Theoretical and Experimental Investigation of Elliptical Heat Pipe Flat Plate Solar Collector

K. Sivakumar, N. Krishna Mohan, and B. Sivaraman

***Abstract*—In the present study, the performance of an elliptical heat pipe solar collector has been investigated experimentally and theoretically. Solar flat plate collector, consists of transport tubes; but in a heat pipe solar collector, it is replaced by heat pipes and, it works as a heat exchanger in which evaporator and condenser section exist in same unit. The objective of the study is to find out the outlet temperatures of the heat pipe and condenser section and overall output condenser temperature, to find out the efficiency of the flat plate solar collector with this modification. Experimental and theoretical analysis of the effect of L_c/L_e ratio of the heat pipe and different cooling water mass flow rates and different inlet cooling water temperature have been analyzed, and presented in this paper. Elliptical heat pipe with stainless steel wick has been fabricated and used as transport tubes in the collector. Copper tube has been used as working fluid of the heat pipe. These heat pipes were fixed to the absorber plate of the solar collector and the performance of elliptical heat pipe solar collector is studied and results were compared.**

***Index Terms*—Heat pipe, flat plate solar collector, solar water heater.**

Nomenclature

A	area, (m^2)
D	diameter, (m)
a	semi-major axis length of elliptical tube, (m)
b	semi-minor axis length of elliptical tube, (m)
C_p	specific heat capacity,(J/kg K)
E	overall effectiveness
F'	collector efficiency factor
h	convection heat transfer coefficient, (W/m^2 K)
I	insolation, (W/m^2)
k	thermal conductivity, (W/m K)
L	length, (m)
m	mass flow rate, (kg/s)
n	number of heat pipes
ntu	number of transfer unit for one heat pipe
NTU	number of transfer units for n heat pipe
Nu	Nusselt number
Q	heat transfer rate, (W)
R	thermal resistance (K/W)
T	temperature, (^oC)
U	heat transfer rate coefficient, (W/m^2 ^oC)
e	effectiveness
($\tau\alpha$)	transmittance–absorptance product
λ	latent heat, (J/Kg)

Subscripts

a1	absorber for a single heat pipe
a	ambient
c	condenser
e	evaporator
hp	heat pipe
i	inlet, inside
l	liquid
L	absorber surface to ambient
o	outlet, outside
p	pipe
v	vapour

I. Introduction

This research was conducted at Chidambaram 11°24’N 79°42’E, situated in the southern part of Indian peninsula of India, near the equator, the sub- tropical region of the northern hemisphere near the Coromandel coast where the prevailing hotness perceived mostly throughout the year except small rainy and winter season.

Heat pipe is a device for heat transmission. It consists, as a rule, of the heat pipe filled with a certain amount of a suitable pure working fluid. The heat is transferred as latent heat energy by evaporating the working fluid in a heating zone and condensing the vapor in a cooling zone, the circulation is completed by return flow of the condensate to the heating zone through the inner wall of the container [1] schematic diagram of the heat pipe is shown in Fig.1. Hussein et al.[2] studied the performance of wickless heat pipe flat plate solar collector having different cross section geometries and filling ratios. They investigated the water filling ratio corresponding to the flooding limit of the elliptical cross section and referred that it is very close to 35% for circular section significantly improves the performance of wickless heat pipe flat plate solar collectors at low water filling ratios. Riffat etal.[3] studied thermal performance of a thin membranes heat pipe solar collector and the heat processes in different areas of the collectors were investigated and represented.

