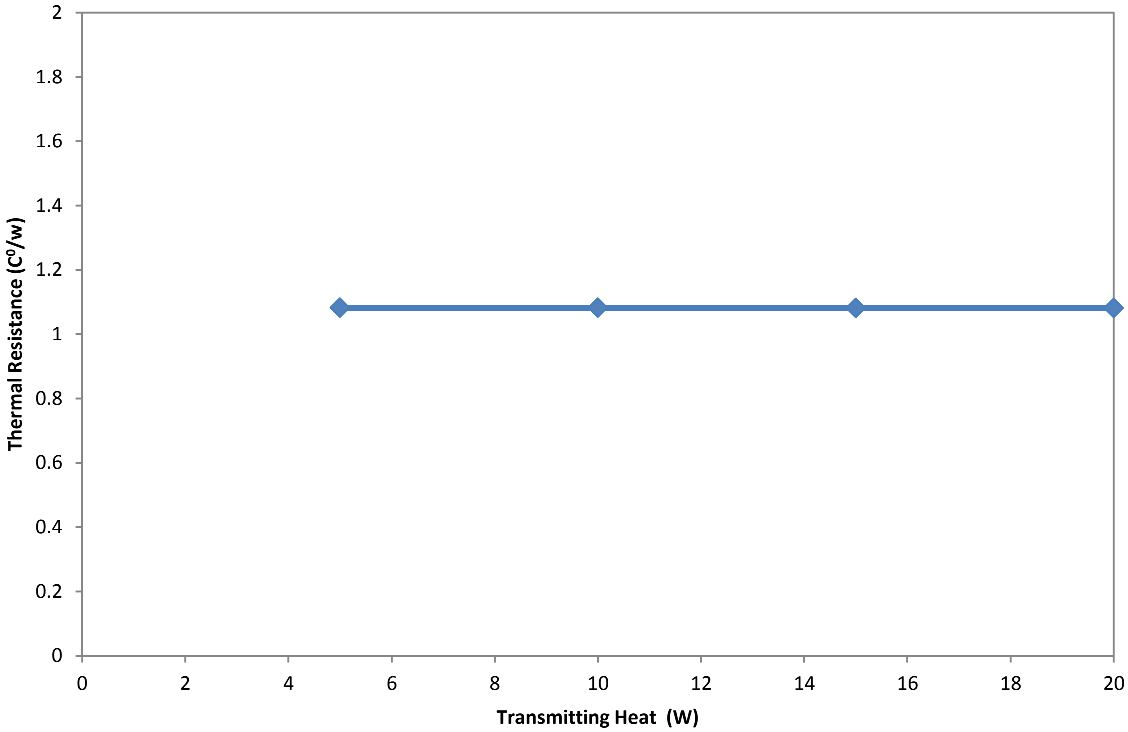

Fig. 8 Thermal resistance of heat pipe with variation of transmitting heat power

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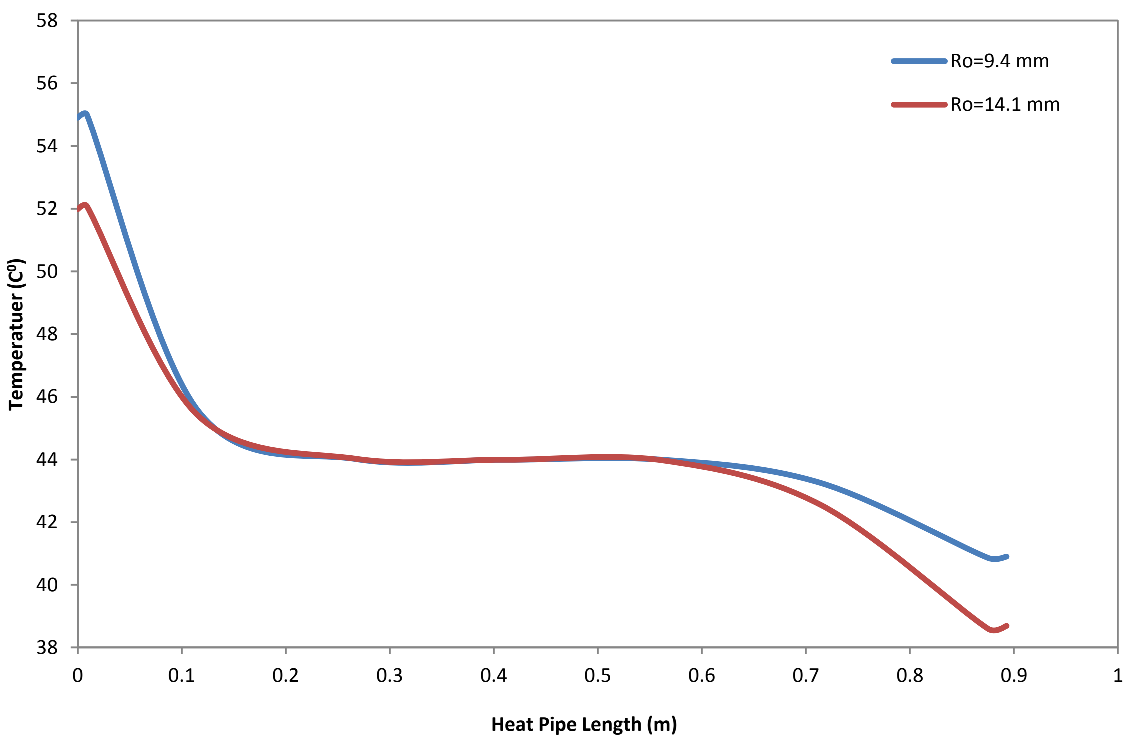

