THE UNIVERSITY OF ADELAIDE.

Department of Mining, Metallurgical and

Chemical Engineering.

HEAT TRANSFER

in the

CLIMBING FILM EVAPORATOR

with particular reference to

THE VARIATION OF LOCAL HEAT TRANSFER COEFFICIENTS.

T.O. Penman, B.Sc. (Eng.), B.E.

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ACKNOWLEDGEMENT.

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SUMMARY

The heat transfer to liquids boiling inside vertical tubes has been investigated with particular reference to the region where the climbing film has been developed. Four different sized tubes and five liquids have been used. In addition the work of other investigators has been examined.

As a result of these studies, it has been concluded that the main resistance to heat transfer, throughout the climbing film region, is at the liquid-vapour interface. The rate of heat transfer is dependent mainly on the rate at which liquid drops can be transferred from the climbing film to the vapour stream.

It has been shown that the following equation represents the action of the climbing film, and also the action which is found in the falling film evaporator and the horizontal tube evaporator when annular flow has been developed:-

$$\frac{h\sqrt{D}}{c_L\sqrt{\rho_L \sigma g_c}} = 0.012\left(v_B\sqrt{\frac{\rho_V D}{\sigma g_c}}\right)^{0.5}$$

Equations have also been presented for conditions which are found at the entry to the tube.

I certify that this thesis contains no material which has been accepted for the award of any other degree or diploma in any university and that, to the best of my knowledge and belief, the thesis contains no material previously published or written by another person, except where due reference is made in the text of the thesis.

CHAPTER 1

INTRODUCTION

The driving force in evaporation is the temperature difference between the heating surface and the fluid being heated. As the heating surface temperature rises above the saturation temperature, heat transferred to the liquid appears as latent energy of vaporisation, and vapour bubbles are formed.

For bubbles to form and grow the liquid must be superheated, the degree of superheat required being greater as the bubble radius is smaller (1). The formation of bubbles is made easier by the presence of irregularities on the heating surface. Bubbles are important in boiling heat transfer not only because of the heat as latent heat which they carry to the surface, but because they push a quantity of hot liquid from the heating surface into the fluid stream (2). Engelberg-Forster and Greif (3) showed that the quantity of heat in the liquid thus transported can be about 100 times the quantity transferred as vapour.

When a cold liquid is heated, local boiling precedes net vapour generation. Steam bubbles are formed on the heating surface and these condense and disappear in the bulk of the liquid (4). Local boiling is of importance because the formation and collapse of bubbles close to the heating surface disturb the boundary layer on that surface. The coefficient of heat transfer between the surface and liquid is then much greater than it would be for normal convective heat transfer (5).

As the temperature of the liquid reaches the saturation temperature, the bubbles which form on the heating surface migrate to the surface of the liquid. This is known as "Nucleate boiling". As the temperature difference between the surface and the liquid increases, so less favourable sites for nucleation become active. Eventually with further increases of temperature difference the whole surface is blanketed with vapour and the heat transfer coefficient, which increases as nucleation increases, decreases rapidly. This is known as "film boiling". When the heat source is a constant heat flux, as with electric heating, the surface temperature rises until mechanical failure occurs. When the heat source is of constant temperature, such as when a medium like condensing steam is used, the heat flux stabilizes at some value less then the maximum.

When liquids are boiled inside vertical tubes conditions are different in that no free surface is available. Consequently at the onset of boiling individual bubbles from nucleation points are diffused through the bulk of the fluid. These bubbles grow and coalesce until the volume of vapour is such that the bubble diameter approximates to the diameter of the tube and slugs of vapour alternate with slugs of liquid. The velocity of these vapour slugs is relatively slow and liquid can run down from the top of the slug to the bottom (6). Eventually the liquid separating the slugs is dispersed and a core of accelerating vapour drags up the tube a film of slower moving liquid. This region is known as the climbing film region. Industrial plant items which use this principle are known as Climbing Film Evaporators, a name Reavell states (8) which is preferable to the Long Tube Vertical Evaporator.

Failure or burn-out can occur through dry wall conditions in two ways. Either film boiling occurs or more commonly, because of continual evaporation from the liquid film and entrainment in the vapour, the film becomes so thin that it is circumferentially discontinuous.

Thus when liquids boil inside vertical tubes the following zones, all of which may be found in one tube, must be differentiated:-

a) The zone where bubbles are relatively uniformly dispersed through the liquid - homogeneous flow of bubbles and liquid.

b) The zone where slug type flow predominates.

c) The climbing film zone.

d) The zone where the film is discontinuous.

Work which is described in this thesis has been concentrated on the climbing film zone as this is the zone which predominates in industrial climbing film evaporators. It will be shown that the climbing film action can be expressed by the following equation:-

$$\frac{h\sqrt{D}}{c_L\sqrt{\rho_L \sigma g_c}} = 0.012\left(v_B\sqrt{\frac{\rho_V D}{\sigma g_c}}\right)^{0.5}$$

The left hand side represents the ratio of total heat transferred to the resistance of the liquid surface to the creation of new surface by evaporation. The right hand side - the Weber number - represents the ratio of turbulent momentum of the vapour to the resistance of the liquid surface to the creation of new surface by the vapour tearing at the surface.

The above expression, which correlates all experimental work done within ± 25% (with the exception of one result) over a wide range of conditions, also shows good agreement with other authors.

Equations are also presented to enable the heat transfer coefficients at the commencement of boiling to be predicted.

CHAPTER II

REVIEW OF PREVIOUS WORK

The climbing film evaporator was patented by Paul Kestner in 1899 and for many years was designed by empirical methods (7).

From 1936 onwards Badger and his associates (9, 10, 11, 12) carried out experiments in tubes 12 - 20 feet long and generally 1.75" ID. They used condensing steam as the heating medium and introduced the feed cold. A travelling thermocouple was used to determine the point at which boiling was thought to commence. At first only the overall heat transfer coefficient was investigated but subsequently thermocouples were imbedded in the walls to find the boiling film coefficient. The feed was then (10) introduced at a temperature 5 to 10 F° above the boiling point at the exit and in addition a quantity of sparge steam was injected into the bottom of the tube.

It was Badger who first reported (9) the zones of froth, slugs and climbing film, and although tube wall and liquid temperatures were measured up the tube, only average values of the heat transfer coefficient were used in the empirical expressions which were evolved. No attempt was made to separate these zones.

Kirschbaum (13, 14, 15) worked on a copper tube 1.5m long and 30mm ID, using condensing steam as the heating medium. He also introduced the feed cold and had a travelling thermocouple to find the extent of the non-boiling zone. He used wall thermocouples to determine the temperature difference between wall and liquid (ΔT) and thus calculated the boiling heat transfer coefficient (h). He found that $h \propto \Delta T^{0.6}$.

He then (16) used a 4 m tube 40mm ID, and found that the boiling heat transfer coefficient increased rapidly at the onset of boiling to a maximum and then decreased. The decrease he explained as being caused by the vapour blanketing the walls. Unfortunately insufficient data are available to enable the proportion of feed vaporized to be calculated, but it does seem likely, in view of the low values of ΔT at which this occurred that it was because of insufficient liquid being present to wet the wall. Much higher values of ΔT would be required for film boiling. Generally half his tube length was required to bring the feed to its boiling point. In some of the results presented for higher heat fluxes it is possible to trace two, and possibly three, slopes of the graph of ΔT against tube length. These may represent some of the zones described in chapter I.

Rumford (17) heated a cold feed in a 9ft. tube 0.5" ID. with hot water in a jacket. By using the technique suggested by McMillen and Larson (18) he found values for the boiling film coefficient, after correcting for the non-boiling zone. These values he considered too high to be reasonable, and he deduced that much of the heat required for evaporation was provided in the lower part of the tube. He concluded that the division of the tube into boiling and non-boiling zones was unwarrantable, and finally presented his results as if the fluid had been boiling throughout the length of the tube. It is possible, however, that his thermocouple, which was enclosed in a glass tube, may have given false readings of the temperature inside the tube. However Guerrieri and Talty observed that about 10% of the total evaporation always occurred in the local boiling zone. Rumford also concluded that above a certain vaporization, boiling film coefficients are not affected by changes in the feed rate.

Sternen (20) examined the heat transfer coefficient to boiling water inside a 16mm ID pipe. He plotted heat transfer coefficients against B, the steam content (percent evaporated) and found that for any one set of experimental conditions, the coefficients remained constant as B increased, and then rose sharply. He correlated only the sections of his work where the coefficient was constant, with a ratio of Nusselt numbers for boiling and equivalent non-boiling conditions. It is possible that these represented the zones up to the climbing film zone and the rising part of the curve, the climbing film. Unfortunately so little information is presented with the paper that very few conclusions can be drawn.

Coulson and Mehta (7) used a $5\frac{1}{4}$ ft tube 0.43" ID to boil water, sucrose solutions, and iso-propanol, with hot water in the jacket. They introduced the feed at its boiling point and attempted to measure the tube wall temperatures with imbedded thermocouples. The high velocity of the heating water in the annulus made the leads too difficult to maintain and consequently they had to use average values of wall temperature and heat transfer coefficient. Their results and Kirschbaum's (14) agreed very closely where comparable data were available. At constant temperature differences they found that $h \propto w_f^{\,0.25}$, a variation which was not found by Rumford (17) or Turner (21) and that $h \propto \sigma^{-1.4}$. In this surface tension correlation they pointed out that the exponent was probably wrong as the surface active agent they used might have affected other properties of their fluid as well as surface tension. They also showed that with a constant feed rate the heat transfer coefficient increases as the quantity of vapour generated increases.

In general the authors had not enough data to present a general correlation. They attempted to compare their results with those of Insinger and Bliss (22) which were obtained for nucleate boiling outside a 1.26" OD tube 6" long submerged in various liquids. Coulson and Mehta found their own coefficients very much higher. Now if the heat transfer coefficients found outside a tube can be related to those inside the tube it would seem that Coulson and Mehta's results could only be compared with those of Insinger and Bliss if nucleate boiling conditions prevailed in the whole length of their tube. As the results were higher we can assume that another mechanism, possibly that of the climbing film, gives higher heat transfer coefficients.

McNelly (23) investigated heat transfer to water and several solvents in tubes of diameters from ½" to 1" and 5½ ft. long. The apparatus used was essentially that used and described by Coulson and Mehta (7) but the hot water in the jacket was pressurised and consequently higher temperature differences were obtained. Again only average values of the boiling heat transfer coefficient could be evaluated. McNelly reported that he found that the annulus heating liquid entered with a spiral which only gradually straightened, consequently giving heat transfer coefficients for the water that were not constant up the tube.

He also reported that for low to moderate values of temperature difference $h \propto (\Delta T)^{0.53}$ and forced convection heat transfer prevails, while for higher values of ΔT, nucleate boiling predominates and

$h \propto (\Delta T)^{2.2}$. Both these conditions were reported when annular flow had been developed. During the region of forced convection heat transfer, higher coefficients would be obtained for small temperature differences than with nucleate boiling. For higher feed rates, higher heat transfer coefficients would also be obtained. In nucleate boiling, the feed rate would have no effect. McNelly presented a semi-empirical correlation for the forced convection line.

McNelly's results were very similar to those obtained for nucleate boiling by Rohsenow (24) who correlated data from Cichelli and Bonilla (25), Cryder and Finalborgo (26), Rohsenow and Clark (2) and Kreith and Summerfield (27). Cichelli and Bonilla obtained their results with horizontal flat plates submerged in several solvents. They found that the heat transfer coefficient increased as the pressure increased and that nucleate boiling ceased to be stable near the critical pressure because the surface tended to get vapour-bound. Cryder and Finalborgo used a $1\frac{1}{4}$" dia. brass tube to heat water under various pressures. Rohsenow and Clark boiled water on a 9" tube 0.18" ID under pressures up to 2000 psia, and concluded that the increase in the heat transfer rate obtained with surface boiling was due mainly to the agitation of quiescent regions of liquid adjacent to the heated surface by the bubbles generated there. Kreith and Summerfield used very high heat fluxes of up to 1,500,000 BTU/hr., sq. ft. with cold water in tubes 0.587" OD, 17" and 38" long.

They concluded that the Colburn equation (28) correlated their data only until the surface temperature exceeded the boiling point of the liquid. Above this, the surface temperature was a function of the pressure of the liquid. This conclusion was confirmed by McAdams and his co-authors (29) who used an electrically heated tube as the inner surface of an annulus, again with heat fluxes up to 1,500,000 BTU/hr/sq ft. Schweppe and Foust (30) investigated heat transfer to water in tubes 8½" long with fluxes up to 300,000 BTU/hr/sq ft. They found that the heat transfer coefficient for non-boiling feeds approached those they obtained for boiling feeds.

Schweppe and Foust reported that they found no variation of column behaviour with time, confirming the earlier work of Insinger and Bliss (22) who investigated this carefully. Staker (31) however found that a period of up to two hours was necessary before equilibrium was reached especially with solvents at high heat fluxes. He also found a hysteresis effect with nucleate boiling in that the heat transfer coefficient was higher if he reduced his heat flux to test conditions then if he increased it. Staker explained this by considering activation of nucleation sites. Once a site has been activated by a high temperature difference it remains activated although the temperature difference has dropped below the value at which that particular site nucleated.

Activation is possibly caused by small air pockets trapped in small surface irregularities. Staker argued that the occluded air is more easily removed at low pressures and it has been established (25, 32) that for any given heat flux the temperature difference decreases as the pressure increases. Griffith and Wallis (33) who experimented with prepared nucleation sites, found it impossible to initiate boiling from prepared sites if the liquid had been degassed before starting. They surmised that the liquid would then have dissolved any occluded gas in the cavities. They also found that sites could be deactivated by increasing the pressure of the system sufficiently to make the liquid subcooled and then releasing the pressure. They argued that the cavities thus deactivated have become full of liquid. Pike, Miller and Beatty (34) found that air and carbon dioxide dissolved in water reduced the temperature difference required for a given heat transfer coefficient by very large factors. Westwater (35) stated that a gas-filled cavity with a sharp apex serves as a nucleation site because the gas can easily gain vapour and grow to a bubble of critical size. Then, even if the gas molecules are gradually removed in escaping bubbles, vapour remains because the liquid cannot fill the tip against the surface tension forces. Hence the site remains active.

Cathro (6) and Cathro and Tait (36) described the study of cine-films taken of water and methanol in a glass climbing film evaporator. When the climbing film action starts, a thick uneven film is left on the wall. This climbs upwards under the thrust of the vapour but falls back when the vapour flow momentarily ceases. In this unstable zone of alternate rise and fall, there is a high degree of slip between liquid and vapour, and nucleate boiling is suppressed, as it is when climbing film conditions are fully developed. Cathro stated that bubbles could be seen in slow motion shots to be forming in moving masses of liquid and this condition is contrary to that required for nucleate boiling. Yet with values of temperature difference of the same order that Cathro used, McNelly stated that nucleate boiling predominated. It must be pointed out that McNelly carried out his experiments in silver tubes and his statement about nucleate boiling was an inference drawn from his results. It is also possible that Cathro was unable to observe nucleation points because the bubbles were swept away while they were still very small. Against this is his description of a rising and falling film which presumably had momentarily still periods when nucleation points could have been seen had they been there.

