EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER FROM A VERTICAL ROD-BUNDLE INTO TWO-PHASE FLOW

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بحث عملى فى انتقال الحرارة من مجموعة قضبان رأسية فى سريان ثنائى الطور

خلاصة : الهدف من هذا البحث هو دراسة عملية انتقال الحرارة وتأثير تغير بعض العوامل مثل: درجة تحت التبريد عند المدخل ، الضغط، معدل التدفق ، والفيض الحرارى على معامل انتقال الحرارة من حزمة قضبان رأسية الى سريان ثنائى الطور.

لهذا الغرض تم تصميم وتنفيذ دائرة اختبار يتكون مقطع الاختبار فيها من حزمة قضبان ثلاثية ومثلثية الشكل ورأسية الوضع بطول ٢ متر.

من تحليل النتائج يتضح ان معامل انتقال الحرارة فى منطقة الغليان النووى المشبع وتحت التبريد يزداد بزيادة الفيض الحرارى ونقص درجة تحت التبريد ومعدل التدفق. وقد امكن الحصول على علاقة بسيطه لحساب معامل الانتقال الحرارى لسريان ثنائى الطور منخفض الجفاف اثناء الغليان النووى أو الغليان تحت التبريد.

ABSTRACT

The aim of this work is to investigate the process of heat transfer and the effect of varying different parameters as : inlet subcooling , inlet pressure, mass flow rate and heat flux on heat transfer coefficient in two-phase flow in the region of low quality nucleate boiling in a vertical rod bundle. To carry out the experimental study a test rig has been designed and constructed . The test section is a vertical three tube-bundle connected electrically in parallel. They are arranged in triangular shape. From the evaluation of the obtained experimental results, it is concluded that the two-phase heat transfer rate can be improved by raising the operating heat flux (in the region of nucleate boiling). In addition, boiling heat transfer rate increases by decreasing both inlet subcooling and mass flow rate. Finally a simple correlation is obtained for prediction of the heat transfer coefficient in the region of nucleate boiling with low steam quality.

INTRODUCTION

Two-phase flow is the most common flow of fluid in nature. Blood flow, pneumatic conveyance of granular solids, boiling of liquids and condensation of vapor are only a few examples of two-phase systems. Among the four types of the two-phase-flow, gas-liquid one component flow is the most important. The design of the recent steam generators, refrigeration equipments, water cooled nuclear reactors, and other major items of chemical and power plants is dependent upon the knowledge of the heat transfer processes in the two-phase one component flows such as convective boiling or condensation. This type of flow combines the characteristics of a deformable interface, the compressibility of one of the phases and interfacial phase changes , which complicate the processes of this flow type. The phenomenon of two-phase flow is highly empirical one, and any design problem is to be solved either with the aid of experimental data from similar system or by some correlations computation methods that are based on relevant measurements.
In literatures [2,4,6,9] , good correlations are published for the determination and prediction of the heat transfer coefficient for the different regimes of boiling heat transfer for various heating surface geometries and different liquids specially for pool boiling [3,8]. A lot of these correlations are developed for single heated tubes. Although there are good research activities for rod-bundles, still there is a lack for satisfactory information about the process of heat transfer from rod-bundles in two-phase flow.

Considering a vertical tube heated uniformly over its length with a low heat flux and fed with subcooled liquid at its base, Fig.(1) shows the expected idealized form of the flow pattern and the variation of surface and liquid temperatures in the various regions [2]. The published correlations for prediction of the heat transfer coefficient cover both the nucleate boiling region and the two-phase forced convection region, where both mechanisms are assumed to occur over the entire range of the correlation and that the contributions of both mechanisms are superpositioned. The two-phase heat transfer coefficient is then given as follows:

$$h_{TP} = h_{NcB} + h_{c}$$

where h_{TP} is the local heat transfer coefficient, h_{NcB} is the contribution due to nucleate boiling and h_{c} is the contribution due to forced convection.

The convective component could be given by the Dittus-Boelter equation based on the two-phase Reynolds and Prandtl numbers. The nucleate boiling component is expressed by the Forster and Zuber correlation corrected with a suppression factor suggested by Chen [2].

Surface conditions' affect the process of nucleation [2,6]. It is reported that the smooth surface requires higher superheat to transfer

the same heat flux as a rough surface.
Effect of the flow direction has been studied by Bartolini,et.at [7] in single tube, and it was found that the heat transfer regimes was similar to that in pool boiling.

EXPERIMENTAL TEST LOOP

To investigate-experimentally- two phase heat transfer and pressure drop through vertical heated rod-bundle, an experimental test loop has been designed and built. Schematic layout of the experimental loop is shown in figure (2). This loop consists mainly of test section (4), water condenser (6), preheater (2) and pump (1). Distilled water enters to the test section through a one way valve. A mixture of saturated vapor and saturated water leaves the test section to the water cooled condenser. City water is used for condensation. The condensate is wormed by preheater and return again to the test section. Test section is heated electrically by direct current supplied from a welding rectifier unit. The direct current passes to the test section through cable terminal (12). The inner surface temperatures are measured on different levels along the length of one tube of the test bundle by means of movable thermocouple. The inner surface temperature is then corrected to obtain the outer surface temperature.
From the theoretical evaluation of the experimental error, it is estimated that the total experimental error in measuring the heat transfer coefficient may rise to about 5% .

RESULTS AND DISCUSSION

The obtained experimental results concerning the two-phase heat transfer coefficient along the considered three-rod bundle which simulates nuclear fuel elements in light water reactor are presented and evaluated in this section.

The effect of various parameters, such as, inlet pressure, inlet subcooling , heat flux and mass flow rate, on the surface temperature, heat transfer coefficient is discussed in this section. Finally, a comparison is made between experimental results and previously obtained data.

All experiments of this work are carried out in the nucleate boiling , low heat flux, and low quality region. The experiments are carried out in the following operating conditions which are allowed by the test facility:

	Experimental range	
	Minimum	Maximum
Heat flux, W/cm^2	4.45	6.6
Inlet pressure $.10^{-5}$ $N/m2$	1.52	2.11
Mass flow rate, Kg/hr	500	900
Inlet temperature, °C	92	106
Inlet subcooling, °C	9.5	26
Outlet dryness fraction, %	0.0	1.05
Total pressure drop, N/m^2	18000	18750

The above allowed operating conditions cover the following heat transfer regimes:
1-single-phase forced convection 2- subcooled nucleate boiling
3- saturated low quality nucleate boiling .