Fig. 9 Heat pipe wall temperature distribution with respect to variation of radius.

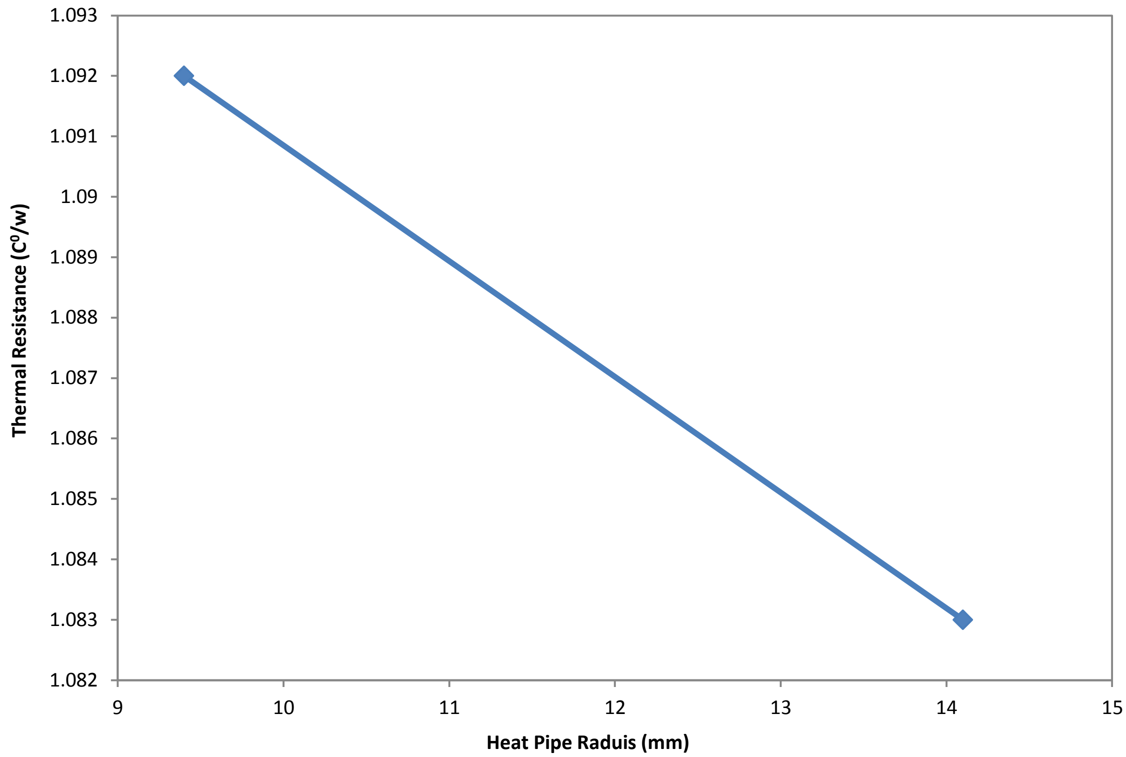

Fig. 10 thermal resistance of heat pipe with to variation of radius.

