A Heat Pipe Performance With Heptane as a Working Fluid

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ABSTRACT

The heat pipe is a device that efficiently transfers heat from one end to another. It has been widely applied in electronic system cooling and heat spreading applications due to its superior heat conductivity. Many parameters affect the performance of the heat pipe which are worthwise to be investigated. For this reason an experimental rig was designed and constructed to study the important parameters such as the heat load, the working fluid charge and the angle of inclination. Calibrated thermocouples were inserted into the heat pipe outside surface, along and around it, to measure and to show the temperature distribution profile. Heptane was used as a working fluid. Results show that the maximum conductivity obtained was more than one thousand times that of stainless steel solid bar which is the material of the heat pipe container. Also, results of the experimental work show a good agreement with that obtained from theoretical and empirical correlations derived by other researchers especially when the power input is lower than (1000)W where after a dry out condition can be clearly seen.

Keywords: heat pipe; wick; filling ratio.

دراسة عملية لأداء أنبوب حراري باستخدام الهيبتان كمائع تشغيلي

الخلاصة

أنبوب الحرارة جهاز كفوء وفعال جداً لنقل الحرارة من طرف إلى أخر، يستخدم في الوقت الحاضر بصورة واسعة في تبريد المنظومات الإلكترونية وانتشرت تطبيقاته الحرارية بسبب موصليته الفائقة. العديد من العوامل تؤثر على أداء أنبوب الحرارة التي تستحق ذلك الجهد في البحث والتحقق فيها. لهذا السبب تم تصميم الجهاز المختبري وتصنيعه لدراسة تأثير العوامل الحاكمة المختلفة، كالطاقة المجهزة لمبخر الأنبوب الحراري،كمية شحن السائل التشغيلي وزوايا الميل المختلفة. ثبتت متحسسات حرارية معيّرة على طول وحول سطح أنبوب الحرارة لقياس وبيان توزيع درجات الحرارة. أستخدم الهيبتان في هذا البحث كسائل تشغيلي .أظهرت النتائج التجريبية أن الموصلية الحرارية عالية وهي بحدود أكثر من ألف مره بقدر توصيلية قطعه صلدة من الفولاذ التي صنع منها الأنبوب الحراري. كانت النَتائج التجريبية للعمل الحالي متفقة بصوره جيدة مع نتائج معادلات عملية ونظرية لباحثين آخرين، خاصه عند الطاقة المجهزة الاقل من (1000w) بعدها يمكن ملاحظة ظاهرة الجفاف بوضوح .

NTRODUCTION

There has been renewed interest in the use of heat pipes for thermal management due to increasing heat flux requirements and thermal constraints in many industrial applications. The thermal performance of a wicked heat pipes is typically characterized by both its maximum heat transport rate and its effective thermal resistance. The heat pipe operates on a closed two phase cycle. Figure (1) shows a schematic of a typical heat pipe set up. Inside the container there is a liquid under its own pressure that enters the pores of the capillary material, wetting all internal surfaces. When the heat is supplied to the evaporator, this equilibrium breaks down, as the working fluid evaporates, causing the liquid at that point to boil and enter a vapor state. The vapor, which then has a higher pressure in the evaporator than that in condenser, vapor moves inside the sealed container to a colder section where it condenses. Thus, the vapor gives up the latent heat of vaporization. In other words, evaporation and condensation result in transfer heat from the input to the output end of the heat pipe.A considerable experimental and theoretical works have been done on the application and design modification for improving heat pipe performance. Shwin-Chung et al. [1] in their work presented visualization of the evaporation/boiling process and thermal measurements of operating horizontal transparent heat pipes. Kempers et al.[2] presented in their study the heat transfer mechanisms in the condenser and evaporator sections of a copper-water wicked heat pipe with three layers of screen mesh. Bilegon and Fetcu. [3] Have conducted an experimental study of the heat transfer characteristics of the gravity –assisted aluminum extruded heat pipes.

Yahya [4] carried out an experimental study concerned with the development of an indirect type of solar cooker using a heat pipe for transporting energy from the focal spot. Based on the results of the heat pipe, a complete solar cooking system was developed and tested to find its effectiveness. Frank [5] conducted an experimental investigation of improved injection lances with heat pipe having two wraps of stainless steel mesh as a wick structure in the heat pipe. The present work investigates experimentally heat transfer characteristics performance of HP. Tests for different orientations, different working fluid charges and a range of heat loads were carried out. One of the most important design parameters involved in different applications of two-phase flow heat pipes, is the heat transfer coefficient. This parameter was calculated from the experimental results of this work and compared with well known theoretical and empirical correlations of Rohsenow and Imura [6, 7, 8]. The comparison between the present work and the above correlations was agreed well especially at power input is less than (1000)W.