The outer surface temperature and heat transfer coefficient distribution along the bundle tubes are shown in figures (3) to (6) for different operating conditions.
Figure (3) illustrates the behavior of outer surface temperature and heat transfer coefficient (at $P_{in} = 1.96 \times 10^5\ N/m^2$, $T_{in} = 103$ °C and W = 700 Kg/hr) along the test section for different heat fluxes. It is remarkable that the outer surface temperature and heat transfer coefficient increases with the increase of heat flux. It is clear that the outer surface temperature increases slowly with increasing heat flux which is physically accepted. This can be explained by increasing bubble formation with heating. For the allowed operating conditions the local heat transfer coefficient , given by :

$$h = q''/\Delta T_{sat}$$

increases by a factor of 1.15 when the heat flux is raised by a factor of 1.35.
Also, it is to notice that the outer surface temperature increases with increasing (z/L) up to the end of one phase length, as z measured from the lower end of the bundle tube. In the saturated nucleate boiling region it becomes nearly constant. The high temperature at the end of test section, shown in Fig.(3), can be explained as due to the end effects. Copper bus-bars at flow inlet works as cold sink and some

conduction flow through the hot heating tubes to the relatively colder sink in opposite direction to the flow. This lead to an increase in the heat transfer coefficient and decrease in outer surface temperature in the flow direction at the test section inlet. The last thermocouple at the outlet of the test section registers higher reading than that preceding one in most runs. This is because near the outlet of the test section some of the vapor bubbles form a thin film between the working fluid and heating tube. The resistance to heat flow becomes greater causing a rise in the outer surface temperature at the test section outlet. Referring to Fig.(1), it is concluded that the behavior of the wall temperature in the considered rod bundle has the same trend as for single tube.

The curve in figure (4) indicates that the outer surface temperature increases with the increase of inlet pressure as expected because higher pressures require higher saturation and higher surface temperatures than that at low pressures. Consequently heat transfer coefficient and steam quality increases with the decrease of inlet pressure. It is clear from above table that the range of the allowed operating pressure is small, which gives no clear picture for the effect of the inlet pressure.

Figure (5) shows that the outer surface temperature increases with the decrease of inlet subcooling or with the increase of inlet temperature at constant inlet parameters. Also it is to notice that heat transfer coefficient is improved with decreasing inlet subcooling. This is because the rate of increase of bulk temperature is higher than the rate of increase of outer surface temperature. Consequently the temperature difference decreases and heat transfer coefficient increases for the same value of heat flux.

The results show that the outer surface temperature decrease with the decrease of mass flow rate. It is also clear- as shown in Fig.(6)- that the heat transfer coefficient and steam quality increases with decreasing mass flow rate.
Figure (7a) illustrates the variation of the surface superheating for different mass flow rates. This figure illustrates the variation of surface superheating , ΔT_{sat} with the operating surface heat flux ,q'' . According to the figure, it is clear that for the same heat flux as the mass flow rate increases to one and half of its initial value there is a small increase in surface superheating. This means that the heat transfer coefficient increases by a small value with decreasing mass flow rate, for the same value of heat flux.

The behavior of the heat transfer coefficient for two-phase flow as obtained experimentally is presented in Figs.(3) to (6). It is clear that the heat transfer coefficient rises very slowly along the boiling length of the test section and has an average value of $2.5 * 10^{3}$ W/m2.°C .
Takahashi et al [5] performed an experimental test for the pool boiling heat transfer from a horizontal plane heater with distilled water as the test fluid. The obtained results are compared with the

data presented in this work as shown in Figure (8), which indicates the same trend with a small deviation. The outer surface temperature distributions along the heating tube for different boiling types (pool, flow and natural circulating boiling) are illustrated in Fig. (9). The outer surface temperature for pool boiling decrease as (z/L) increase. In contrast to the situation of pool boiling the outer surface temperature remains nearly constant in natural circulating and flow boiling. This can be explained as due to the fluid temperature. In pool boiling the entire fluid acquires the saturation temperature, where in the flow boiling the fluid temperature is constant only in the region of nucleate boiling.

The heat transfer correlations for prediction of the heat transfer coefficient in nucleate boiling are often complicated [2,4,6] . Many dimensionless relationships have been published which attempt to find a relationship in the following form [2,6]:-

$$Nu = f(Re, Pr) \quad ,$$

by analogy with the relations applicable for single phase flows. From the obtained experimental data plotted in Fig. (10), the following simple correlation which satisfies the above form is found :

$$Nu = 0.053 \, Re_{TP}^{0.8} \, Pr_{f}^{0.33}$$

Because of the narrow range of variation in the Reynold's number allowed by the test facility, validation of the above relationship requires more experimental examination. It can be used for prediction of the heat transfer coefficient in the low quality nucleate boiling region.

CONCLUSIONS

From the previous discussion, the following conclusions can be drawn:

i) The outer surface temperature of the heating tubes increases in the direction of flow in one-phase flow region and remains nearly constant in the two-phase flow region where the fluid boils.

ii) The local heat transfer coefficient in one-phase flow region increases at interance length, but it decreases in the flow direction until the boiling begins then it remains nearly constant.

iii) The local heat transfer coefficient increases about 15 % of its original values when the operating heat flux is raised by a factor of 1.35 at mass flow rate of 700 Kg/hr and inlet pressure of 1.96×10^5 N/m^2.

iv) Variation of the mass flow rate has a negligible effect on the measured heat transfer coefficient. It was found that increasing the mass flow rate from 500 to 700 Kg/hr leads to about 6 % decrement in the heat transfer rate at inlet pressure of 1.86×10^5 N/m^2 and heat flux of 6.38 W/cm^2.

v) According to the data obtained for the operating conditions allowed by the test facility ,the rate of heat transfer is improved with decreasing inlet subcooling .

vi) The average boiling heat transfer coefficient along the two-phase flow length is found to be $2.5 * 10^3$ W/m2°C .

