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ARAB REPUBLIC OF EGYPT
ATOMIC ENERGY ESTABLISHMENT
REACTOR & NEUTRON PHYSICS DEPARTMENT

BURNOUT HEAT FLUX IN NATURAL FLOW BOILING

By

M.M. HELAL, M.A. DARWISH,
S.I. MAHMOUD

1978

SCIENTIFIC INFORMATION DIVISION
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CONTENTS

ABSTRACT

Twenty runs of experiments were conducted to determine the critical heat flux for natural flow boiling with water flowing upwards through annuli of centrally heated stainless steel tube. The test section has concentric heated tube of 14 mm diameter and heated lengthes of 15 and 25 cm. The outside surface of the annulus was formed by various glass tubes of 17.25, 20 and 25.9 mm diameter. System pressure is atmospheric. Inlet subcooling varied from 18 to 5°C. Obtained critical heat flux varied from 24.46 to 62.9 watts/cm^2.

A number of parameters having dominant influence on the critical heat flux and hydrodynamic instability (flow and pressure oscillations) preceeding the burnout have been studied. These parameters are mass flow rate, mass velocity, throttling, channel geometry (diameters ratio, length to diameter ratio, and test section length), and inlet subcooling.

Flow regimes before and at the moments of burnout were observed, discussed, and compared with the existing physical model of burnout.

INTRODUCTION

The nature of flow boiling in vertical channel, burnout conditions, parameter affecting the boiling crisis, and the hydrodynamics instability usually preceeding the burnout are very essential to be known for designing boiling water reactors.

The aim of the study is (i) to determine the critical heat flux for upward natural flow boiling of water in annuli, (ii) to study the parameters of dominant influence on the critical heat flux and the hydrodynamic instability, and (iii) to observe the flow regimes before and at the moment of burnout in order to compare them with the existing physical models on burnout.

Twenty runs of experiments to determine the critical heat flux were carried out on an experimental loop, designed and constructed specially to this purpose. The experimental loop is reported in ref.(1) Results, for natural flow boiling in 15 and 25 cm length test sections having 1.85, 1.43, and 1.23 diameter ratios and different conditions of mass flow rate and inlet subcooling are presented.

APPARATUS

A schematic layout of the experimental loop is shown in Fig.(1). The entire apparatus consists of an annualr test section of centrally heated stainless steel tube and glass jacket, condenser, electrically heated preheater, throttling valve, and cooling water supply system. Distilled water passes through the test section by natural convection where it boils and a portion of it changes to vapour leaves the test section to the condenser where the vapour losses its latent heat and subcooling of the entire fluid occurs. The liquid flow from the condenser to the preheater through throttle valve in the downcomer to regulate the flow and to study the effect of throttling on the flow instability. The pressure in the loop equal to the height of the water in the feed water tank. For further details of the loop we refer to a previous report.

BURNOUT DETECTION

Burnout detection is based on the measurement of the test section outer surface temperature by chromel-alumel thermocouple. The thermocouple output was continueously recorded on Honeywell single point temperature recorder. The recorder was equipped with an adjustable micro switch to trip the power supply to the test section when the wall temperature reached a predetermined value (200°C).

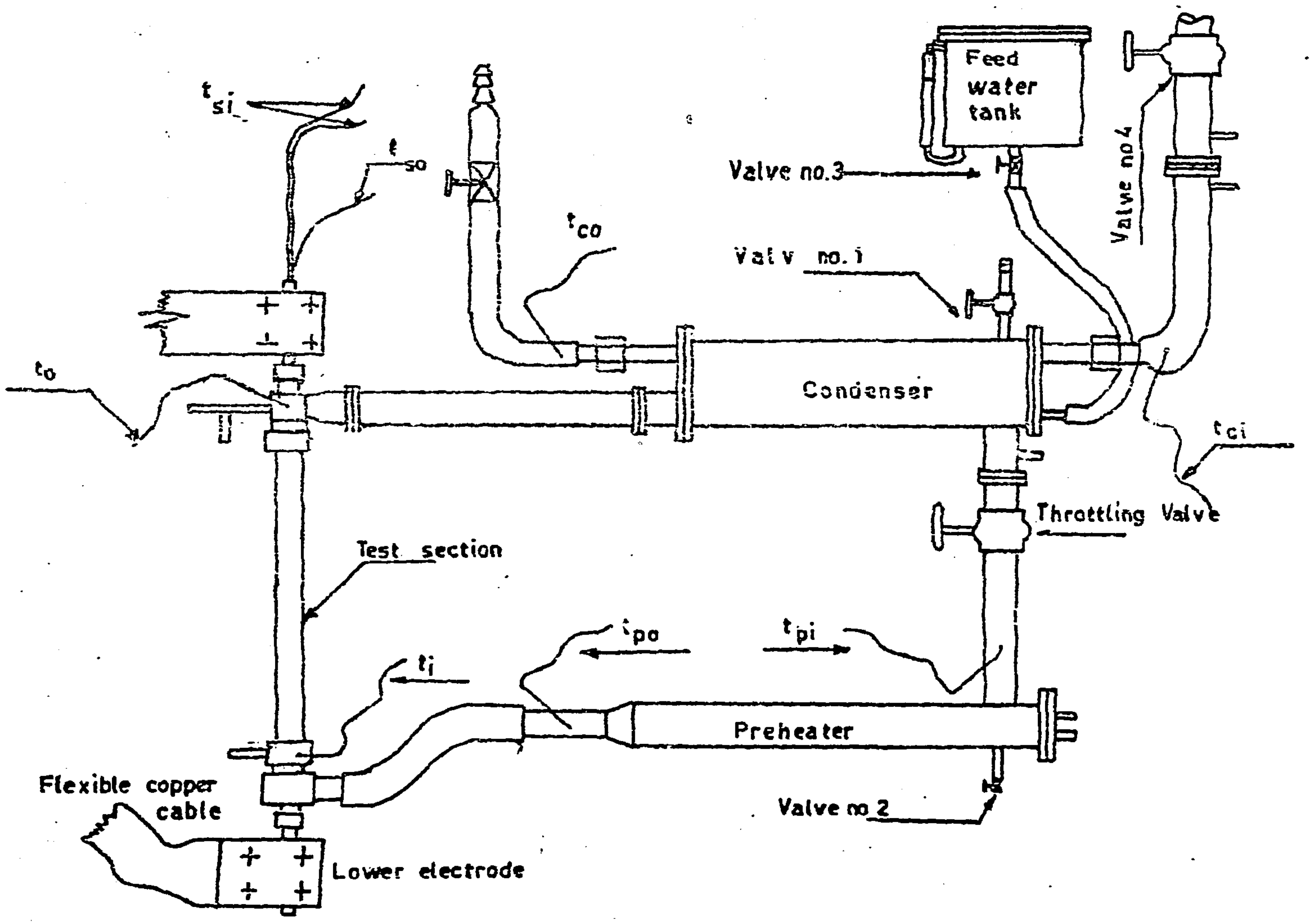

Fig. (1) A Schematic Layout of the Experimental Loop.