EXPERIMENTAL APPARATUS

An Experimental investigation was performed in order to evaluate the performance of the heat pipe, by varying the important parameters such as power supply, inclination angle on the transport behavior in the heat pipe charged with an amount of a high-urity n-heptane (99.5%) as a working fluid. Heptane is a hydrocarbon fluid (C_7H_{16}) with density of (0.612) g/cm^3 and a boiling point of (98) ^oC at atmospheric pressure, as shown in Table (1). To accomplish this goal, an experimental rig was designed and constructed consisting of the heat pipe body made from stainless steel as shown schematically in Figure (1) with length of(1200)mm, outside and inside diameters of (48)mm and (43)mm respectively. The heat pipe was heated by electrical coils clamped on the evaporator section and it is cooled by water flowing through a jacket along the condenser. Between the evaporator and condenser there is a well insulated (adiabatic zone). The pipe is lined with capillary structure of three layers of stainless steel screen wire mesh. Twenty two calibrated thermocouples (chromel – alumel; type K) used in temperature measurement were fixed in equally spaced 110mm, and around the external circumference. The sheaths of the thermocouples were fully insulated as shown in Figure (1). The temperature was read directly. A pressure gauge is mounted on the top of the heat pipe for measuring the inside pressure. Heat losses by radiation and convection to the surrounding were minimized by applying wrapping thermal insulation directly to the outside surface of the heaters (glass wool and asbestos insulator with thermal conductivity of the $k_{ins.}$=0.03 W/m.^{o}C ; $k_{as.}$ =0.11 W/m.^{o}C, respectively) as shown in Figure (1). An accurate wattmeter covering the anticipated power range, and a variac were incorporated in the heater circuit to record the exact power supplied as shown in Figure (1). The heat output from the condenser was calculated from the temperature rise of the cooling water and its flow rate. A rotameter (Metering float type; with flow rate rang 0 -1000 ml/min.) is used to measure the amount of cooling water flowing through the condenser jacket. The heat pipe was charged with a working fluid through valve 1,as shown in Figure(1). Before operation, all the materials used in the heat pipe must be cleaned by trichloroethylene, acetone and dematerialized water. Cleanliness achieves two objectives; it ensures that the working fluid will wet the materials and that no foreign matter is present which could hinder capillary action or create incompatibilities [3].

RESULT AND DISCUSSION

Start-up operation

The startup operation HP needs some time in order to discharge excess liquid and to stabilize the vapor-liquid interface at the top (i.e. small fluctuation of temperature with time). Generally, whole startup process needs about two hours, as shown in the Figure (2).

Effects of Filling Ratio and Power on temp. distribution In this study the quantity of the working fluid in the HP is defined by the ratio of the working fluid to the entire volume of the evaporator. Three different fluid charges (25%, 50% and 75%) were considered.Figures(3)and(4) show the variation of the temperature along the heat pipe for three different filling ratios and heating loads rate of(250) and (1500) Watt, at different inclination angles respectively. It is obvious that the temperature of the evaporator is almost constant, particularly, at lower power and starts to increase with the increase of power from the bottom of the evaporator to a maximum value and decreases until the adiabatic section, where more temperature significant gradient can be noticed, due to boiling of working fluid forming a blanket of bubbles in the middle of the evaporator which in turn means an increase in temperature. The wall temperature distribution along the condenser is almost constant. It can also be seen that the temperature increases when the filling ratio decreases.According to Figure (5) the heat input to the HP does not exceed 1000W otherwise the dry out condition will happen. As expected, the evaporator wall temperature increases with the power input and it remains relatively isothermal at lower heat transfer rates. From Figure (6), at higher heat transfer rates, the temperature at the end of the evaporator is still lower than the rest of the temperatures along the evaporator. The reason of this lower value is probably due to the fact that the bottom end of the evaporator was not completely insulated, in addition to the heat transfer by conduction along the tube, resulting in heat leak from the evaporator body. The slope of the temperature traces for these locations is significantly smaller than the rate of rise of the other evaporator body temperatures.

Effects of the inclination angle

The heat transfer by boiling in the evaporator section shows larger local differences depending on the inclination of the HP. The intention here to gain better insight into the transport processes of HP by observing local phenomena in different parts of the device especially the effect of the inclination of HP. This behavior is observed as shown in figures (7)to(9). Those figures show the temperature against the distance along HP for three values of filling ratios. Thus, they show that the temperature of the evaporator is higher when the inclination angle (β) is $(25)^o$ and decreases when β equals$(50)^o$ and $(75)^o$ and starting to increase when the angle becomes $(90)^o$. As the inclination angle decreases from vertical ($\beta=90^o$), the contact area between the heating surface and the working fluid increases which in turn gives an increase in heat transfer coefficient. While for the filling ratio (25)% the temperature decreases as the inclination angle increases from $(25 \text{ to } 90)^o$. This case is obviously shown in all of the previous figures**.**

Effects of Filling Ratio on the heat transfer coefficient

Figure (10) represents the average temperature of the evaporator wall against the heat transfer coefficient for different filling ratio charges at a given inclination angle. The large value of the heat transfer coefficient, which can be seen from the Figures, is at (50)% charge for all power inputs at angles(90^o, 75^o and 50^o) while when the HP was tilted by an angle $(25)^o$, as shown, the higher values of heat transfer coefficient are at (75)% filling ratio. This is due to that the liquid still within the evaporator zone(heating zone) which in turn means more heat transfer area between liquid and the heating surface or in other word there is an increase in the heat transfer coefficient. When the filling ratio more than 75% with inclination less than 90^o, flooding limit may occur since the working fluid will pass the upper end of the evaporator section. So, it can be said that the heat transfer coefficient reached the maximum value about (2880) $W/m^2.{}^oC$ at (75) % filling ratio when HP was tilted at angle $(50)^o$.

Comparison of the Exper. Results with Theoretical and Empirical Correlations

The experimental results were compared with the correlations of Rohsenow and Imura [7, 14].The theoretical model assumed implicitly that the HP is positioned vertically. From the measured data of wall temperature distribution, vapor temperature and thermal load, the experimental heat transfer coefficient in the evaporator ($h_{exp.}$)can be evaluated using the following equation:

$$h_{\exp.} = \frac{Qav.}{p \cdot D_i \cdot l_e \cdot (T_{w,av.} - T_{v,av.})} \qquad \text{--- (1)}$$

Where T_w and T_v represent the average temperature value of evaporator surface and adiabatic section respectively.In a previous study [2], it was found that nucleate boiling is the dominant mechanism in the evaporator, when the filling ratio is higher than (30) %. Therefore, the following two correlations based on nucleate boiling were chosen to compare with the experimental data.