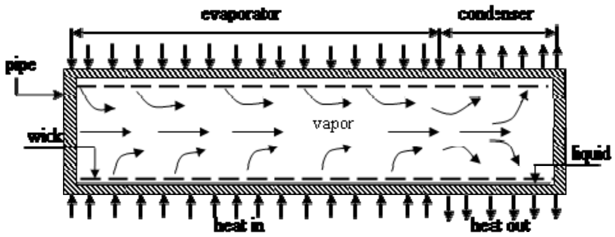

Fig.1. Heat pipe schematic

Azad [4] studied the thermal performance of heat pipe

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solar collector and the heat pipe solar collector is investigated theoretically and experimentally, and the optimum ratio of evaporator length and condenser length is also discussed. Hussein [5-6] studied theoretical and experimental investigation of wickless at pipes flat plate solar collector and optimum cooling water mass flow rate and number of heat pipes. Thermal behavior of flat plate heat pipe solar collectors was studied by various researchers [7-12]. Abdul-Jabber [13] studied the performance of locally made flat plate collectors used as part of a domestic hot water system. They found the temperature distribution along the flow direction and the thermo siphon mass flow rate. Kulkarni et al. [14] observed that there exists a minimum as well as a maximum storage volume for a given solar fraction and collector area. Akhtar [15] studied the computation of glass cover temperatures and obtain the glass cover temperatures. Maatouk knoukhi [16] studied the theoretical approach of flat plat solar collector taking into account the absorption and emission within glass cover layer and the thermal behavior and head loss from the solar collator. Elshazly et al [17] studied the heat transfer by free vertical and inclined elliptical tube. For the elliptical tube a constant heat flux and with different orientation angles has been used.

In the present study an elliptical heat pipe solar collector was designed constructed and investigated theoretically and experimentally under different water mass flow rates and inlet cooling water temperature. Also the effect of ratio of evaporator length to condenser length also discussed.

II. Theoretical Analysis

Fig 2 shows the flat plate elliptical heat pipe solar collector. In the solar collector five numbers of heat pipe was used. The manifold is us made up of galvanized iron when water is flow us through the manifold it picked up heat from the heat pipe condenser.

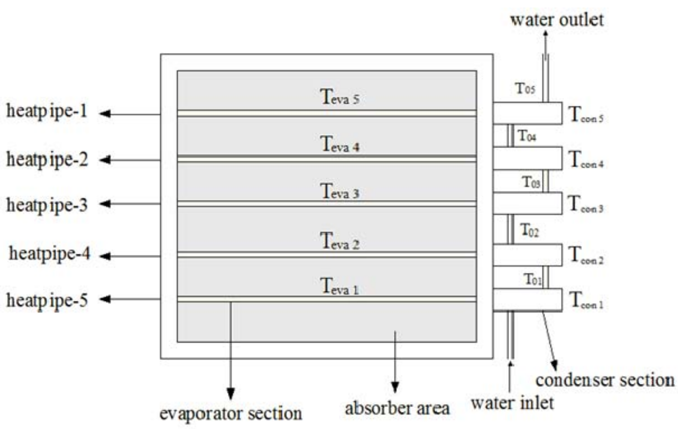

Fig. 2. Flat plate Elliptical heat pipe solar collector

1) In a collector with ‘n’ heat pipes, the water flows from one condenser to another.
2) The outlet temperature of the first condenser becomes the inlet water temperature of the second one, and so on.
3) A single panel of flat plate heat pipe collector can be considered as an array of heat pipes connected to a manifold.
4) The heat loss coefficient between the collector and the ambient is assumed to be constant.
5) The actual heat transfer rate for a single heat pipe Q_{hp_1}, which is the thermal energy transfer from evaporator to the condenser section, may be written as,

A. *The Actual Heat Transfer Rate for a Single Heat Pipe Q_{hp1}*

$$Q_{hp_1} = \frac{A_{hp1}(T_{hp1} - T_{c,01})}{\Sigma R_{hp}} = A_{hp1} U_{hp} (T_{hp1} - T_{c01}) \quad (1)$$

$$U_{hp} = \frac{1}{\Sigma R_{hp}} \quad (2)$$

The heat transfer between a single heat pipe and the cooling liquid may be expressed as

$$Q_{c1} = A_{c1} U_{c.01} \left(T_{c.01} - T_i\right) = Q_{hp1} \quad (3)$$

From equation (1) and (3) condenser outer surface temperatures may be written as

$$T_{c,01} = \frac{(T_i + PT_{hp1})}{(1+p)} \quad (4)$$

where

$$p = \frac{U_{hp} A_{hp}}{A_{c1} U_{c.01}}$$

where ‘$U_{c,01}$’ is the overall heat transfer co-efficient and can be written as

$$U_{c.01} = \frac{1}{\frac{t_p}{k_p} + \frac{1}{h_{c,01}}} \quad (5)$$

t_p and k_p are the known parameters and heat transfer co-efficient $h_{c,01}$ can be find from the following Nusselt number

$$N_u = \frac{h D_{hyd}}{k} \quad (6)$$

Or

$$h_{c,01} = \frac{N_u k}{D_{hyd}} \quad (7)$$

Hydraulic diameter for ellipse

$$D_{hyd} = \frac{4 \times \pi a b}{\pi \sqrt{2\left(\left(a^2 + b^2\right) - \left(a - b\right)^2 / 2\right)}}$$