Cathro also described the effect of adding 1% Teepol solution as a feed to his evaporator. When the liquid was boiled a stable froth was formed and liquid and vapour climbed the tube at the same velocity. Clearly the nature of the action of the fluids in the column had changed and hence caution must be used in claiming any particular effect of surface tension on the climbing film. This confirmed a suggestion made by Kirschbaum (14) that foam caused the change in heat transfer. Stroebe, Baker and Badger (10) added small quantities of a soapless detergent to their column and increased the boiling film coefficients 2 to 4 times. Morgan, Bromley and Wilke (37) investigated the effect of surface tension on nucleate boiling. They found that whereas 0.1% and 1% solutions of their detergent had essentially the same surface tensions, the more concentrated solution gave higher heat transfer coefficients, both solutions giving greater coefficients than water. The authors explained this anomaly by suggesting that as far as surface tension was concerned, a surface ages. The actual surface tension of a newly created surface would be different from the measured value and it is this actual value which determines the boiling characteristic. The authors did not explain why only one of these 3 liquids should age in this way and possibly the explanation may be in a radical alteration in the behaviour of the liquid, rather than in a trend. It was also reported by Morgan, Bromley and Wilke that the critical temperature difference reduced in value as the surface tension lowered. Coulson and Mehta (7) experimented with Teepol solutions of 0.01% and 0.1% concentration and some of their curves duplicated those presented by Kirschbaum (14), both groups of workers finding that the heat transfer increased as the surface tension decreased. Coulson and Mehta presented graphs which showed that the temperature from the bottom to the top of the tube dropped much more for boiling Teepol solutions than for the corresponding water runs. This suggests that there was a much higher pressure drop in the tube - 3.2 psi for Teepol and 1.2 psi for water - again indicative of a different action rather than a trend. In addition Insinger and Bliss (22), Bonilla and Perry (38), Rohsenow (24), Forster and Zuber (39) and others (10, 23, 40, 41) all claimed in their equations that the heat transfer coefficient increased as the surface tension decreased. On the other hand the reverse effect is noted by Kutateladze (42-) and Nakagawa and Yoshida (43) as reported by Westwater (35). Westwater commented that if a surface active agent is present in the liquid in the form of suspended particles these would act as nucleation points.

A limiting factor in the behaviour of many reported experimental climbing film evaporators is burn-out (6, 10, 23, 29, 30, 31). As has already been stated, with electrical heating, mechanical failure can occur and hence burn-out is a condition which should be avoided. McNeillis, in a review article (44) described the four most important causes of burn-out:-

1) The occurrence of film boiling in the tube.

2) When the rate of turbulence promotion fails to meet the demand of a sudden increase in heat flux, and local failure occurs.

3) The occurrence of sonic choking when the film appears to boil itself dry.

4) When hydraulic instability is caused by fluctuating flow and the flow is at a minimum during a single fluctuation. Most of

the work done on burn-out has been concerned with type 1) especially where high heat fluxes have been used with sub-cooled liquids. Gunther (4) presented an equation where the heat flux at burn-out is directly proportional to the difference between the saturation temperature and the actual temperature of the liquid, and proportional also to the square root of the fluid velocity. Bonilla (45) reviewed the information available which again is mostly of sub-cooled boiling. Gambill and Greene (46) compared the burn-out of sub-cooled liquids which are pumped with a vortex flow through small diameter pipes. Bernath (47) listed 10 reasons for the wide scatter of the data in the literature and presented an equation for the variation of burn-out with pressure and velocity. Cathro (6) commented from his photographic studies that when approximately 80% of the feed had been evaporated the top of the tube became dry. Bennett, Collier, Pratt and Thornton (48) found that beyond 65% evaporation the heat transfer rate decreased and approached a value set by the dry steam coefficient while Yoder and Dodge (49) evaporating Freon 12 found a similar result at 50%. There was no suggestion that there was any fluctuation of flow by Cathro and hence it would seem that a fifth type of boiling should be added to McNeillis's list of causes of burn-out - where the film is thinned because of excessive evaporation and entrainment, and finally becomes discontinuous.

Most of the work on pressure drop in liquids being evaporated in tubes has been done under conditions of high heat flux and high feed rates. Because of the extremely complicated nature of the problem, most equations which have been derived are empirical. Lockhart and Martinelli (50) considered cases where either or both the gas and liquid film would be in either turbulent or viscous flow. They produced a parameter, $^1/_{X_{tt}}$, for the case where both liquid and vapour are in turbulent flow, which is a strong function of the weight fraction of the vapour, modified by viscosity and specific volume terms. This has been the basis of several investigations (48, 52, 53, 54). Harvey and Foust (55) in their analysis made the assumptions that the vapour-liquid mixture was homogeneous and that there was no slip between vapour and liquid. This clearly renders such work inapplicable to the climbing film evaporator. Bennett and Thornton (56) investigated the pressure drop for air-water mixtures in the climbing film region and used friction factors which were expressed as functions of the liquid and gaseous Reynolds numbers. Because there are no additional complications of heat transfer and mass transfer by evaporation, the approach to the problem of pressure drops in climbing film evaporators could be more easily investigated this way.

Guerrieri and Talty (19) observed that the linear velocity of a two-phase mixture strongly influenced the heat transfer coefficient. Piret and Isbin (57) found variation of heat transfer with velocity. The value of velocity they used was a log mean value between the feed inlet and the vapour-liquid outlet assuming it to be homogeneous. The relationship they presented was an empirical Dittus-Boelter form modified by a surface tension group. Yoder and Dodge (49) found that one of the most important variables affecting the film coefficient was the percentage of vapour in the mixture and that the coefficient did not change markedly with varying heat flux, flow rate and pressure for a given fraction of vapour. They were however mostly investigating dry wall conditions and these observations are valid only over a range of from 30 - 50%.

Little appears to have been done in investigating the relationship between liquid velocity and vapour velocity. Untermyer (58) presented graphs of steam velocity and the velocity of steam relative to the water, against saturation temperature. This showed that at atmospheric pressure the steam velocity would be approximately 72ft/sec and the water 8 ft/sec. Lotten (53) also uses the slip ratio as a parameter in 2 phase boiling but is more interested in pressure drop. Collier and Lacey (54) in their review of spray cooled reactor systems show that the heat transfer coefficient increases as the vapour quality increases. They also report that coefficients of the order of 20,000 - 40,000 Btu/(hr sq ft F^{o}) have been obtained at pressures of 1000 psi. These authors also differentiate between the climbing film region and a spray dispersed region, although they claim no clear boundary between the two, nor is there any apparent difference in the way that the heat transfer coefficient increases in these regions.

As a result of the literature survey which was made at the start of the project, it was felt that the following points required investigation:

1) The manner in which the heat transfer coefficient changes with vapour velocity, heat flux, feed rate and film thickness.

2) The effect of tube diameter.

3) The effect of tube length.

4) The variation of the heat transfer coefficient with different materials, and hence different physical properties.

5) The effect of pressure, should the change of material produce insufficient evidence to enable the nature of the boiling to be analysed.

The literature which appeared while the experimental work was proceeding and the results of that work did not seriously affect this programme.

CHAPTER 3.

APPARATUS.

A diagram of the apparatus is shown in figure 1 and a photograph in figure 2.

The Tube

The work, as originally intended, was to extend the field investigated by Cathro (6). As a start a 6 ft long tube was envisaged so that there was a reasonable certainty that the climbing film would be fully developed for the majority of runs. The tube diameter would be a variable, but to start with a $\frac{1}{2}$" ID tube was selected. The tube material was the subject of much thought, depending as it did on the method which would be used for heating. It was eventually decided that the tube would be cold drawn copper. This would give a surface of reasonable smoothness and uniformity, yet not so polished that nucleation would be suppressed. It would have some relationship to the type of surface which would be met with in industry.

Four sizes of tube were investigated and they had dimensions as follows:

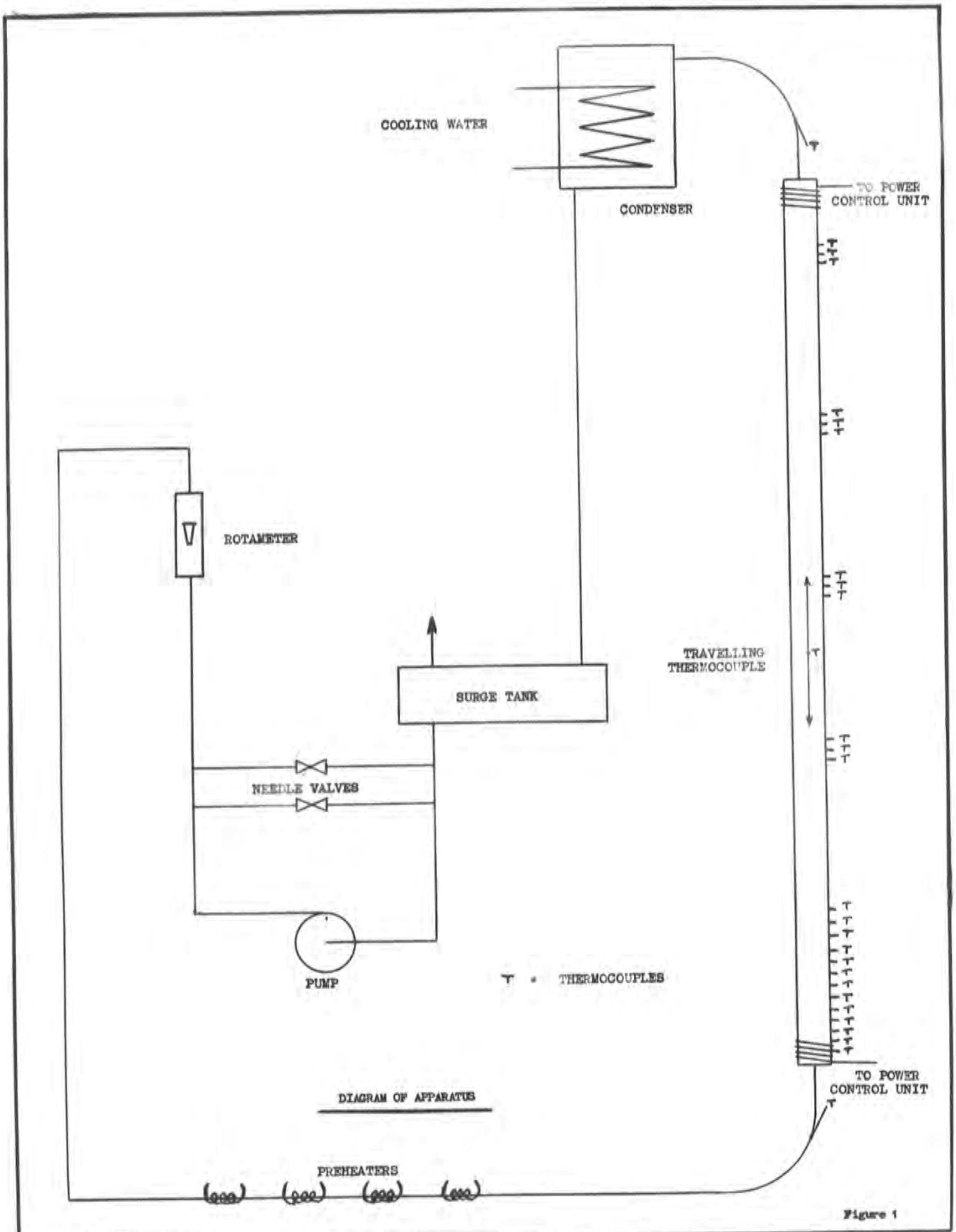
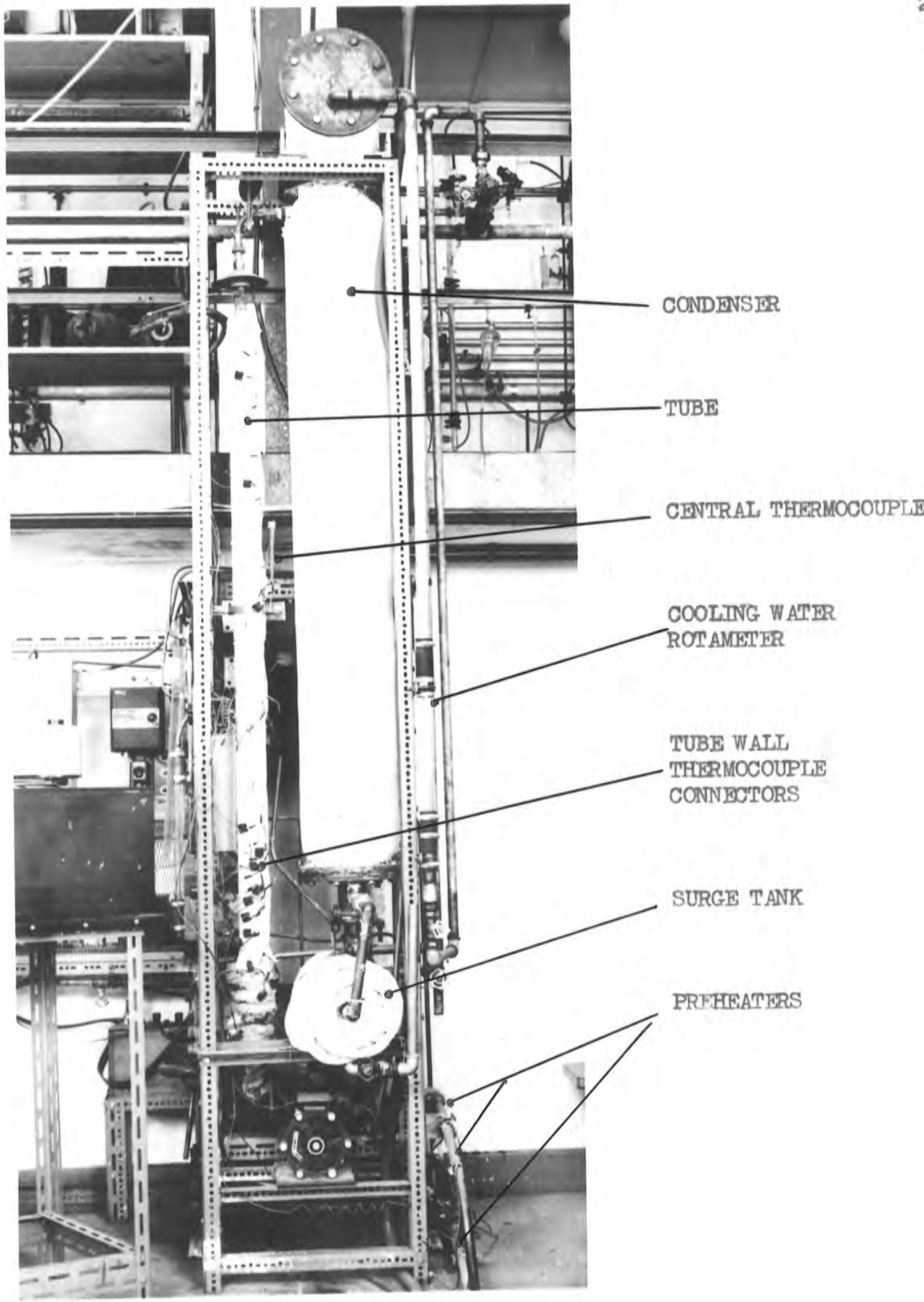

COOLING WATER
CONDENSER
TO POWER CONTROL UNIT
ROTAMETER
TRAVELLING THERMOCOUPLE
SURGE TANK
NEEDLE VALVES
PUMP
T = THERMOCOUPLES
TO POWER CONTROL UNIT
DIAGRAM OF APPARATUS
PREHEATERS

Figure 1



PHOTOGRAPH OF THE APPARATUS Figure 2

Nominal Size	Heated Length	Inside Diameter	Outside Diameter	Remarks
$\frac{3}{8}$"	72"	0.375"	0.500"	Max I.D. 0.376"
$\frac{1}{2}$"	68"	0.500"	0.751"	I.D. ± 0.0005"
$\frac{3}{4}$"	75"	0.747"	1.006"	Min I.D. 0.746"
1"	75"	0.996"	1.249"	I.D. ± 0.0005"

Before the tube was assembled into the apparatus, it was thoroughly cleaned. This was done by pulling through with flannelette soaked in trichlorethylene. This was continued until the cloth, which was renewed for each pulling through, remained clean. The tube was then pulled through with dry flannelette and finally inspected visually.