$$h_{Rohsenow} = \frac{q_e^{2/3}}{\frac{C_{sf} \times h_{fg}}{C_{p,l}} \cdot \left\{ \frac{1}{h_{fg} \times m_l} \left[\frac{s_l}{g \cdot (r_l - r_v)} \right]^{0.5} \right\}^{1/3} \cdot \mathrm{P}_r^{1.7}} \qquad \text{........(2)}$$

$$h_{\mathrm{Im}\,ura} = 0.32 \cdot \frac{r_l^{0.65} \cdot k_l^{0.3} \cdot C_{p_l}^{0.7} \cdot g^{0.2} \cdot q_e^{0.4}}{r_v^{0.25} \cdot h_{fg}^{0.4} \cdot m_l^{0.1}} \left[\frac{P_{v.}}{P_{atm.}} \right]^{0.3} \qquad \text{--- (3)}$$

Figure (11) shows the comparison of heat transfer rate between the experimental work (Equation.1) and the theoretical correlation (Equtuons.1 and 2)for tilt angle$(90)^o$ and filling ratios(25%, 50% and 75%) .It is clear from the figures that the experimental results of heat transfer coefficient well agreed up to power input 1000W with that calculated by the theoretical and empirical correlations specially when the filling ratio is (50)% and (75)%. For filling ratio (25)% experimental results of heat transfer coefficient are slightly lower than that obtained by theoretical and empirical correlations.From the figure it can be seen that the phenomenon of dry out occurs. Dry out can be seen from a sudden sharp drop in wall temperature. When the power exceeds (1000)W. However, the effect of the above phenomenon can be explained as follows: with further increase in power input, the small bubbles coalesced in the capillary wick forming vapor patches acting as an isolating blanket preventing the heat flow from heaters to the working fluid. These processes impede the liquid flow in the wick and the evaporation zone of liquid would dry out gradually, limiting the heat transfer process. Although the heat can be removed from the evaporator, the durable dry out region extends and the average heat transfer coefficients decrease rapidly [13].It is worthwhile to point out the value of C_{sf} included in Eq.1 ranges between 0.0025 and 0.015[14]. From the present work it was found that the value of 0.0075 is suitable to the present work which makes it well greed with the theoretical and empirical correlation.

CONCLUSIONS

From the present work, the following conclusions can be extracted:

1-The temperature distribution along the HP wall in the evaporator section is oscillating within (±9) degree centigrade as shown in Figure (2). The measured temperature along the condenser shows lower values. This drop of temperature is expected because of the internal resistances due to boiling and condensation.

2-The average outside temperature of evaporator section is low when the filling ratio is (50%) and the inclination angle is $(50)^o$.
3-The experimental results indicate that the filling ratio and the heat flux have the important effects on the heat transfer performance. The optimal performance of the HP was found when the filling ratio ranged between (50%and75%), at $(50)^o$ inclination angle, while the minimum performance was found when the filling ratio was (25)% and $(25)^o$ inclination angle.
4- Maximum thermal conductance (which reciprocal of the thermal resistance) of the heptane HP was found to be about 1050 times that of a stainless steel piece of the same size.
5-When the power input in HP exceeded (1000)W, the phenomenon of dry out occurred, indicated by the sharp rising of the out side wall temperature of the evaporator.
6-The experimental heat transfer coefficient results well agreed with the theoretical and empirical correlations, especially when the power input is lower than 1000W.
7-From the experimental results obtained, $\mathbf{C_{sf}}$ which depend upon fluid and surface is found to be (0.0075) when using heptane as a working fluid.

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Nomenclature

A	area,	m^2
C_p	specific heat,	J/kg.°C
C_{sf}	Constant, exper. data	
D	diameter,	m
FR	filling ratio	%
g	gravitational ,	m/s^2
h	heat coefficient,	W/m^2K
h_{fg}	Latent heat,	kJ/kg
HP	heat pipe	
k	conductivity,	W/m.°C
L	pipe length,	m
P	pressure,	bar
Pr	Prandtl number	$\mu.C_p$/°C
Q	heat transfer rate,	W
q	heat flux,	W/m^2
T	temperature,	^oC

<u>Greek symbols</u>

β	inclination angle from horizontal line	o
μ	viscosity,	Ns/m^2
ρ	density,	kg/m^2
σ	surface tension,	N/m

<u>Subscripts</u>

a	adiabatic
av.	average
exp.	experimental
c	condenser
e	evaporator
v	vapor (adiabatic section)
w	evaporator wall

Table (1) Properties of Haptane [6].

Properties	Data
Chemical symbol	C7H16
Purity	95% (high- purity)
Toxicity	Toxic
Flammability	Flammable (sever fire hazard)
Pressure	atmospheric
Temperature(^{o}C)	100
Latent heat (kJ/kg)	319.6
Liquid density (kg/m^3)	612
Liquid thermal conductivity(W/m.^{o}C)	0.133
Liquid viscous (cP)	0.21
Liquid surface tension (N/m $\times 10^2$)	1.28
Melting point Temperature(^{o}C)	-98
Boiling point Temperature(^{o}C)	98
Useful point Temperature range (^{o}C)	0 - 150