The fluid flow in the manifold is considered to be fully developed laminar flow. It is assumed that the flow inside the condenser of the heat pipes is thermally developed and therefore under constant heat flux boundary condition at the wall, the Nusselt number may be written as

$$N_u = 5.331$$

The outlet water temperature from the first condenser, $T_{c,01}$, can be calculated from thermal resistance analysis. Heat from the evaporator section to the condenser section may be mathematically modeled by a number of thermal resistances

$$\Sigma R = R_{e,p} + R_{e,wick} + R_{e,i} + R_{c,i} + R_{cp} \tag{8}$$

B. Heat Pipe Evaporator Resistance

Within the evaporator section, the thermal resistances which account for temperature drops are container wall, wick conduction resistances and internal resistance at the evaporator. These can be expressed in term of film coefficient h_e.

$R_{e,p}$ – Thermal resistances across the thickness of the container wall (e-(heat pipe) evaporator section, p-at the container wall)

$$R_{e,p} = \frac{\ell n \dfrac{d_o}{d_i}}{2\pi\, k_p\, l_e} \tag{9}$$

$R_{e,\,wick}$ - Thermal resistances across the thickness of the wick thickness (e- (heat pipe) evaporator section)

$$R_{e,wick} = \frac{\ell n \left(\dfrac{d_{o,wick}}{d_{i,wick}} \right)}{2\,\pi\, k_{wick}\, L_e} \tag{10}$$

Screen mesh structure is used in the evaporator section. The effective thermal conductivity of the saturated wick may be calculated from the following equation

$$k_{wick} = \frac{k_l\left[k_l + k_s - (1-\varepsilon_{wick})(k_l - k_s)\right]}{k_l + k_s + (1-\varepsilon_{wick})(k_l - k_s)} \tag{11}$$

$R_{e,i}$ - is the thermal resistances that occurs at the vapour liquid interfaces in the evaporator and may me written as, (e-evaporator, i- at liquid –vapour interface)

$$R_{e,i} = \frac{2}{\pi\; h_e\; d_i\; L_e} \tag{12}$$

where

$$h_e = \frac{k_l}{t_{wick}} \tag{13}$$

Since, for a wick lined wall as in the case with evaporator, the film coefficient is approximately equal to the thermal conductivity of the fluid divided by the wick thickness

C. Heat Pipe Condenser Resistances

The vapour condenses on the inner section of the condenser releasing the latent heat of condensation. The heat must then be conducted through the container wall surface. The resistance associated with the conduction process through the pipe wall is

(c – heat pipe condenser resistance, p – at the wall)

$$R_{c,p} = \frac{\ell n \dfrac{d_o}{d_i}}{2\pi\, k_p\, L_c} \tag{14}$$

D. Thermal Resistances Associated with the Condensing Process is Given by

$$R_{c,i} = \frac{1}{h_{c,i}\;\; \pi\, d_i\;\, L_c} \tag{15}$$

The condensing film co-efficient may be obtained from the Nusselt analysis for film wise condensation

$$h_{\,c,i} = 0.728 \left[\frac{g\, \rho_l (\rho_l - \rho_v) k^3 \lambda}{d_i\, \mu_l\, \Delta Ti} \right]^{1/4} \tag{16}$$

E. To Find T_{hp}

The rate of useful energy gained or collected and heat loss to the ambient may be modeled according to Hotel-Willer equation, for single heat pipe as follows

$$Q_{u,1} = A_{a1} F' \left\lfloor I(\tau\alpha)_e - U_L (T_{hp1} - T_a) \right\rfloor \tag{17}$$

Useful energy extracted in the form of heat by fluid flowing in the heat exchanger can also be expressed as

$$Q_{u1} = mC_p \left(T_{o1} - T_i\right) \tag{18}$$

By equating equation (17) and (18)

$$A_{a1} F' \left\lfloor I(\tau\alpha)_e - U_L (T_{hp1} - T_a) \right\rfloor \;=\; mC_p \left(T_{o1} - T_i\right)$$

put

$$(ntu)_{hp1} = \frac{A_{a1}\, F'\, U_L}{m\, c_p}$$

We get

$$T_{hp1} = T_a + \frac{I(\tau\alpha)}{U_L} - \frac{\left(T_{o1} - T_i\right)}{(ntu)_{hp1}} \tag{19}$$