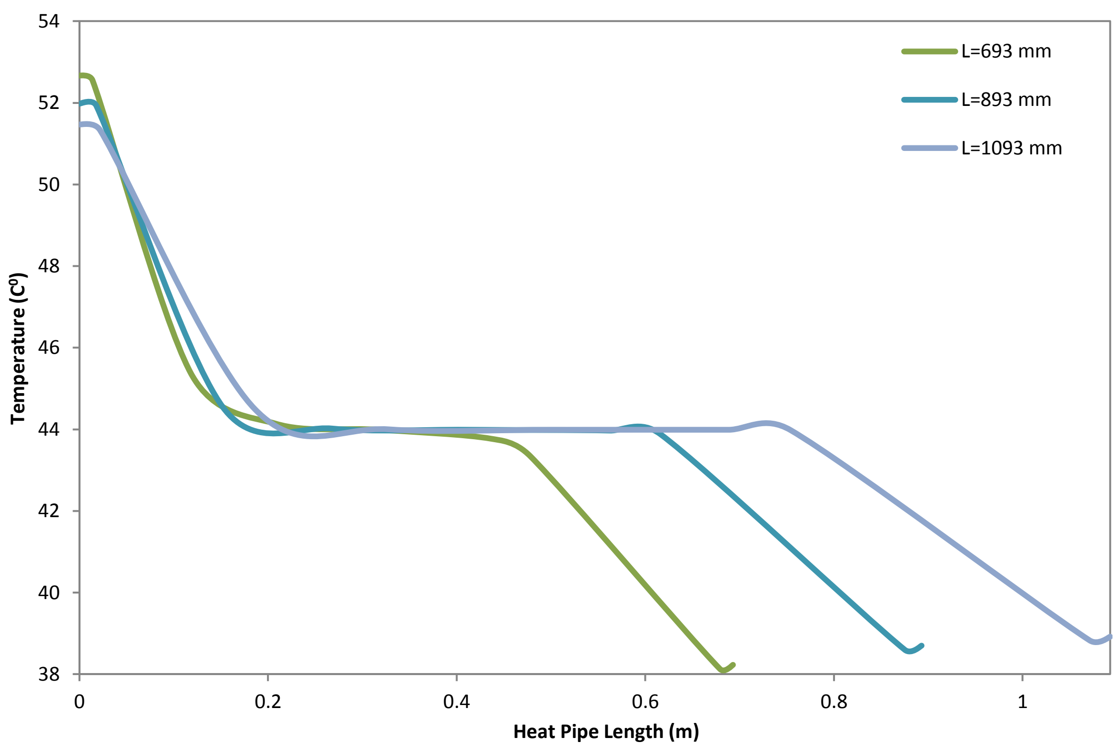

Fig. 11 Heat pipe wall temperature distribution with respect to variation of length.