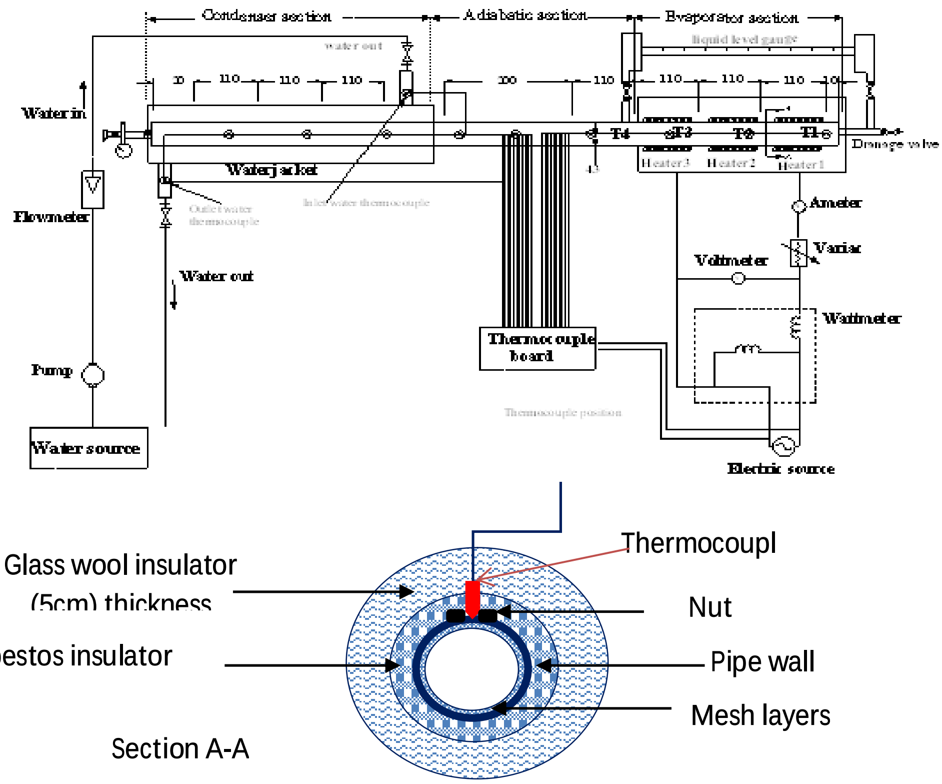
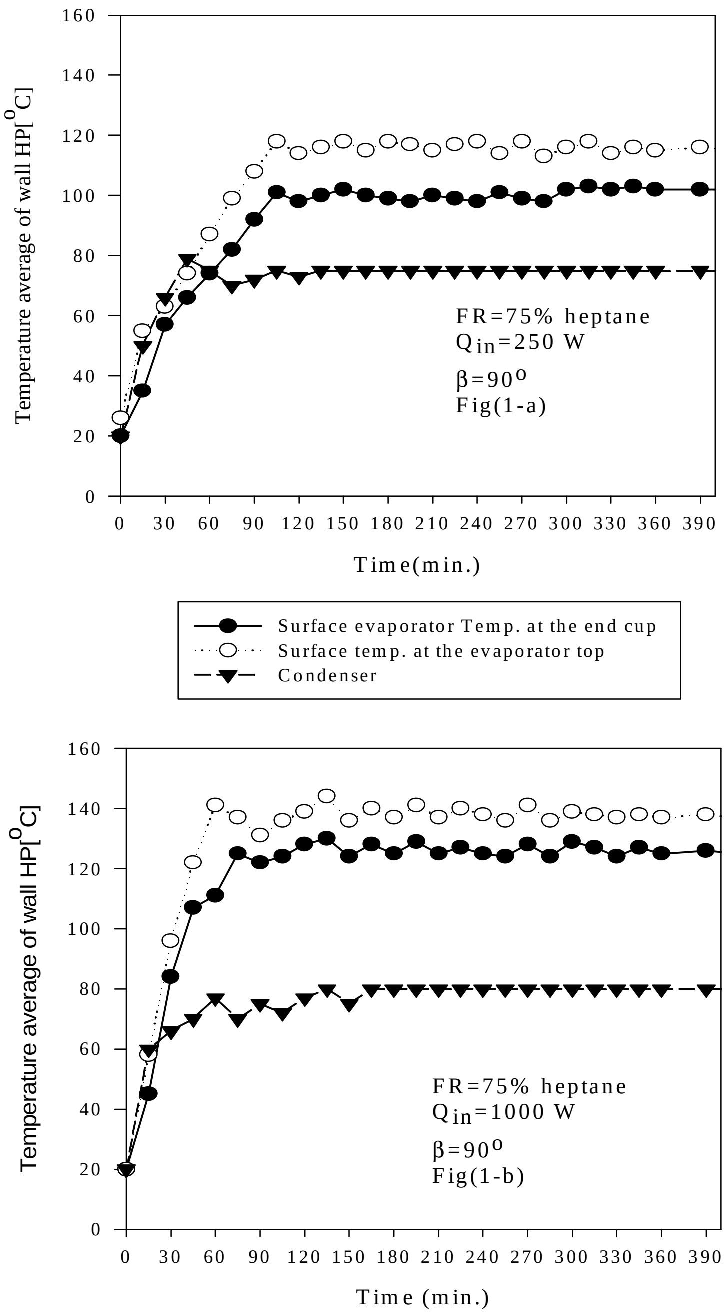
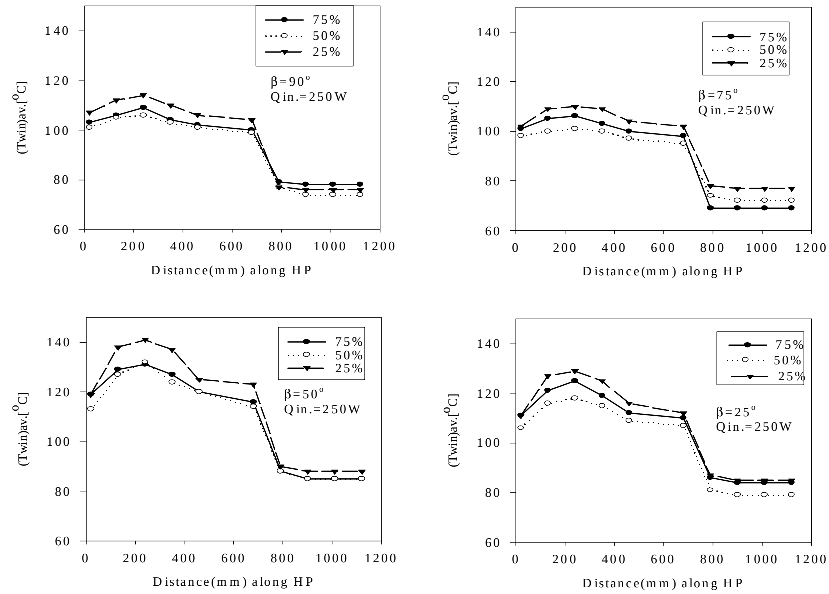


