MECHANICAL ENGINEERING

Mansoura Engineering Journal, (MEJ), Vol.21, No. 1, March 1996 M.1

EXPERIMENTAL STUDY OF BOILING HEAT TRANSFER IN A THIN FILM ON HORIZONTAL TUBE

دراسه عملية لانتقال الحراره بالغليان فى طبقة رقيقة على أنبوبة أفقية

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خلاصه : فى هذا البحث تم دراسه تجريبيه لانتقال الحراره بالغليان على انبوبه افقيه و تم دراسه تأثير كل من الفيض الحرارى و معدل السريان و شكل مقطع الانبوبه و ارتفاع سقوط الماء على الانبوبه . ولاتمام هذه الدراسه تم تصميم و تنفيذ دائره اختبار حيث يسخن السطح الخارجى للانبوبه المختبره عن طريق مرور بخار بداخلها و يرش الماء على السطح الخارجى لهذه الانبوبه بواسطه موزع حيث تتكون طبقه رقيقه من الماء حول الانبوبه من الخارج فيحدث الغليان على السطح الخارجى لهذه الطبقه الرقيقه. و كان من نتائج هذا البحث تحديد القيم الافضل للعوامل المؤثره على عمليه انتقال الحراره و استنتاج معادله تجريبيه لمعامل انتقال الحراره صالحه فى المدى المدروس فى هذا البحث . تم دراسه انتقال الحراره لانابيب ذات قطر ١٢ , ١٩ , ٣٨ مم و كان مدى رقم رينولدز من ٣٠٠ – ١٣٠٠ ، ومدى فرق درجات الحراره من ١٠ – ٣٥ م و مدى الفيض الحرارى من ٥٠ – ٢٨٠ كيلو وات / م٢

ABSTRACT

Boiling heat transfer process in a thin film on horizontal tubes is , experimentally. investigated . The effect of the operating parameters on the boiling heat transfer coefficient is studied and accordingly the optimum operating conditions is deduced. The operating parameters such as heat flux. mass flow rate, feed water height and test tube configuration are considered.

To carry out this experimental work, a suitable test rig has been constructed such that the effect of these operating parameters can be investigated. The test tube is heated by a heating steam flowing inside it in a stratified flow pattern. A perforated distributer sprays a thin water film on a test tube forming dripping flow.

The considered ranges of the investigated parameters are heat flux from 20 to 300 kW/m^2 , mass flow rate from 0.02 to 0.1 kg/m.s. height to diameter ratio from 0.1 to 1 . and tube diameter from 12.2 to 38 mm . The effect of test tube configuration in case of circular . horizontal elongated oval shape and vertical elongated oval shape is studied. It is found that , the heat transfer coefficient in case of circular cross sectional tube is the best one .specially .compared with horizontal elongated oval shape.

1- INTRODUCTION

Boiling in a thin film on horizontal tube is very different from that of nucleate pool boiling. In a thin falling film on horizontal tube, bubble nucleation takes place with rapid bubble growth and then sliding around the circumference after the bubble detaches nucleation sites. The bubble diameter is greater than the liquid film thickness. The liquid spray on the horizontel tube increases the convective contribution and results in early removal of the adhering bubbles. This mechanism enhances local rate of boiling heat transfer according to the number and size of the bubbles. Although boiling heat transfer process in a thin film on horizontal tubes was studied theoretically and experimentally by a number of works [1-10]. it is of need to investigate some more parameters such as tubes configuration and other operating parameters on boiling heat transfer coefficient ,specially. from experimental point of view.

Two methods of heating the tube surface are used. First one is by wet steam to satisfy the constant temperature as in [1] and the second one by electrical heater to satisfy constant heat flux as in [3].