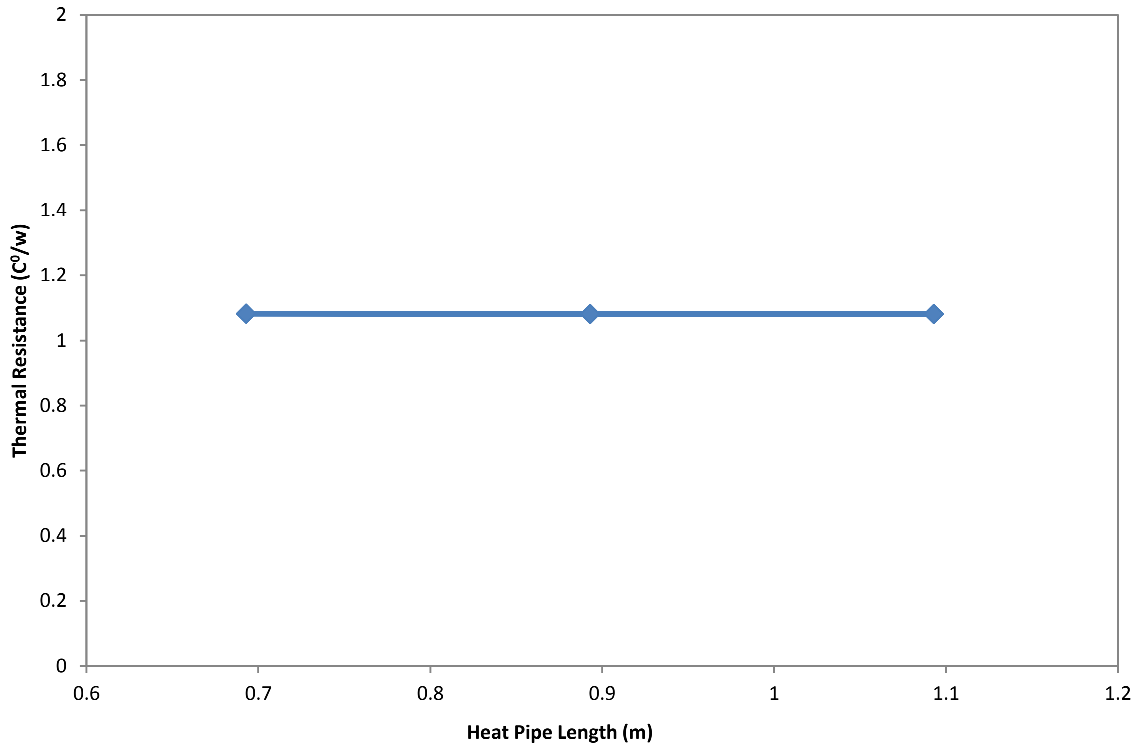

Fig. 12 Thermal resistance of heat pipe with variation of length

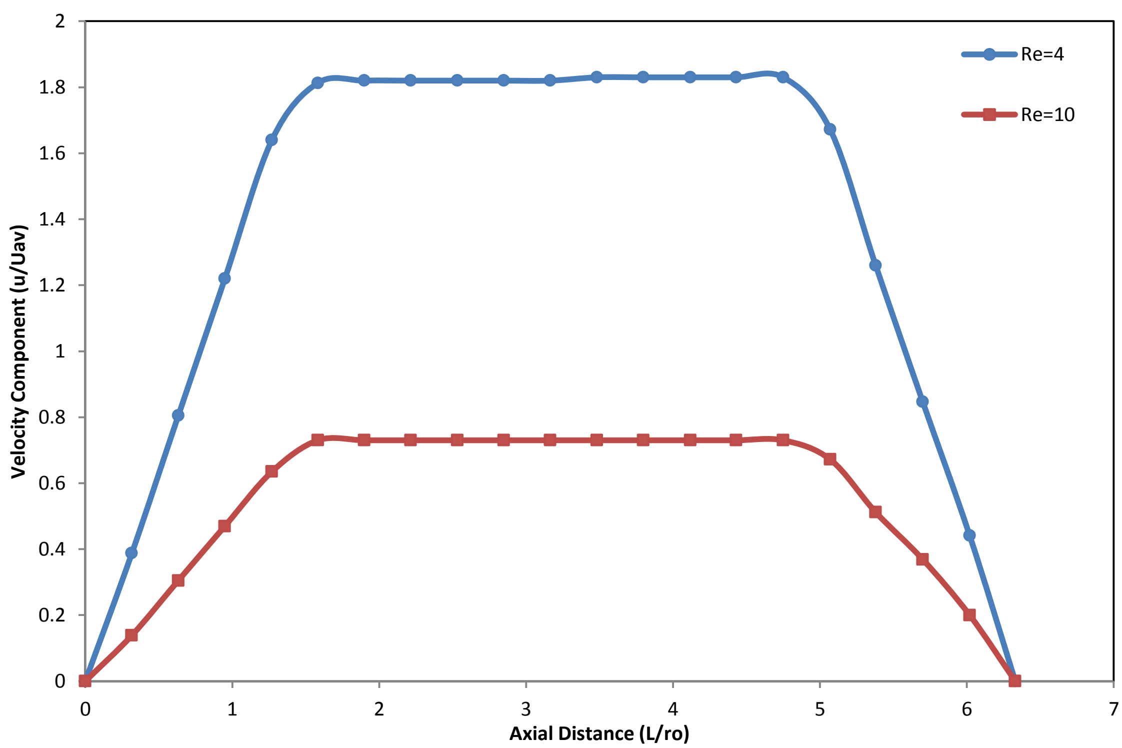