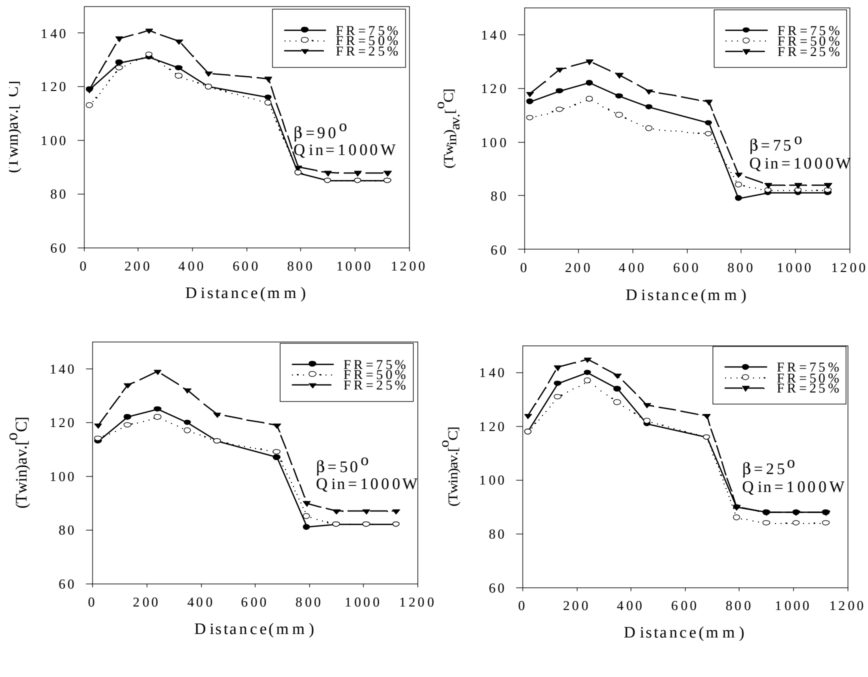
Figure (1) The experimental rig diagram.



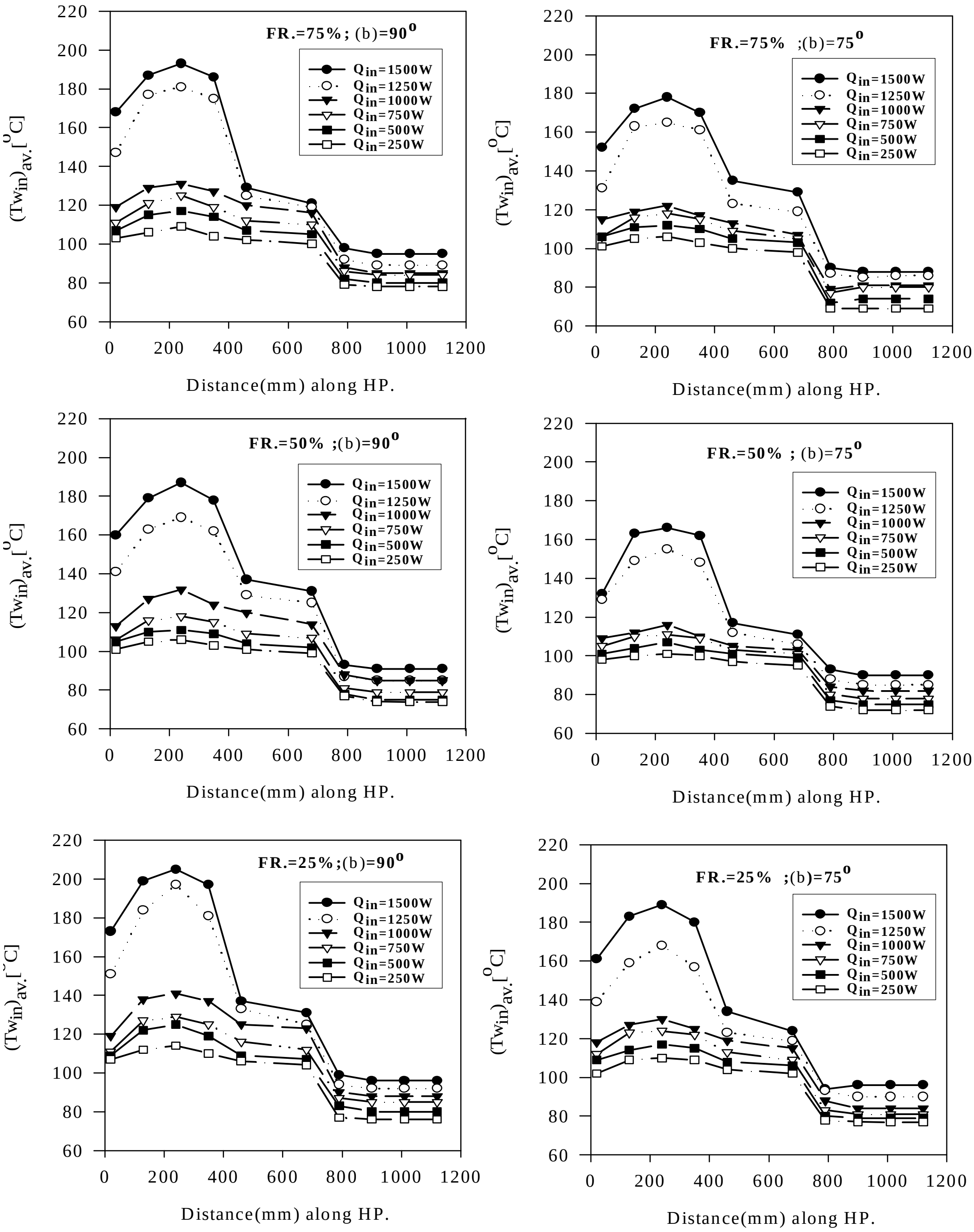
Figure(2):Temperature variation of the HP evaporator and condenser surface with time in the startup process: (a) at 250W ; (b) at 1000W.