RESULTS AND DISCUSSION

Twenty runs were carried out to find the critical heat flux for natural flow boiling in 15 and 25 cm length test sections having 1.85, 1.45 and 1.25 diameter ratio at different conditions of mass flow rates and inlet subcooling. The inlet subcooling varied from 5 to 18 °C.

Heat Flux- Δt Curves :

During each run the heat flux was increased gradually causing the fluid to flow naturally. The surface temperature was measured by the thermocouple located 1 cm from test section upper end (the region where the burnout is expected) at each step of power increase. The difference between the recorded surface temperature and fluid saturation temperature Δt (°C), was plotted against the heated flux (watt/cm^2) as shown in Figs. (2) to (6) for different conditions of inlet subcooling, diameter ratios, and test section lengths. The resulted curves are portions of the Nukiyama curve but for flow boiling. The heat transfer coefficient (q" divided by Δt) curve is shown in Fig.(2). Two regions are observed on these curves. First q" increases slowly with Δt, and h increases slowly (natural convection region with small bubble formation). Second, q" increases rapidly due to bubble streams from surface causing considerable agitation and turbulence. This increases h considerably in the nucleate boiling region before the occurrence of burnout.

For 1.25 diameter ratio, Fig.(3) shows that the mass flow rate for 25 cm length test section was higher than that of 15 cm length, and the q - Δt curve for the longer test section was displaced above the curve of short test section. This was expected from the forced convections typical boiling curves explained by reference$^{(2)}$. At this case where (D_2/D_1= 1.25), the flow was highly unstable for both lengths of test

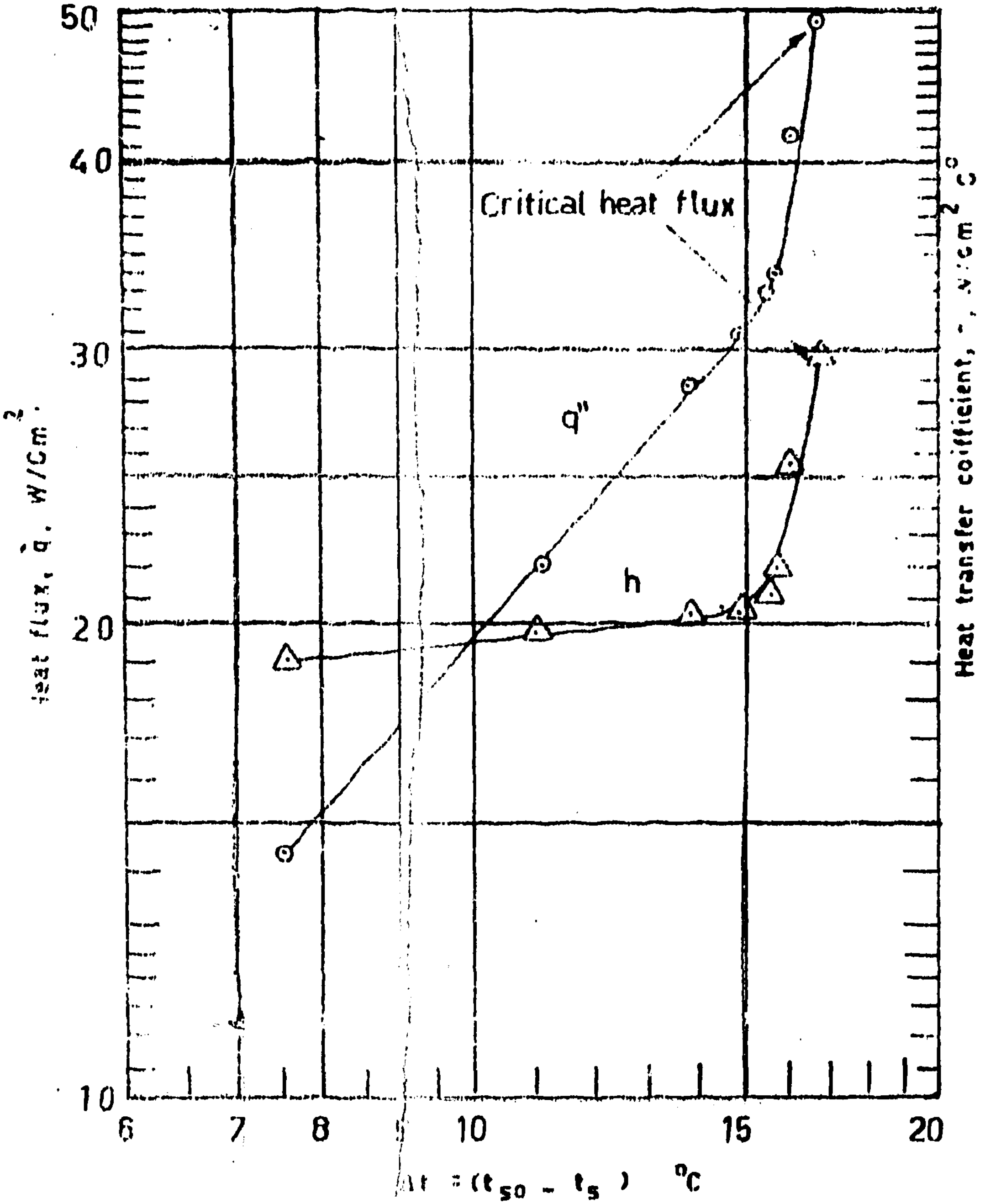

Fig. (2) Heat Flux versus Temprature Difference for L=25 cm t = 95 ..100°C and D_2/D_1 = 1.85