Fig.13 distribution of axial velocity near the wall

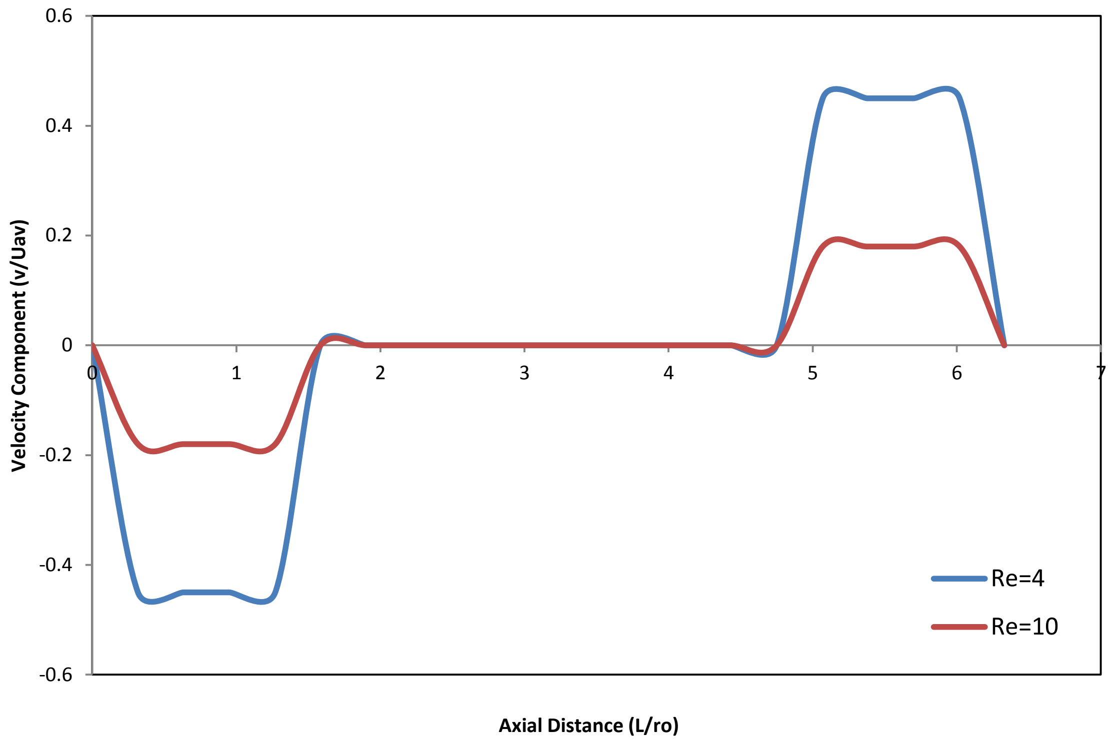

Fig.14 distribution of redial velocity at the wall