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NOMENCLATURE

Symbol	Definition	
C_p	Specific heat at constant pressure.	(kJ/kg.c)
D_e	Equivalent diameter.	(m)
G	Mass velocity.	(kg/m2.s)

g	Gravitional acceleration.	(m/s2)
h	Heat transfer coefficient.	(Kw/m2 °c)
K	Thermal conductivity.	(W/m °c)
L	Test section length.	(m)
P	Absolute pressure.	(N/m2)
q"	Heat flux.	(W/cm2)
T	Temperature	(°C)
ΔT	Temperature difference.	(°C)
T_b	Liquid bulk temperature	(°C)
u	Velocity.	(m/s)
v	Specific volume.	(m^3/Kg)
W	Mass flow rate.	(Kg/hr)
X	Steam quality.	(%)
z	Distance measured from test section bottom.	(m)
z/L	Length of test section ratio.	(dimensionless)

Greek letters

α	Steam void fraction.	(dimensionless)
β	Volumetric quality.	(dimensionless)
ρ	Density.	(Kg/m^3)
$\bar{\rho}$	Average density of homogeneous fluid.	(Kg/m^3)
μ	Dynamic viscosity.	(N.s/m^2)
$\bar{\mu}$	Mean viscosity of homogeneous fluid.	(N.s/m^2)
σ	Interfacial tension.	(N/m)

Dimensionless Numbers

Nu_f = Nusselt number. ($h_{TP}\,D_e / K_f$)

Pr_f = Prandtl number. ($C_p\,\mu / K_f$)

Re = One-phase Reynolds number. ($G\,D_e/\mu$)

Re_{TP} = Effective two-phase Reynolds number. $(G\,D_e/\mu_f)\{((1-X)\,F^{1.25}))$

Subscripts

f	Liquid.
g	Vapor.
sat	Saturation.
TP	Two-phase.
W	Wall
b	liquid bulk

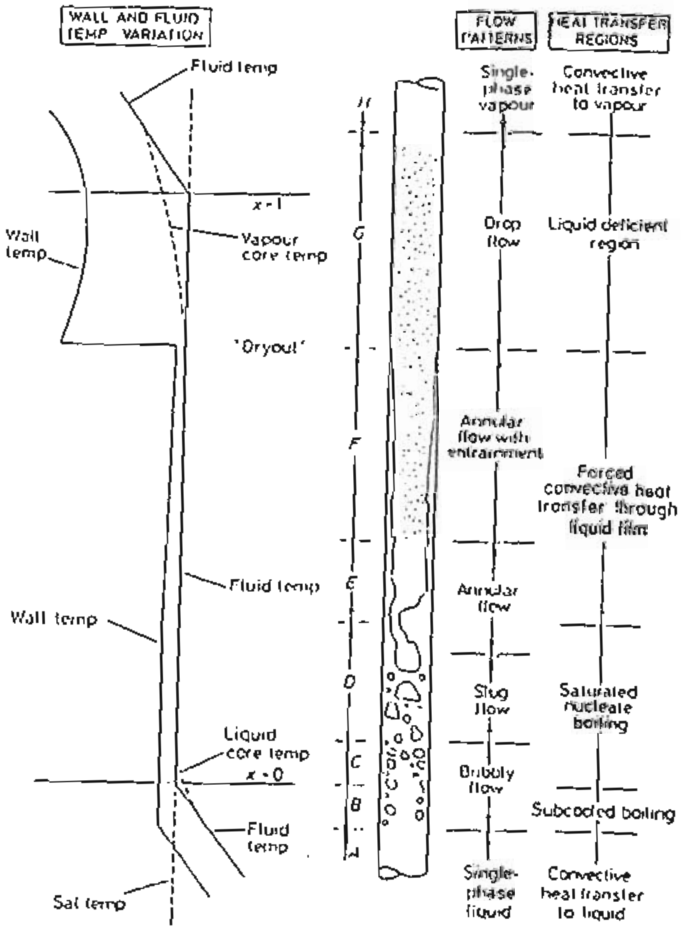


FIG. 1 Relationship between wall temperature variation and the flow and boiling regimes. From Collier (1972)

1 Pump.
2 Preheater.
3 Heating coil.
4 Test section.
5 Equalizer tank.
6 Condenser.
7 Water tank.
8 Orifice meter.
9 C.W.I.
10 C.W.O.
11 Vent valve.
12 Cable terminal.
13 Manometer.
14 Drain valve.
15 One way valve.
P Pressure gauge.
T Temperature probe
C Cooling coil.