Heating Methods.

It was decided that the heat flux would be of the order of 100,000 BTU/(sq ft)(hr), thus effectively doubling Cathro's range. Various ways of heating the tube were considered. Rumford (17), Cathro, Coulson and Mehta (7), and McNelly (23) had all used hot water in the jacket but as has already been described the system has disadvantages. There is doubt that the heat transferred would be uniform (23) and an effective way of measuring tube wall temperatures had not been found. In addition it seemed unlikely that even with pressurized hot water a high enough flux would be achieved. "Dowtherm" and other heat transfer fluid media were considered, but the cost of the specialized equipment would have been prohibitive. Condensing steam was next considered, but it seems reasonable that for a 6 ft vertical condensing length the steam coefficient would change from top to bottom as the condensate film became thicker. Also it has been suggested (59) that when drop wise condensation occurs on the steam side nucleation may more easily be initiated on the boiling side. In addition the problem of measuring the tube wall temperature seemed difficult. Turner (21) has subsequently described how he had to abandon this method of measuring wall temperatures. He found that the reading depended on the soldered junction and the thickness of the condensate layer in its neighbourhood. Having six jackets each one foot long was considered but at that stage it was felt that the discontinuity which would be inevitable between each jacket was a potential experimental error and should be avoided. The building of a small furnace around the tube to be gas- or oil-fired, or electrically heated was also considered but abandoned, as it was felt that the difficulties of maintaining uniform temperatures or heat fluxes were too great.

Electrical means of heating were next considered. Many workers (27, 30, 47, 60, 61) have used the tube itself as the resistance to a d.c., but this did not appear to be attractive financially as a large and expensive transformer would be required as well as rectifying equipment. Besides considerable doubt was felt that accurate values of the wall temperatures would be obtained. It was then decided that the heat would be applied by means of a winding wrapped around the tube and operated at normal mains voltage. The possibility of winding the tube in six lengths of one foot each was also considered. However it was felt that there would be difficulty in matching each winding, and there would be more control required on the tube than would be feasible and that there was also the probability that the discontinuity at the end of each section would give a heat flow pattern which would vitiate the accuracy of any results. Consequently it was decided that the winding should be continuous.

First attempts were made by threading Nichrome wire 13 gauge, 111 ft. long, through asbestos spaghetti. This was wound onto the tube by mounting the tube in a lathe and feeding the insulated wire on under tension from the saddle which was engaged on the screw feed, set to the required pitch. A lathe steady was used to prevent the tube bowing under the tension which was applied to the wire. As the tube wall thermocouples were spaced at 12 inch intervals, the steady was made stationary. At 12" intervals the lathe was stopped, the steady advanced 12" and the winding continued.

This winding did not prove satisfactory. Generally the insulation was insufficient to withstand the higher voltages required to produce the high heat fluxes. It was surmised that resistance break-downs were caused at points which had been weakened when the wire penetrated the insulation during the threading process. Also the wire had to be in close contact with the tube throughout its length. Any excessive gap prevented the heat from being conducted away and failure often occurred because of melting. For this reason, the junction between the conducting leads and the heating wire was continually failing and the use of this method was abandoned.

Investigations were then made into the use of "Pyrotenax" heating cable. "Pyrotenax" consists of either conducting or heating wire insulated by packed magnesium oxide held in a copper sheath. It is proposed to use heating cable type 180/1K which is rated at 20 watts per foot for radiation in still air. It was reasoned that the limiting factor was the temperature that the core reached, and that if intimate contact could be maintained between the sheath and the conducting surface, this rating could be safely exceeded.

To ensure better heat transfer and to make the cable cover completely the length of the heated section, it was decided to flatten the "Pyrotenax". The cable was fed on to the tube in the same manner as the asbestos covered wire, but in addition passed through two rollers spaced 0.140" apart. The rollers were mounted on springs (like a small edition of a domestic wringer) and the cable as wound on to the column was a strip 0.205" wide instead of 0.177" dia. The ends of the "Pyrotenax" were connected to the conducting cable inside the barrel. Care was taken when the cable was assembled that the heating length inside the barrel was as short as possible. The barrel was silver soldered along its length to a copper lug which in its turn was silver soldered to the tube at right angles to the tube direction. Thus the short length of the heating cable inside the barrel was in intimate contact with a cooler surface and no trouble was experienced with cables assembled in this manner.

While the "Pyrotenax" was being wound, the tube and the cable were liberally coated with "Fryolux". This is a water based suspension of solder and flux. When the tube had been assembled, a power supply was connected to the "Pyrotenax" and heat applied by passing an electric current through the winding. About 1.5 Kw were used, controlled by an auto-transformer ("Variac"). This heat was sufficient to melt the solder, and thus the "Pyrotenax" was soldered to the tube along its length. In addition it was hoped that any slight irregularities in the winding were smoothed out. Excess solder and flux were removed by wiping the hot tube with a damp cloth.

When the cable was tested for electrical resistance from core to sheath, a low value was found. Visual examination of the "Pyrotenax" revealed several cracks in the cable. Much time was spent in attempting to repair these. An oxy-acetylene flame on low heat was used to warm the cable in the vicinity of the crack starting away from the crack. This would drive any moisture which had diffused into the magnesia insulation of the cable out through the opening. The metal around the opening was then cleaned and the hole sealed with a high temperature solder. After several cracks had been sealed in this manner it was realized that the rollers had work-hardened the sheath and that the bending the cable had undergone in being placed on the tube had caused the metal to crack. The cable was virtually useless, therefore, and a fresh length of "Pyrotenax" was obtained.

The second length of "Pyrotenax" was pre-rolled with the same rollers as were used in the first attempt. The cable was not fitted onto the tube, however, but loosely coiled inside a stainless steel tank, 3 ft in diameter. The ends of the "Pyrotenax" were connected to a power supply regulated by a "Variac" and a current passed through the coil. The cable was held at an estimated 300 - 350°F for two minutes and at the instant that the current was switched off, water, poured simultaneously from five buckets, flooded the cable in the tank, thus annealing the sheath. The cable was then wound on to the heating tube in the manner described for the asbestos covered wire without further incident.

During the first runs made with this tube it was observed that at the high powers the wall thermocouples had a temperature distribution which seemed illogical. The absence of any kick from the wall thermocouples when the power was switched on suggested that the a.c. or control system was not inducing stray voltages into the thermocouple system. The thermocouple switch was also investigated, but the same pattern of readings was obtained when point positions were reversed. As an additional check a different switch was also tried. The tube was then removed and placed in position upside down. An unexpected pattern of temperature readings was again obtained. Hence it was concluded that the trouble was in the thermocouples and the heating tubes. Close examination of the tube was made and it was then noted that at certain points the flattened side of the cable was not hard against the tube. The cable had a corner only on the tube and consequently the temperature of the cable was varying along its length thus heating the tube unevenly. The tube was tested with "Thermocrom" pencils. At low heat loads the irregularities were not detectable but they were noticeable at higher fluxes. The decision was made to wind a new tube without flattening the cable. This time a conducting type of "Pyrotenax" had to be used and was found satisfactory. The table below gives the lengths and types of cable used for winding 6 ft lengths of tubes of the diameters given.

TUBE		CABLE				
I.D.	O.D.	Type	Dia	Length	Ohms/ft	Resistance.
0.375	0.500	102/1	0.099	115 ft	0.0318	3.65
0.500	0.751	102/1	0.099	150	0.0318	4.75
0.747	1.006	196/1K	0.193	116	0.044	5.1
0.996	1.249	196/1K	0.193	140	0.044	6.15

It will be noted that only two types of cable were used. This limitation was imposed by the small range of "Pyrotenax" which is available in Australia. The letter "K" which follows the type number denotes that the conductor is made of Kumanal, a material whose resistance does not change with temperature. The other cable core was copper whose resistance does change. It is not a matter of concern that the resistance of the cable would change with temperature but it is if one end or part of the cable changes with respect to the other.

Let us assume that at one end of the tube the cable conductor temperature is 270°F (132°C), and at the other 250°F (121°C).

Now $$R_{T_2} = R_{T_1}\left(1 + \alpha_{T_1}(T_2 - T_1)\right) \qquad (48)$$

where R is the resistance in ohms, T is the temperature in °C, and α_T is the temperature coefficient of electrical resistance at temperature T.

For copper, $$\alpha_{T_1} = \frac{1}{\frac{1}{0.00393} + (T_1 - 20)}$$

$$= \frac{1}{254 + (121 - 20)}$$

$$= \frac{1}{355}$$

$$R_{T_2} = R_{T_1}\left(1 + \frac{1}{355}(132 - 121)\right)$$
$$= R_{T_1}\left(1 + \frac{11}{355}\right)$$
$$= 1.031\ R_{T_1}$$

So provided that the conductor temperature difference from one end of the tube to the other is of the order of 20 F° then the change in resistance, heat flux, and heat transfer coefficient is of the order of 3%. This accuracy is well within the usual limits in this type of work and is acceptable.

Lagging the Heating Tube:

It was decided that the heat losses to atmosphere from the column could best be stopped by surrounding the tube with a cylinder which would be maintained at the temperature of the windings. Clearly the diameter of this cylinder should be as small as possible to minimise convective losses. Because of the difficulties of threading such a cylinder over thermocouples and heating cable connections, the smallest practical diameter was found to be 5". 3 cylinders of asbestos pipe each two feet long, were prepared by winding nichrome heating wire of suitable resistance in notches on the outside. Each cylinder had 3 thermocouple groups, each group consisting of a thermocouple inserted with asbestos cement in a hole 1/16" deep in the outer surface, a thermocouple similarly buried so that its junction was within 1/16" of the inner surface and a third thermocouple projecting ½" into the air gap between the cylinder and the heating tube.

All these thermocouple groups were made midway between two coils of the compensating winding and the top and bottom stations were made 6" from the ends so that end effects would be minimized.

The three cylinders were separated from each other by asbestos discs turned in a lathe to fit the tube closely and to provide recesses for the ends of the cylinders. The cylinder windings were connected to power supplied from suitable variacs. The outer wall thermocouple and the air space thermocouple were connected in opposition to each other through the thermocouple switch and the potentiometer. The tube was placed under an arbitrarily selected load of 6 Kw with an arbitrarily selected water flow rate of 700 lb/hr. The compensating winding variacs were adjusted until all thermocouples in the centre of each cylinder were balanced. The other thermocouple groups were then checked and found to be out of balance. Any attempt to balance them resulted in the balanced thermocouples being put out of balance. In case external heat losses were causing these differences the compensating windings were lagged with asbestos mats. This did not improve the balancing and the experiment was abandoned.

It was then decided to lag the tube with 85% magnesia pipe lagging. This was done with 3 sections, two feet long, 1" internal diameter and 1" thick, secured in position by metal bands. A 240 volt a.c. supply was connected across the tube winding through a 2 Kw variac and an arbitrary load of 25 watts was placed on the tube. The tube was left for three hours to ensure that thermal stability had been established and the temperature of the windings and the air temperature were noted thrice at 15 minute intervals. This was repeated for other power loadings. A combined radiation and convection coefficient was then calculated so that if the winding and air temperatures are known, the heat losses from the column can be easily calculated. It was noted that because of conduction along the pipe (there being no water flowing) the heat losses were in fact measured from a greater area than the effective area between the two fluid stream thermometers. This was allowed for in the calibration calculations.

Vapour Liquid Separation:

Other workers have measured the degree of evaporation obtained in their tubes by splitting the vapour and liquid in a separator and measuring the two masses. Examination of Cathro's results (6) showed that by measurement he obtained evaporation rates up to 30% different from those calculated from heat input. It was his practice to use the calculated rate and to use the measured rate as a check.

It was therefore decided that little purpose was to be achieved in separating the vapour and liquid. In addition, difficulty would be found in measuring their rates accurately if the system was worked under pressure. Accordingly, the stream from the top of the column fed directly to the condenser.

Condenser:

The condenser was designed for the maximum expected heat load of 30 Kw and in case it was decided to investigate the climbing film under higher pressure it was designed for a working pressure of 250 psi, and a temperature of 400°F. The shell was made of 12" I.D. steam piping 1' - 6" long. The bottom was sealed with a 1½" thick plate welded in a suitable manner, and the top was fitted with a flange and a bolted cover designed to B.S. pipe flange table H.

Two ¾" N.B. steam pipe inlets with a male ¾" B.S.P. thread were welded into the bottom, one centrally and the other at the circumference. Two similar inlets were welded into the side wall near the bottom, one radially and the other tangentially. Four more connections were similarly placed at the top of the condenser. These connections thus provided means of connecting feed pipes, vents, and instruments to the condenser in a variety of ways. In the cover plate were welded 2 $\frac{3}{4}$" N.B. steam pipe connections which projected through both sides of the plate and which were threaded both sides as well. These were designed for the coils of the condenser and the cooling water connections. After fabrication the condenser was galvanised inside and out.

16 ft of $\frac{1}{2}$" I.D. copper tubing were wound into a helix and connected to the underside of the cover plate with a union. The coils were shaped so that if future work demanded, the coil could be lengthened with an inner helix.

As the experimental work proceeded, it was found that the condenser was near its limit. Before working on solvents a condenser which was taken from another piece of equipment was suitably modified and installed. It consisted of 15 copper tubes each 6 feet long, 0.5013" O.D. and 0.375 I.D. The condenser was installed vertically, with vapour in the shell and water in the tubes. The vapour was introduced at the top and the water passed counter-currently from the bottom.

When the condenser was commissioned it was found that conditions in the tube were unstable. The flow of cooling water had to be adjusted so finely, and the thermal lag of the system was so large that it was found impossible to keep the tube pressure constant and the feed at its boiling point.

The condenser was then modified to give co-current flow of cooling water and a rotameter was used to keep the cooling water constant. This proved to be most successful.

Preheaters:

The design of the preheaters depended upon the nature of the circuit which was to be used. Two possibilities were considered: an open circuit and a closed loop.

If an open circuit were used a large vessel would be needed with installed heaters to bring the contents to within a few degrees of boiling. This would entail either long delays at the beginning of experimental work or a continual heat load to keep the contents hot. It was felt, too, that the storage of large quantities of solvents at elevated temperatures was a health and safety hazard and consequently the alternative of a closed loop was considered.

The condenser was arranged so that the condensate fed directly to a 2 gallon surge tank from which a pump would feed it through preheaters to the tube.

The preheaters were made the experimental pre-runners of the heating tube, and various methods were tried to find the most effective way of applying heating cable to the piping from the pump to the heating column. At this stage the method of using asbestos spaghetti over heating wire on the column had been tried and shown to be unsuccessful. "Red Fibre" tubing was then tried as an alternative to asbestos covered wire wound around the tube. It had the apparent advantages that both its electrical and thermal conductivities were higher than asbestos. A tube length 1" O.D., $\frac{7}{8}$" I.D. was cut with a spiral screw-thread 0.025" x 0.030" at 20 turns to the inch, and wound with Nichrome wire to give a resistance of 57.6 ohms. This on 240 volts gave a heater of 1 kw. Mains water was passed through the tube at an unmeasured fast velocity and the power switched on. The tubing was observed to soften as the temperature of the wire increased and the experiment was abandoned. The use of "Pyrotenax" heating cable was then examined.