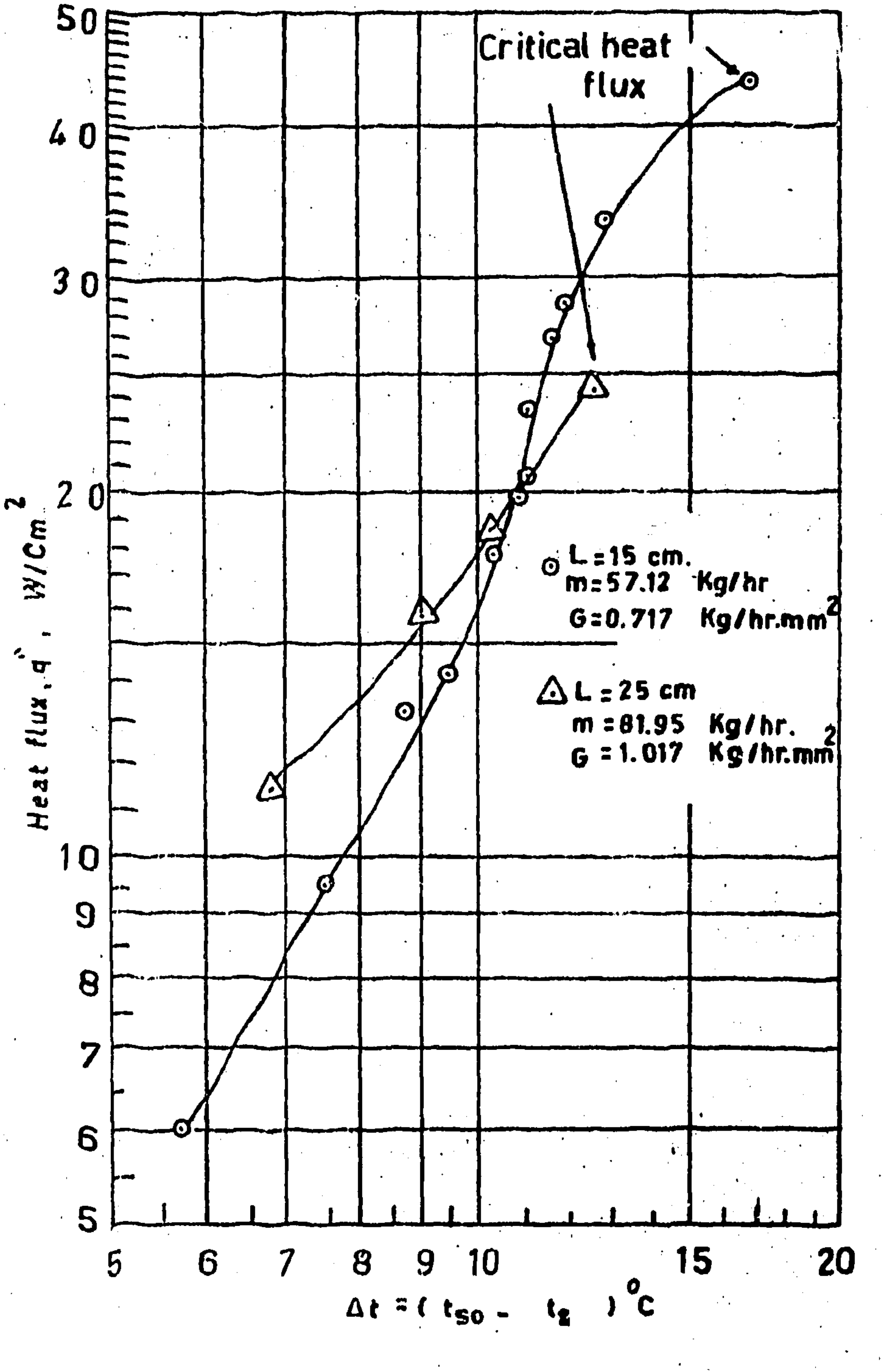

Fig. (3) Heat Flux versus Δt for $D_2/D_1 = 1.23$ and Inlet Temperature = 95 – 100 °C

Section specially at the onset of burnout. The burnout occurs for the longer test section (25 cm length) at lower heat flux and higher flow rate than those of shorter test section (15 cm length)

Fig.(4) shows that when D_2/D_1 was increased from 1.23 to 1.43, the flow was more stable and the mass flow rates at the burnout onset as well as the critical heat flux were higher for the longer test section (25 cm length) than the shorter one (15 cm length). The same results are expected for forced flow boiling.

For the same length (15 cm) and inlet temperature (80-85 C), the increase of diameter ratio from 1.43 to 1.85 increases the mass flow rate from 73.6 to 160.3 kg/hr at the moment of burnout. The critical heat flux was higher in the case of $D_2/D_1 = 1.23$ (see Fig.5)

For the same length (15 cm) and the same diameter ratio (1.85), Fig.(6) shows that the critical heat flux for high subcooling (84 C inlet temperature) is higher than for small subcooling case (97 C inlet temperature). This is understood since higher portion of heat dissipated from the test section is used to raise the fluid temperature from subcooled to saturation and/or superheated liquid temperature in the high subcooling case.

Mass Flow Rates:

The mass flow rate ,m , through the test section was measured by the preheater which was used as calorimeter. At each power step increase, the mass flow rate and the mass velocity were calculated and plotted against the heat flux Fig.(7) to (10).

