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Boiling Heat Transfer in Closed Loop Heat Pipe.

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BOILING HEAT TRANSFER IN CLOSED LOOP HEAT PIPE
"إنتقال الحرارة بالغليان فى أنبوب حرارى ذى دائرة مغلقة"

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خلاصة البحث:

فى هذا البحث أجريت دراسة معملية لدراسة إنتقال الحرارة بالغليان فى أنبوب حـرارى ذى دائــرة مغلقة وذلك مع إستخدام وسط مسامى فى المبخر. ولإتمام هذا الغرض تم تصميم وتنفيذ دائرة إختبار معمليـة لمقارنة عملية إنتقال الحرارة بالغليان عند إستخدام الوسط المسامى المشبع بالماء النقى مقارنة بالمــاء النقـى فقط كمانع تشغيل وذلك عند تغيير الفيض الحرارى من ١٦٥٠٠ إلى ٥٦٢٠٠ وات/م'. وقد تم تغيير المسامية للوسط المسامى عن طريق إستخدام كرات من الحديد المطلى بطبقة مـن النيكـل بأقطـار ٤,٧٣ ، ٣,٩٤ ، ٣,١٢ ، ٢,٤٧ مم. حيث زودت دائرة الإختبار بأجهزة القياس المطلوبة لتسجيل القياسات الخاصـة بـدرجات الحرارة والضغط والسريان وذلك لحساب معامل إنتقال الحرارة بالغليان للأنبوب الحرارى. وقد تم حسـاب معامل التحسين فى معامل إنتقال الحرارة بالغليان عند إستخدام الوسط المسامى المشبع بالماء النقى فى المبخر مقارنة بالماء النقى فقط.

وقد أظهرت النتائج أن درجة حرارة سطح المبخر تقل مع إستخدام الوسط المسامى مقارنة بالماء فقـط وبالتالى فإن معامل إنتقال الحرارة بالغليان للأنبوب الحرارى يزيد مع إستخدام الوسط المسامى وكذلك يزيـد مع زيادة الفيض الحرارى. أيضا أظهرت النتائج أن معامل التحسين فى معامل إنتقال الحرارة بالغليان يزيـد مع نقصان المسامية حتى قيمة معينة وهى ٠,٤ يبدأ بعدها فى الإنخفاض. وقد وجد أن أفضل ظروف تشغيل للأنبوب الحرارى المختبر هى إستخدام وسط مسامى له مسامية ٠,٤ عند فيض حـرارى ٣٨٦٠٠ وات/م، ليعطى أكبر قيمة لمعامل التحسين فى معامل إنتقال الحرارة بالغليان وهو حوالى ٢,٥. وقـد تمـت المقارنـة بالأبحاث السابقة حيث أعطت نفس الإتجاه.

Abstract

An experimental study for the boiling heat transfer in a closed loop heat pipe was achieved by using saturated porous medium instead of pure water in the evaporator section. An experimental test loop equipped with the required measuring devices was designed and constructed to assess the effects of applied heat flux, porosity of porous medium in the evaporator section on the boiling heat transfer process. The tested porous media are balls made from iron coated with nickel. Four porous media are tested, each one consists of balls of diameter 4.73, 3.94, 3.12, and 2.47 mm respectively. The experimental measurements of temperature, pressure and volume flow rate are recorded and manipulated to calculate the boiling heat transfer coefficient and the enhancement factor for the boiling heat transfer coefficient when using porous media instead of water only in the evaporator section of heat pipe.

The obtained experimental results show that, the evaporator wall temperature was reduced by using porous media compared with water only. Accordingly, the boiling heat transfer coefficient increased for closed loop heat pipe when using porous media and increased also with increasing heat flux. The enhancement factor of the boiling heat transfer coefficient increased with decreasing porosity up to the value of 0.4 then it decreased. The optimum operating conditions for the studied closed loop heat pipe when using porous media in the evaporator section at porosity about 0.4 and heat flux about 38600 W/m^2. There, the enhancement factor takes the highest value equal to about 2.5. Comparison with the previous work gave the same trend.

1. INTRODUCTION

Traditional air cooling for electronic components is predominant for lower values of heat dissipation (lower than 0.05 W/cm^2). Higher heat flux rates (up to 100 W/cm^2) required efficient methods for cooling by increasing heat dissipating area which is in most cases difficult to be achieved. Therefore compact closed loop two-phase thermosyphon heat pipe is the optimum solution for cooling of high heat dissipating electronics. Moreover, many researchers have used porous coating or rough surfaces to enhance nucleate boiling and increase the thermal performance of the system.

Loop thermosyphon heat pipe consists of four components in a loop. These components are evaporator, rising tube, condenser and falling tube. Heat is added through the evaporator where vaporization of working fluid takes place. The vapor then passes through the rising tube to the condenser, where the vapor releases its latent heat and gets liquefied. The liquid flows toward the evaporator section through the falling tube due to gravity. The basic advantage of the compact closed loop two-phase thermosyphon heat pipes is the ability to dissipate high heat fluxes due to the latent heat of vaporization and condensation at much lower temperature difference between the evaporator wall and the working fluid. This leads to reduce the weight and volume with smaller heat transfer area compared with other systems. Closed loop thermosyphon heat pipe have numerous practical applications like solar heaters, energy storage, geothermal energy, heat rejection from nuclear reactor core, cooling of internal combustion engines, turbine blades and computers. Therefore design of these engineering applications is very important in modeling of various kinds of thermosyphon heat pipes.

Launay et al. (2007) presented a literature review on loop heat pipe based on the most recent published experimental and theoretical studies. This review was carried out to investigate how various parameters affect the loop heat pipe operation-characteristics. The loop heat pipe thermal resistance and maximum heat transfer capability are affected by the choice of the working fluid, the fill charge ratio, the porous wick geometry and thermal properties, the sink and ambient temperature levels, the design of the evaporator and compensation chamber, the elevation and tilt, the presence of non condensable gases and the pressure drops of the fluid along the loop.