P.K.Tewari et al. [5] studied the nucleate boiling in a thin film on a horizontal tube at atmospheric and sub-atmospheric pressures. The experiments are carried-out using distilled water and Sodium Chloride Solutions (35000 and 50000 p.p.m.) in a pressure range 60-100 KN/m^2. Boiling occurs on the outer surface of the 19 mm outer diameter stainless steel tube which is heated by the dry saturated steam. The boiling heat transfer coefficient increases with the film flow rate for atmospheric and sub-atmospheric pressures in turbulent flow. The boiling heat transfer coefficient decreases with decreasing the saturation pressure. An improvement in boiling heat transfer is noted in case of horizontal tube-falling film compared to nucleate pool boiling.

Heat transfer for saturated falling-film evaporation on a horizontal tube has been analytically and experimentally, studied by M. C. Chyu and A. E. Bergles [6] . The effect of film flow rate, liquid feed height and wall superheat are investigated

S. Sideman and A. Levin [8] performed an enhancement in heat transfer coefficient in the horizontal tube film type evaporator-condenser water desalination units by utilizing circumferentially grooved on the external tube side. The analyzed grooved shapes are square-edged ,triangular and circular grooved. The square-edged groove. with a straight or modified bottom. was the most efficient shape in the flow of 300<Re<1000. Various grooved surface shapes was seen to have the best operational advantages of flow rate and heat transfer.

2- EXPERIMENTAL TEST LOOP

An experimental test rig has been designed and built. A layout of the test rig used in this work is shown in Fig. (1). This apparatus consists of a circulating pump (2), preheater (3), main heater (4). test chamber (5) and condensers (8.9.10). The pump is used to circulate the water through its loop. Water is heated in an electrically operating preheater and then in main heater. In preheater water is just warmed up and then it is heated to saturation temperature corresponding to chamber pressure. in main heater. This temperature is controlled by justifying the amount of heating steam in main heater. Heating steam is condensed inside a first secondary condenser(9) and then is accumulated in feed water tank (1).

Water is passed to the distributer in the test chamber to be dripped over the outer surface of the test tube. Heating steam is passed inside the test tube to obtain constant surface temperature during the experiments and a thin film of liquid is maintained at saturation temperature to insure boiling of the spraying water over this surface. Heating steam is condensed inside the tested tube imparting its latent heat to evaporating liquid film on the outer surface of the tube. Test chamber has two facing pyrex-windows for visual observation of the boiling phenomenon and liquid falling film along the tested tube . The liquid feed height was adjusted by moving the distributer up and down along slots on the side walls of the test chamber. Heating steam leaves test chamber and enters the second secondary condenser (10). The mass flow rate of heating steam is measured by collecting the condensate of second secondary condenser through a by-pass. Otherwise this condensate accumulated in feed water tank. Liberated vapor from the outer surface of the test tube is passed to the main condenser (8). City water is used for condensation process. Pressure ,temperature and mass flow rate of a thin liquid film, liberated vapor ,heating steam and city cooling water are measured in different required points in test loop.

3- RESULTS AND DISCUSSION

Boiling curve for saline and distilled water for laminar and turbulent flows is shown in Fig.(2). It is clear that, distilled water and Sodium Chloride Solution with salinity 40000 p.p.m. (close to the salinity of sea water on the Mediterranean and Red sea) has a little effect on the boiling heat transfer coefficient for laminar and turbulent flows, this is physically expected only for fresh evaporators. For this reason distilled water is used as working fluid in the experimental work instead of saline water.

Fig. (3) indicates that an increase in heat flux leads to an increase in the average value of boiling heat transfer coefficient for laminar and turbulent flows. As physically expected, increasing heat flux causes a rapid bubble population (on the number and size of the bubbles) and a rapid detachment from its nucleation sites. This bubble agitation causes a turbulent perturbation in the water film.