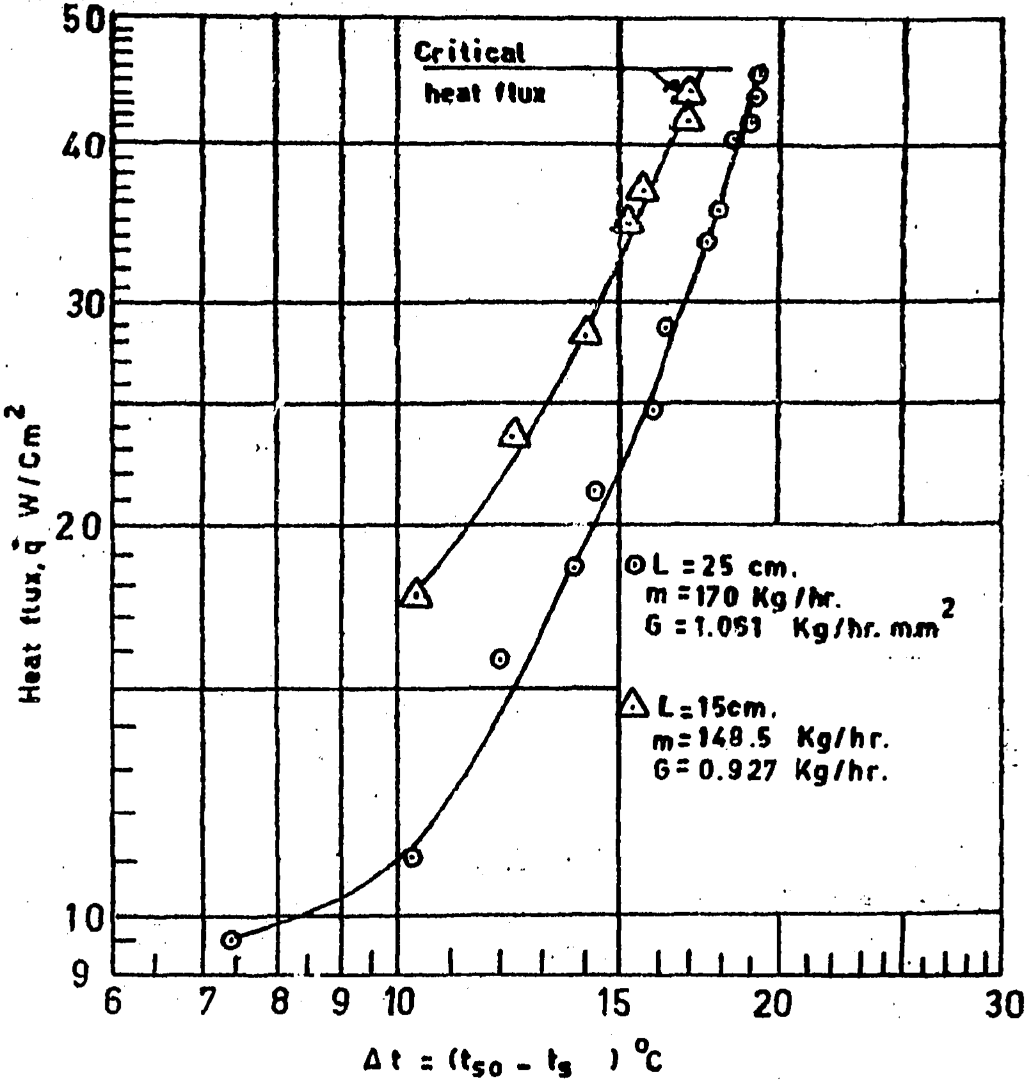

Fig. (4) Heat Flux versus Δt for $D_2 / D_1 = 1.43$ and Inlet Temprature = 95 _ 100 °C

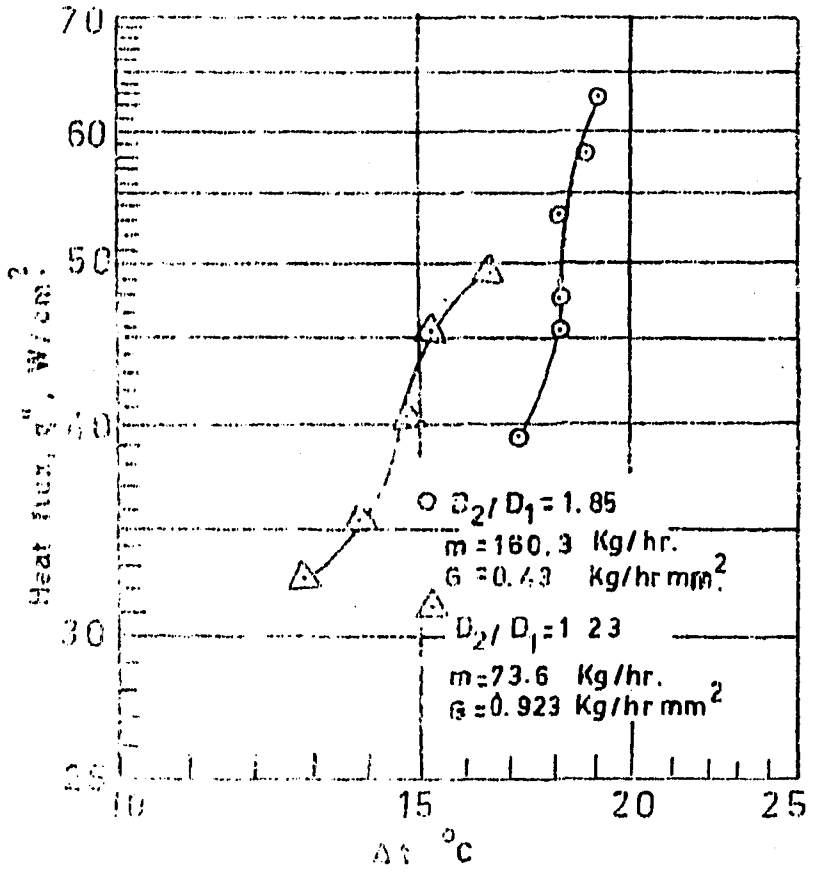

Fig (5) Heat Flux versus Δt for L = 15 cm and Inlet Temperature = 80_85 °C.

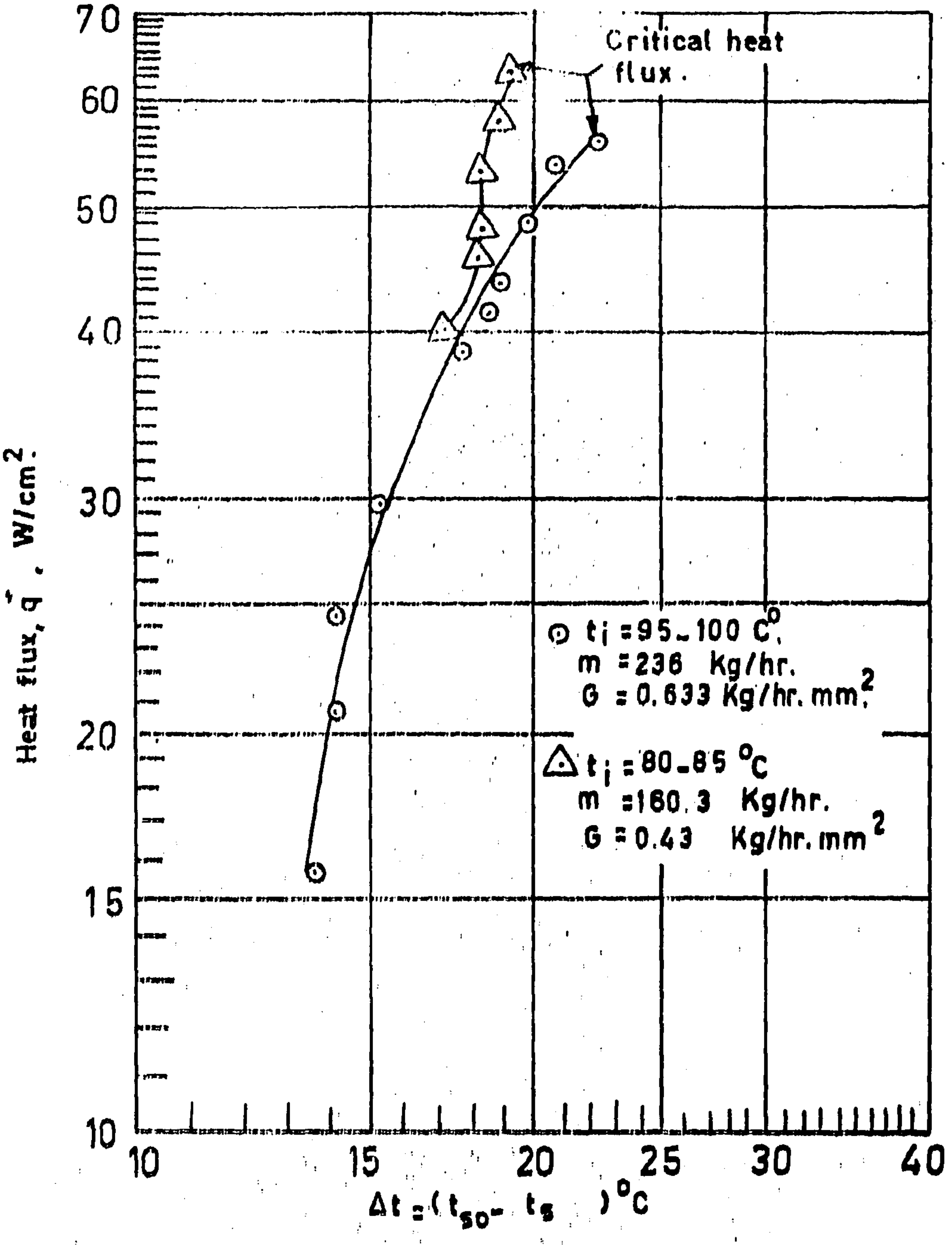

Fig. (6) Heat Flux versus Δt for $D_2 \neq D_1 = 1.85$ and L = 15 cm.