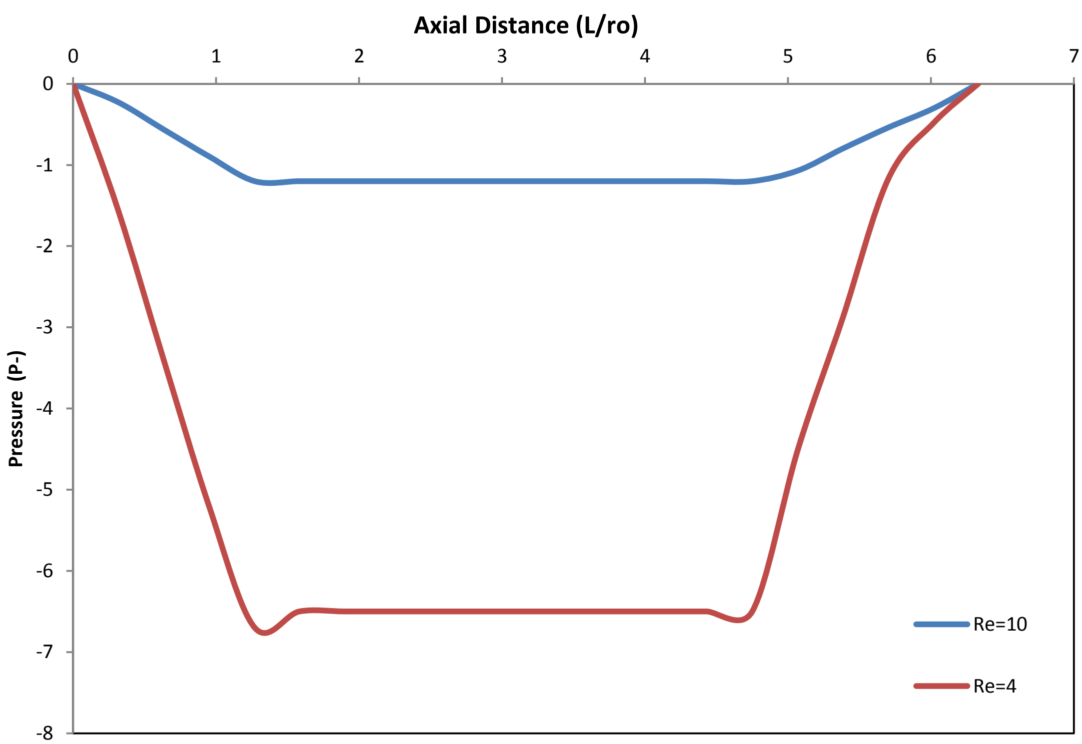

Fig.15 pressure distribution in the heat pipe along its length

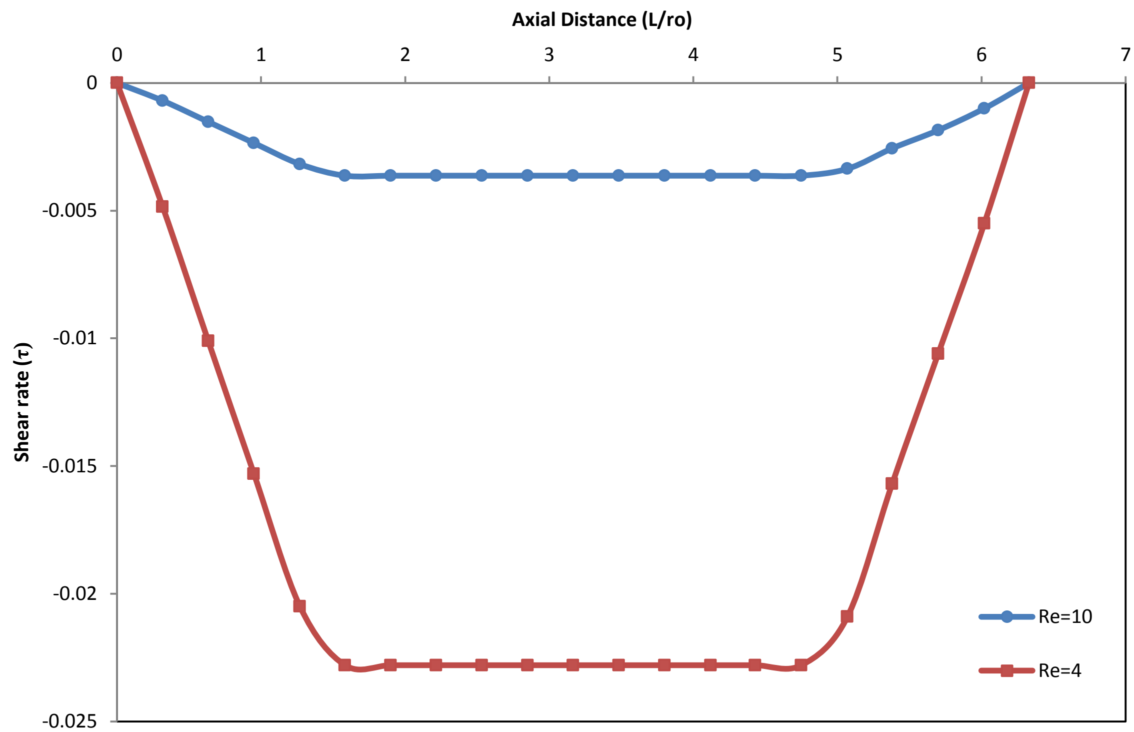

Fig. (16) distribution of shear rate near the wall