It is clear that the heat flux required to obtain the same wall superheat is higher for turbulent flow than for laminar flow. Accordingly, values of the average boiling heat transfer coefficient and average Nusselt number are higher for turbulent flow than for laminar flow. The average boiling heat transfer coefficient increases by about 37% for laminar flow when the heat flux is increased from 80 kW/m^2 to 240 kW/m^2 and increases by about 20% in turbulent flow for the same conditions. Also, it is observed that for high heat flux (more than 200 kW/m^2) the effect of Reynolds number is limited due to the rapid increase in bubble occulation and coalescence at the tube surface.

The effect of variation of feed water height on boiling curve is investigated for laminar and turbulent flow, as shown in Fig. (4). Nusselt number is plotted against dimensionless feed water height for a certain value for heat flux (100 kW/m^2), as shown in Fig. (5). It is found that Reynolds number has a little effect on Nusselt number, specially at $H/D_o=1.0$. This means that for $H/D_o=1.0$, low Reynolds number can be applied in the commercial units, which reduces the pumping power requirements. It is found that the boiling heat transfer coefficient has a maximum value for laminar and turbulent flows when the dimensionless feed height is in the range 0.4–0.6.

Fig.(6) illustrates the effect of the tube diameter on the boiling heat transfer coefficient for laminar and turbulent flow. The average boiling heat transfer coefficient for 19 mm tube diameter is the highest one, when the falling film occurs from a certain distance on the tube.

In the experimental work the circular tube shape is changed to oval shape with different vertical to horizontal axis ratio, which is called aspect ratio, E. For aspect ratio less than unity, the behaviour is near to a horizontal plate and is called horizontal elongated oval shape. Also, vertical elongated oval shape has aspect ratio greater than unity and its behaviour is near to a vertical wall. For horizontally elongated oval shape the condensate thickness increases sharply from the inflection point (at $\phi=\Pi/2$), while for a vertically elongated one the increment in the condensate film thickness is rather moderate for most of the circumferential distance. This causes a reduction in the boiling heat transfer coefficient for the horizontal elongated oval shape than for the vertical elongated oval shape as shown in fig. (7). The boiling heat transfer coefficient decreases for aspect ratio higher than 3, as shown in Fig. (7). For higher aspect ratio the vertical elongated oval shape is close to vertical flat plate. For vertical flat plate the evaporation heat transfer coefficient based on constant wall temperature is given by [12]:

$$h_v=(4/3)^{1/3} \left(k^3 g/ \nu^2 \right)^{1/3} \left(4 \Gamma / \mu \right)^{-1/3} \qquad (1)$$

Average boiling heat transfer coefficient obtained from the experimental results is greater than that obtained from Equation (1) by about 40% in laminar flow, which gives an advantage for utilizing circular tube in practical desalination units.

Using circular tubes improve the boiling heat transfer coefficient than that of the horizontal elongated oval shape for laminar and turbulent flow, as shown in Fig. (8). The average boiling heat transfer coefficient for the horizontal elongated oval shape decreases by about 25 % than for horizontal tube if the aspect ratio equal to 1/3 in laminar flow and decreases by about 33% in turbulent flow. The thermal and hydraulic behaviour of this shape is close to that of horizontal plate. Hence utilization of the horizontal circular tube is recommended rather than horizontal flat plate or horizontal elongated oval shape.

Fig. (9) shows that the difference in the boiling heat transfer coefficient for circular and vertical elongated oval shape is not exceed 6% for laminar and turbulent flow. Also it is found that the optimum value for aspect ratio is 3 which give the highest values for boiling heat transfer coefficient.

Also, a fair agreement is found between the present experimental results and that obtained by S. Sideman et al. [8], as shown in Fig. (10). The deviation between both them is about 14 %, where the present values are higher.