The curves plotted in Fig.(7) and (8) have the same characters for different lengths and diameter ratios if the inlet temperature was close to the saturation temperature (small subcooling). The mass flow rates for large cross sectional areas (i.e. large D_2/D_1) are higher than for small diameter ratios since less pressure loss due to friction is encountered. The increase of heat flux increase the buoyancy force and consequently increases the mass flow rates. Further increase in the heat flux causes the boiling to start extensively. Bubbles formation decrease the mean density and the corresponding velocity increases considerably. This increase the pressure losses (mainly acceleration losses) and the flow increases with smaller rate than before. Further increase in power causes frictional and acceleration losses to increase considerably and offset the buoyancy force. The flow starts to decrease after reaching a maximum. Further increase in the heat flux causes the burnout to occur and the channel cannot be effectively cooled by the fluid any longer. This holds true for 25 cm length test section with 1.23, 1.43 and 1.85 diameter ratio, and 15 cm test section with 1.43 and 1.85 diameter ratios.

For 15 cm length test section with 1.23 diameter ratio, the critical heat flux had a high value corresponding to small mass flow rate. During this run the regime was froth and unstable and the burnout character differs than the previous cases.

For large inlet subcooling ($\Delta t = 15\text{-}20$), 1.43 and 1.85 diameter ratios, and 15 cm length test section, Fig.(9) and (10) show that , m ,increases slowly with q" bubble flow exits up to certain q". More increase in q" changes the flow pattern to highly unstable piston flow and burnout occurs. It seems that burnount occurs as a