A cryogenic loop heat pipe has been developed for future aerospace applications by Mo and Liang (2006). Their device has demonstrated to be able to operate in liquid –nitrogen temperature range using nitrogen as working fluid and to transfer large amount of heat over long distance with very small temperature difference. Also, the application of loop heat pipes as two-phase thermal control devices for space missions has been considered and successfully used in many spacecraft, as mentioned by Riehl and Dutra (2005)

A short review of heat pipe was presented by Vasiliev (2005) which testifies that heat pipes are very efficient heat transfer devices which can be easily implemented as thermal links and heat exchangers in different systems to ensure the energy saving. Miniature loop heat pipe can be used for electronics cooling, as mentioned by Pastukhof et al., (2003). Closed loop rectangular thermosyphon was studied using experimental and two-dimensional numerical method for uniform wall temperature by Basaram and Kucuka (2003). It was found that the two-dimensional model provides meaningful results with the experiments for calculating the heat transfer rate.

The effect of using threaded surface in the evaporator section was studied experimentally by Khodabandeh and Palm (2002) using R134a and R600a as working

fluids. It was found that with threaded surface, the temperature difference in the evaporator decreased to about half for R134a and even more for R600a. The enhancement of heat transfer in the threaded tube was obtained due to a larger number of active nucleation sites.

Osman et al., (2002) studied experimentally heat transfer performance of a gravity assisted heat pipe filled with porous medium using R-11 as the working fluid. The results indicate that, the evaporative heat transfer coefficient and overall conductance of the heat pipe are improved with decreasing particle to heat pipe diameter ratios and increasing particle to fluid thermal conductivity ratios for the same packing material diameter.

Muraoka et al., (2001) analyzed the operational characteristics and limits of a loop heat pipe with flat porous elements in the condenser and evaporator. Liao and Zhao (1999) tested loop heat pipe with a spherical glass bead diameter ranging from 0.55 to 1.99 mm. A small pore size was recommended to obtain maximum thermal load. Crowe et al. (1993) carried out experimental and theoretical investigations to study the performance limits of a copper heat pipe with copper porous wick using either water or R-11 as a working fluid. It was found that, shorter adiabatic section length gives better performance. It was also concluded that, the maximum heat flow rate was obtained for heat pipes having low porosity wick.

Loop thermosyphon heat pipe normally uses a single component working fluid, usually water, because water has a high latent heat of vaporization and high working temperatures range (40 °C to 200 °C). According to the author knowledge it is observed that, little work was concerned with enhancement for boiling heat transfer by filling evaporator section with porous medium. Therefore, this work presents an experimental study for boiling heat transfer inside evaporator section for closed loop heat pipe. Moreover, the effect of input heat flux, and porosity on the process of boiling heat transfer of closed loop heat pipe was examined.

2. EXPERIMENTAL TEST LOOP

A schematic diagram for the experimental test loop is shown in Fig. (1). It consists of a closed loop heat pipe, a heating unit and a cooling unit. Closed loop heat pipe consists of evaporator section (4), rising tube or vapor line (6), water cooled condenser (5) and falling tube or condensate line (13). Pure water is used as a working fluid. The vapor generated at the evaporator section is naturally moved to the condenser through the vapor line. Then, the condensate returned back, through falling tube, to the evaporator by gravity.

The evaporator section is made of a hollow Pyrex tube inner diameter 62 mm, outer diameter 74 mm, and height 60 mm. Lower and upper metal bases are assembled with Pyrex tube with eight bolts, as shown in Fig. (2). Lower base and upper one are made of iron and coated by nickel layer to protect it from rust and other environmental effects. The specified amount of pure water was charged through the falling tube. The evaporator section was filled with pure water to a level equal to 20 mm. In case of using porous media the level of water plus balls is equal to 20 mm also. Porous medium was balls made from iron coated with nickel to protect them from rust and other environment effects. Four different ball diameters were tested; 4.73, 3.94, 3.12, and 2.47 mm. The corresponding porosity for porous media are calculated as mentioned in Appendix (A) and found to be 0.66, 0.54, 0.4 and 0.27. The evaporator section is heated from the lower metal base by an electric heater 500 W rated power, 220 V. Heating element is a copper rod 25 mm in diameter and 110 mm length. A hollow tube of Teflon encloses electric heater to insure that the generated heat moves only in the axial direction toward the lower surface of the evaporator lower metal base. Variable voltage power supply (variac) is used to

control and adjust the electrical power supplied to the electric heater.

The cooling unit is a closed condenser (shell and tube type). The generated steam from evaporator section flows through the inside condenser tubes, which made from copper and have 8 mm inner diameter and 1 m long. While cooling water flows in the shell side. Cooling water was pumped from constant head water tank to pass in the shell side, which made from clear acrylic. The basic dimensions of the shell are 40 mm width, 150 mm length and 160 mm height. To ensure minimum heat loss to the surroundings, a layer of 50-mm of glass wool thermal insulation followed by additional aluminum foil sheet is wrapped on the outer surface of the whole parts of loop heat pipe, except the Pyrex tube in the evaporator section to visualize the boiling phenomenon inside the evaporator.

3. EXPERIMENTAL MEASUREMENTS

Three thermocouples of k-type are attached inside the evaporator section. One is fixed in the middle of working fluid layer (pure water) to measure its temperature. The second one is suspended in the space above the water surface, to measure the generated steam temperature. These two thermocouples are inserted inside the evaporator section to measure the saturation temperature for the working fluid of the loop heat pipe. The third one welded on the inner surface of the evaporator lower metal base at radius of 15.5 mm to measure the inner wall temperature. The evaporator radius was 31 mm, therefore the choice for fixation of this thermocouple at that position gave an indication to the average value for surface temperature of the evaporator lower metal base. These three thermocouples are induced in the evaporator upper metal base through six small diameter copper tubes welded in it (where each wire of thermocouple enters through one of these tubes). These tubes are bended several times and injected by adhesive material (Defecon) to prevent leakage.