Figure(3):Variation of average wall temperature along the HPat different heptane filling ratios and orientations for a given power input.



Figure(4) Variation of average wall temperature along the HP at different filling ratios and orientations at a given power input.



Figure(5) Variation of average wall temperature along the HP at different heating loads and filling ratios for a given orientation.

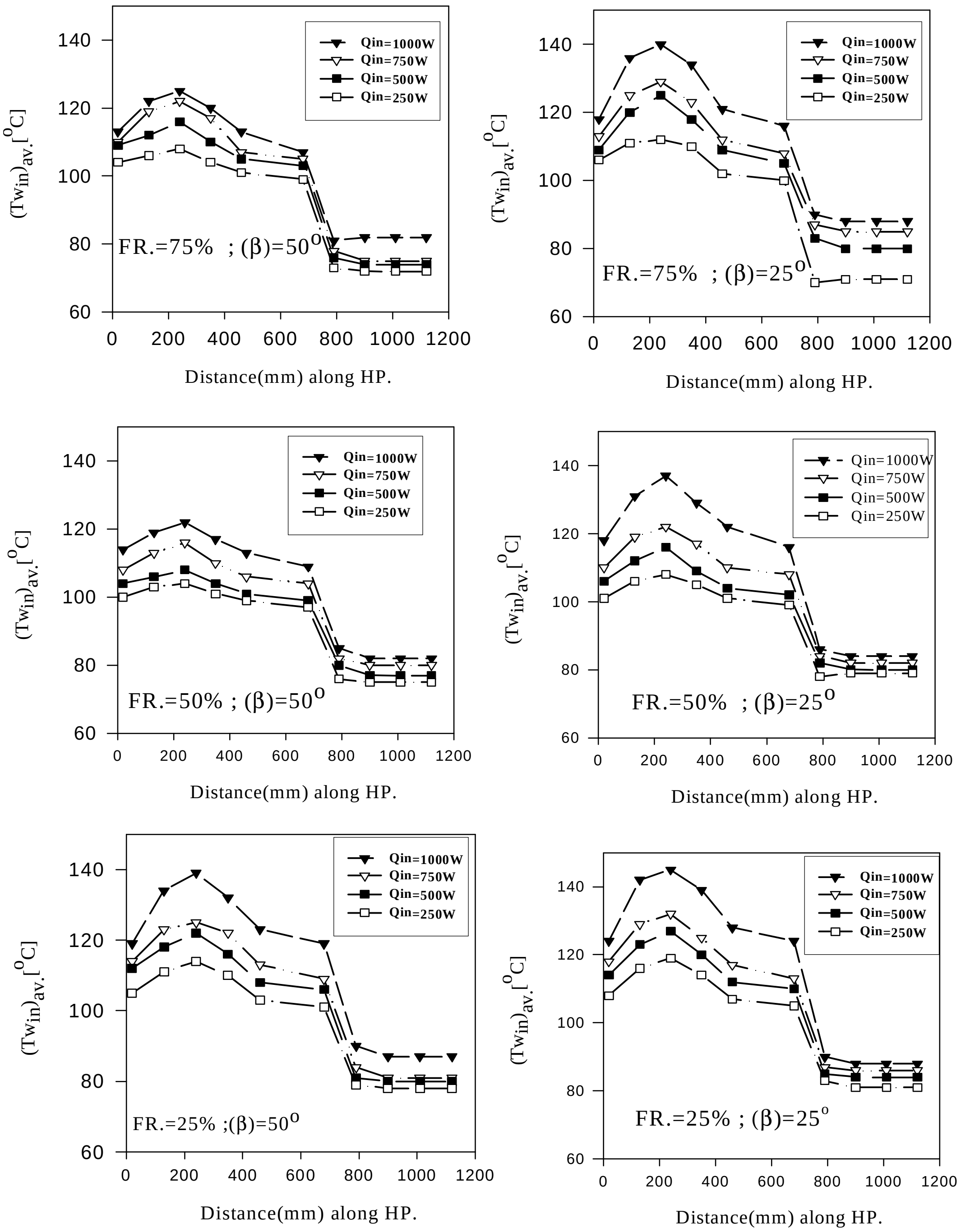
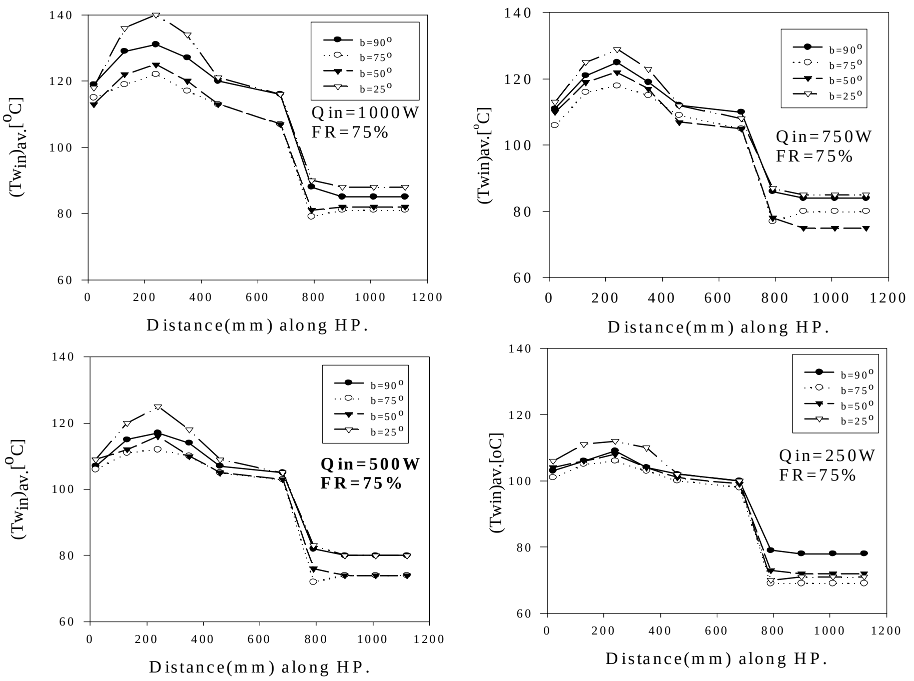
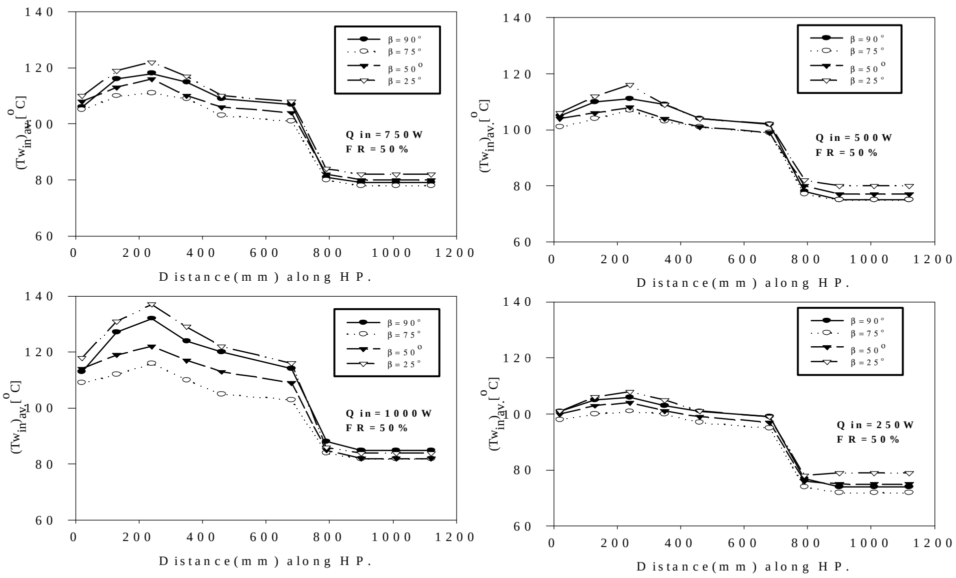