From the experimental data empirical correlations is obtained for average Nusselt number over a wide range of the studied operating parameters. The technique used for derivation of the following correlations is the dimensional analysis for the different factors affecting the boiling heat transfer process:

i- Laminar flow ($Re<750$):

a-For $(H/D)<0.5$

$$Nu=0.48\ Re^{-0.08}\ Pr^{0.85}\ (H/D)^{0.15}\ K_{q''}^{0.15} \qquad (2-a)$$

b-For $(H/D)>0.5$

$$Nu=0.4\ Re^{-0.08}\ Pr^{0.85}\ (H/D)^{-0.1}\ K_{q''}^{0.15} \qquad (2-b)$$

ii- Turbulent flow ($Re>850$):

a-For $(H/D)<0.5$

$$Nu=1.6*10^{-7}\ Re^{0.1}\ Pr^{0.85}\ (H/D)^{0.1}\ K_{q''}^{0.5} \qquad (3-a)$$

b-For $(H/D)>0.5$

$$Nu=1.35*10^{-7}\ Re^{0.1}\ Pr^{0.85}\ (H/D)^{-0.2}\ K_{q''}^{0.5} \qquad (3-b)$$

Where $K_{q''}=(q''\ D^3\ \rho^2\ )/\mu^3$

Fig. (11) shows a comparison between the experimental Nusselt number and the above derived Nusselt number. The maximum deviation between the experimental and derived Nusselt number is about 20% .

CONCLUSIONS

Boiling heat transfer in a thin film on horizontal smooth tubes has been studied to obtain the optimum operating conditions. The effect of heat flux , film flow rate , feed water height and test tube configuration on average boiling heat transfer coefficient have been investigated. The following conclusions are drawn:

1- Distilled water and Sodium Chloride Solution have the same values of the boiling heat transfer coefficient in laminar and turbulent flow .
2- The average boiling heat transfer coefficient increases with increasing heat flux .
3- It is found that the optimum ratio for height to diameter ratio lies between 0.4 to 0.6. Also it is found. that Reynolds number has a little effect on Nusselt number at $H/D_o=1.0$. which reduces the required pumping power.
4- Circular tubes improve the boiling heat transfer coefficient than the horizontal elongated oval shape .The vertical elongated oval shape (at the optimum aspect ratio .E=3) for laminar and turbulent flows improve the boiling heat transfer coefficient than circular tube by about 6% .
5- Comparison between the present results with other previous results shows a good agreement.

6- From the experimental results empirical correlations are obtained for Nusselt number in laminar and turbulent flow over a wide range of the operating parameters.

ACKNOWLEDGEMENT
The authors wishes to express his gratitude to Prof. Dr. H. M. El-Saadany and Prof. Dr. M. M. Mahgoub for his continuing advice and encouragement.

NOMENCLATURE

D	Diameter	m
E	Aspect ratio	—
g	Accelration due to gravity	m/s^2
h	Average boiling heat transfer coefficient, defined as $h=q''/\Delta T_{sup}$	$W/m^2.°C$
H	Water feed height	m
H/Do	Height to diameter ratio	
K	Thermal conductivity	$W/m.°C$
L	Tube length	m
Q	Heat input from heating steam	W
q"	Heat flux. defined as $q''=Q/(2\Pi R_o L)$	W/m^2
R_o	Outer tube radius	m
T	Temperature	°C
ΔT_{sup}	Superheat temperature difference (T_w-T_v)	°C

Greek symbols:

Γ	Mass flow rate per unit length per one side of the tube	kg/m.s
δ	Film thickness	m

μ	Dynamic viscosity	$N.s/m^2$
ν	Kinematic viscosity	m^2/s
ρ	Density	kg/m^3
ϕ	Angle of inclination	Rad.