Two thermocouples are attached to the outer surface of the rising tube and falling tube to indicate the generated steam temperature and condensate temperature respectively. Also, two thermocouples are used to measure inlet and outlet cooling water temperatures to the condenser, as shown in Fig. (1).

Bourdon tube pressure gauge ranging from 0 to -1 bar was fixed at the top of heat pipe to measure the inside vacuum pressure. The absolute value for pressure inside closed loop heat pipe was fixed along the experiments at value equal to 0.12 bar (the corresponding saturation temperature was 49.5 °C).

Volume flow rate of cooling water for condenser was adjusted to the required value and measured by using a calibrated tank and stop watch.

The experimental apparatus was allowed to operate until the fluctuation in temperatures was about ±0.1 °C. Then, steady state condition was reached and the required measurements of temperature, pressure, and volume flow rate were taken. The root-mean-square random error propagation analysis was carried out in the standard fashion using the measured experimental uncertainties of the basic independent parameters. The error analysis is done for the average values of the calculating parameters. The experimental uncertainties associated with these measurement techniques were estimated to be approximately equal to 5.5 % for boiling heat transfer coefficient.

4. DATA REDUCTION

At steady state, the total input heat from the electric heater to the evaporator section (Q_t) divided into useful heat which flows through the evaporator and rejected in the condenser (Q_{us}) and the remaining amount of heat transferred to the surroundings as heat loss (Q_{loss}).

The total input heat can be determined as;

$$Q_t = (V^2 \cos \varphi)/R \qquad (1)$$

Where: V, cos φ and R are the applied voltage across electric heater, power factor and electric resistance for heater respectively.

The rejected heat transfer to cooling water in the condenser of closed loop heat pipe can be calculated as;

$$Q_{us} = m^{\cdot}\, Cp\, (T_o - T_i) \qquad (2)$$

Where m', Cp, T_i and T_o are the amount of cooling water mass flow rate, specific heat of cooling water, inlet and outlet cooling water temperatures respectively. Cooling water properties are calculated at average temperature ($T_{av} = (T_i + T_o)/2$).

Then, the amount of heat loss from loop heat pipe to the surrounding air (Q_{loss}) can be determined as the difference between input heat and useful heat as;

$$Q_{loss} = Q_i - Q_{us} \qquad (3)$$

Heat flux (q") can be calculated from the following equation as;

$$q'' = Q_{us} / A_{ev} \qquad (4)$$

Where; A_{ev} = Inside surface area for evaporator lower metal base ($A_{ev} = \pi\, d_i^2/4$). Also, d_i is the evaporator lower base diameter.

The experimental boiling heat transfer coefficient (h_b) was defined by the ratio of heat flux (q") and the temperature difference between the evaporator average wall-temperature ($T_{w,ev}$) and the saturation temperature for the working fluid (T_{sat}) as;

$$h_b = q'' / (T_{w,ev} - T_{sat}) \qquad (5)$$

The enhancement factor is defined as the ratio of the boiling heat transfer coefficient when using porous medium in the evaporator section ($h_{b,\varphi}$) to the boiling heat transfer coefficient when using pure water only (h_b);

$$EF = h_{b,\varphi} / h_b \qquad (6)$$

5. RESULTS AND DISCUSSIONS

Evaporator wall-temperature was measured and presented in Fig. (3), for evaporator when using porous medium for four tested ball diameters (different porosities) and compared with evaporator when using pure water only. It is observed that, evaporator wall temperature was reduced when using porous medium than pure water. This leads to enhance boiling heat transfer process when using porous medium than pure water. Also, it is observed that, evaporator wall temperature increases with increasing heat flux for the examined two cases when using porous medium and pure water only. From visual observation, by increasing heat flux the liquid in contact with the evaporator wall will become progressively heated and bubbles will form more nucleation sites. This leads to increase in the evaporator wall temperature.

The main aim of the research workers in the field of boiling heat transfer are to transmit the largest heat flux by applying the smallest temperature difference between the heating surface and the boiling liquid and to bring the critical heat flux to the highest possible value. Various means have been developed with this aim in mind, including the use of saturated porous medium instead of pure water only. Clearly, more nucleation sites would become active at a given superheat. Boiling curves for saturated porous media gave higher heat flux than pure water only (for the same temperature difference: $\Delta T_{ev} = T_{w,ev} - T_{sat}$), as shown in Fig. (4). The enhancement in the boiling heat transfer when using porous medium is attributed due to combination effects, including thinning the thermal boundary layer on the evaporator wall, increased mixing (thermal dispersion) and direct conduction through the porous matrix. For the tested porous media, the amount of boiling heat transfer was found to increase with decreasing

porosity up to $\varphi=0.4$ (ball diameter was reduced to 3.12 mm) after that value the trend was reversed. This reduction in the amount of boiling heat transfer (when using small ball diameter, d=2.47 mm) may be attributed by more increasing in the direct conduction through the porous matrix and more thinning in the boundary layer thickness causing coalescence and increasing for bubble departure diameter. This leads to increase in surface temperature for the same heat flux.

The effect of heat flux on the boiling heat transfer coefficient was illustrated in Fig, (5) for different values of bed porosity. It is clear that, as expected boiling heat transfer coefficient increases with increasing heat flux for the same bed porosity. But the rate of increasing is not equals for all bed porosities. Boiling heat transfer coefficient increases with decreasing porosity and take higher values at $\varphi=0.4$. After this value for porosity the boiling heat transfer coefficient was decreases.

Enhancement factor was plotted against heat flux as shown in Fig. (6). Enhancement factor increases with decreasing porosity and take higher values at $\varphi=0.4$. After this value for porosity the enhancement factor was decreased. .It is noticed that, for higher values of porosity at $\varphi=0.66$ the enhancement factor decreases with increasing heat flux. But for porosity lower than $\varphi=0.66$ the enhancement factor increases with increasing heat flux to reach the maximum value and then it decréases again. This trend was noticed clear at $\varphi=0.4$ and the enhancement factor takes highest value equal to 2.5 at q"=38600 W/m², after this value enhancement factor decreases.