F. To Find T_{01}

In the condenser section of a single heat pipe heat exchanger, cold fluid is in cross flow with vapour flow inside the heat pipe. However, since the vapour inside a heat pipe is almost at constant temperature, its specific heat, C_p, and capacity rate, C_L will by definition be equal to infinity and as a result $C_{p/}$ C_L=0. Therefore, the effectiveness-NTU-equation for this condition will be as follows (16):

$$\varepsilon_1 = 1 - e^{-(ntu)_{c1}} \quad (20)$$

where $(ntu)_{cl} = \dfrac{A_{c1}\, U_{c,o1}}{mc_p}$

$$\varepsilon_1 = \frac{T_{01} - T_i}{T_{c,01} - T_i} \quad (21)$$

Eqn. (21) may be rearranged in the following form:

$$T_{o1} = T_i + \varepsilon_1 \left(T_{c,01} - T_i\right) \quad (22)$$

The condenser water temperature at the outlet of a single heat pipe is obtained from substituting $T_{c,01}$ into eqn. (22), yielding:

$$T_{o1} = T_i + \varepsilon_1 \left(\frac{P}{1+P}\right)\left(T_{hp1} - T_i\right) \quad (23)$$

Substituting T_{01} in T_{ph1}, that is, equation (23) in (19), we get

$$T_{hp1} = \frac{T_a + \dfrac{I(\tau\alpha)}{U_L} + \dfrac{\varepsilon_1 T_i}{(ntu)_{hp1}}\left(\dfrac{p}{1+p}\right)}{1 + \dfrac{\varepsilon_1}{(ntu)_{hp1}}\left(\dfrac{p}{1+p}\right)} \quad (24)$$

where $(ntu)_{hp1} = \dfrac{F' A_{01} U_L}{mc_p}$

For a collector with 'n' heat pipes as shown in fig, the water flows from the condenser of one heat pipe to another. The outer water temperature of the first condenser becomes the inlet water temperature of the second condenser. For a collector with 'n' heat pipes the final temperature can be calculated from the following equation.

$$T_{0\,(n)} = T_{0\,(n-1)} + \varepsilon_n \left(\frac{P}{1+P}\right)\left[T_{hp\,(n)} - T_{0\,(n-1)}\right] \quad (25)$$

G. Thermal Analysis of Heat Pipe Array

The final water outlet temperature for an array of 'n' heat pipes may also be obtained by a different approach, for overall effectiveness of 'n' condenser, one fluid in series and the other fluid in cross flow. Referring to fluid in series, the overall effectiveness 'E' can be written as

$$1 - E = \left(1-\varepsilon_1\right)\left(1-\varepsilon_2\right)\left(1-\varepsilon_3\right)\ldots\left(1-\varepsilon_n\right) \quad (26)$$

Assuming equal effectiveness for all condensers. Eq. (26) may be written as

$$1 - E = \left(1-\varepsilon_1\right)^n \quad (27)$$

The overall effectiveness in term of temperature is given by

$$E = \frac{T_o - T_i}{T_{hp} - T_i} \quad (28)$$

The water outlet and heat pipe temperature of the collector array are

$$T_o = T_i + E\left(T_{hp} - T_i\right)\left(\frac{P}{1+P}\right) \quad (29)$$

And

$$\bar{T}_{hp} = \frac{T_a + \dfrac{I(\tau\alpha)_e}{U_L} + \dfrac{E\, T_i}{(NTU)_{hp}}\left(\dfrac{p}{1+p}\right)}{1 + \dfrac{E}{(NTU)_{hp}}\left(\dfrac{p}{1+p}\right)} \quad (30)$$

where

$$(NTU)_{hp} = n(ntu)_{hp1} \quad and \quad NTU_c = n(ntu)_{c1} \quad (31)$$

TABLE I: HEAT PIPE SPECIFICATIONS

Heat pipe material	Copper
Working fluid	Methanol CH_3 OH
Total length of pipe (L_t)	1.00m
Evaporator length (L_e)	0.65m, 0.75m, 0.85m
Condenser length (L_c)	0.35 m, 0.25m, 0.15m
Major axis of elliptical pipe (a)	26 X 10^{-3} m
Minor axis of elliptical pipe (b)	16.7 X 10^{-3} m
Thickness of pipe (t)	2.0 X 10^{-3}m