Subscripts:

o	Orifice, outer, initial
sup	Superheat
v	Vapor
w	Wall

Dimensionless numbers:

Nu	Nusselt Number (hD/K)
Pr	Prandtl Number ($C_p.\mu/K$)
Re	Reynolds Number ($4\Gamma/\mu$)

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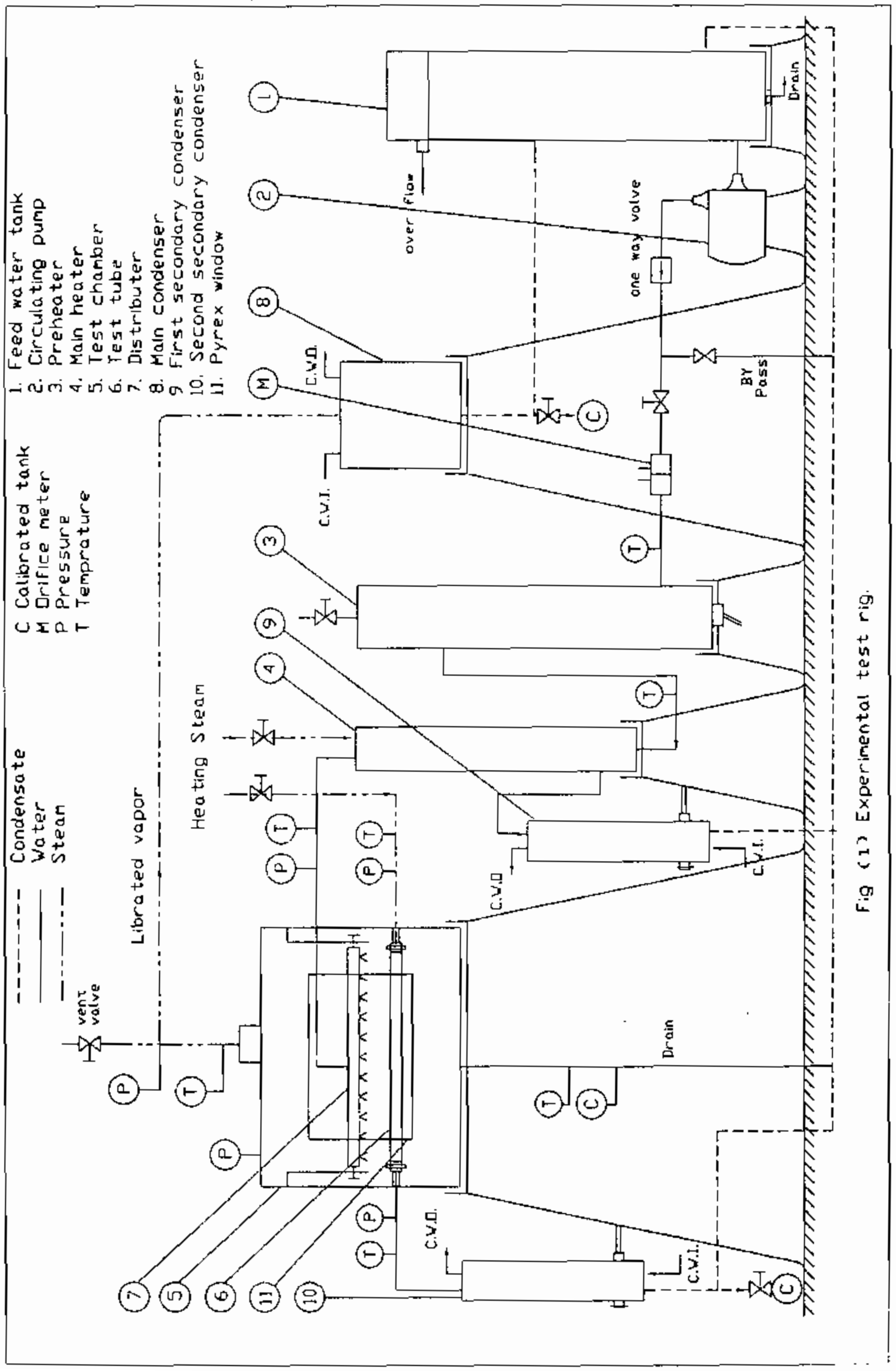

Fig (1) Experimental test rig.