Boiling heat transfer coefficient was plotted against porosity as shown in Fig. (7) for different values of heat flux. Boiling heat transfer coefficient increases with decreasing porosity for the same heat flux due to increasing the number of nucleation sites, reduction in bubble departure diameter, increase departure

frequency and decreased tendency to coalescence. But decreasing porosity than: $\varphi=0.4$ coalescence was increased and this cause a thermal shield around the surface. Therefore, evaporator surface temperature was increased and consequently boiling heat transfer coefficient was decreased.

The enhancement factor was plotted against porosity for different values of heat fluxes as shown in Fig. (8). It is clear from this figure that, the optimum values for the operating conditions at q"=38600 W/m² and $\varphi=0.4$ gave the highest value for enhancement factor which equals to 2.5.

Comparison between the present work, when using porous medium at $\varphi=0.66$, and the previous work which done by Osman et al. 2002 is illustrated in Fig. (9). Their heat pipe filled with porous medium using R11 refrigerant as working fluid. Rod bars packed bed of mild steel at rod to pipe diameter=0.2465 was filling the whole heat pipe (which used by Osman et al. 2002). It is observed that, comparison between the present work and the previous work gave the same trend.

CONCLUSIONS

Experimental study for the boiling heat transfer in a closed loop heat pipe was achieved by using saturated porous medium instead of pure water in the evaporator section. Experimental results indicate that, evaporator wall temperature was reduced when using porous media compared with water only. Also, the boiling heat transfer coefficient increased for closed loop heat pipe when using porous media and increased with increasing heat flux. The responsible mechanism for the improvement in the boiling heat transfer characterized by a reduction in bubble departure diameter, increased departure frequency, reduction in coalescence and increased in the number of nucleation sites. The enhancement factor in the boiling heat transfer coefficient increased with decreasing porosity up to the value of 0.4 then it decreased. The optimum operating

conditions for closed loop heat pipe when using porous media in the evaporator section at porosity about 0.4 and heat flux about 38600 W/m^2 gave the highest value for the enhancement factor equal to about 2.5. Comparison with the previous work gave the same trend.

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NOMENCLATURE

A_{ev} : Lower base evaporator surface area, m^2
Cp : Specific heat for cooling water, J/kg.K
d, : Evaporator lower metal base diameter, m
EF : Enhancement factor, -
h_b : Boiling heat transfer coefficient, W/m^2.K
m' : Water flow rate, kg/s
R : Electric resistance, Ω
Q : Heat transfer rate, W
q'' : Heat flux, W/m^2
T : Temperature, °C
V : Heater voltage, V
φ : Porosity, -
$Cos\varphi$: Power factor, -

Subscripts:
av : average
b : boiling
ev : evaporator
i : inlet, inner
loss : loss
o : outlet, outer

sat   : saturation
t   : total
us   : useful
w   : wall

Appendix (A)

The porosity is measured, experimentally, for the studied porous media by using a graduated tank (have the same inner diameter like evaporator of the loop heat pipe) filled with balls to a specified height of 20 mm. The water gradually filled the cavities inside the graduated tank and between the balls. The amount of water added to the tank was measured. Then water volume can be obtained. The total volume is defined as the volume of balls and volume of water.

Porosity φ for the specified volume can be obtained from the following equation;