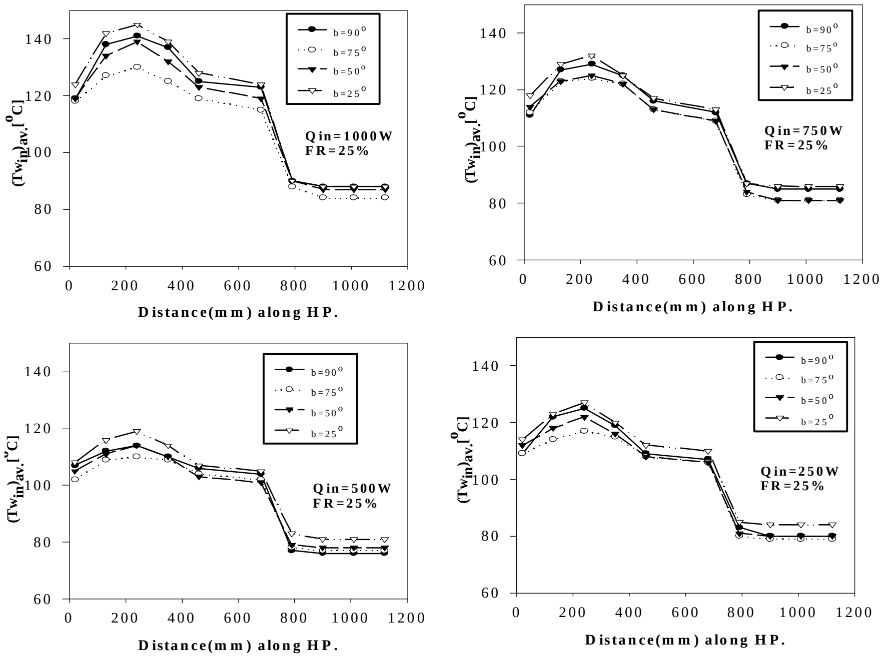
Figure (6) Variation of average wall temperature along the HP with different powers and filling ratios at a given orientation.