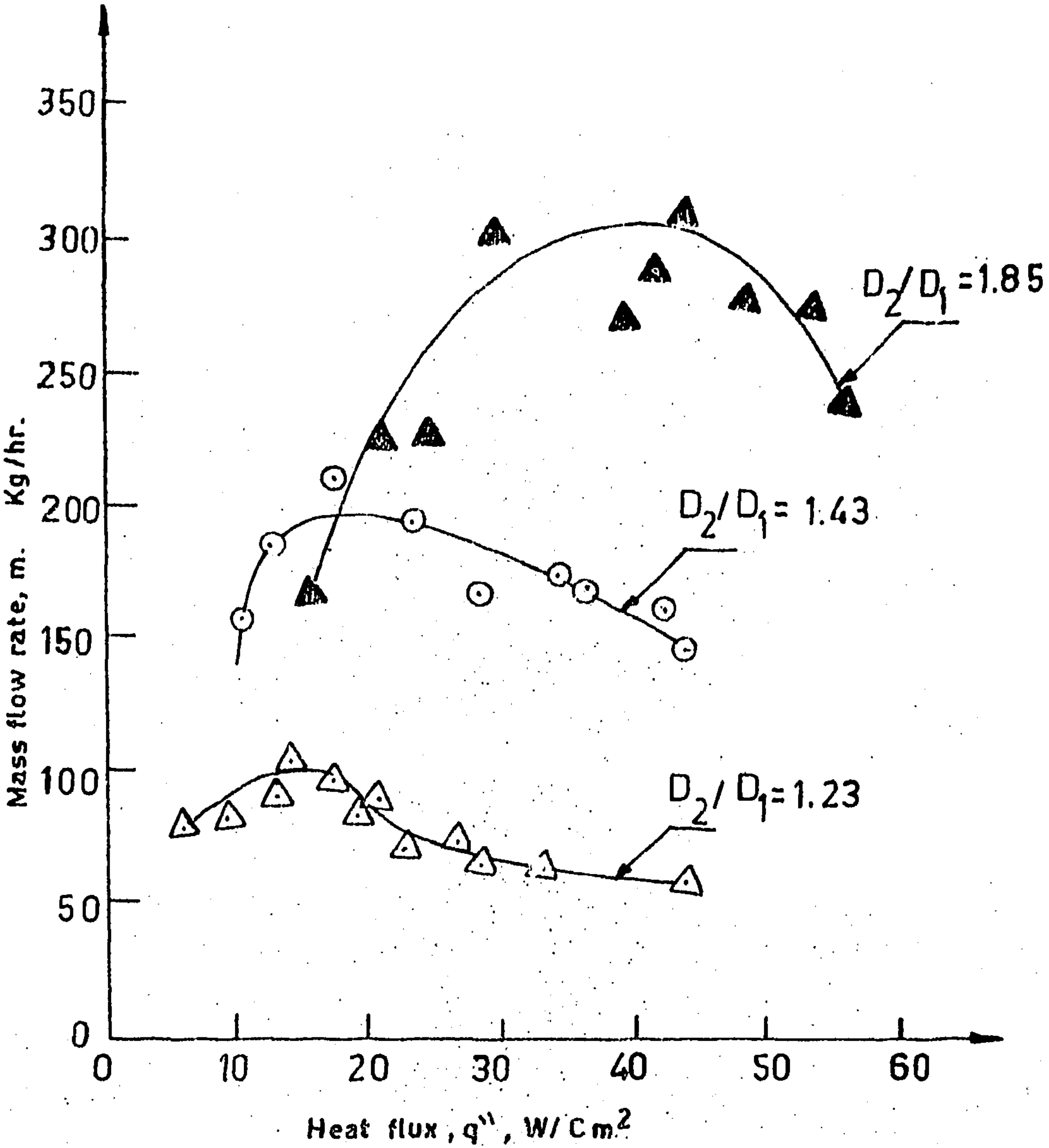

Fig.(7) Mass Flow Rate versus Heat Flux for L =15 cm and t_i = 95 - 100 °C

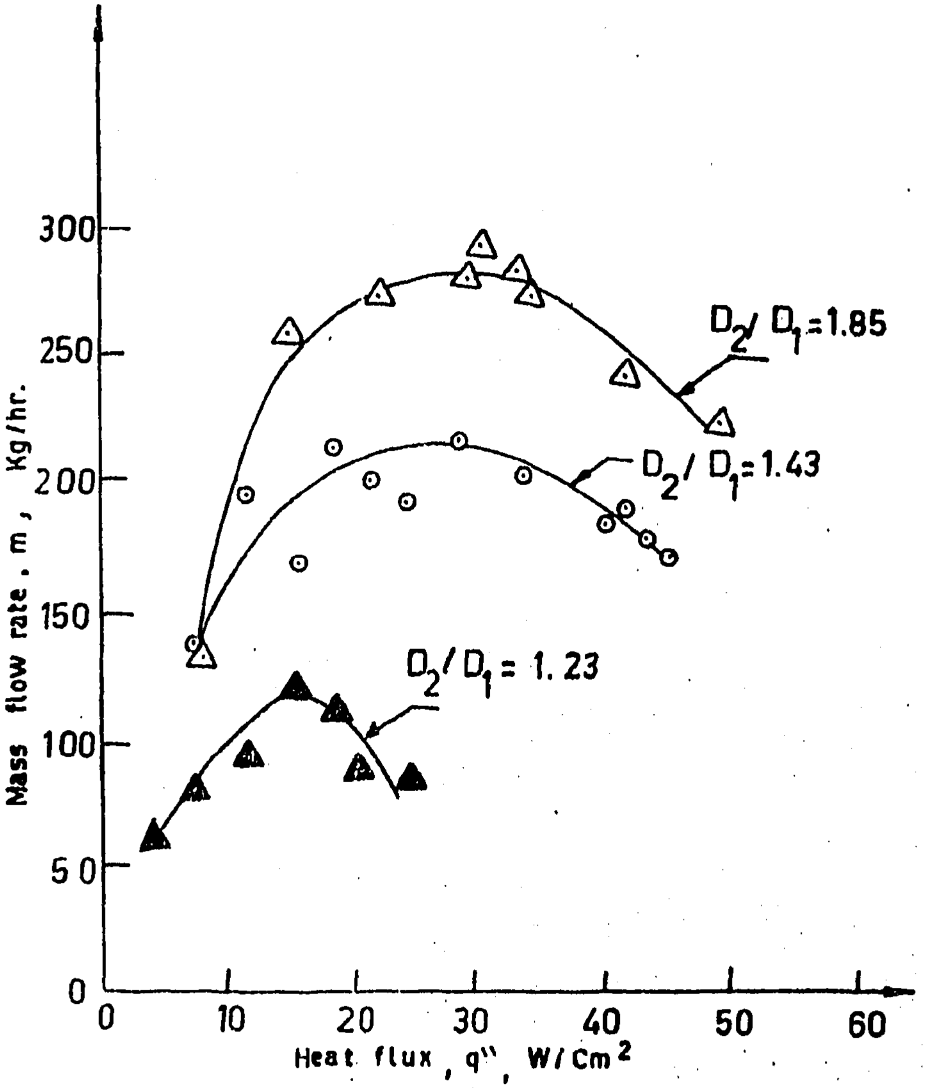

Fig. (8) Mass Flow Rate versus Heat Flux for L = 25 cm and t = 95-100°C

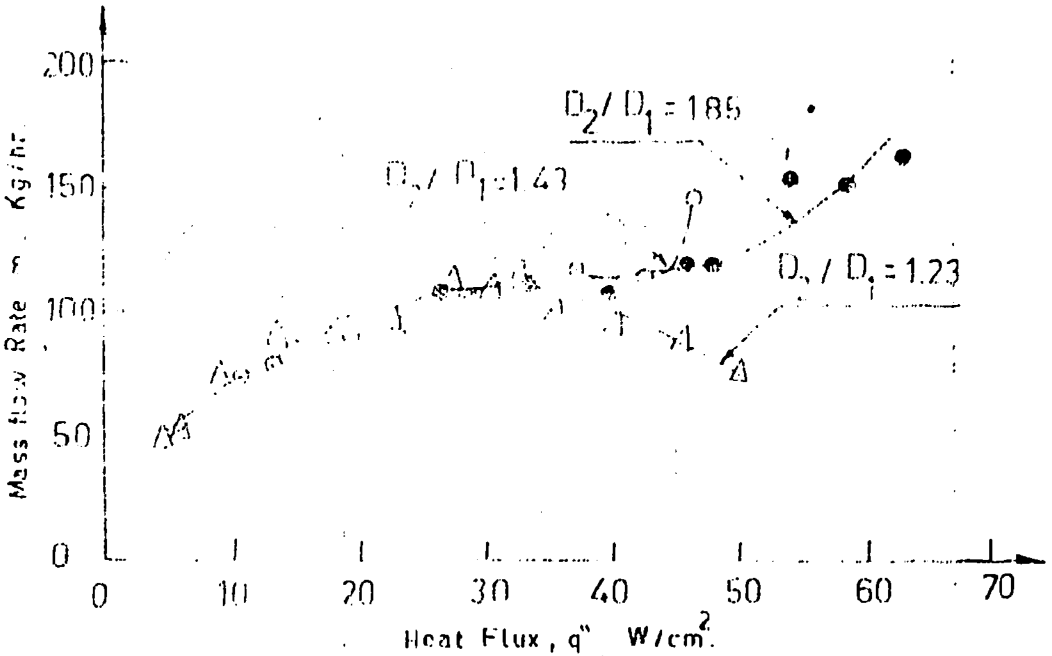

Fig.(9) Mass flow Rate versus Heat Flux for L = 15 cm and t_i =80_85°C.