$$\varphi = 1 - (\text{Volume occupied by balls/Total volume})$$
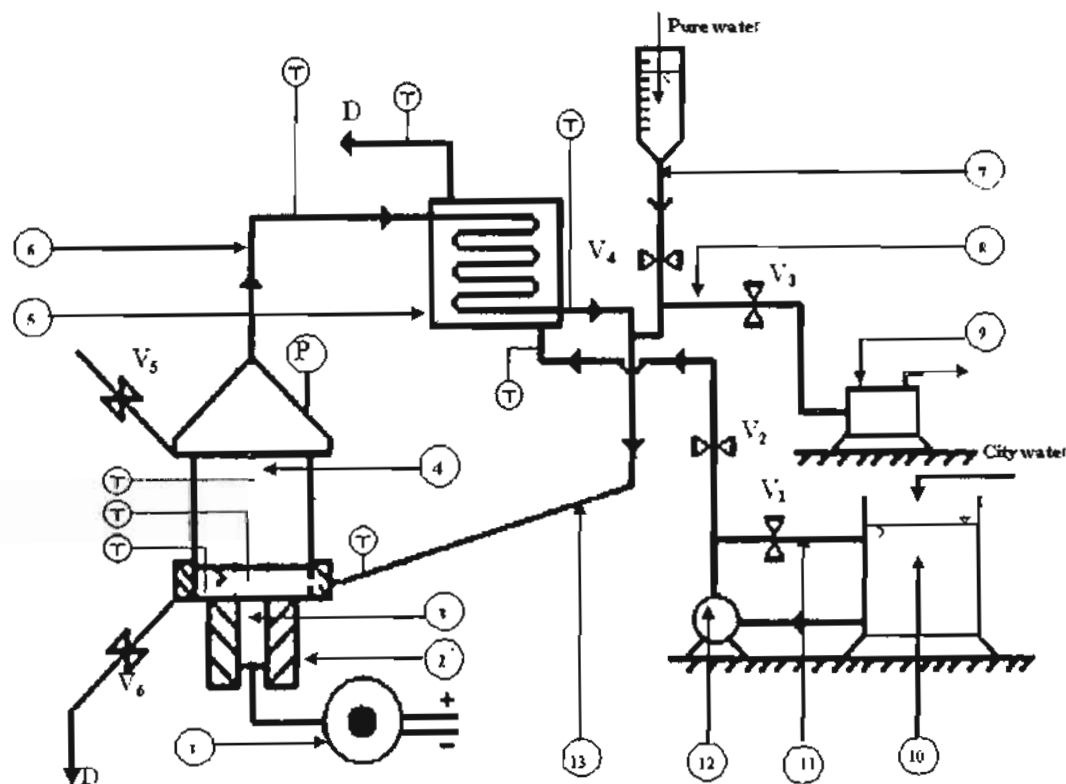
1. Variac. 2. Teflon Tube. 3. Electric Heater. 4. Evaporator Section. 5. Water cooled Condenser. 6. Vapor Line (Rising Tube). 7. Charging Line. 8. Extracted Air line. 9.Vacuum pump. 10.Water Tank. 11. By pass. 12. Water pump. 13. Condensate line (Falling Tube). D: Drain. P: Pressure Gauge. T: Temperature. V: Valve.
Fig. (1) Schematic diagram for the experimental test loop.
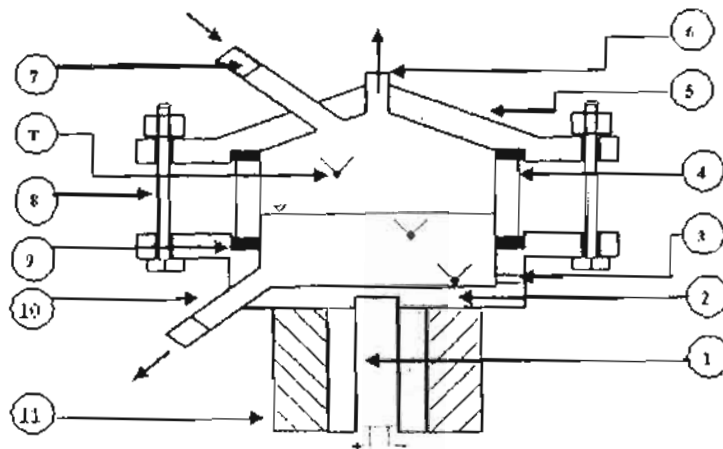
1. Electric Heater.　　2. Lower Metal Base.　　3. Condensate Inlet.　　4. Pyrex Tube. 5. Upper Metal Base.
6. Generated Steam Outlet. 7. Charging port of Porous Media. 8. Fixing Bolts. 9. Seal. 10. Delivery port of
Porous Media. 11. Teflon Tube. T Temperature.
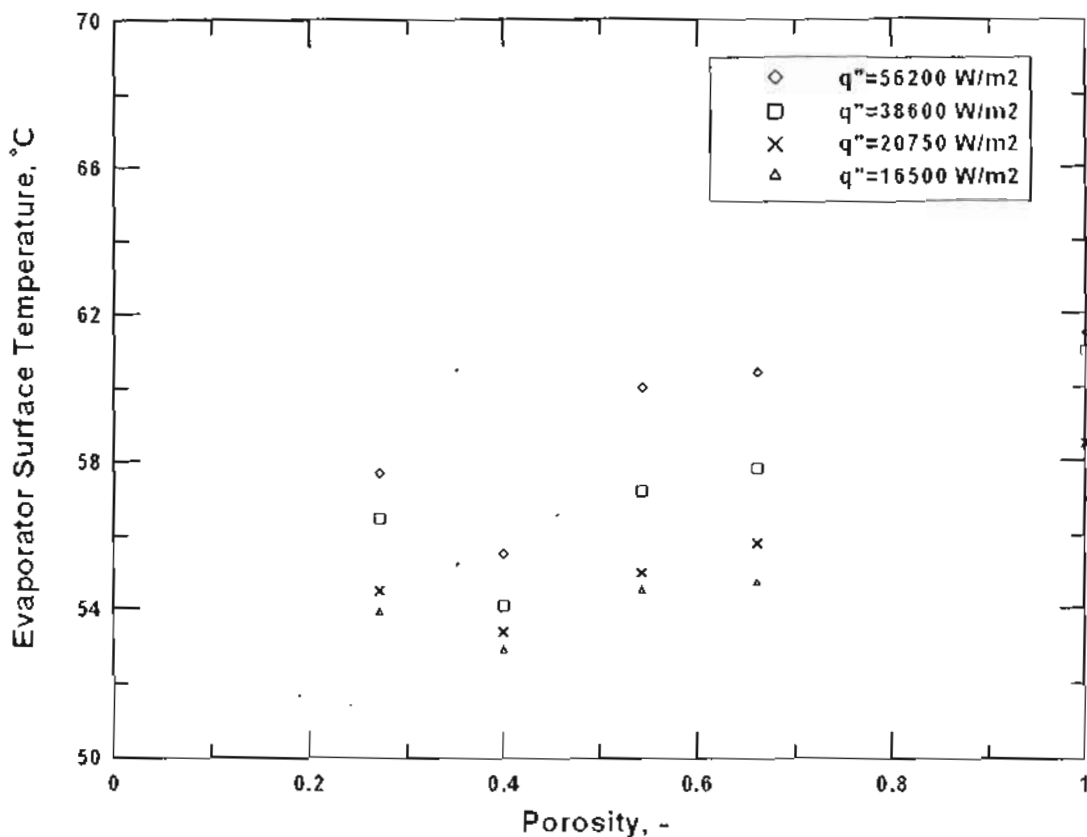
Fig. (2) Details of the evaporator section.