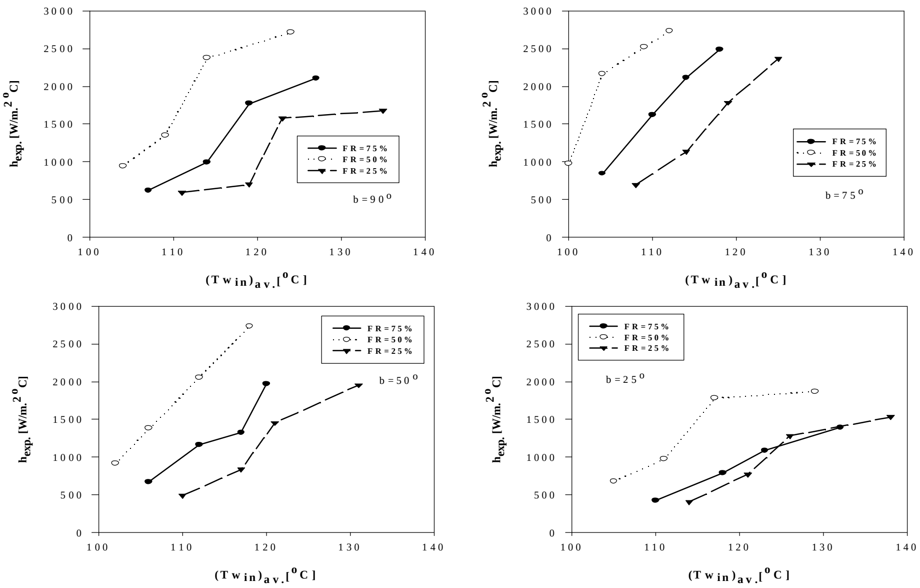
Figure(7) Variation of average wall temperature along HP with different orientations at a given power input and filling ratio.



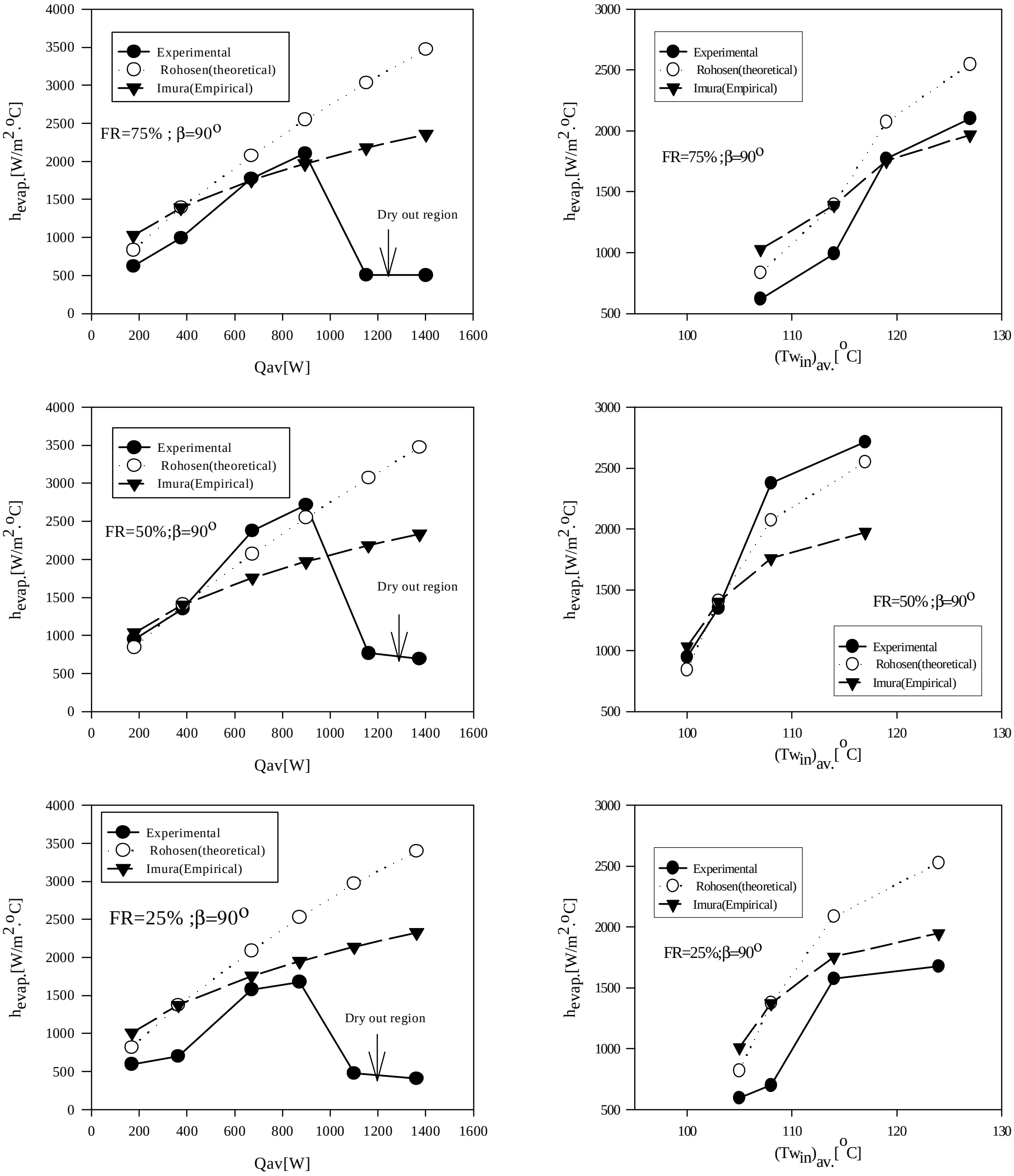
Figure(8) Variation of average wall temperature along HP with different orientations at a given power input.



Figure(9):Variation of average wall temperature along HP at different orientationsat



Figure(10) Heat transfer coefficient vs. average wall temperature of evaporator at a given power input different filling ratios and a given inclination angle.



Figure(11) Heat transfer coefficient vs. average heat input for experimental, theoretical and empirical correlations at inclination angle 90° and diff. FR.