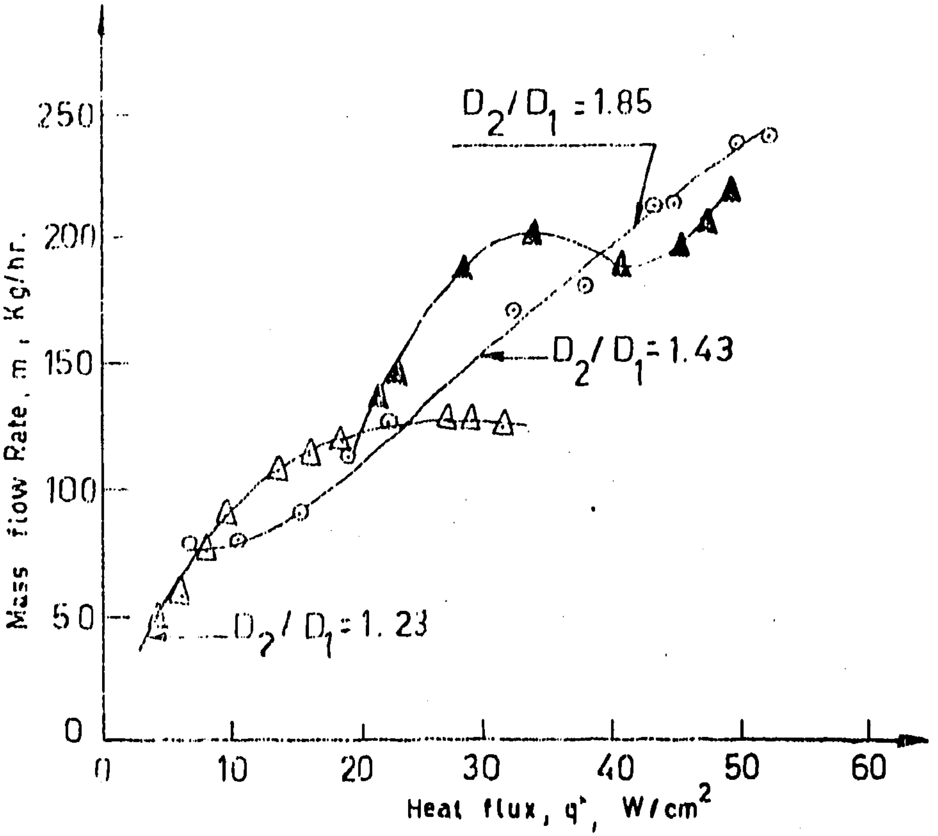

Fig.(10) Mass Flow Rate versus Heat Flux for L = 25cm and t_i = 80 _85°C result of increasing the thermal resistance and not due to mass flow deficiency. For small diameter ratio ($D_2/D_1 = 1.25$) and both test sections, the mass flow rate increases gradually then starts to decrease due to frictional increase in two phase flow. The burnout occurs after successive vapour flow followed by liquid coverage all over the test section. Small mass flow or dry out of the liquid film attached to the heater are the cause of burnout. For high subcooling at the inlet, it is noticed that the instability preceeding the burnout is vigorous. This instability should have more considerations than burnout. For all cases studied, the critical heat flux increases with the increase of subcooling.

Since the circulation is natural the mass flow rate could not be controlled to certain value (i.e. mass flow rate is dependent value). The mass flow rate was increased or decreased by adjusting the throttling value openings. At the moment preceeding the burnout, slug flow existed and give rise to mass flow oscillation. Data obtained show that for constant inlet enthalpy, length, and diameter ratio, the critical heat flux increases with increasing mass velocity Fig.(11). More data on the effect of throuttling is available in reference (5).

Temperature and Pressure Oscillation Before Burnout:

Temperature oscillation was observed in the slug flow regime before the burnout. Successive temperature increase during vapour passage followed by surface quenching during water slug passage cause this oscillation. As explained by Fiori et al(4), critical heat flux or burnout will result when the temperature rise (Δt), due to a dry spot exist at the burnout location and grows as the vapour passes over is greater than the temperature drop (Δt quench) resulting from the quench.

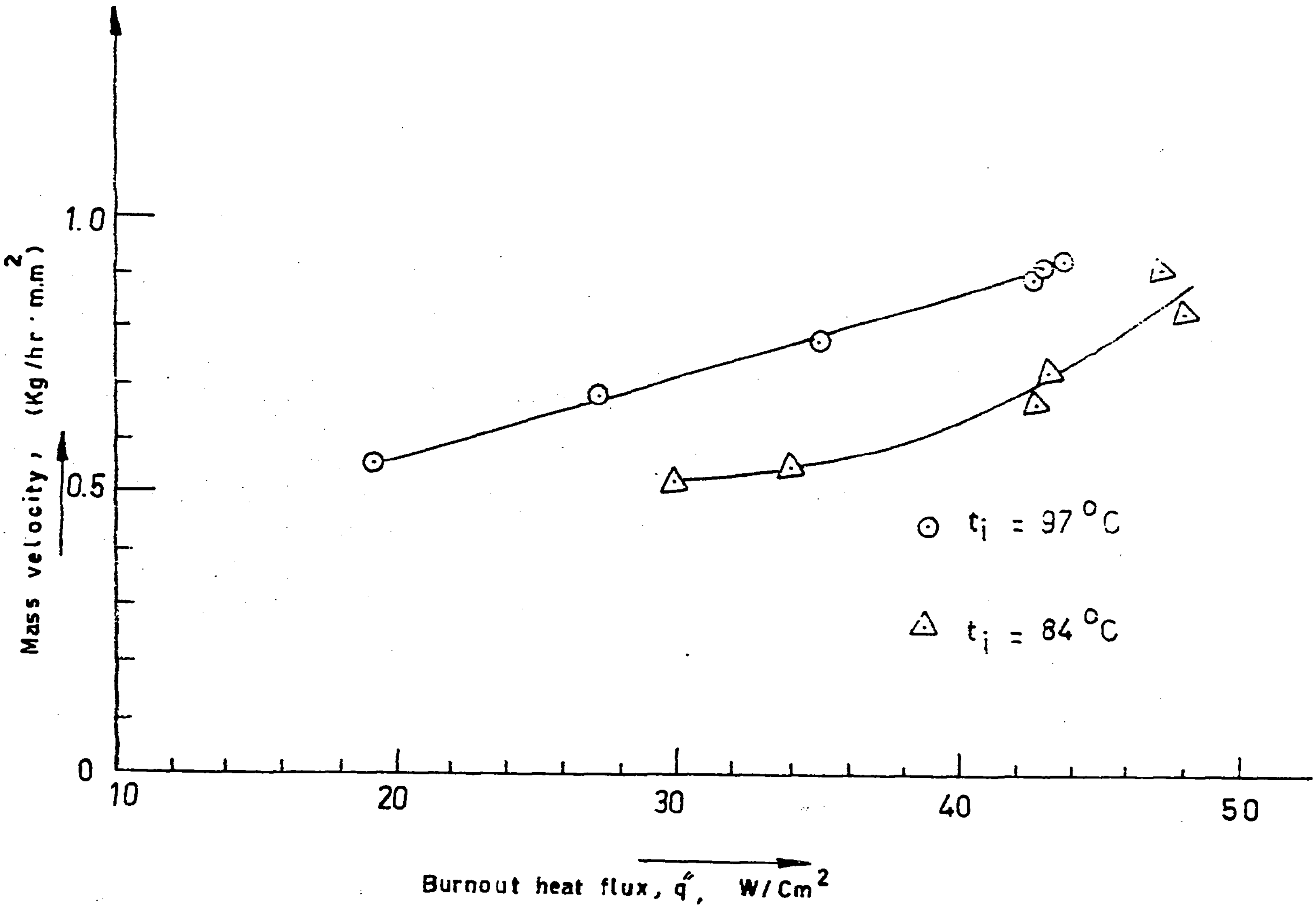

Fig.(11) Effect of Throttling on Burnout Heat Flux.
$L = 15$ cm, $D_2 / D_1 = 1.43$.

The amplitude of static pressure at the outlet generally increases as the heat flux increased and reached maximum at the onset of burnout. This amplitude increased as the diameter ratio decreased. The glass jacket used for 1.23 diam. ratio was exploded before burnout and was changed to brass tube giving the same diameter ratio. The explosion occurs as a result of large vapour formation in narrow space with no access to flow in immediatly.