Fig.(3) Measured evaporator surface temperature versus porosity for different
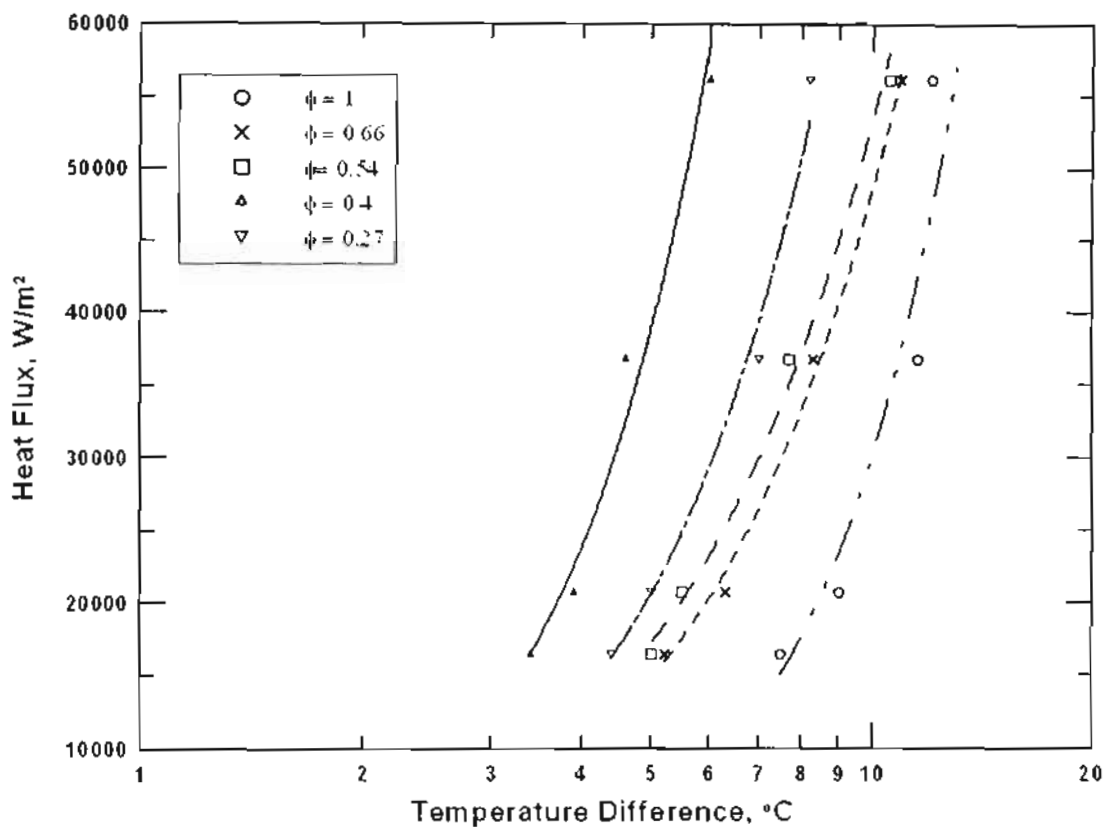values of heat flux.

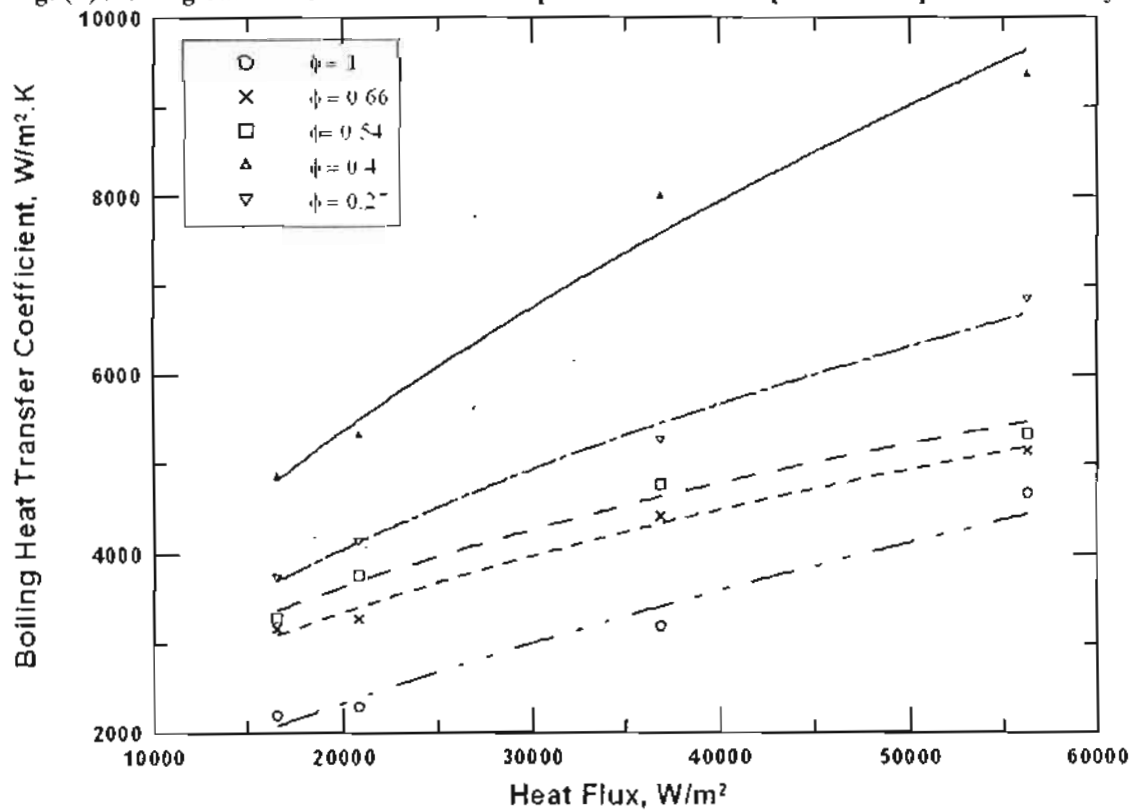Fig. (4) Boiling curves for saturated fluid porous media compared with pure water only.