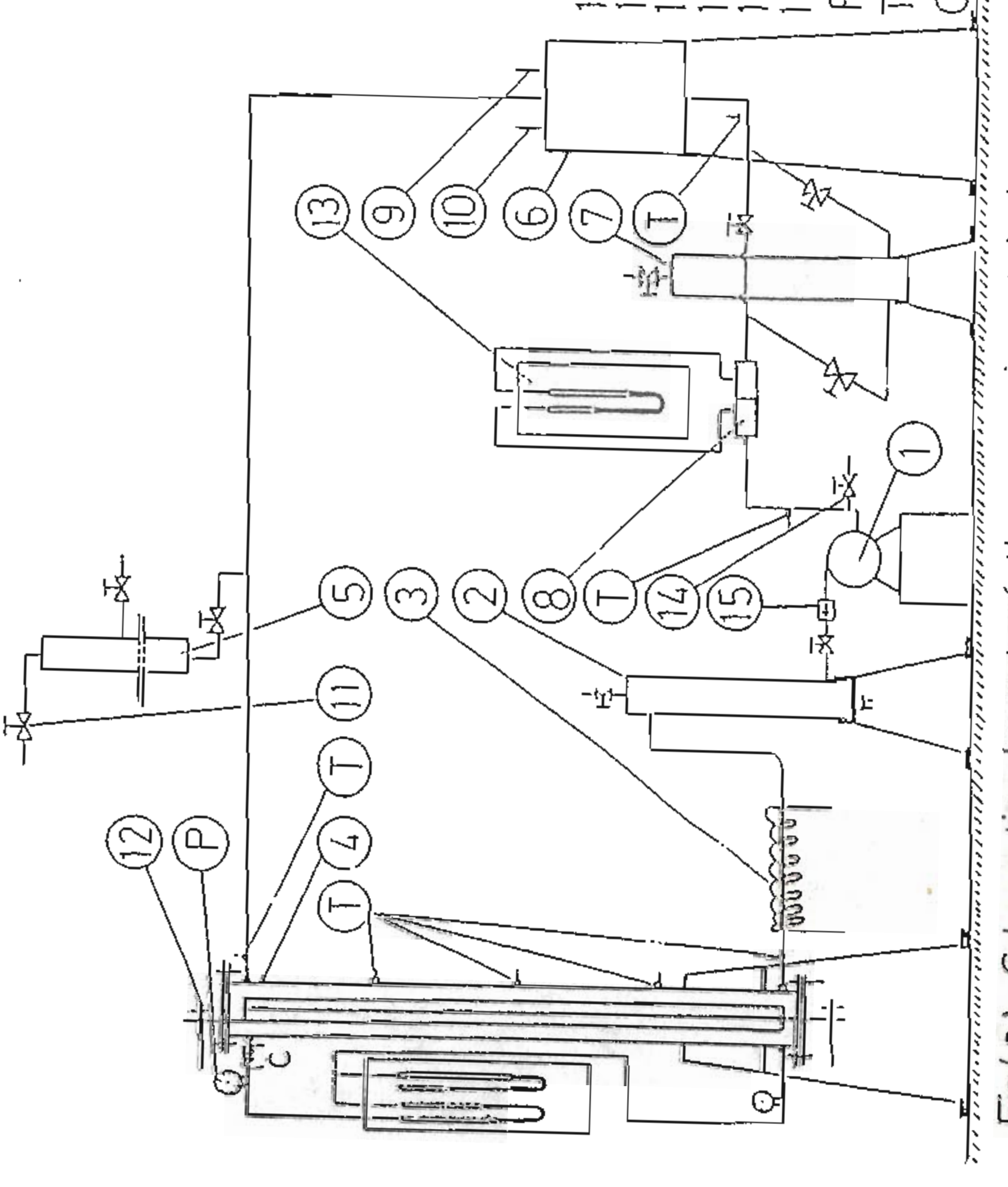


Fig.(2) Schematic layout of the experimental loop.

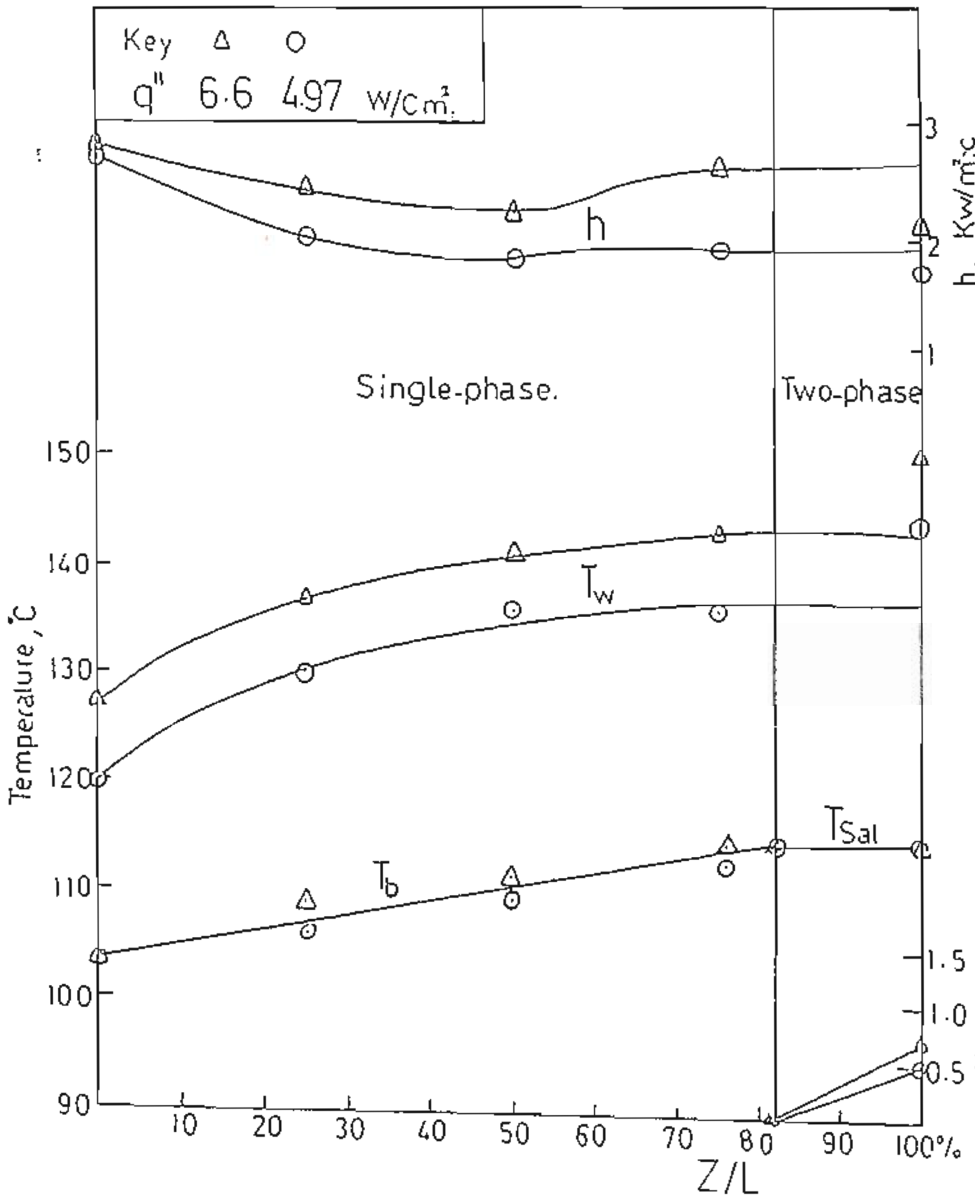


Fig.(3) Influence of heat flux on temperature and heat transfer coefficient for W = 700 Kg/hr T = 103 °C and $P = 1.96 \times 10^5$ N/m²