Flow Pattern Observation Before and at Burnouts

As an example, a certain run will be considered to report the flow pattern during an experiment. In the run where 25 cm length test section with 1.43 diameter ratio and inlet temperature close to saturation, the power increase induced the flow and heat is transferred by natural convection for very short time. Further increase in q causes bubbles formation close to the surface. Coalesces of bubbles formed occur by power increase and form vapour slugs followed by water slugs. Most of the time, liquid droplets were suspended in the vapour slugs. The velocity of water droplets is lower than the vapour velocity and the following liquid slug succeeds in catching these water droplets. The frequency of vapour and liquid slugs increase by increasing q and the cycle took about half of the second when $q = 41.7$ Kcal/cm^2. hr.

As q increases to 43.6 Kcal/cm^{2}hr, large pressure oscillations occur and the time of water coverage to the test section was shorter than the corresponding time for vapour converage further increase in q causes the burnout to occur.

It was noticed that the temperature of the test section region where the burnout occured continued to increase beyond the predetermined temperature assigned to trip the power. This can be explained as a result of stable vapour clot formed at the upper and of the test section and acts as thermal insulator. The flow pattern beyond the burnout is shown in Fig. (12). The outer surface temperature measured by the thermocouple is lower than the inner surface temperature. After the power cutt off, heat transfer by conduction from the inner surface to the outer surface increased, for moments, the outer surface temperature. The vapour clot formed at the upper and of the test section was surrounded by water. Bubbles originated at point (a) due to the heat conducted through the tube after tripping the power. These bubble moved upward in the liquid surrounding the vapour clot and close to the liquid clot interface. These bubbles didnot emerga in the vapour clot. This may indicates that the pressure of the vapour clot is higher than the pressure of the bubbles. Effective cooling (at point a) by evaporation) causes the vapour clot to decrease gradually. It may be notice that the temperature increase beyond the cut off temperature depends largely on the size of this vapour clot.

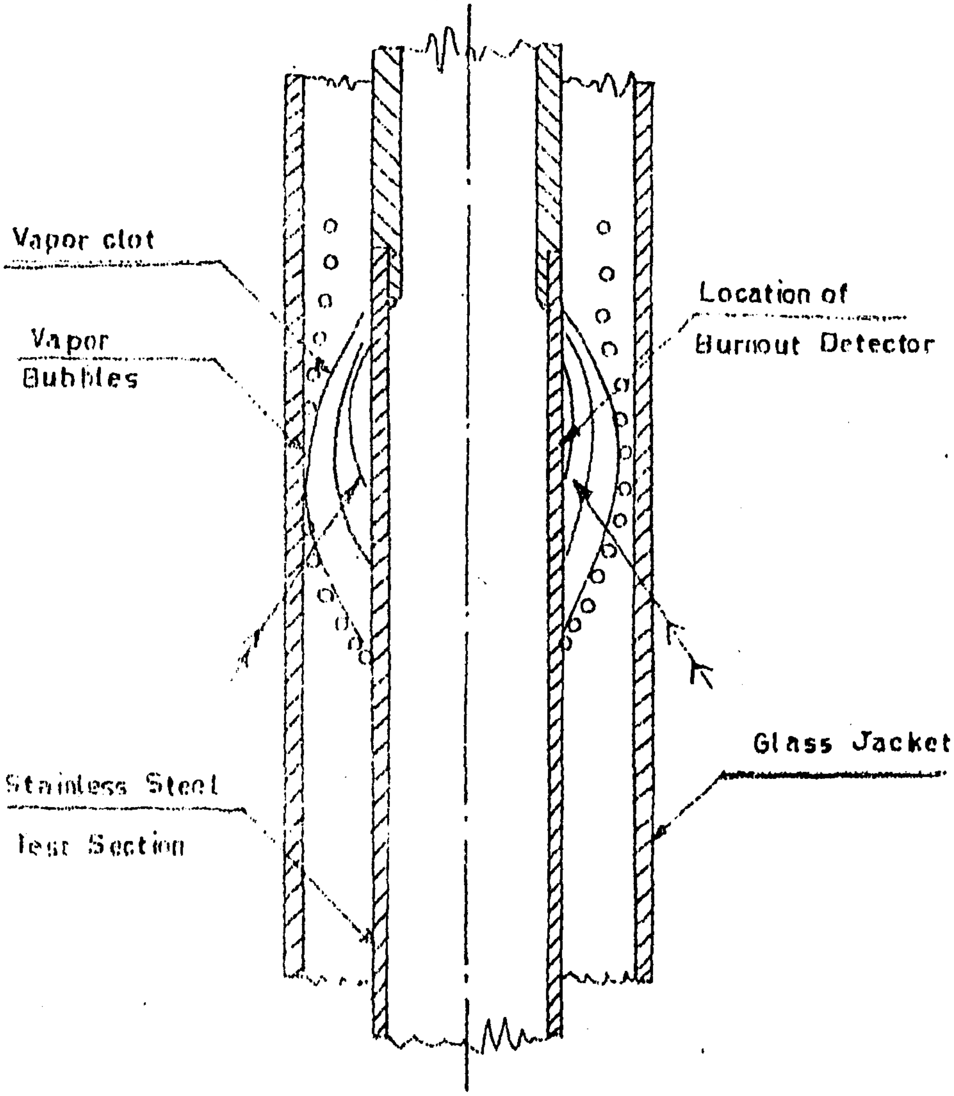

Fig. (12) Flow Pattern Beyond Burnout.

CONCLUSIONS

The heat flux at the moment of power tripping was denoted as the critical heat flux. The obtained critical heat flux varied from 24.46 to 62.9/watt/cm^2

The following conclucions can be derived from the performed experiments:

1- The critical heat flux increases with the increase of inlet subcooling.

2- The critical heat flux increases with increase of mass flow rate at the moment of burnout.

3- The mass flow rate increases as the diameter ratio is increased and consequently the critical heat flux gets higher.

4- The increase of the test section length results in higher flow rate and a decrease in the exit steam quality and decrease in the temperature level along the test section. As a further result, the critical heat flux decreases.

5- Piston of successive vapour and liquid slugs gives successive temperature rise (during vapour slug) and temperature decrease (during liquid slug by quenching) These temperature oscillations precede the temperature jump at the moment of burnout.

6- Flow instability precedes the burnout and has great influence on the value of the critical heat flux. Vigorous oscillations occur for small diameter ratio ($D_2/D_1 = 1.23$) and 15 cm test section length and cause premature burnout.

7- Burnout occurs as a result of mass flow rate defficiency and / or increasing thermal resistance between the heated surface and the flowing fluid.

8- A vapour clot surrounding the burnout location exists at the moment of burnout. The temperature jump beyond the prefixed temperature at the power cut of depends largely on the volume of this vapour clot.

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