"Pyrotenax" has been described in the section on "The Heating Tube". A 4 kw winding was made by manually wrapping 37 feet of "Pyrotenax" type 104/1K closely around a ¾" copper tube. The ends were securely held in place by jubilee clips of suitable size. This winding length had a resistance of 14.4 ohms. The arrangement was tested by passing mains water through the tubes with a 240 volt power supply connected directly across the end connections. It was noted that the copper sheath of the "Pyrotenax" was oxidising rapidly so the test was abandoned.

To prevent this oxidation a separate tube was pre-coated with "'Ot Spot", a proprietary preparation sold for the prevention of oxidation of motor cycle engine cylinder heads. The cable was also painted with it, as it was being wound onto the tube. After this cable had been tested for one hour, signs of oxidation were observed and the test was discontinued. Another tube was then wound with cable after treatment in a similar manner with "Silvasheen", a proprietary suspension of aluminium in a bitumastic base. This was tested for periods of 3 hours or more and no sign of oxidation was observed.

The "Silvasheen" coated tube was, however, not entirely satisfactory as it was noticed that sounds of intense boiling activity were coming from one end of the tube only. This suggested that uneven heat transfer was taking place. Measurements of temperature at various points on the winding were made with a contact pyrometer. Results obtained were not satisfactory as readings perforce were taken on a point of the coil with a small flat surface and reproducibility was difficult. A trend in the readings though confirmed the suggestion that the heat transfer was uneven.

Another length of "Pyrotenax" of suitable resistance for 2 kw was then taken and after fluxing, tinned along its whole length. The length of tube onto which it was to be wound was also tinned, and the cable was then wound onto the tube mounted in a lathe. The steady turning obtained this way enabled a much more even winding to be produced. The end lugs were laid flat and silver soldered to the tube. Finally the cable connections were connected to a power supply from a 2 kw auto-transformer ("Variac") and the tube heated up until the solder melted. When the tube had cooled the cable had been effectively soldered to the tube along its length. The subsequent test with water flowing in the tube appeared to show uniform heat transfer along the length of the tube. This was confirmed by the uniform change of colour obtained when testing with "Thermochrom" pencils.

A second 2 kw winding was then assembled as before but this time "Fryolux" solder was used to fix the cable to the tube. The details of this method have already been described in the section "The Heating Tube". This winding was tested and found to give uniform heating along its length.

The apparatus was assembled with four 2 kw windings, two of which were connected in parallel to give an effective load of 4 kw. The 2 kw winding at the downstream end was connected through a variac, and hence the loading on the preheaters could be infinitely varied from no load to 10 kw. A distance of approximately 4 ft was kept between the last winding and the inlet to the tube to ensure that there was no possibility of vapour not being in equilibrium with the liquid at the entrance to the tube.

<u>Pumping:</u>

It was estimated that a maximum of 1000 lb/hr of feed would be needed, and that this feed rate should be easily adjustable whilst the apparatus was running. A $\frac{3}{4}$" "Globe" gear pump driven through a variable speed gear box with a 10 : 1 variation in output speeds was used in conjunction with a 1 H.P. 1440 rpm squirrel cage motor and suitable pulleys to give the required range.

So that the inside of the tube should not be contaminated with grease all grease cups were removed from the gear pump before it was installed and the pump was washed grease-free with solvents. The pump was very slow running and the wear rate proved negligible.

To enable finer adjustments to be made than could be obtained with the variable speed gear box, two by-passes were installed around the pump: one with a $\frac{1}{4}$" needle valve and the other with a $\frac{1}{8}$" needle valve. These enabled fine and very fine adjustments of flow rates to be simply made.

Flow Measurement:

The measurement of the flow rate while the system was also under pressure was considered. A rotameter in a stainless steel body with a magnetically linked follower of the float was considered but the expense and delivery delays enforced consideration of an orifice system.

Two orifice plates were made, both to B.S. specifications, one $\frac{1}{8}$" dia. and the other 0.08" dia. Vena contracta taps were made with $\frac{1}{8}$" dia. copper tubing. A bypass was fitted around the orifice plate so that the measuring fluid would not be forced from the system if too fast a flow rate was used.

A mercury under water manometer was found to give reasonable differentials under the expected flow conditions. Several systems of reading the differential were tried. A graphite rod was turned approximately $\frac{1}{4}$" dia. and held through an insulated bush inside one arm of a U-tube fabricated from $\frac{3}{4}$" M.S. piping. By measuring the resistance from the top of the graphite rod to the mercury, the position of the mercury level could be quickly calculated. After several hours operation, the indicated flow rate was found to be inaccurate. Examination revealed that the mercury had occluded onto the graphite and lowered its resistance, consequently showing a higher level than in fact prevailed. The graphite rod was then replaced by a high resistance wire wound on to a wooden rod. The wire was kept in position by laying it into notches prepared in the rod. This device also failed through globules of mercury being entrained on the wire and in the grooves. Further experimentation was obviated by the location of a pair of high pressure boiler sight glass fittings and tubes. These were installed so that zero was obtained at the bottom of the downstream glass and the top of the up-stream glass.

During the work with solvents, it was noticed that vapour locks were occurring which caused the abandonment of several runs. As it had become clear by this time that it would not be necessary to run the tube under pressure, the orifice assembly was replaced by a rotameter of suitable size.

Both the rotameter and the orifice assemblies were carefully calibrated by timing weighed quantities of the feed. The tube was disconnected at the base and the line from the preheaters was connected to a tank standing on a weighing machine. When smaller flow rates were being calibrated, the feed was run into a graduated flask of suitable size and the time for a particular volume was measured. As far as it was practical, the assemblies were also calibrated for fluids at temperatures higher than ambient. It was not possible to approach boiling point in these calibrations as the surge tank was of limited capacity, and in the case of the solvents fumes were a hazard to health and safety as well. The results of the calibration runs were plotted on log-log paper and produced straight lines. The readings which would be expected at the temperature of the runs made above ambient were then calculated, plotted and compared with the values found. In all cases the lines coincided. The calibration line was then calculated for a temperature 10°F below the boiling point and plotted. As the recycling feed would be only slightly cooled after being condensed it was considered that this would represent an average value of the temperature of the measured fluid.

Fluid Stream Thermocouples:

The inlet and outlet fluid stream thermocouples were made from 22 S.W.G. Chromel/Alumel wire. These were inserted into the fluid stream on the bends away from the line of the heating tube. It was thus hoped to minimize any chance of error because of radiation from the tube wall. The thermocouples were inserted into the fluid stream through ¼" dia. branches silver soldered onto the bends. After experimentation, a successful gland was made by threading three fibre washers onto the two thermocouple wires. The washers were drilled with two holes for the wires, the holes being made with the finest drill bit available and located ⅛" apart. The three washers were squeezed together by a female threaded cap screwing onto the male thread of the branch, the strain being taken up by a brass washer either side of the fibre washers.

The temperature of the fluid stream up the tube was measured with a thermocouple, one wire of which came from a branch at the top bend, in line with the tube and the other from a branch in the bottom bend. Similar glands were provided for these two branches as has been described above. Originally two porcelain beads were threaded onto either side of the junction and held in position by a small knot in the wire. As the beads were greater in diameter than the junction, if any turbulence caused the wire to vibrate, the beads would strike the wall and thus the thermocouple would not be shorted out.

The thermocouple was placed half way up the tube to measure the fluid stream temperature and was found to be reading high. It was concluded that radiation from the tube wall might be causing this. After many trials the central thermocouple was successfully made by threading two 1" lengths of twin hole porcelin spacers on the wire, securing them by threading wire in the second hole. A very thin silver sheath was then wrapped over the spacers and two thin windings of asbestos string were wound over the silver to act as insulation in case the thermocouple touched the tube wall. The silver foil was kept from touching the bead of the thermocouple by fine asbestos string. This also helped to keep an area free so that fluid could circulate past the thermocouple.

The thermocouple wires passed from the fluid stream through glands in line with the tube. The wires then passed over two 3" dia. pulleys turned from bakelite and were connected to the potentiometer switch through a junction box. Considerable difficulty was experienced in maintaining sufficient tension on the thermocouple to prevent the wires touching the tube wall. Eventually this difficulty was overcome by joining the wire ends to two junction boxes sliding on a small section of rod with a light spring between them to maintain tension on the wire.

Wall Thermocouples:

The tube wall thermocouples were originally inserted into the wall in a similar manner to that described by Staker (31). A slot ½" long, 1/16" deep and 0.040" wide was cut into the wall and the thermocouple made of 28 gauge Chromel/Alumel was silver soldered into position. The pair of wires was then varnished. As the winding was put on the tube, depending on the position of the turn nearest the wires, the pair was either brought out between two turns or around a turn. Subsequent work by Staker suggested that the temperature measured by the thermocouples is that of the "Pyrotenax" surface. Consequently, the later tubes had a 1/32" dia. hole drilled 3/32" deep for each thermocouple and the junction was soldered in to locate it only.

Originally thermocouples were placed at 12" intervals up the column starting with the first, 4" from the bottom of the winding. Each thermocouple had a spare 1" above and 1" below in case of insulation breakdown or mechanical failure. After the experiments with the $\frac{1}{2}$" tube it was felt that a better idea of the temperature distribution would be obtained by placing thermocouples at 1" intervals for the bottom 18" of the tube, but using alternate ones only. The spare thermocouples for those further up the tube were then spaced at ± 2" in case the winding should prove to be slightly uneven, thus vitiating the accuracy of any reading.

All wires from the thermocouples were originally run in compensating leads. In the early stages of operation however, many discrepancies in readings were noted which proved difficult to locate. As part of the investigation into these, all compensating leads were replaced by Chromel/Alumel leads and the inaccuracies appeared to be minimized.

All leads from thermocouples were taken to a 24 point thermocouple switch and readings were made on a Pye Universal Precision Potentiometer type 7565, balanced against a cold junction maintained in a bath of ice and water in a thermos flask.

<u>Calibration of Fluid Stream Thermocouples:</u>

The fluid stream inlet and outlet thermocouples were removed from the apparatus and calibrated. With these two all other thermocouples could be checked in situ, and hence any changes which might be caused by soldering thermocouples in position after calibration would be avoided.

It was proposed to calibrate the fluid stream thermocouples in a bath of glycerine. A 4,000 cc beaker containing glycerine was placed in a bed of slag wool held in a two gallon bucket. A small variable speed laboratory stirrer with a glass impeller was used for agitation and a 1 kw immersion heater with power controlled by an auto-transformer ("Variac") was used for heating. Both these were secured by suitable clamps at one side of the bath. The two thermocouples were located either side of an N.P.L. tested thermometer of suitable range by insertion in holes drilled in a rubber bung. The bulb of the thermometer and the two junctions were kept at the same level and immersed to a depth of about 4".

The readings of the potentials of the thermocouples were taken at 30°F intervals. By suitable adjustment on the power variac, the temperature was slowly brought to the required value and held there for 15 minutes. Three sets of readings were then taken at 5 minute intervals to ensure temperature stability and more readings were taken if any differences in values were noted. The temperature of the stem of the standard thermometer which was not immersed was measured with a non-standard thermometer and the correction was calculated to allow for the incomplete immersion of the thread. When the top of the range for each thermometer had been reached, the bath was allowed to cool down a few degrees. The thermometer was then replaced by the next highest in the range and the temperature brought slowly up to a value which was approximately the same as the highest reading on the displaced thermometer. The procedure for readings was then repeated for the higher range. The bath was then allowed to cool and the procedure was repeated in reverse. Two curves were then plotted, one for each thermocouple, of microvolts against temperature.

It was intended to calibrate the thermocouples in the glycerine bath to a temperature of 420°F which is below the boiling point of glycerine, 554°F (63). However, the fume from the bath made readings difficult and uncomfortable at temperatures above 320°F, and the calibrations with glycerine were discontinued. Investigations were then made into available heat transfer liquids. A suitable material would be non-corrosive, non-toxic, non-inflammable, stable, fluid at normal temperatures, but with a high boiling point, preferably over 600°F. Aroclor 1262 was found to be the best suitable liquid.

Aroclors are chlorinated diphenyls and poly phenyls manufactured and marketed by Monsanto Chemicals (Aust.) Ltd. Aroclor 1262 has a distillation range of 400°- 430°C (750°- 810°F) but has the disadvantage of a high pour point of 37°C (98°F), (64). Hence the glycerine bath was replaced by an Aroclor bath and the readings continual for the required range. After calibration the couples were washed in kerosene, rinsed in trichlorethylene, and finally with water before being inserted into position.

Later when the tube wall thermocouples were being calibrated it was noticed that the reagings on the inlet and outlet thermocouples were diverging rapidly. These thermocouples were removed and examined, when it was noted that the weld had nearly failed on the inlet thermocouple. In case this was a corrosion effect of the Aroclor, two new thermocouples were made and calibrated. For the lower temperatures a bath of water was used and for the higher temperatures a glycerine bath was used. When readings were difficult to make because of fume, the Aroclor bath was used. The time of immersion was kept to a minimum by maintaining the bath steady using a non-standard thermometer. The thermocouple attached to the standard thermometer was then inserted only long enough to ensure that an accurate reading was obtained.

Calibration of the Tube Wall Thermocouples:

The tube was connected to a small boiler made by inserting a 2kw heater in a 2' length of 6" dia. steam pipe. The heater was a normal immersion heater type but was stripped down and cleaned, and then reassembled with silver soldered connections. The top of the tube was connected to a T-piece with a valve and a 0 - 300 psi pressure gauge. The tube and boiler were well lagged with 85% magnesia sections and asbestos mats.

The boiler was filled with enough demineralized water to ensure that the element was adequately covered and a 2 kw Variac was connected to the terminals. When all the air had been driven from the tube, the top valve was closed and the power to the immersion heater was regulated to ensure a steady pressure. The temperature recorded by the standardized feed inlet thermocouple was kept steady for 15 minutes to ensure that the tube had reached thermal stability. The inlet and outlet temperatures were read, the wall temperatures from top to bottom were then read, and finally the inlet and outlet temperatures were again read to ensure that the fluid temperatures had remained constant. This procedure was carried out three times for all readings, which were taken at 30 F° intervals.

A graph of the deviation of each thermocouple was plotted against the readings obtained on the inlet thermocouple and this enabled accurate and quick readings of temperature to be obtained with only one large scale calibration chart.