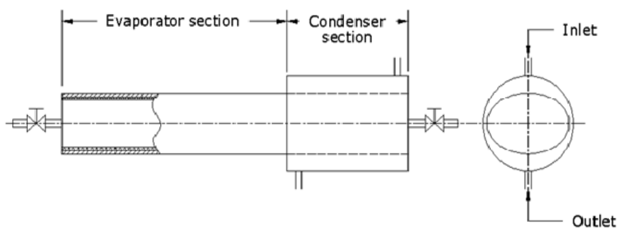

Fig. 3.Cross section view of the elliptical heat pipe

III. EXPERIMENTAL SET UP AND PROCEDURE

In the heat pipe solar collector the sun rays passes through the glass to heat a copper heat pipe. Inside the heat pipe a nontoxic liquid carries the heat to another end of the heat pipe, then the collector unit in which it is transferred through the water heating system. Solar radiation is absorbed by the absorber plate and converted in to heat. Heat pipe then conduct the heat from evaporator section to condenser section water is circulated through the condenser section and the water gets heated up.

Five number of heat pipes have been used in the solar collector and placed over the absorber plate. The cross sectional view of an elliptical heat pipe and a schematic diagram of the test rig are shown in Fig. 3 and 4, respectively. Copper has been used as container material and stainless steel as wick material. Wrapped screen wick structure with two layers of wick has been used in the heat pipe. Methanol has been used as a working fluid of the heat pipe. The specification of the heat pipe is given in Table I. The solar collector is a flat plate collector having dimension of length 0.83 m and breath 0.81m. Thickness of the absorber plate is 2mm, aluminum sheet has been used as absorber material and is painted black paint, and thickness of glass plate is 3.2mm. Experiments were conducted on the designed elliptical heat pipe solar collector with different mass flow rate of 18kg/hr, 24kg/hr, 30kg/hr and 36kg/hr on various days. The condenser and evaporator length varied (ie Lc/L_e ratio 0.1764, 0.3333 and 0.5384) in the experimental study. Experiments were conducted first with L_c/L_e ratio 0.1764 with constant mass flow rate for various days and then mass flow rate has been changed. Similarly, the trials were conducted for L_c/L_e ratio of 0.3333 and 0.5384. The observation of various parameters were measured and recorded from 9.00 am to 4.00 pm.

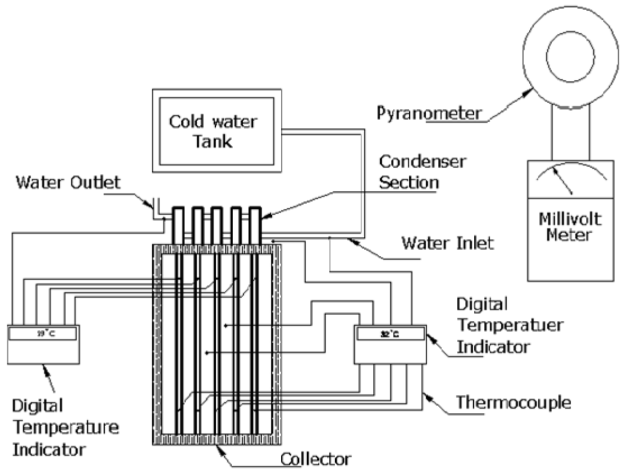

Fig. 4. Experimental setup for Elliptical HPSC

The collector has been placed facing due south with a tilt angle of 11° to the horizontal. The elliptical heat pipes have been brazed to the absorber sheet in such a way that the evaporator section of the heat pipe is in the collector and condenser section is protruding outside the collector. The water circulated through the condenser section at inlet and outlet temperatures were measured by K-type thermocouple. Solar intensity at the test site was measured using EPPLY pyranometer. The heat pipe surface temperature, absorber plate temperatures and glass plate temperatures were measured using thermocouples.

IV. Results and Discussion

The theoretical and experimental results of the elliptical heat pipe solar collector have been compared. A comparison of the collector efficiency for steady state conditions as obtained by outdoor measurements and theoretical model is given in Fig. 5, 6 and 7.