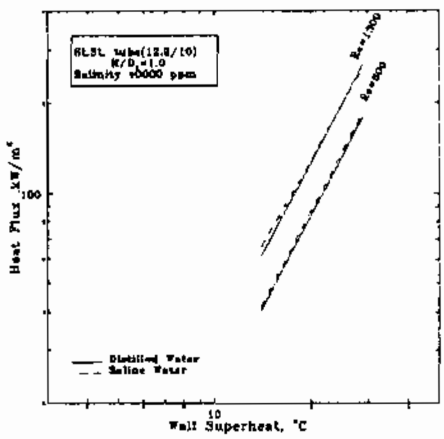

Fig.(2) Boiling curve for distilled and saline water in laminar and turbulent flows

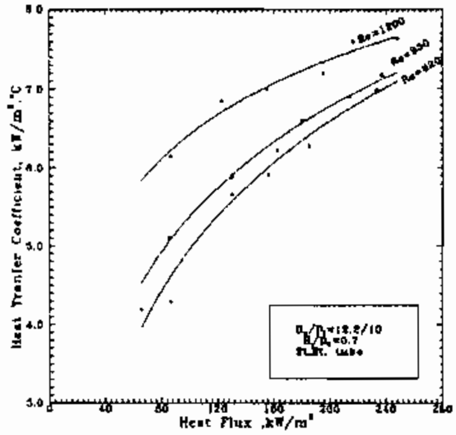

Fig.(3)Effect of heat flux on boiling heat transfer coefficient for laminar and turbulent flows.

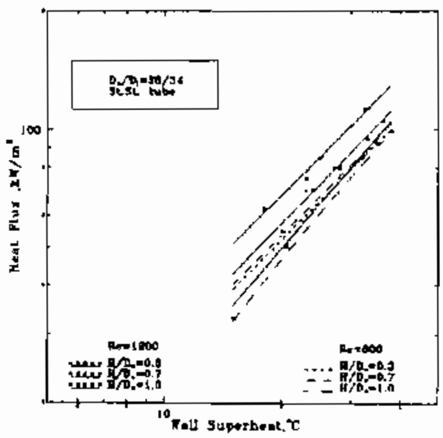

Fig.(4) Boiling curve at different height to diameter ratio for laminar and turbulent flows

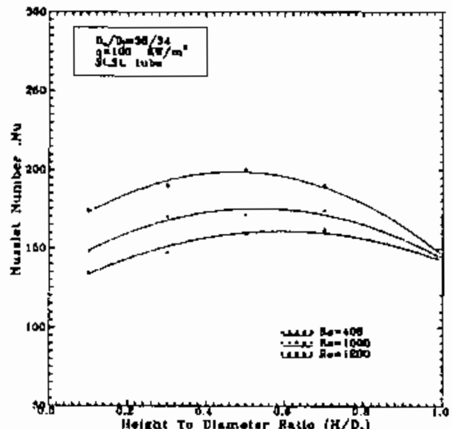

Fig.(5) Effect of height to diameter ratio on Nusselt number for different values of Reynolds number

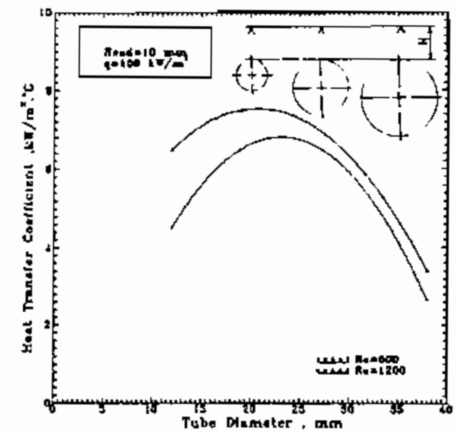

Fig.(6) Effect of tube diameter on boiling heat transfer coefficient for laminar and turbulent flow