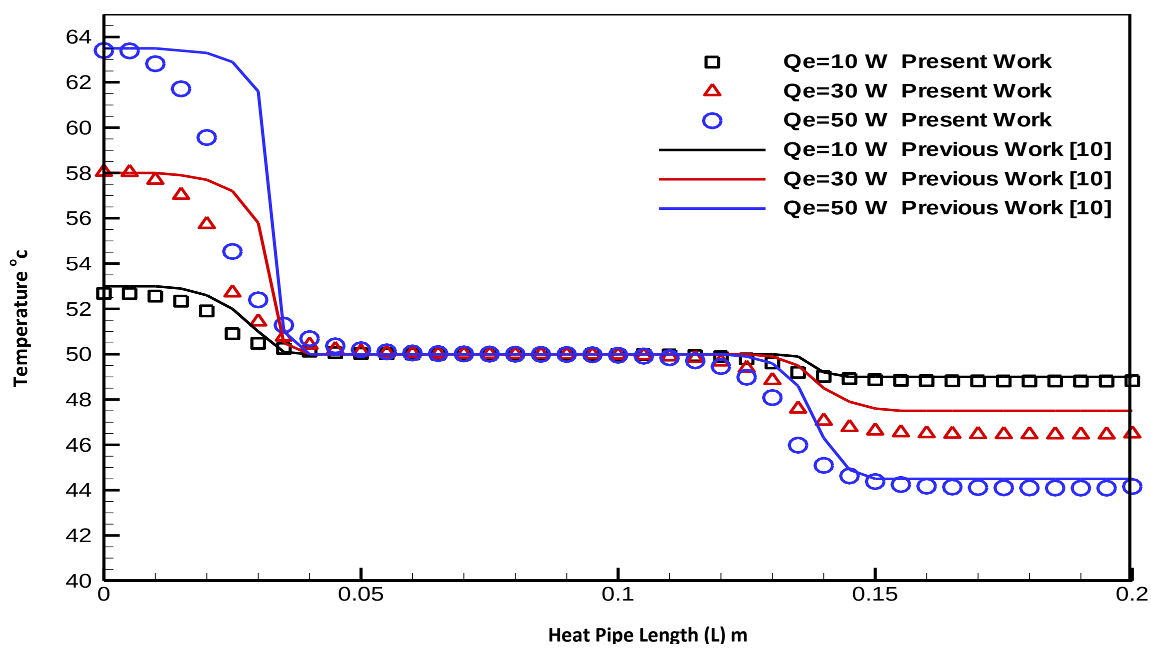

Fig. 17 the comparison with Ref. [10] for temperature along the pipe with deferent heat load