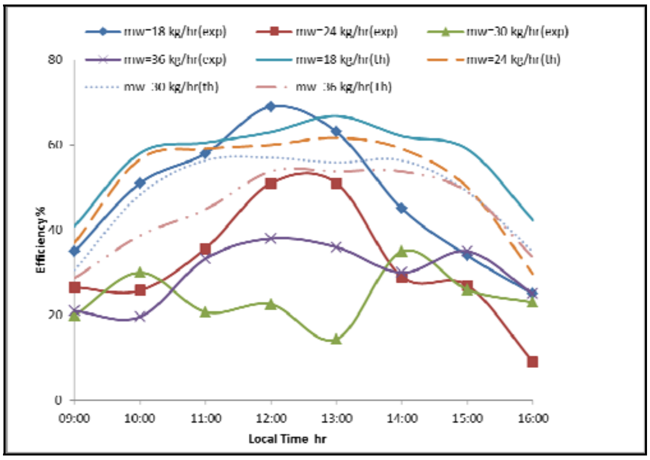

Fig. 5. Instantaneous variation of collector efficiency with local time for 0.1764 Lc/Le ratios

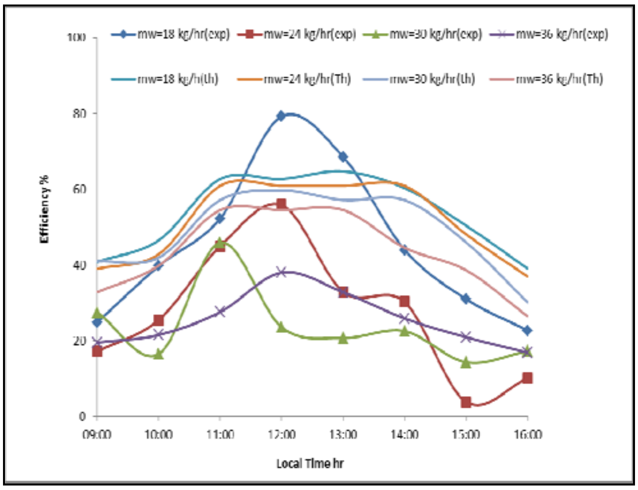

Fig. 6. Instantaneous variation of collector efficiency with local time for 0.333 Lc/Le ratios.

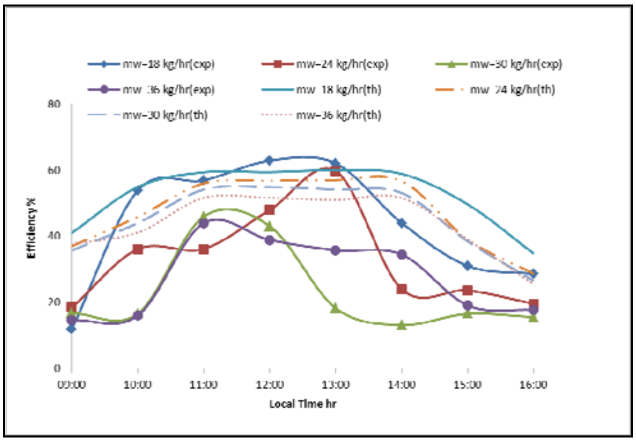

Figure. 7. Instantaneous variation of collector efficiency with local time for 0.5384 Lc/Le ratios.

The theoretical efficiency is given by equation

$\eta_{the]ory} = m\ c_p\ (T_{o(n)} - T_i)\ /AI$

T_o is the calculated water outlet temperature given by equation (29). A comparison of elliptical heat pipe solar collector efficiency for steady-state conditions as obtained by outdoor measurements and by theoretical model given in Fig. 5, 6 and 7. The maximum difference between theoretical and experimental efficiencies occurs at 11:00 a.m and it may be due to fluctuation of solar insolation. Maximum efficiency of around 78% has been achieved during peak hours for a flow rate of 18kg/hr. Fig. 8-10 gives the plot of experimental, theoretical water outlet temperature and insolation on various days. The left scale represent the solar intensity and right scale represent the temperature. The temperature and the theoretical water outlet temperature of heat pipes were plotted in Fig. 11. As expected the heat pipe temperature and the water temperature increase generally from one heat pipe to the next. The temperature of first heat pipe was 55°C and while that of last heat pipe was 65°C. It has been fund that the difference between the theoretical and experimental water outlet temperature is within the acceptable limit. Hence this theoretical estimation has been adopted to simulate the water outlet temperature for various mass flow rate conditions and Lc/Le ratio. The results are plotted as shown in Fig. 12-14.