Fig. (5) Variation of boiling heat transfer coefficient versus heat flux.
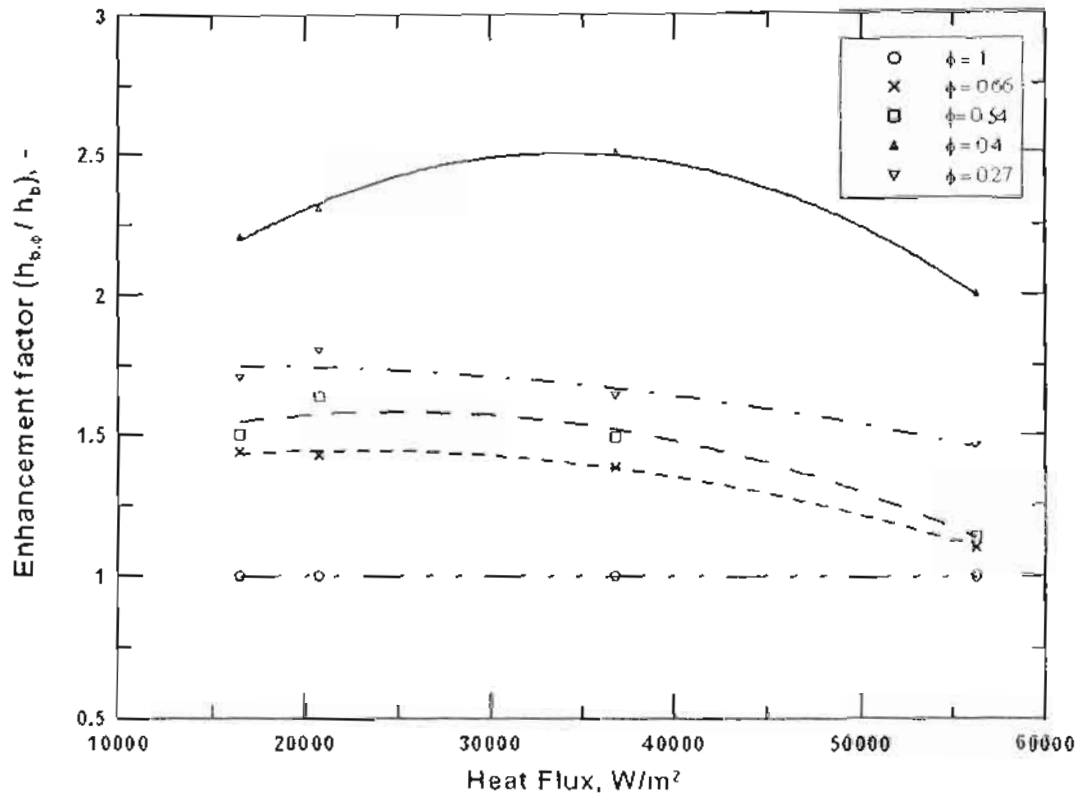
Fig. (6) Enhancement factor for boiling heat transfer coefficient versus heat flux for different porosities.
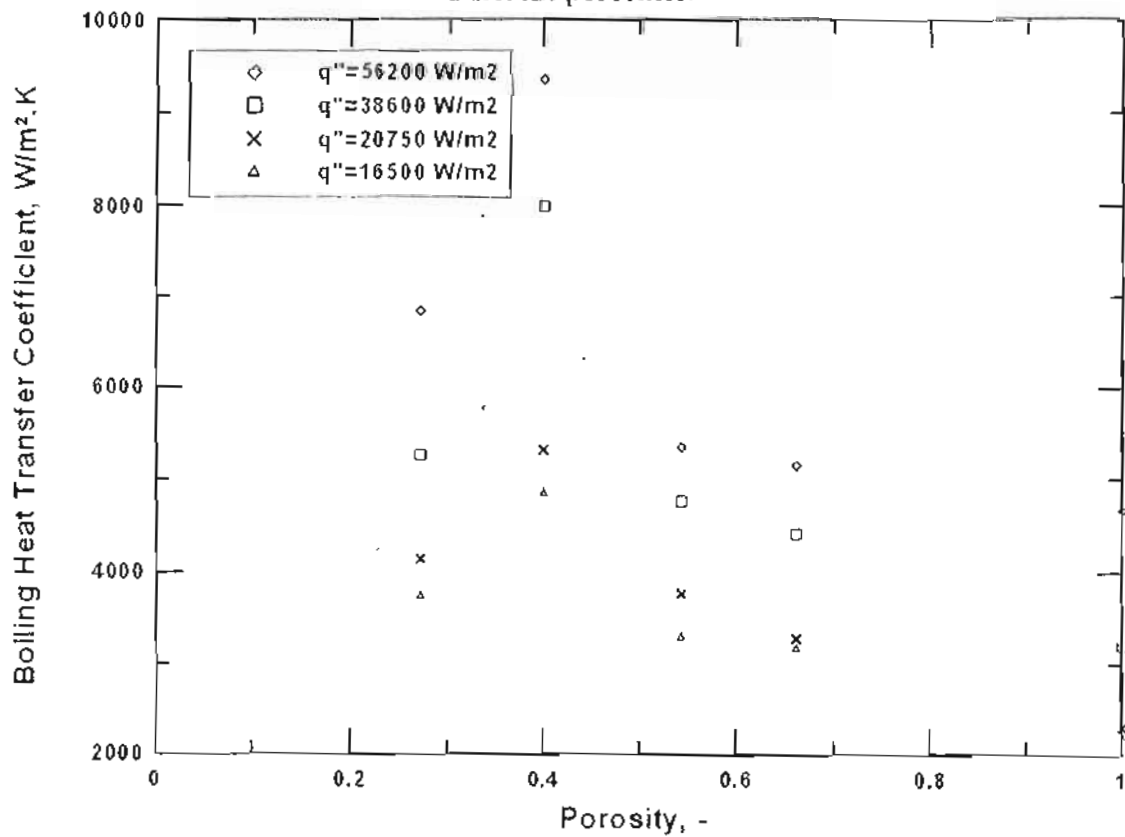