The Power Control Apparatus:

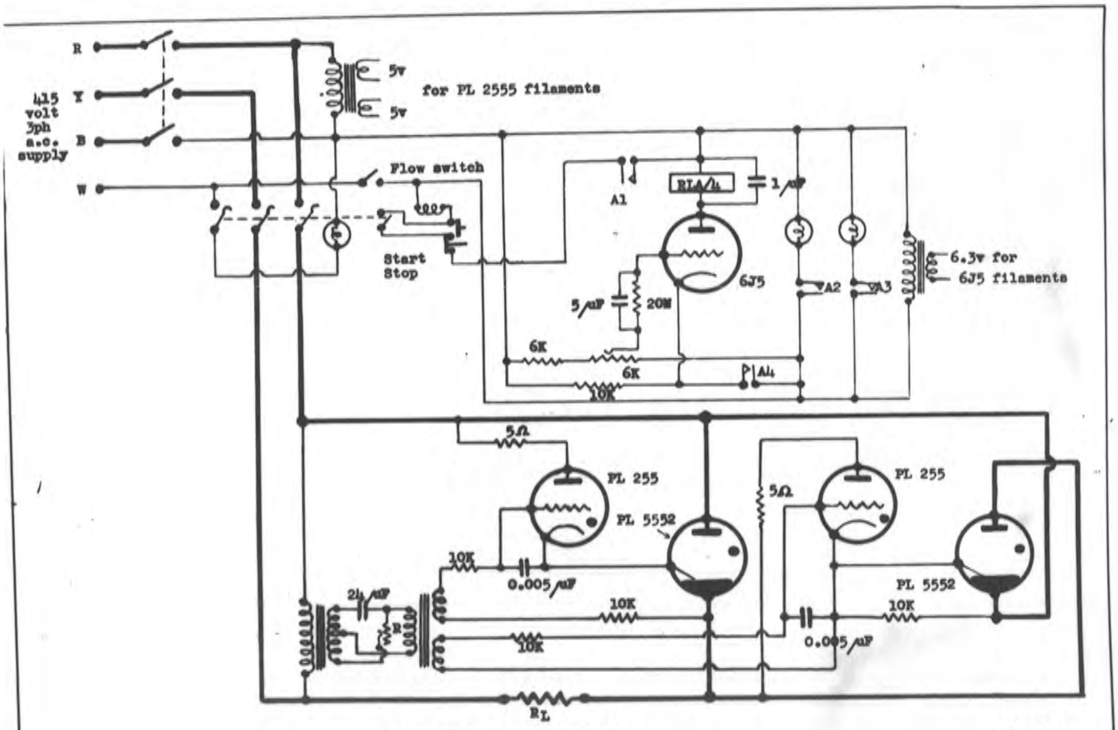
With the heating tube operating under a heat flux of 100,000 Btu/(sq ft)(hr) it was calculated that a power control unit of 30 kw capacity would be required. Conversion of an a.c. to d.c. and subsequent control by a rheostat was considered but preliminary investigations suggested that the equipment costs would be high. Use of a very large auto-transformer was also considered but the largest type available with a reasonable delivery date had a capacity of only 14kw.

The use of grid controlled mercury vapour rectifiers or thyratrons was next considered. Large powers can be controlled by them using relatively small voltages applied to a grid. They have the advantage that when operated on a.c. they are extinguished every half cycle. Firing is effected by the application of a suitable voltage, the magnitude or phase of which can be altered to change the load.

The desired power could have been achieved by using two of the larger thyratrons in anti-parallel but it would have meant that the applied voltage would have had to be of the order of several thousand volts. This approach had two main disadvantages, namely the personal danger in the event of insulation breakdown, and the insulation problems associated with the higher voltages. Also the transformer required would have been expensive.

It was then decided to control the power by using ignitrons which would have to be fired by thyratrons. This system would then operate on 415 volts a.c. While this would not eliminate all personal danger, because any live fault would give 240 volts to earth, it would alleviate the insulation problems and leave the apparatus no more potentially dangerous than most laboratory and domestic equipment. The circuit diagram is shown in Figure 3.

It will be noticed that simple anode ignition of the ignitrons (PL 5552) by the thyratrons (PL255) has been used. It has been reported (65) that too low an anode voltage in the early part of the cycle is likely to give unreliable operation, but this was found



CIRCUIT DETAILS OF POWER CONTROLLER

Figure 3

not to be serious in this case. The arrangement had the advantage that an additional d.c. power supply was not needed to supply the triggering pulse. Thyratrons larger than necessary were used to trigger the ignitrons so that if needed they could be used by themselves to control lower power levels.

The circuit design was straight forward but it was noted that the suggested minimum voltage required to fire the ignitrons is on the conservative side. It is reported (65) that the ignitron ignitor requires 200 volts to fire. An analysis of the system showed that control was being exerted down to 20° which implies that the instantaneous ignitor voltage was 140. The resistance of 5 Ohms in the plate circuit of the igniting thyratrons was selected to limit the ignition current as the ignitron fires. As soon as the ignitron fires the thyratron is extinguished, because its effective plate voltage is equal to the ignitron arc drop. The cooling water for the ignitrons was supplied from the mains. A flow switch was interconnected with the main circuit breaker to guard against water failure. To allow the filaments of the thyratrons to warm before a load is applied, a simple R - C timing circuit was incorporated in the starting circuit to prevent the circuit being closed before the required 5 minutes.

The power delivered to the load can be written in the following form:

$$P = \frac{1}{\pi R_L} \int_{\theta}^{\theta_{max}} (E_{pk} . \sin\theta - 10)^2 . d\theta \qquad (1)$$

where 10 = arc volts drop across the ignitron (measured)

P = power in watts

R_L = load resistance in ohms

E_{pk} = peak voltage of a.c. supply.

An approximate analysis for the phase shift circuit gives

$$\tan\theta = \frac{-RX_c (X_c - 2X_L)}{X_L X_c^{\ 2} + R^2(X_c - X_L)} \qquad (2)$$

where R is the variable resistance in the phase shaft circuit.

With C = 24 μF X_c at 50 c/s = 133 ohms.

and L = 0.53 H X_L at 50 c/s = 166 ohms.

thus $$\tan\theta = \frac{199R}{22078 - 0.248R^2} \qquad (2A)$$

Equation (2A) evaluated gives a set of values as tabulated below:-

R ohms	Tan θ	θ
10	0.09	5°
20	0.18	10°
50	0.46	24°
100	1.02	45°
200	3.28	73°
500	-2.49	112°
1000	-0.88	138°
1500	-0.56	151°
2000	-0.41	158°
2225	-0.37	160°

These values have been plotted on Figure 4. Wave forms of current through a purely resistive load are shown in Figure 5. Measurements of the ignitron delay have been determined from a series of photographs similar to those shown in Figure 5, and these results are also plotted in Figure 4. Close agreement is noted between predicted and actual phase shift, despite the simplification that the thyratron grid loading makes no contribution to the calculated phase shift.

Using the values obtained from Figure 4, and substituting them in equation (1), values for power can be plotted against values for the phase shift resistance. This has been done and is shown in Figure 6.

The power to the load was measured by an Elliott dynamometer type wattmeter which has an accuracy of 0.25% of full scale as laid

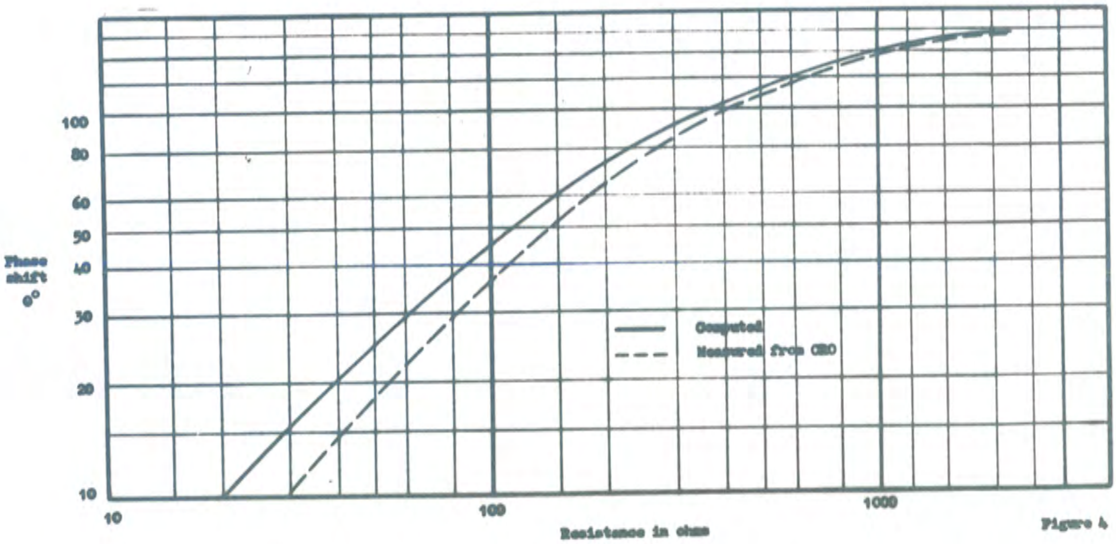
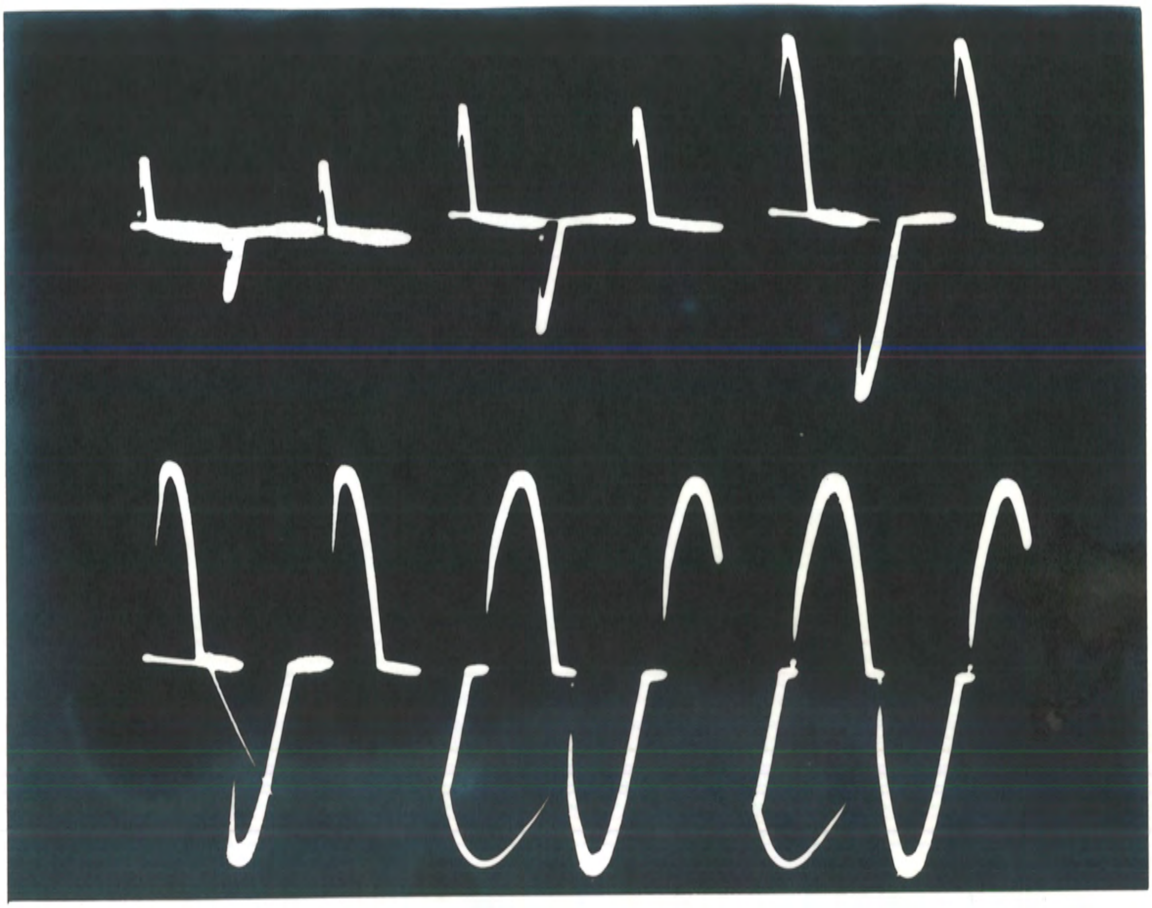

100
80
60
50
40
30
20
10
Phase shift θ°
Computed
Measured from CRO
10
100
1000
Resistance in ohms

Figure 4

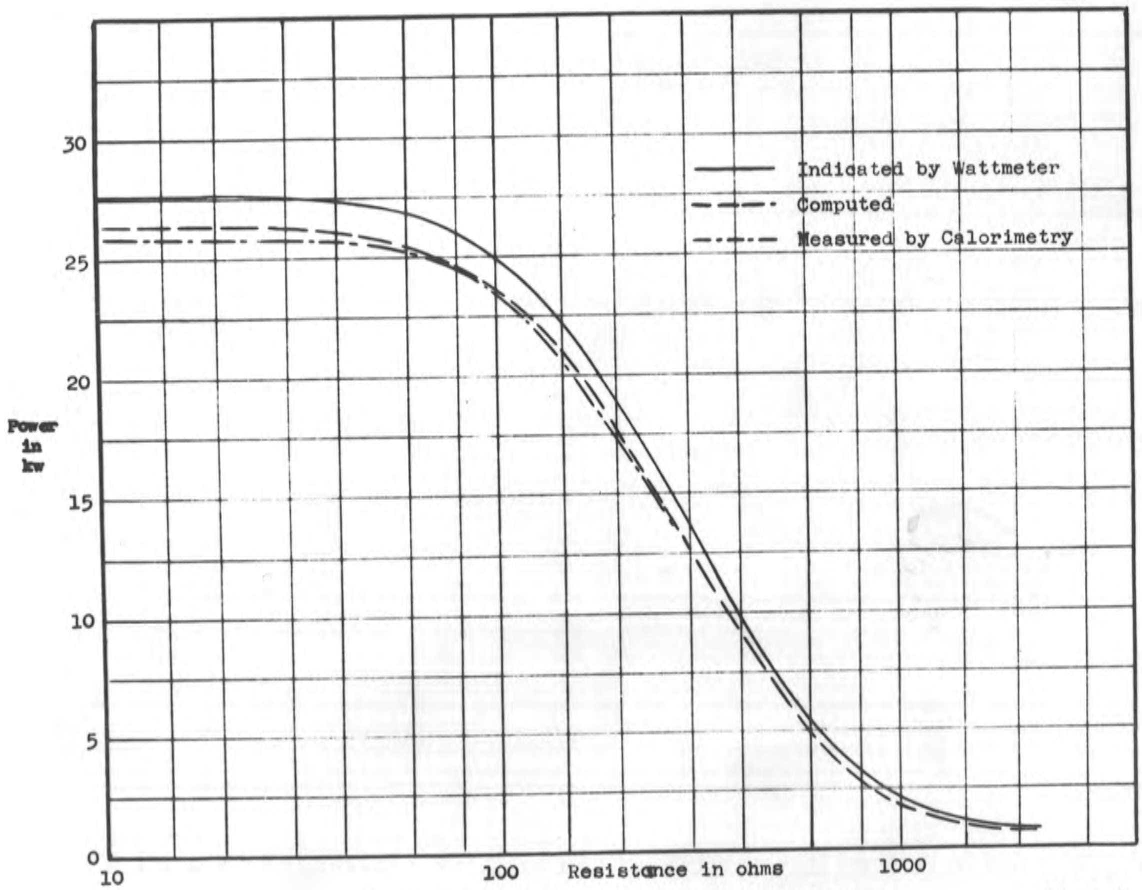


WAVE FORMS THROUGH A RESISTIVE LOAD

a) R = 2230 ohms. b) R = 600 ohms. c) R = 400 ohms.

d) R = 300 ohms. e) R = 80 ohms. f) R = 50 ohms.

Figure 5



CONTROL CHARACTERISTIC OF IGNITRON-THYRATRON CIRCUIT

Figure 6

down by BSS No.89. Because of the high frequency components which would be present in the output waveform at part load it was felt that a more accurate reading would be obtained by connecting the potential leads of the wattmeter to the supply side of the ignitrons. The power reading of the wattmeter is then given by:

$$P = \frac{1}{\pi R_L} \int_{0}^{\theta_{max}} (E_{pk} \sin \theta)^2 \cdot d\theta \qquad (3)$$

while the power dissipated by the load is given by equation (1).