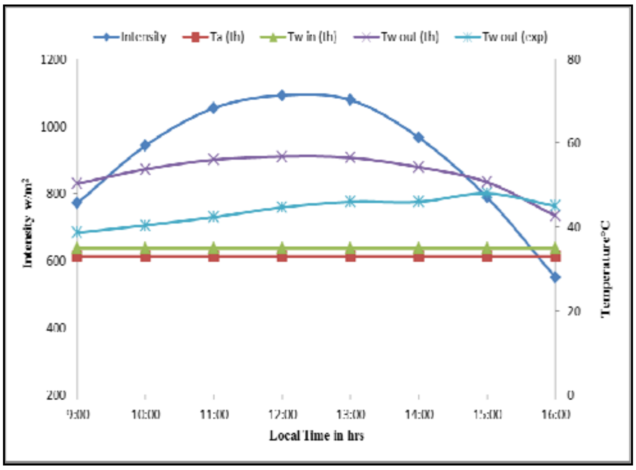

Fig. 8. Variation of Time vs. Temperature and Intensity at Mass flow rate of 18kg/hr for Lc/Le 0.1764

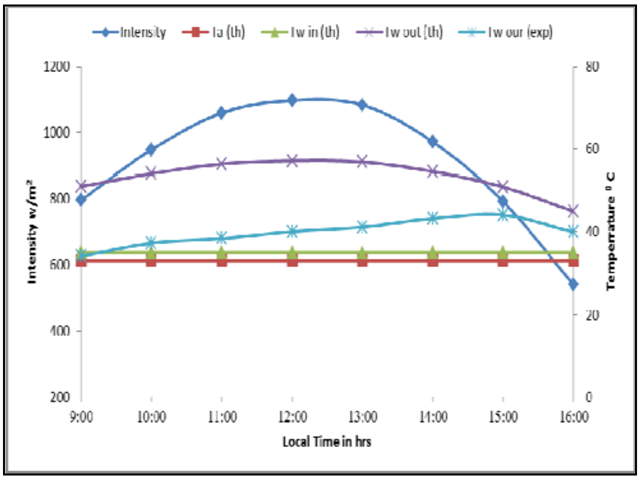

Fig. 9. Variation of Time vs. Temperature and Intensity at Mass flow rate of 18kg/hr for Lc/Le 0.333.

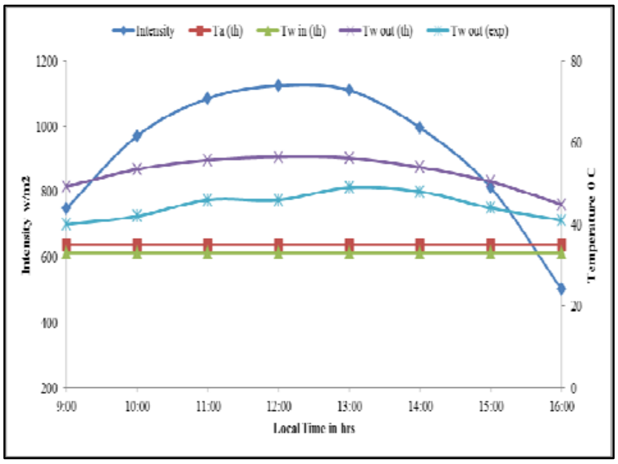

Fig. 10. Variation of Time vs. Temperature and Intensity at Mass flow rate of 18kg/hr for Lc/Le .5384.

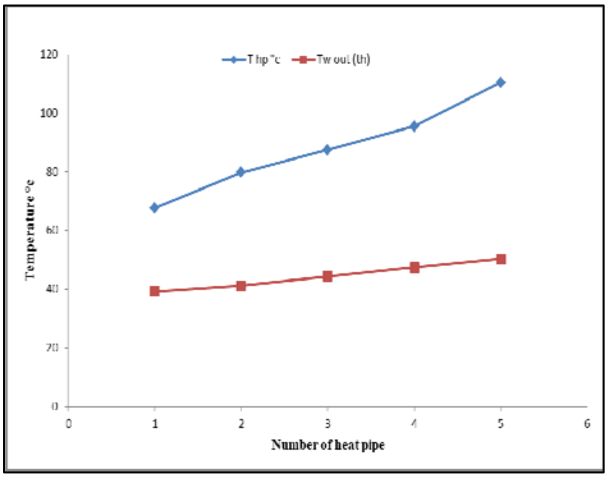

Fig. 11. Temperature distribution vs. number of heat pipe.