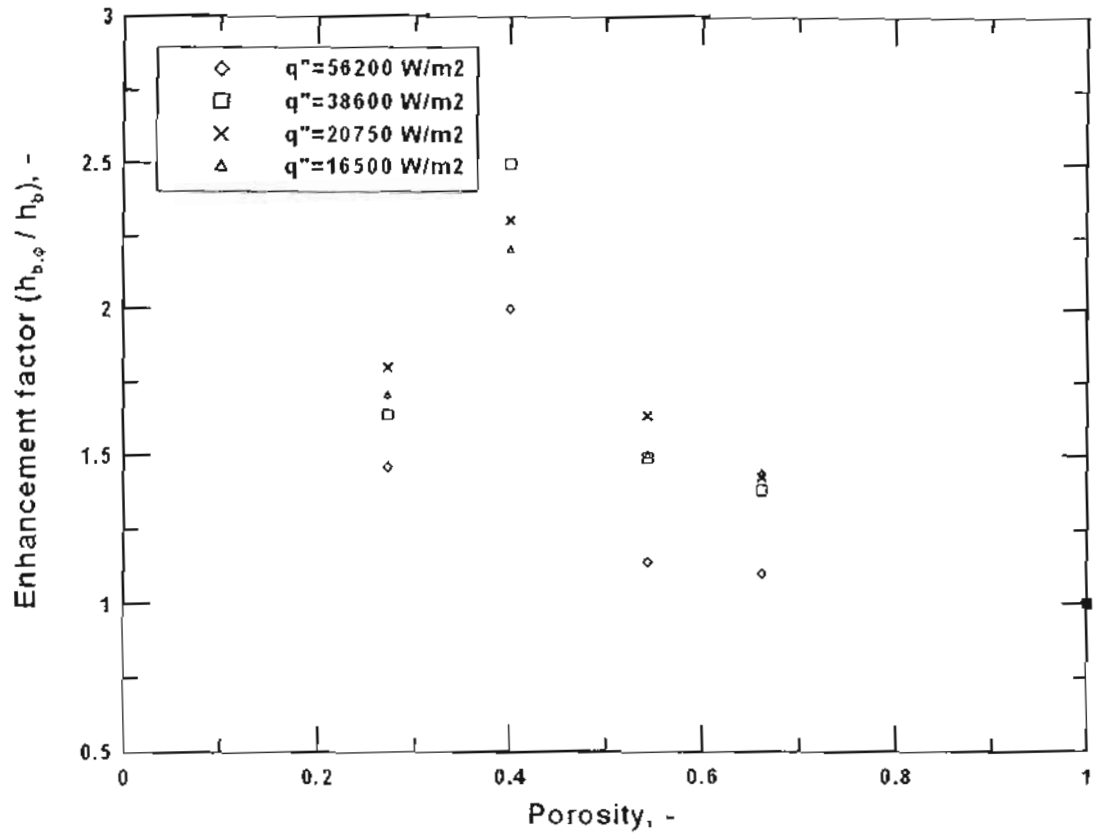Fig. (7) Effect of porosity on boiling heat transfer coefficient.

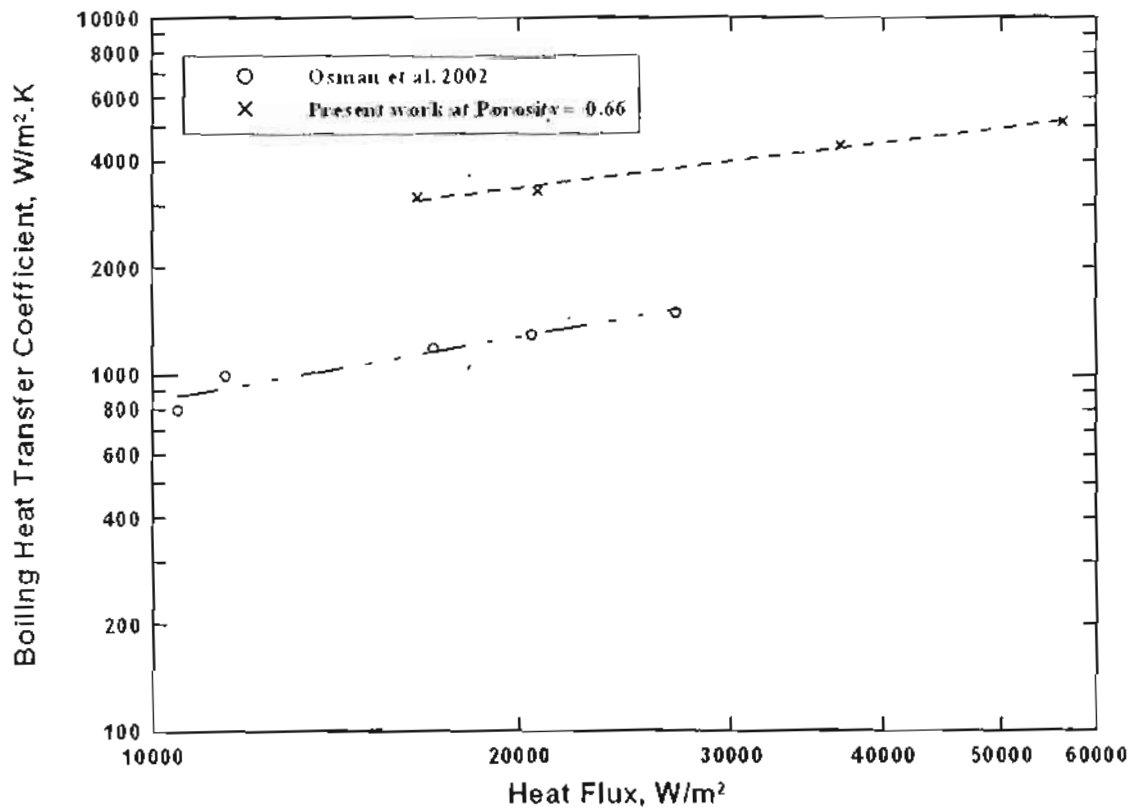Fig. (8) Enhancement factor for boiling heat transfer coefficient versus porosity.

Fig. (9) Comparison between the present work and previous work.