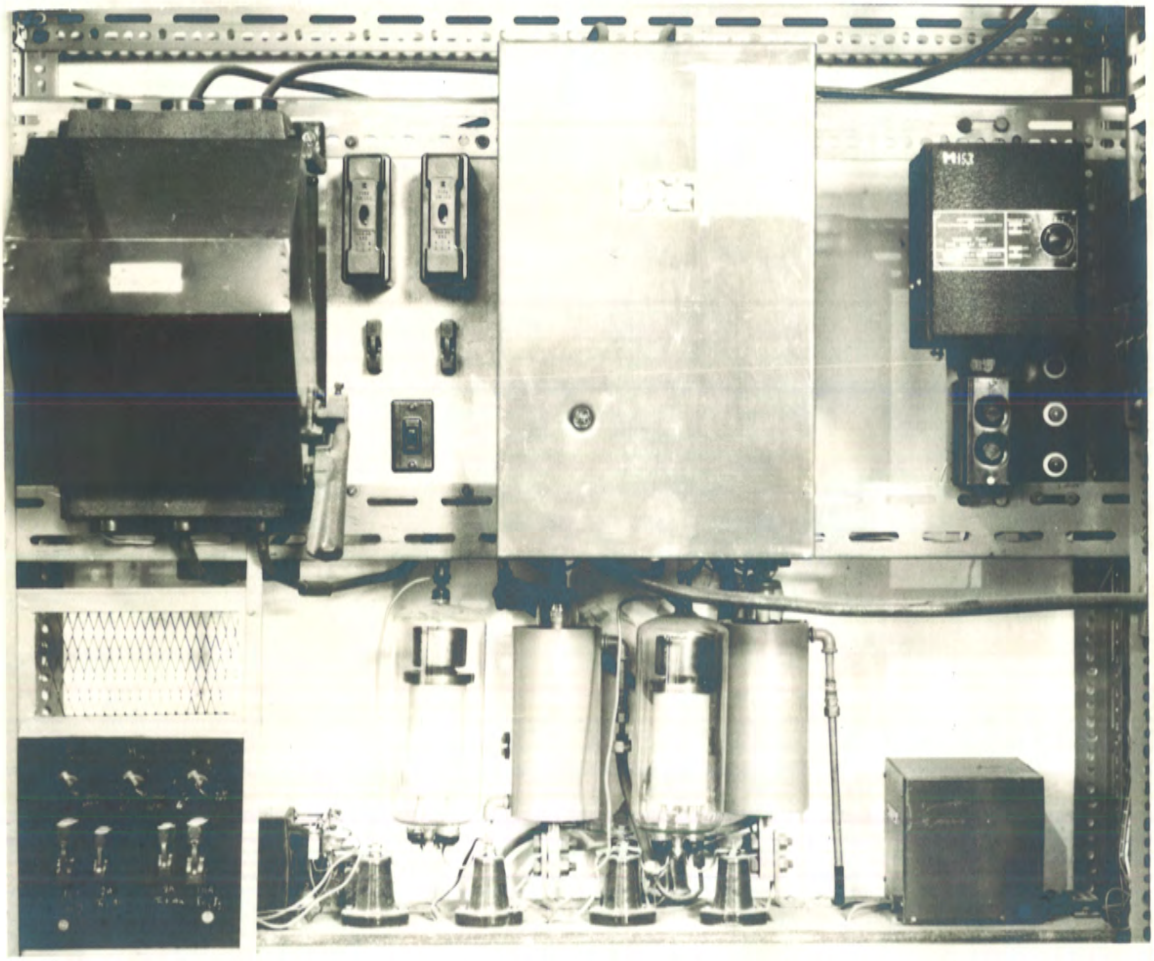
To check these values, an arbitrarily selected water flow rate was sent through the tube and heated by known power loads. The temperature rise of the water was carefully measured and the heat expressed as power. The results of this series of experiments were plotted and are shown in Figure 6. As can be seen, good agreement was found.

In computing the power from equation (1) it was necessary to measure the resistance of the load accurately. This was done by passing a small current through the tube windings and a standard resistance connected in series, and measuring the voltage drop across each with a potentiometer.

The form of the control characteristic is not simple as can be seen, but R in the phase shift circuit was made of three variable resistances, 0-2000 ohms, 0-200 ohms and 0-25 ohms, connected in series. This enabled manual control of the unit to be most satisfactorily and accurately maintained. No attention was needed for the regulation of the load power. The main supply voltage to the laboratory is quite well regulated and small fluctuations were absorbed by the "thermal inertia" of the heating coil and appeared to be of no consequence.

From the manufacturer's data (65) and assuming that a limiting average current of 140 amps governs, then the maximum power which would be obtained from this unit would be 400 kilowatts. Thus it will be seen that the equipment is being run well under full load, but the next smaller ignitron would not have been adequate. Further details of the circuit have been published separately (66).

A photograph of the control apparatus is shown in Figure 7.



PHOTOGRAPH OF THE POWER CONTROL APPARATUS

Figure 7

CHAPTER 4.

EXPERIMENTAL WORK.

Materials:

The liquids used during the experiments were water, methanol, ethanol, chloroform and iso-propanol. The water was obtained from a laboratory still. The methanol was commercial grade containing 98.5% MeOH, with a purity specified by BS.506 (1950). The ethanol was obtained from the Colonial Sugar Refining Co. Ltd., and had a purity of 99.8% with 0.2% water and a trace of benzene as impurities. The chloroform was imported by H. York and Co. Pty., from Bayer. It had a purity of 99.6% with water as the only impurity. The iso-propanol was obtained from Shell Chemical (Australia) Pty. Ltd., and had a purity of 99.9%. All these liquids were assumed to be pure when their physical properties were evaluated.

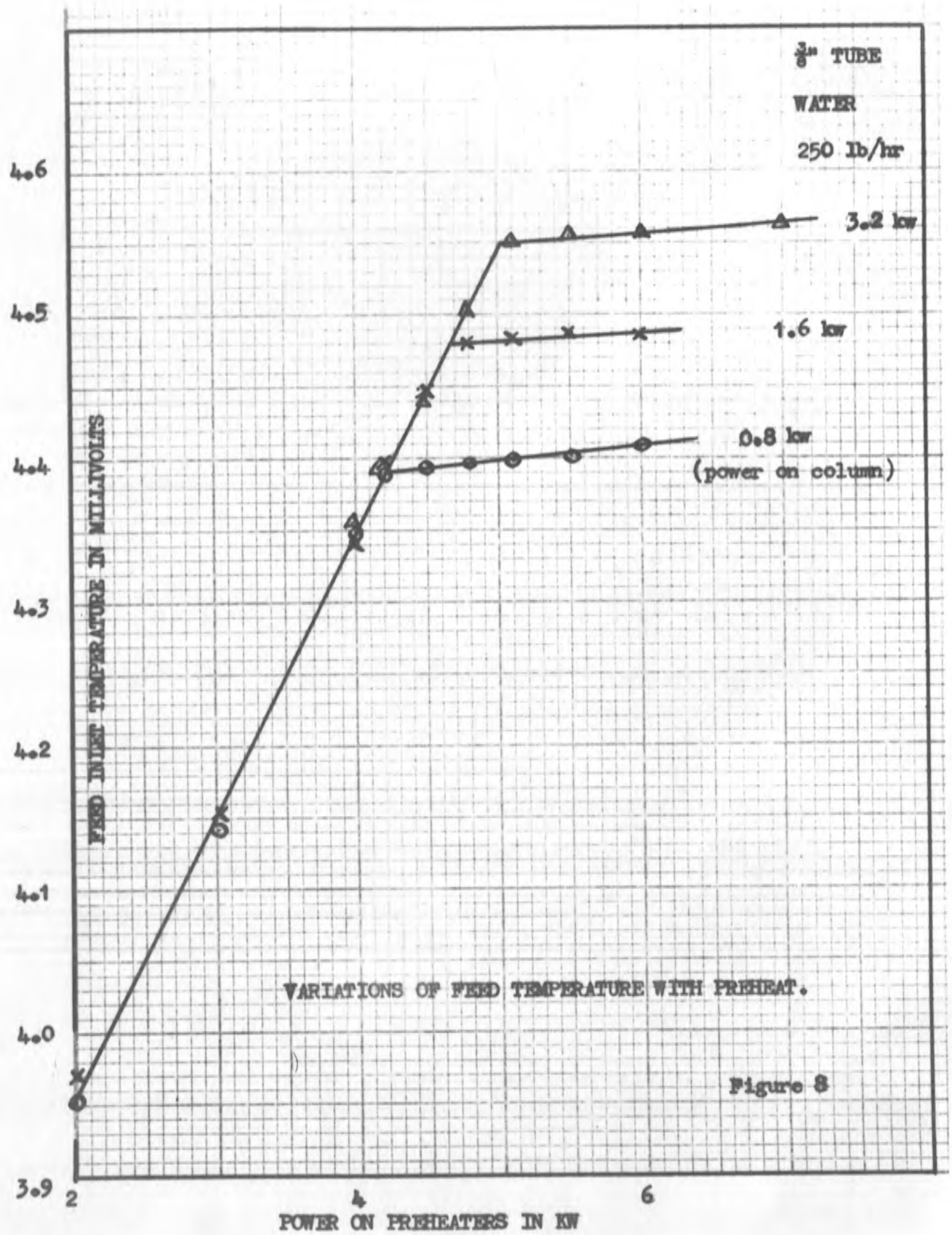
When the apparatus was being changed from one liquid to another the tube was examined to ensure that no change had taken place on the surface. The apparatus was drained and then filled with the new solvent. This would be circulated around with heat on the tube and preheaters for an hour before being drained and discarded. This procedure would then be repeated for three hours and the density of the discarded material checked. This was always found to be the same as the fresh liquid, and the apparatus was assumed to be free of the old material. The apparatus was then filled and the orifice or rotameter calibrated as described in Chapter 3.

Experimental Procedure:

A long series of experiments was made to find a satisfactory technique for predicting when the feed was just boiling at the bottom of the tube, and the following method was evolved. The feed was recycled at the required rate through the apparatus with the required power on the tube and all the preheat switched on. If the run was the first of the day, the apparatus was allowed to warm up for an hour. The preheat was then reduced to, say, 1 kw and the temperature of the feed allowed to come to a steady value. The temperature was read on the travelling thermocouple which had been brought to a position level with the bottom of the winding. The preheat was then raised to 2 kw, and the procedure repeated. This was repeated until it was clear that some vapour was being fed to the tube. If necessary, the preheat power was raised by fractions of a kilowatt. A graph was then plotted of inlet temperature in millivolts against preheat power and a typical one is shown in Figure 8. It will be noted that the graph consists of two straight lines of different slope. The point of interception gives the preheat power for just boiling feed under the set conditions of feed rate and tube power. The line of greater slope represents the increase in temperature as the subcooled liquid approaches the boiling point. The line of lesser slope represents the increase in temperature as the pressure at the bottom of the tube increases. This pressure increases because the quantity of vapour produced in the preheaters adds to the pressure drop which is prevailing in the tube.

It was found that the preheat, when set, proved very stable throughout any run. The feed inlet thermocouple was used at frequent intervals during runs to check that the feed was still at the boiling point. On the few occasions that any deviation was noted, the run was abandoned.

It was considered that the two hours which were occupied in warming the apparatus and finding the feed temperature at the start of each day were sufficient to ensure that any excess dissolved gases were driven from the liquid and the system. The temperature



VARIATIONS OF FEED TEMPERATURE WITH PREHEAT.

Figure 8

of the liquid in the surge tank was such that a slight efflux of vapour prevented any atmospheric air from re-entering the system.

When the tube feed, tube power, condenser cooling water, and feed inlet temperature had all been stable for 15 minutes, the following data were collected:

Tube power

Feed rate

Feed inlet temperature

Tube wall temperatures

Temperatures along the tube measured at 6" intervals

Ambient temperature.

All readings were taken twice except those of column power and feed inlet temperature which were kept constantly under observation. A typical set of results is given in Appendix 1 and the method of calculation is shown.

<u>Experimental procedure to find the effect of tube length:</u>

It has been claimed by several authors (as reported in Chapter 2), that the length of the tube affects the heat transfer coefficient. A short series of experiments was carried out to determine whether this was so or not. It was argued that if extra vapour be fed in at the bottom of the tube then the way in which the heat transfer coefficient varied up the tube would be an indication of the effect of tube length. It was decided that the vapour should be produced by the preheaters.

The experimental procedure was as follows. The preheat was set as described in the previous section and the apparatus allowed to come to steady conditions for 10 minutes. Then an extra 2 kw of preheat were switched on. As soon as conditions had again steadied, readings were taken as before. This was repeated for further increments of preheat power. In two cases, readings were taken with 2 kw of preheat power switched off, in addition to the readings taken when the power was increased.

Experimental procedure to find the film thickness:

A probe was made of a hypodermic needle 0.025" I.D. mounted on a vernier, and projected into the tube through a small hole immediately above the top winding. The needle passed through a small rubber pad which was held in position over the hole by the clamp holding the vernier. This made an effective gland. The needle was advanced until it met the opposite wall and a zero mark made.

Samples were withdrawn from the tube at various points across the tube along a radius, by means of a light vacuum. Any vapour withdrawn or created by the vacuum was condensed by three water cooled Liebig condensers. Each sample was taken over 10-15 minutes to ensure that the flow pattern inside the tube was not disturbed. About 1 cc was collected each time and it was found that 3 - 4 samples were needed for each position to ensure that conditions were steady, not only in the tube but in the sampling apparatus as well. At the conclusion of a series of readings taken across the tube the first (wall) reading was again taken to ensure that conditions had not changed during the run. On the few occasions when a change was found the run was repeated.

Sodium sulphate solution was added to the circulating water, and sodium acetate to the ethanol or methanol, so that a concentration of approximately 50 ppm was obtained. The samples taken through the probe were analysed on a Flame Photometer model A supplied by Evans Electroselenium Ltd. The wall reading was assumed to be entirely liquid and that value was therefore used as the arbitrary full scale reading on the flame photometer. The calibration curve was obtained by successive dilution of a measured quantity of the wall sample with the appropriate solvent introduced from a burette. The lesser concentrations obtained across the tube were assumed to represent the proportion of liquid to liquid plus vapour at these points.

A typical set of results and a type calculation are shown in Appendix 2.

Presentation of results:

Runs were done as tabulated below:

Tube size inch	Material	Feed rate lb/hr	Heat fluxes Btu/(hr,sq ft, F°)
$\frac{3}{8}$	Water	150	6800, 16150, 25100, 33700, 42300.
		250	6800, 11400, 25100, 33700, 42300.
		400	6800, 16150, 29100, 38000, 42300.
$\frac{1}{2}$	Water	75	5740, 13400, 20100, 33600.
		150	2020, 13400, 33600, 53000, 68500.
		250	2020, 13400, 42200, 68500, 86500.
		400	2020, 13400, 33600, 50800, 68,500.
		600	2020, 5740, 13400, 20100.
		750	5740, 13400, 20100.