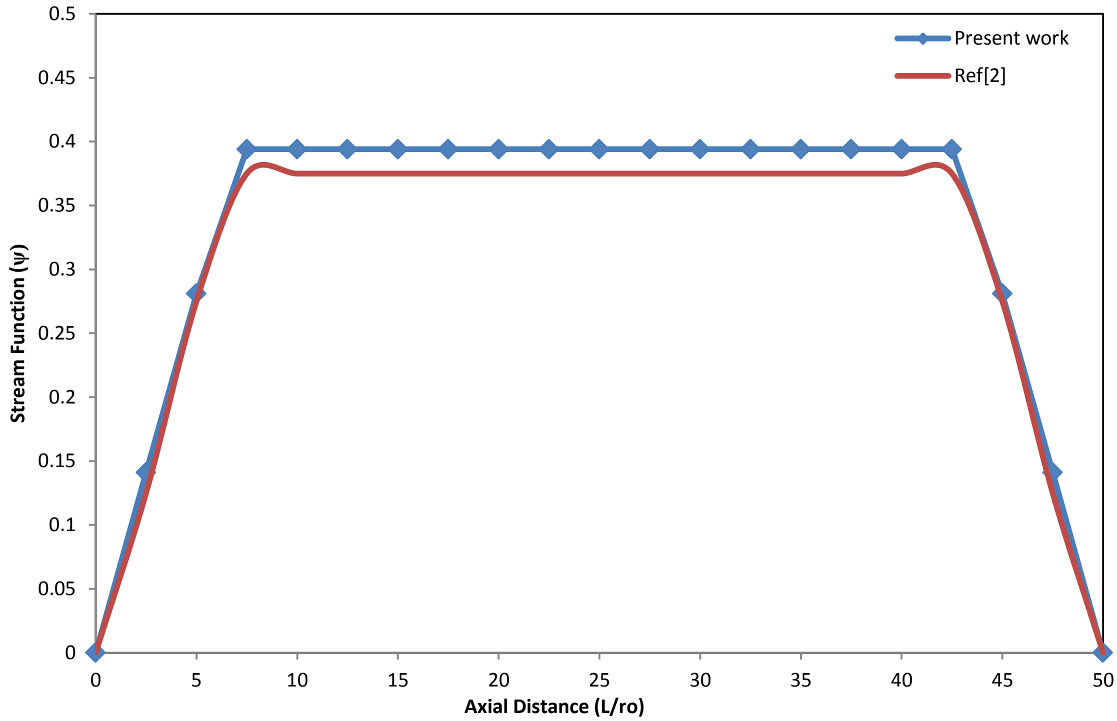

Fig. (18) the stream function calculated in this study versus that of Ref. [2]at the wall for (Re = 4)

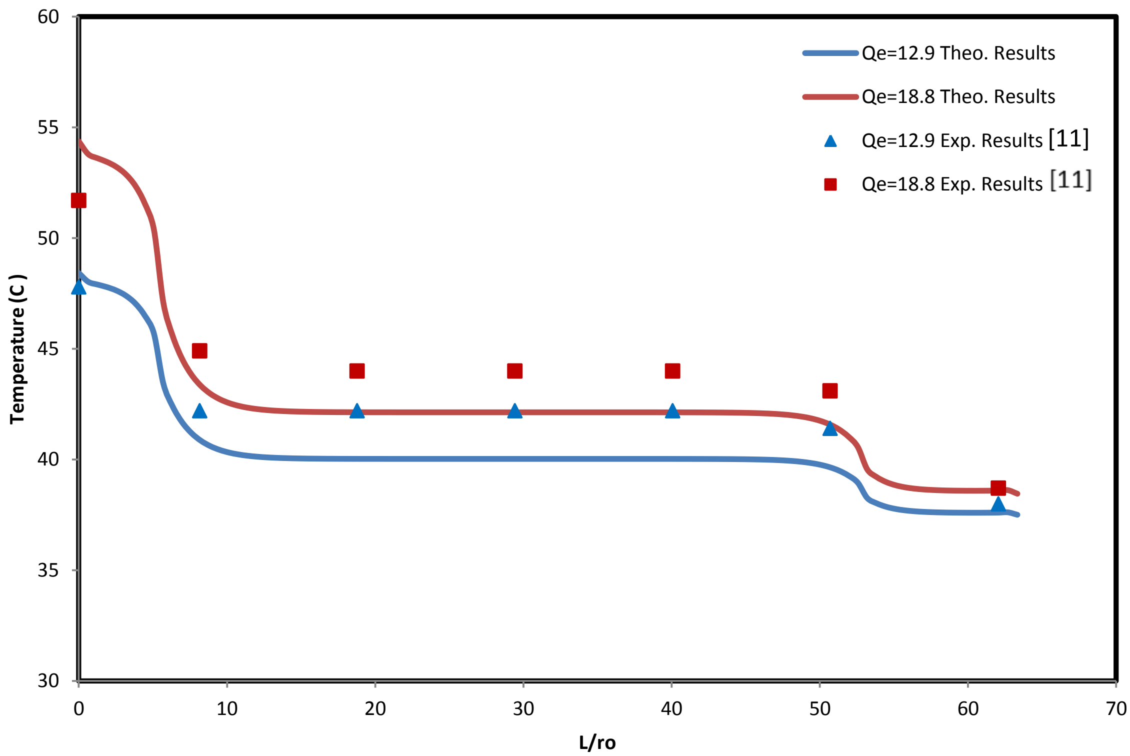

Fig. (19): The comparison of theoretical temperature along the pipe with experimental results [11]