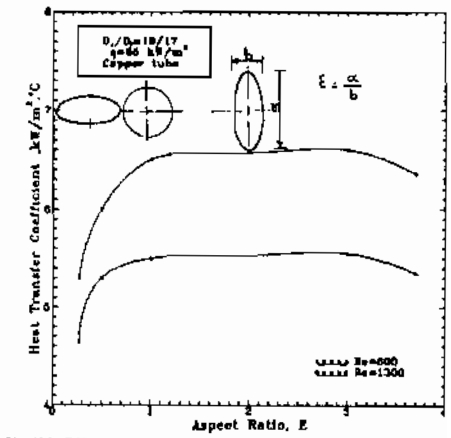

Fig.(7) Effect of aspect ratio on boiling heat transfer coefficient for different values of Reynolds number

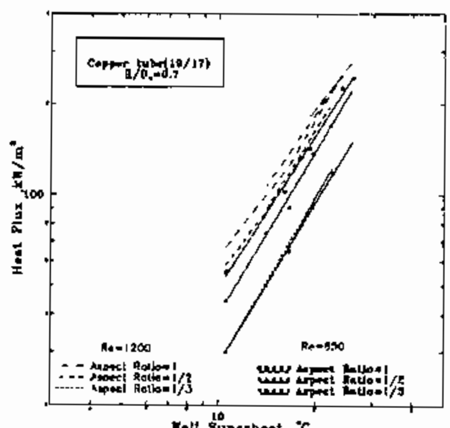

Fig.(8) Boiling curve for circular tube and horizontal elongated oval shape in laminar and turbulent flow

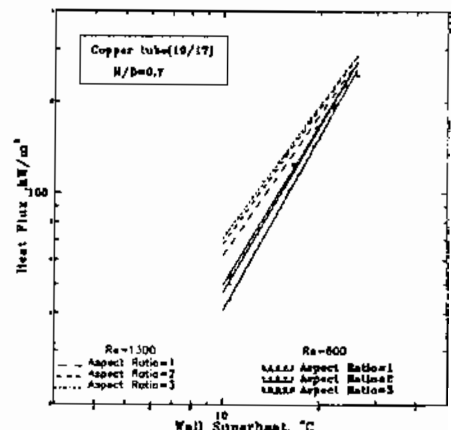

Fig.(9) Boiling curve for circular tube and vertical elongated oval shape in laminar and turbulent flow

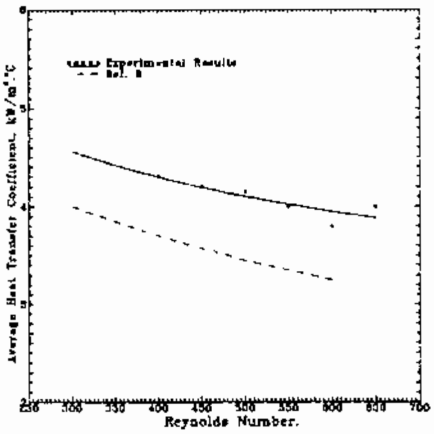

Fig.(10)Comparison between the experimental results with other data (for D_e=38 mm.).

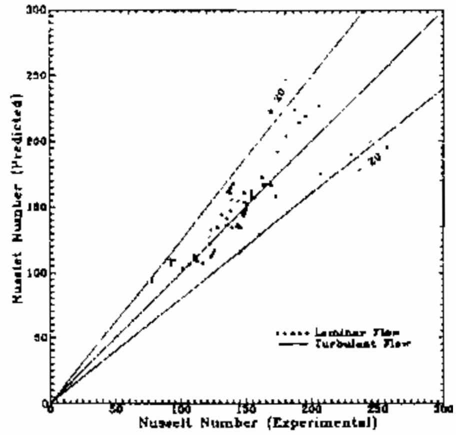

Fig.(11) Comparison between the experimental values of Nusselt number with the predicted values from correlations (2) and (3)