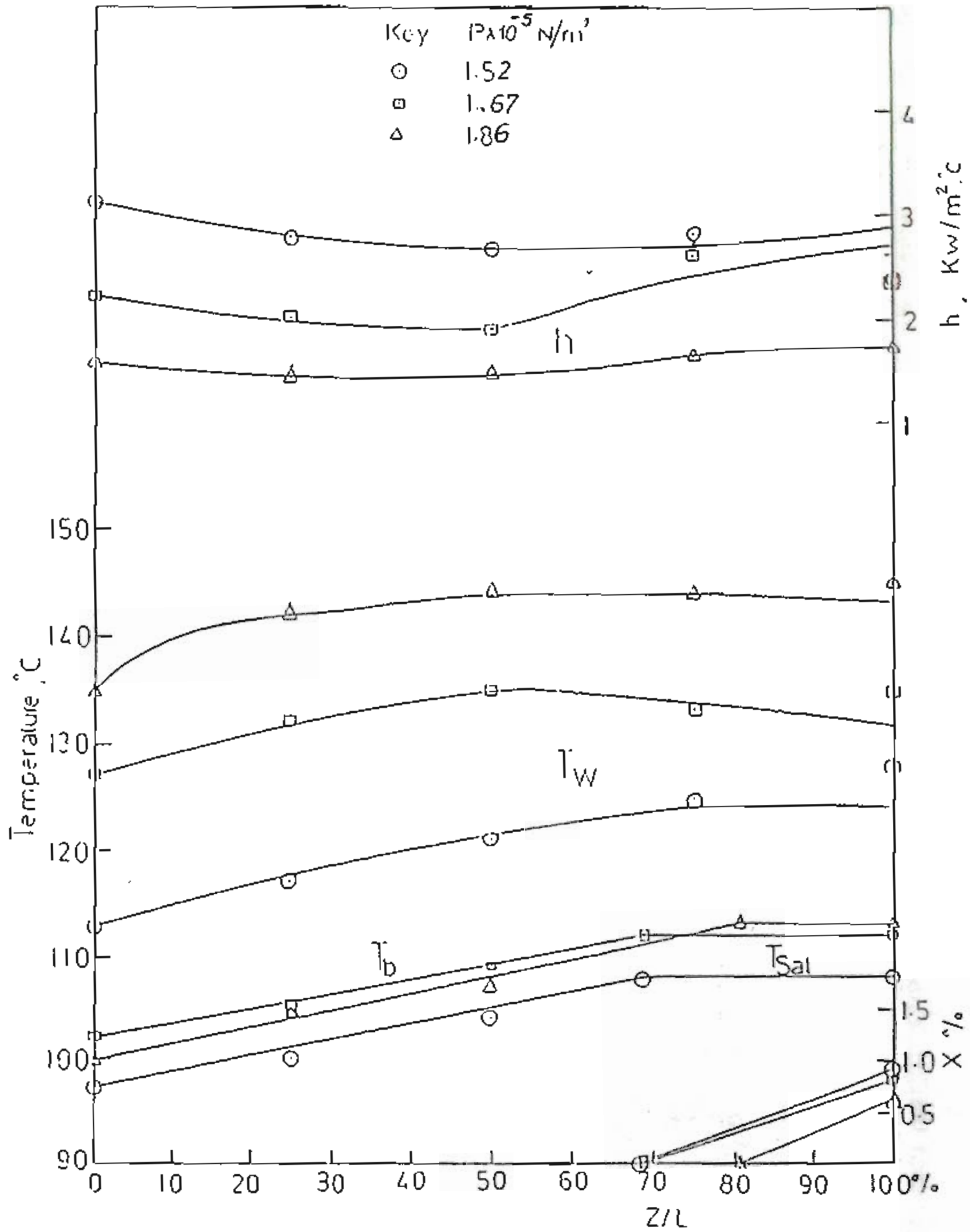


Fig.(4) Influence of inlet pressure for W =700 Kg/hr and q" = 5.45 W/cm².