Tube size inch	Material	Feed rate lb/hr	Heat fluxes Btu/(hr,sq ft, F°)
(cont.)			
$\frac{3}{4}$	Water	150	3270, 7610, 20400, 41800, 52800, 63500, 74500.
		250	3270, 7610, 11950, 20400, 31100, 41800, 52800, 63500, 74500, 85800.
		400	3270, 7610, 11950, 16200, 20400, 31100, 41800, 52800, 63500, 74500.
1	Water	150	2450, 5800, 8950, 15300, 23300, 39600, 55200.
		250	8950, 15300, 23300, 31400, 39600, 55200
		400	2450, 8950, 15300, 23300, 31400, 55200.
$\frac{3}{4}$	Methanol	250	3270, 11950, 20400, 31100, 41800, 52800.
$\frac{3}{4}$	Ethanol	250	7610, 11950, 20400, 25900, 31100, 41800.
		400	7610, 20400.
$\frac{3}{4}$	Chloroform	400	3270, 7610, 11950, 20400, 25900.
$\frac{3}{4}$	Iso-propanol	400	3270, 7610, 11950, 20400, 31100, 41800

The readings taken during these runs are shown in Appendix 3.

Some of the results of the runs at the low heat fluxes were not considered because the wall temperatures and the fluid stream temperatures were fluctuating. This seemed to be caused by pressure flustuations, no doubt the result of slugging action inside the tube. Some of the runs at high heat fluxes were not made with the lower flow rates because of the risk of burn-out.

A complete series of runs covering all feed rates and heat fluxes was attempted with the $\frac{1}{2}$" tube. Burn-out was experienced with the 75 lb/hr feed at a heat flux of 42200 Btu/hr,sq ft, and the tube had to be rebuilt. When a feed rate of 600 lb/hr was attempted at the higher heat fluxes it was found that the indication of just boiling feed conditions could not be found by the method indicated in Chapter 4 and shown in Figure 8. The two straight lines which had hitherto met at a well defined point now merged into each other by means of a gradual curve. It seemed that the pressure drop in the system was being increased by some other factor, such as a sudden increase in entrainment, and the investigation into this phenomenon was deferred. The 400 lb/hr feed rate was found to be the highest which allowed the full range of heat fluxes to be investigated and the 150 lb/hr feed rate was found to be the lowest which could be used so that the tube would not burn out should a sudden fluctuation occur.

It was decided that the heat transfer coefficient, h, calculated as shown in Appendix 1, should be plotted against the vapour velocity. Some authors had plotted h against $\frac{4W_L}{\pi D \mu_L}$ which is a Reynolds number. If this had been done with these results then variations of h would have been plotted against a constant for each feed rate, and it was clear that h was varying up the tube. If, however, h be plotted against vapour velocity then a zero value would apparently prevail at the bottom of the tube. It was felt that a more realistic graph would be obtained if a bulk velocity be used. Values of this were found for each point by the following method:-

1) calculating the total heat input rate to the fluid up to that point,
2) finding the mass of liquid evaporated per unit time,
3) calculating the volume of vapour corresponding to that mass, assuming that the vapour and liquid are in equilibrium at that temperature,
4) calculating the volume rate of the remaining liquid,
5) summing the volume rates of vapour and liquid,
6) dividing by the tube cross sectional area.

Justification for using this velocity may be summed up as follows. In the lower part of the tube the fluid consists of a mass of bubbles and liquid, merging into a region where slugs of liquid and vapour alternate. In these regions it is probable that, unless nucleate boiling predominates, the heat transfer rate is proportional to the velocity of the liquid, and this velocity will probably increase as the proportion of vapour increases. Further up the tube where the climbing film action starts, the volume of vapour is so much greater than the volume of the liquid that the velocity is in effect the vapour velocity within the accuracy of the work. This bulk velocity has been denoted by the symbol v_B. Calculated values of v_B are also shown in Appendix 3.

Graphs of heat transfer coefficient against bulk velocity are shown in Figures 9 to 28. Values of h are in Btu/(hr)(sq ft)(F°); v_B in ft/sec. The points on each line represent arbitrarily chosen positions up the tube generally at 10" increments, as detailed in Appendix 3.

The results of the experiments to find the effect of tube length on the heat transfer coefficient are given in Appendix 4, and the graphs of heat transfer coefficient against bulk velocity are shown in Figures 29 to 32.