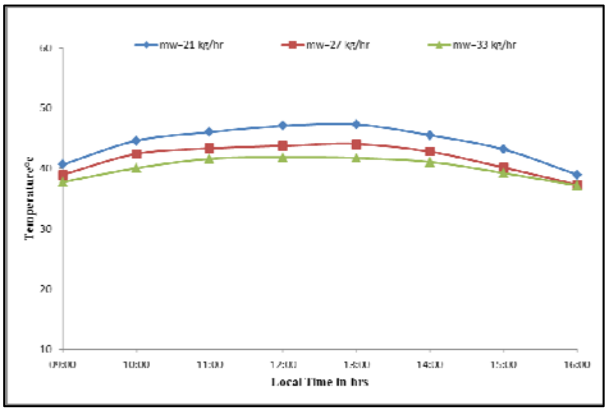

Fig.12. Variation of temperature vs. time for 0.1764 Lc/Le ratios

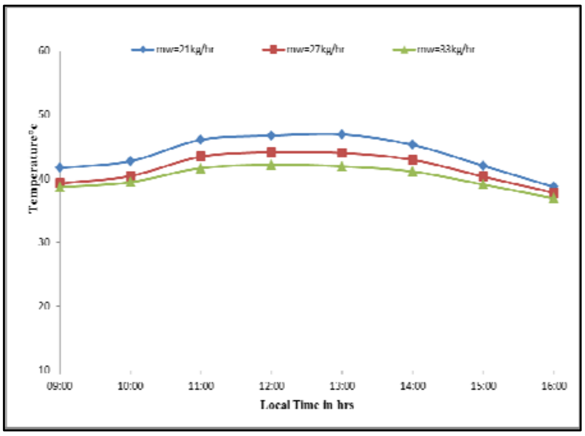

Fig. 13. Variation of temperature vs. time for 0.333 Lc/Le ratios.

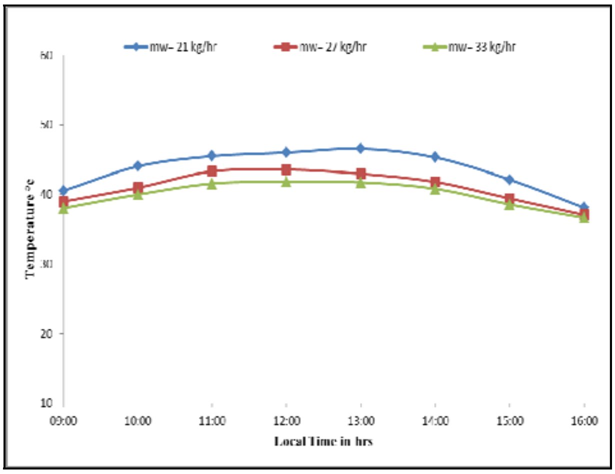

Fig. 14. Variation of temperature vs. time for 0.5384 Lc/Le ratios.

V. Conclusion

This research work was conducted at Chidambaram 11°24'N 79°42'E , situated in the southernmost part of Indian peninsula of India, near the equator, the sub- tropical region of the northern hemisphere near the Coromandel coast where the prevailing hotness perceived mostly throughout the year except small summer and winter season.

The evaporator length and condenser length of 0.65 cm & 0.35 cm, 0.75 cm & 0.25 cm, and 0.85 cm & 0.15 cm respectively and mass flow rate of 18 kg/hr, 24 kg/hr, 30 kg/hr and 36 kg/ hr were taken for the present study.

The equations are formed for the theoretical analysis is accordance to the experimental setup and these equations are tested with experimental values. Then, with the help of computer, equation and values simulations were performed and simulated values are presented in the figure…..

With the help of graphs and tabulated values following inferences are drawn.

Mass flow rate of 18 kg/hr is the best mass flow rate in this experimental setup compare to other mass flow rate. Evaporator and condenser length of 0.65 cm & 0.35 cm performs well compare to other combination of evaporator and condenser lengths.

In general, during 11 am to 2 pm, at this place, the solar intensity is high. Also, the elliptical heat pipe performed well under this condition. The outlet water temperature of experimental and theoretical value, though it follows the same trend, it is very low compare to heat pipe temperature and it shows heavy loss during the exchange of heat from working fluid to water.

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