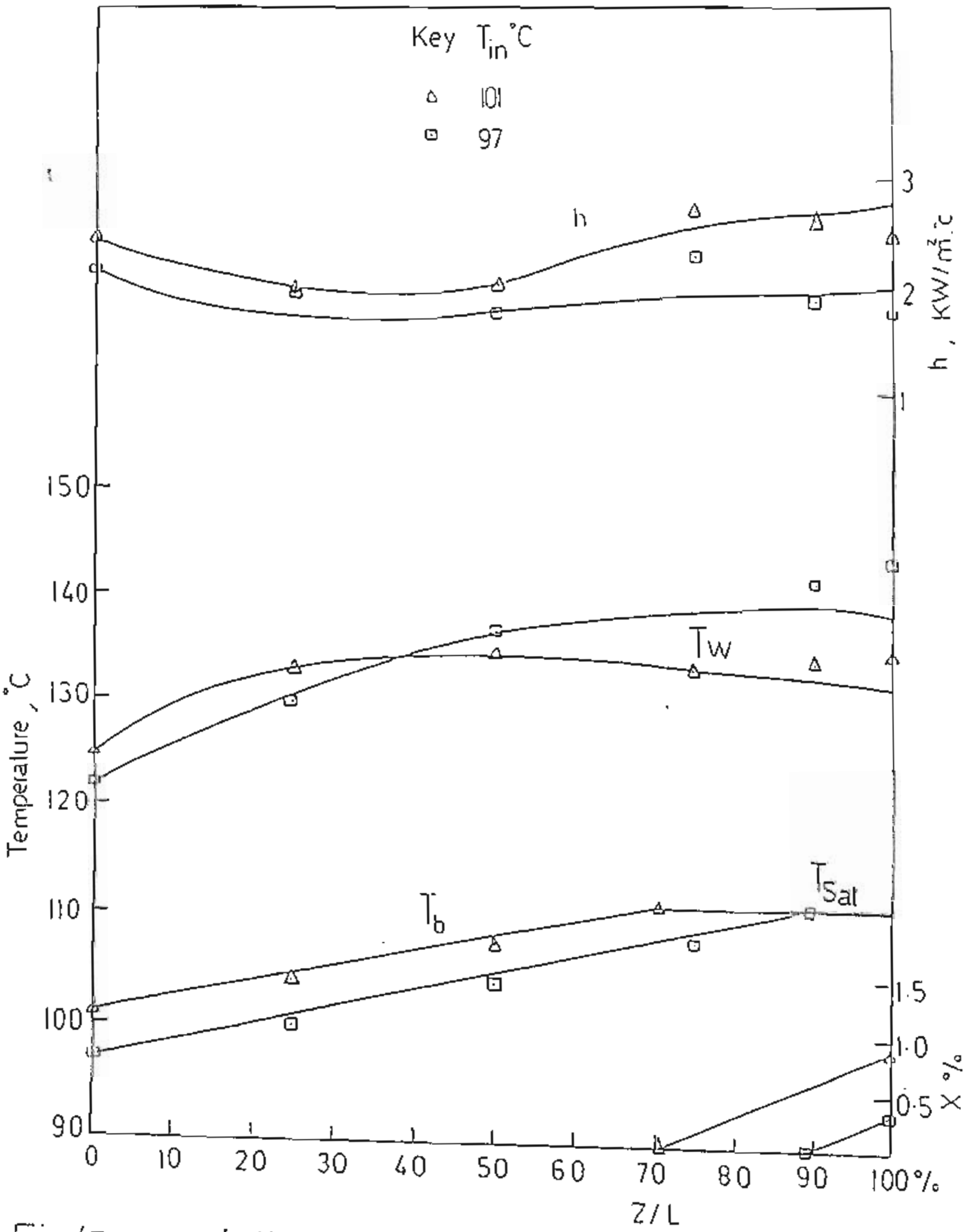


Fig.(5) Influence of inlet subcooling for $P = 1.67 \times 10^5$ N/m^2, W=750 Kg/hr and $q'' = 5.83$ W/cm^2

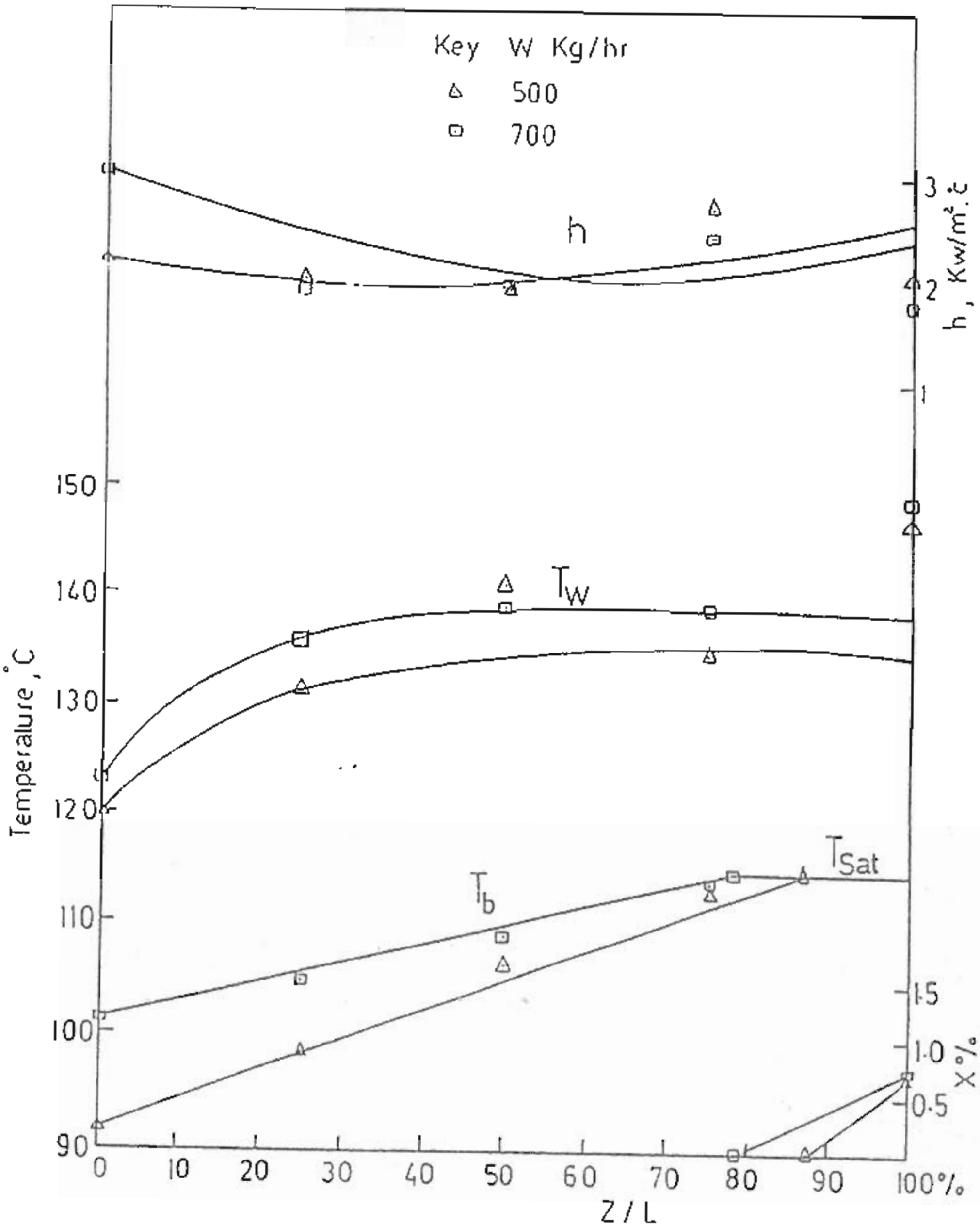
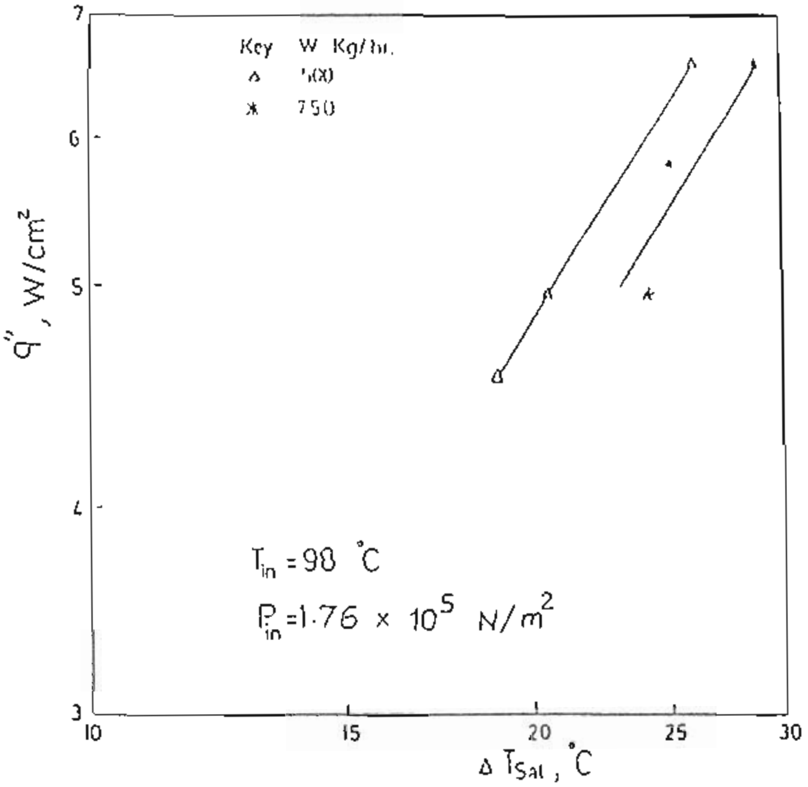


Fig.(6) Influence of mass flow rates for $P=1.86\times10^5$ N / m² and $q''=6.5$ W/cm².



Fig(7a) Surface heat flux versus surface superheating for different mass flow rates.

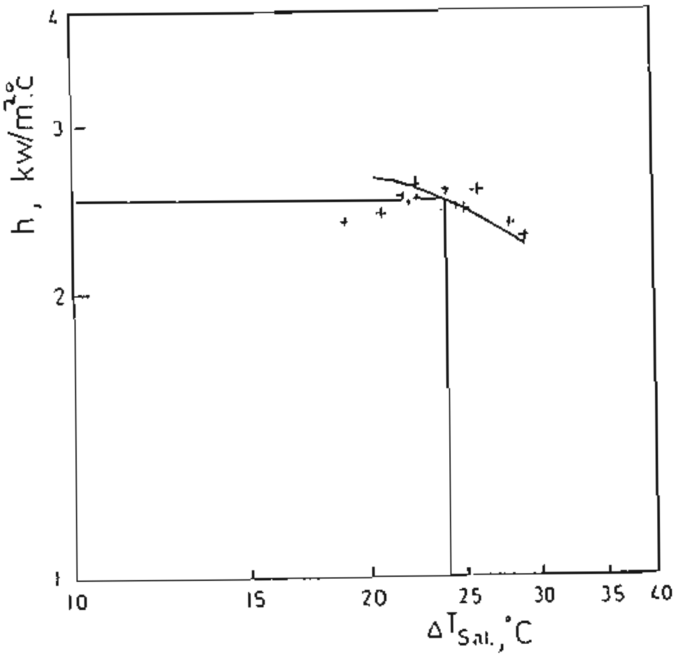


Fig.(7 b) Average heat transfer coefficient

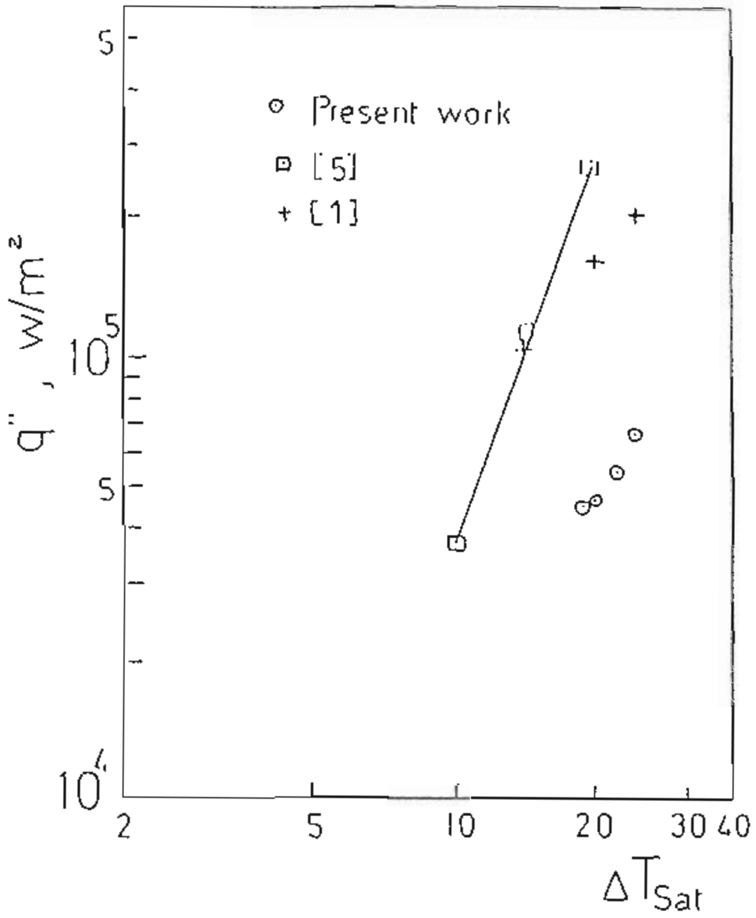


Fig.(8) Comparison of present work with Khattab[1] and Takahashi[5]

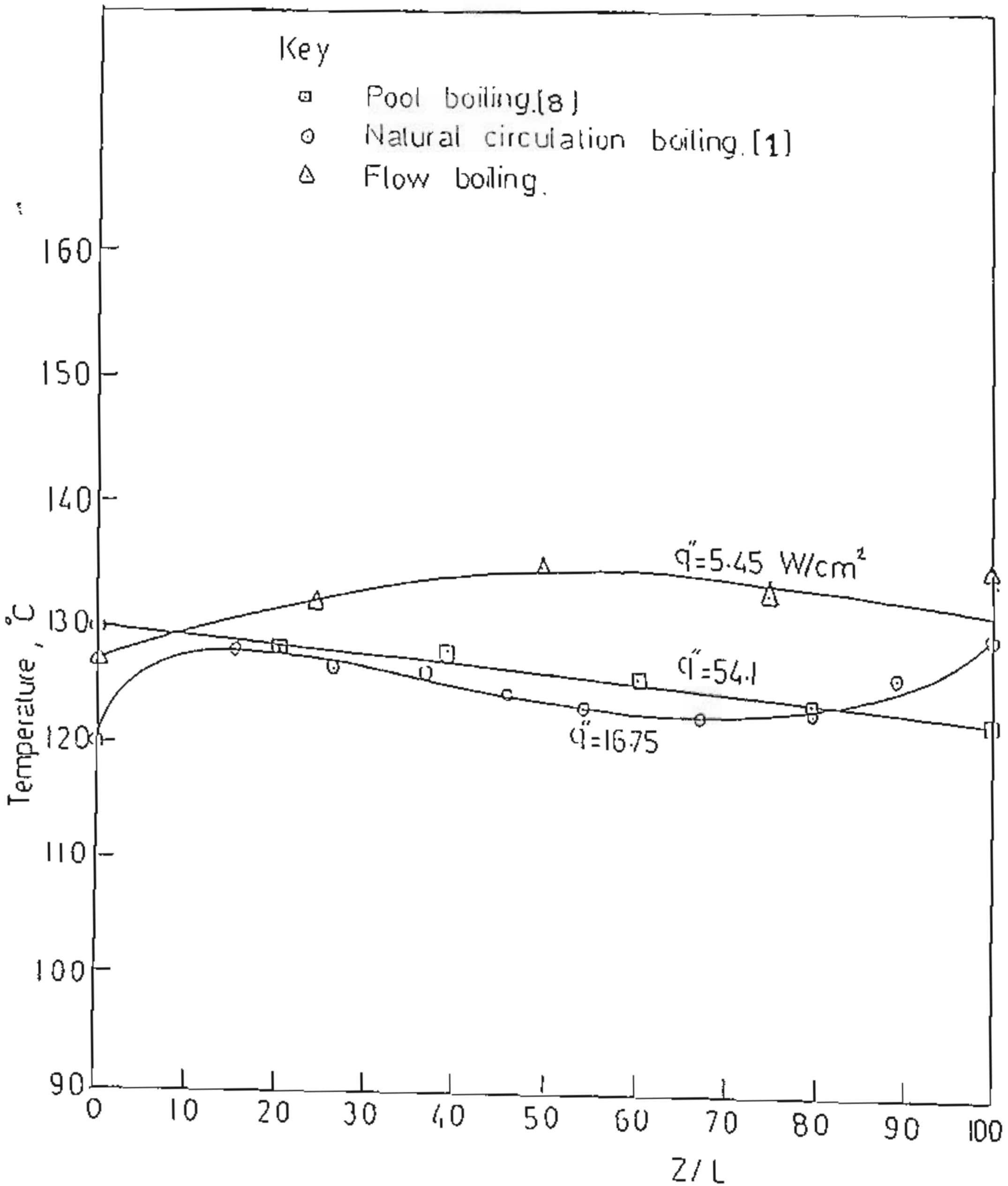


Fig.(9) Axial distribution of the outer surface temperature for different boiling types.

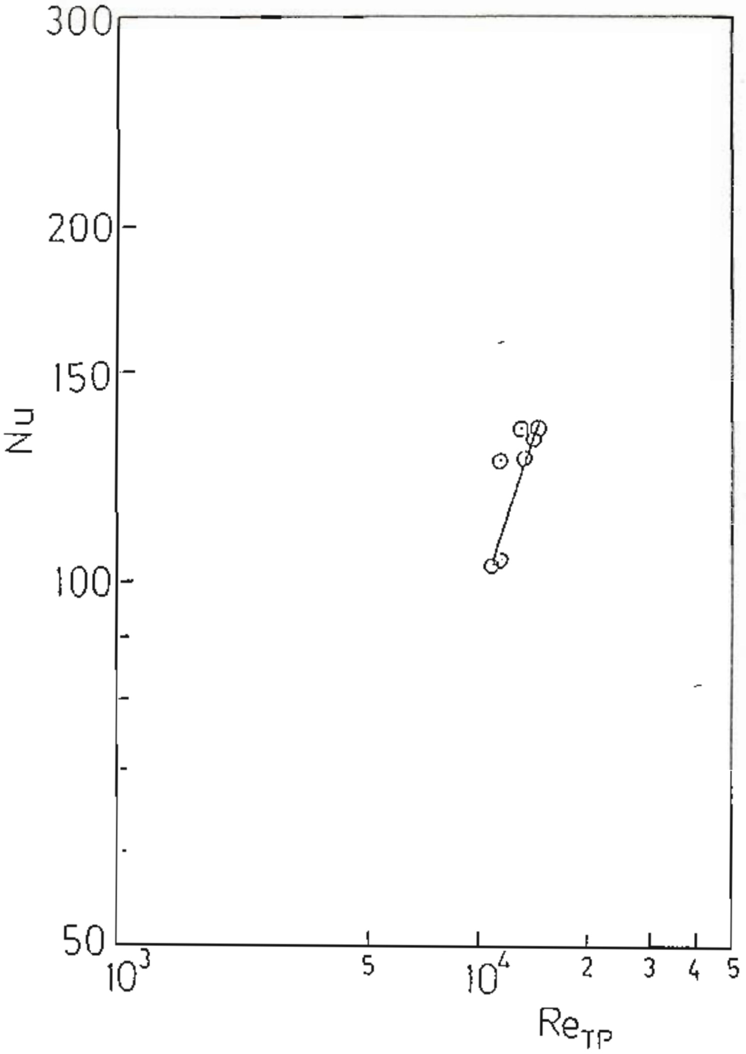


Fig.(10) Dimensionless piesenlation for parallel lube-bundle.