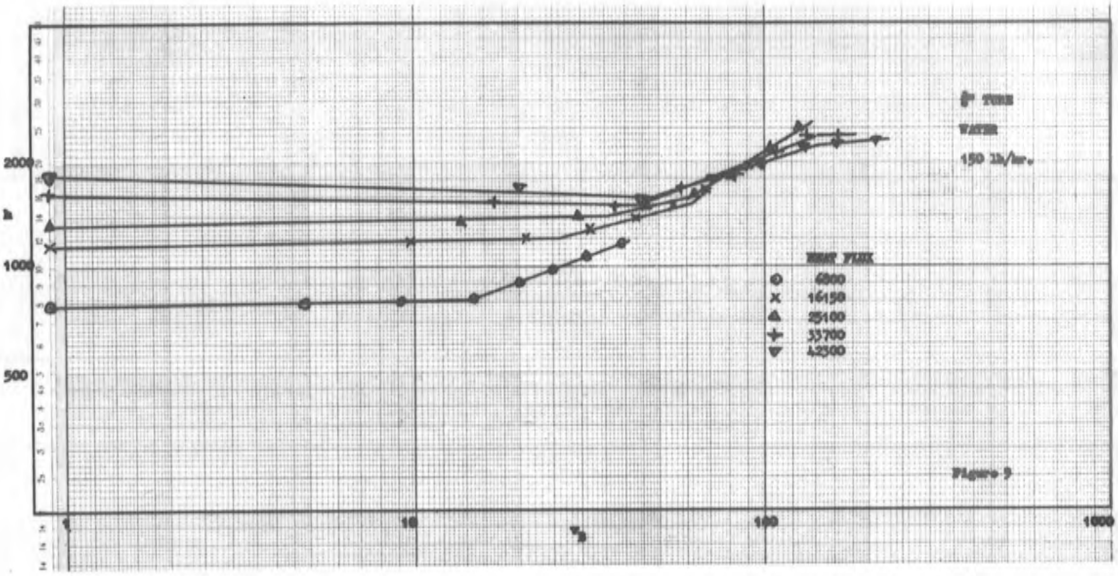


Figure 9

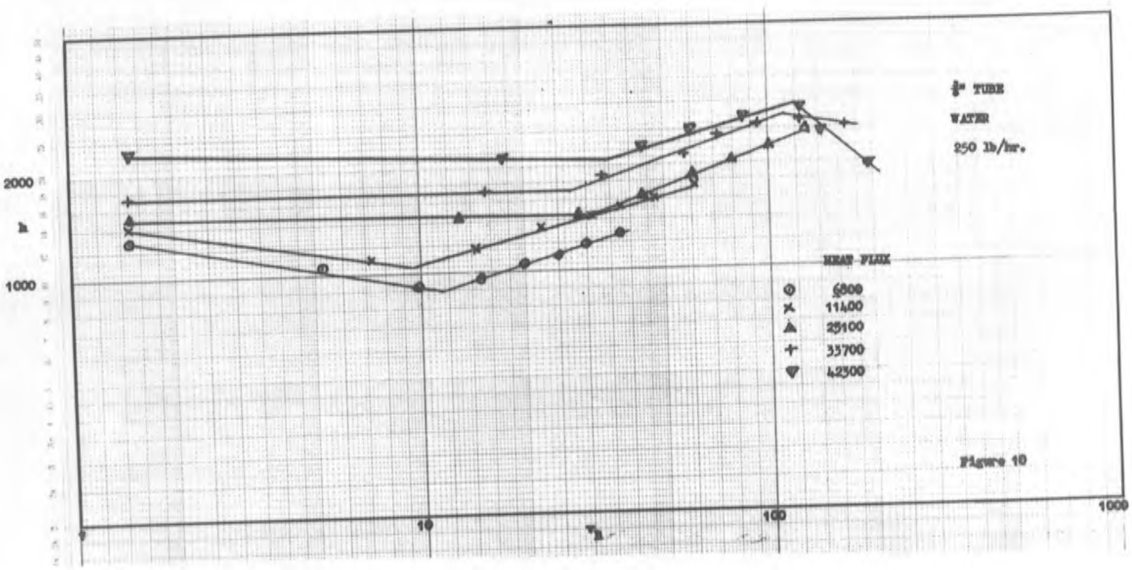


Figure 10

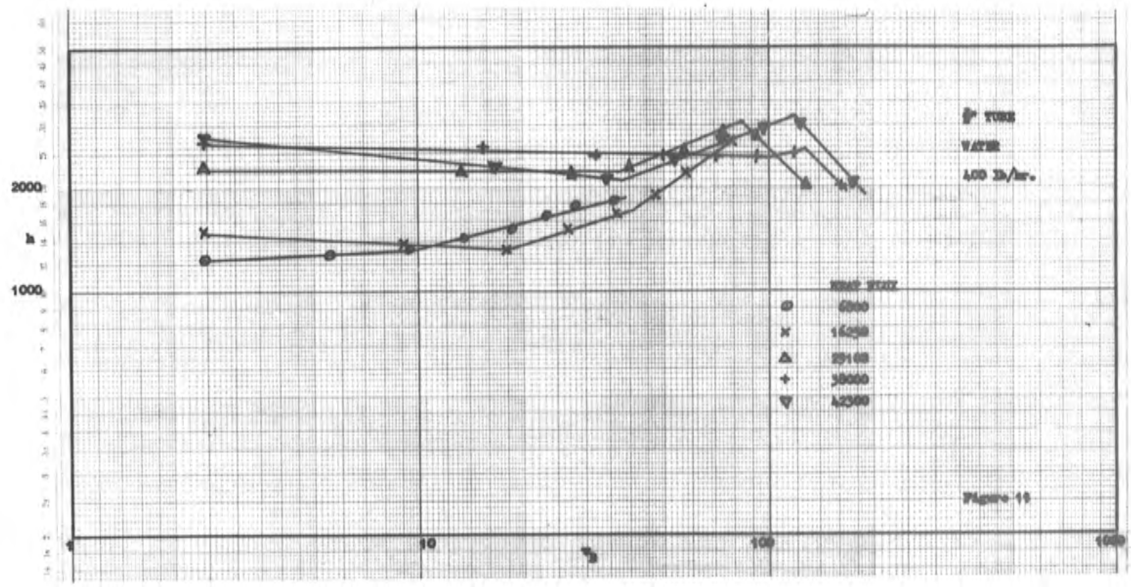


Figure 11

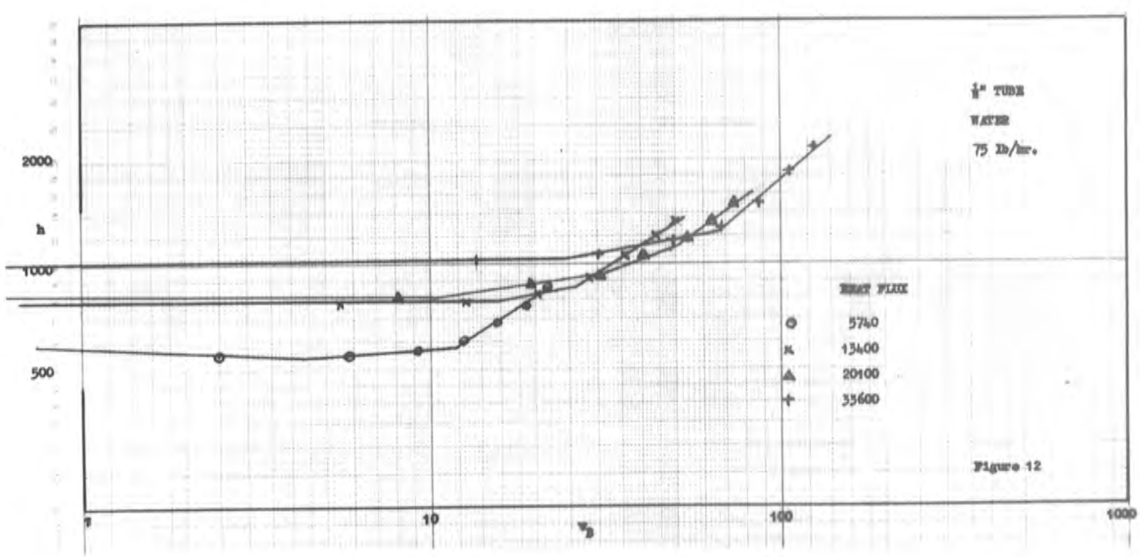


Figure 12

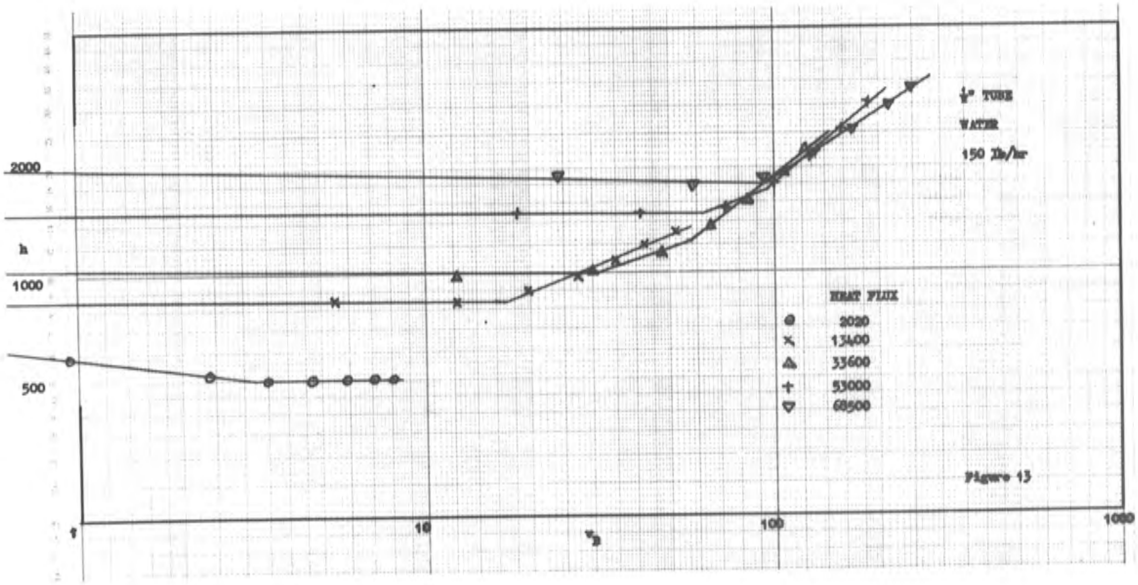


Figure 13

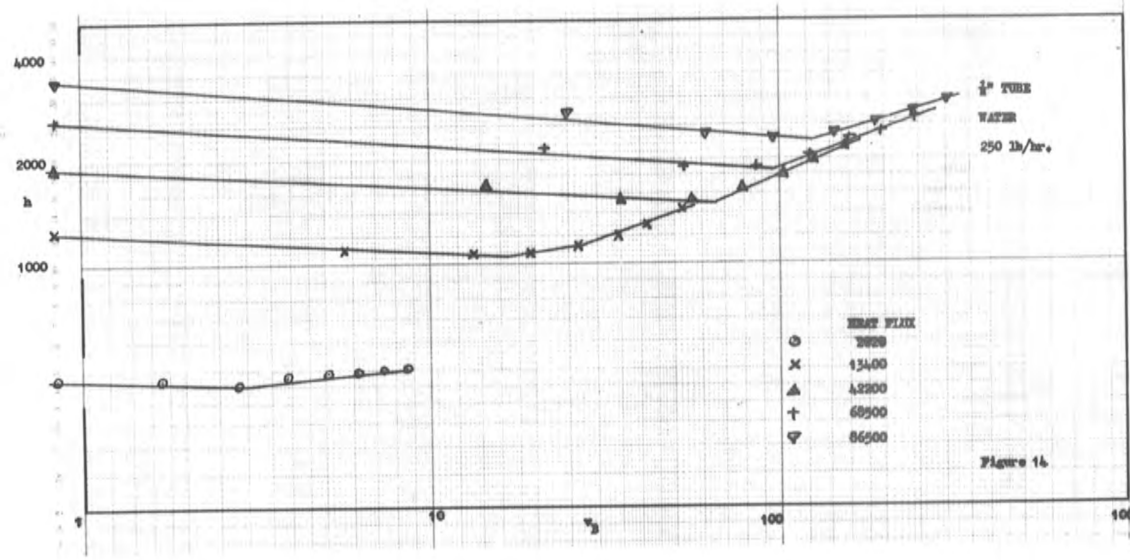


Figure 14

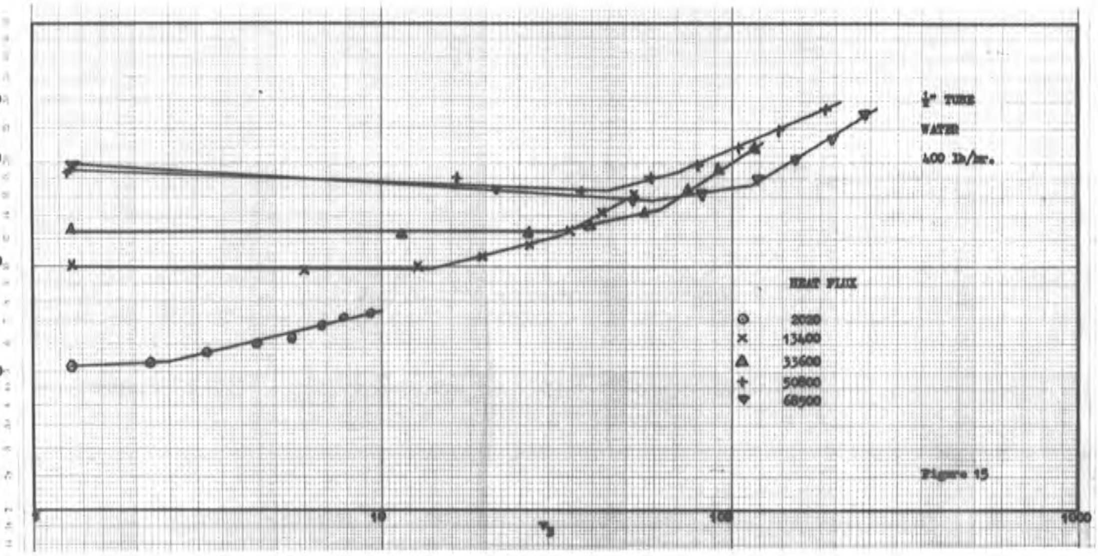


Figure 15

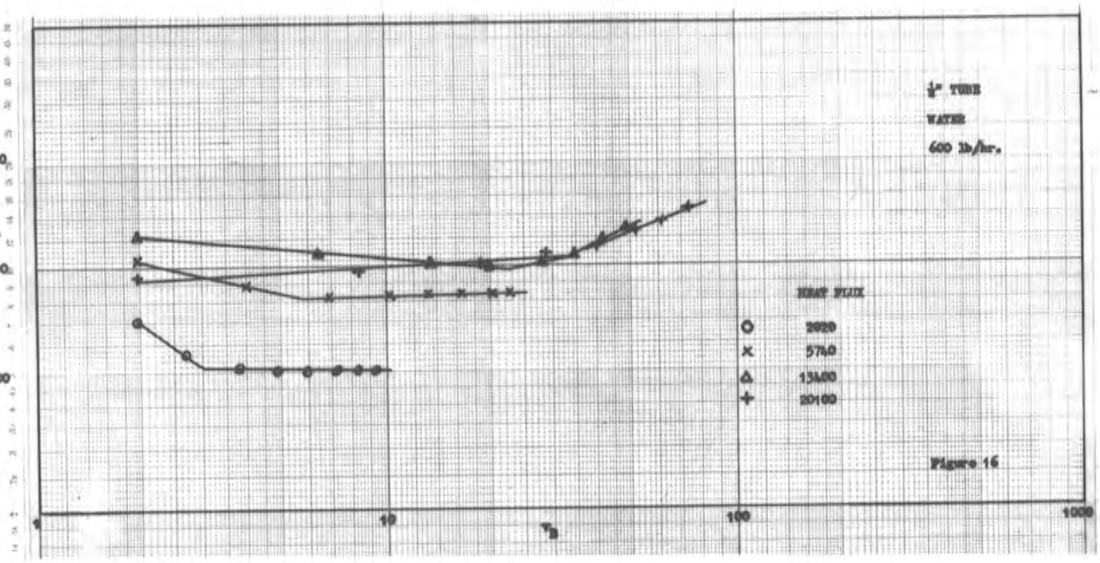


Figure 16

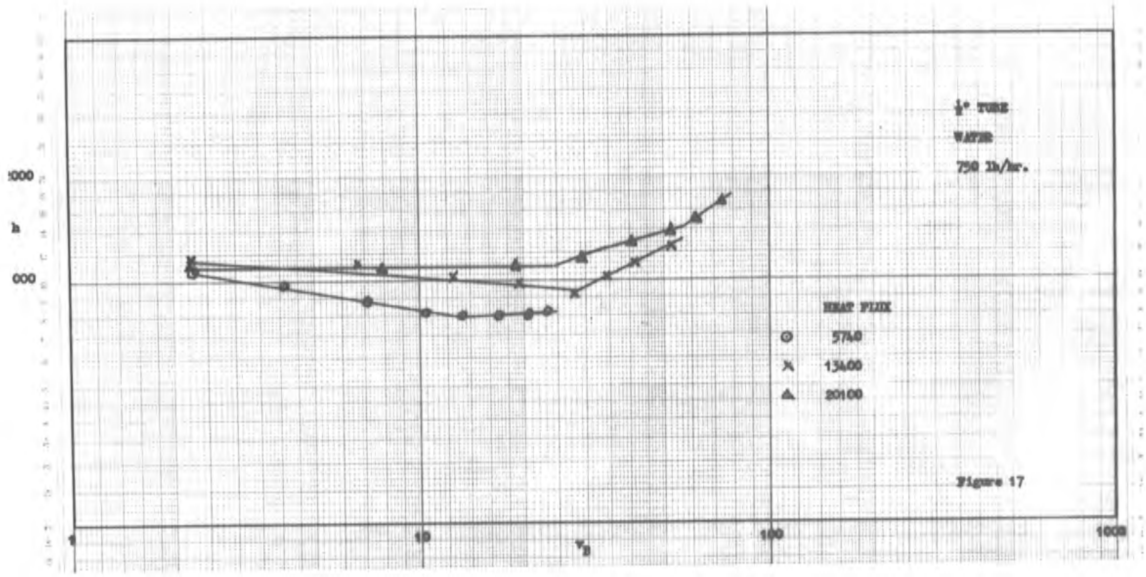


Figure 17

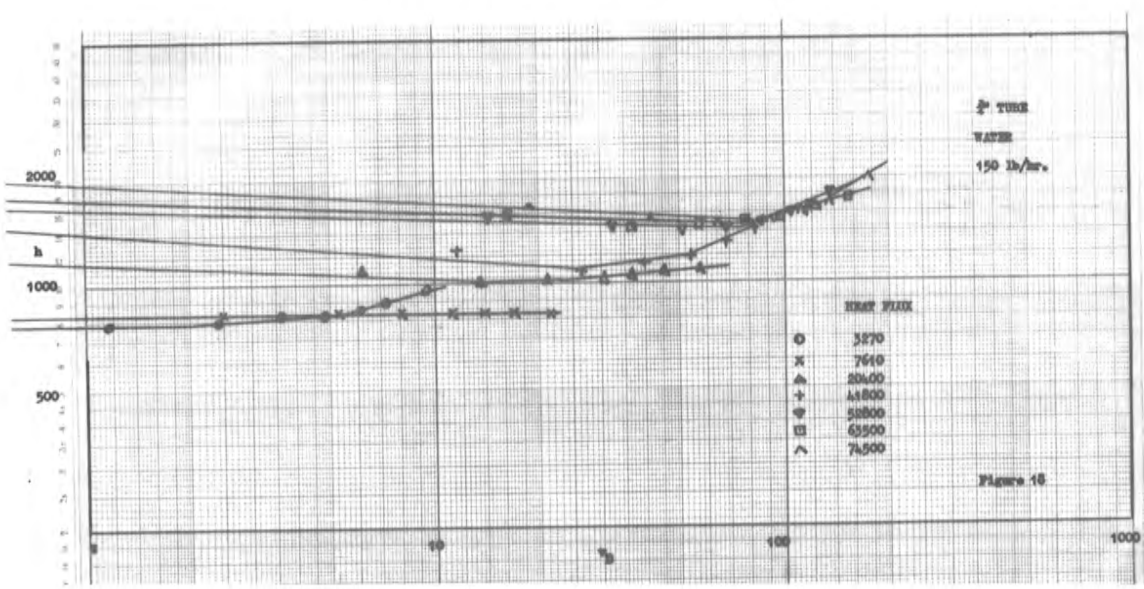


Figure 18

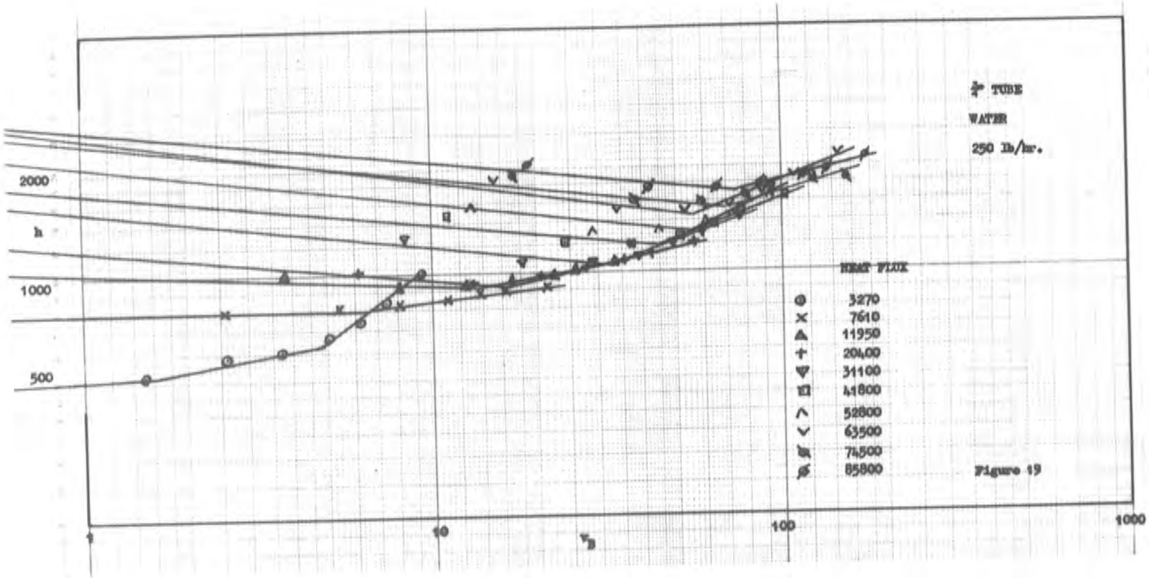


Figure 19

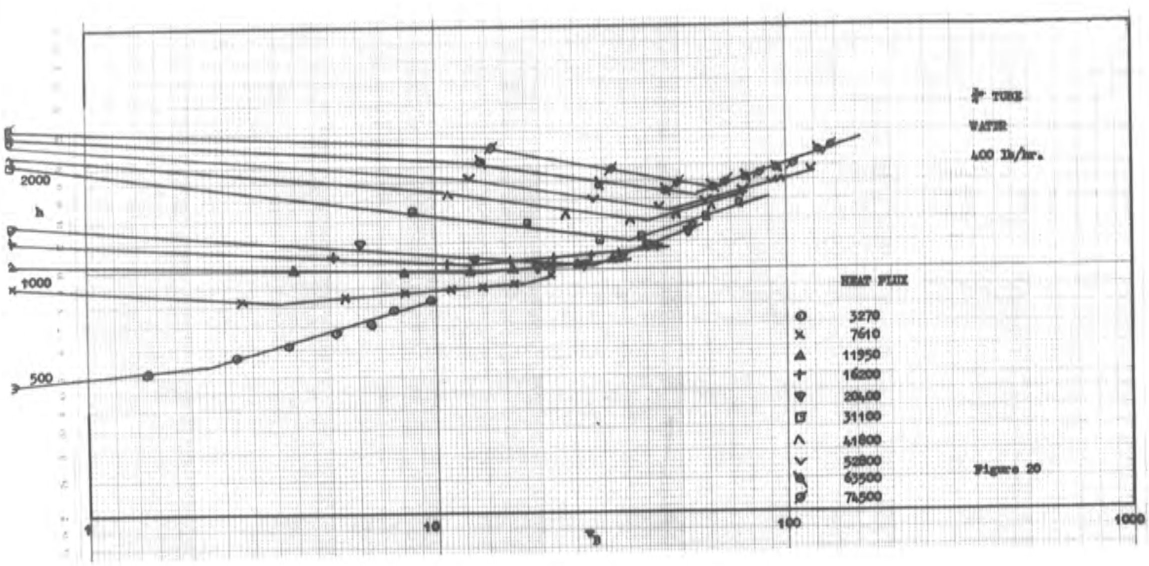


Figure 20

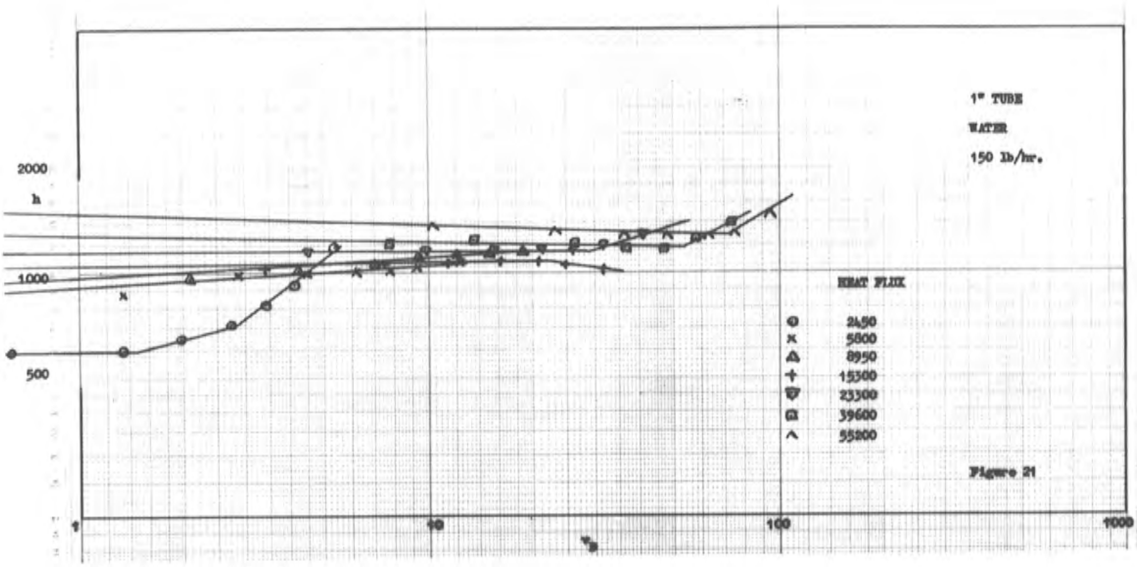


Figure 21

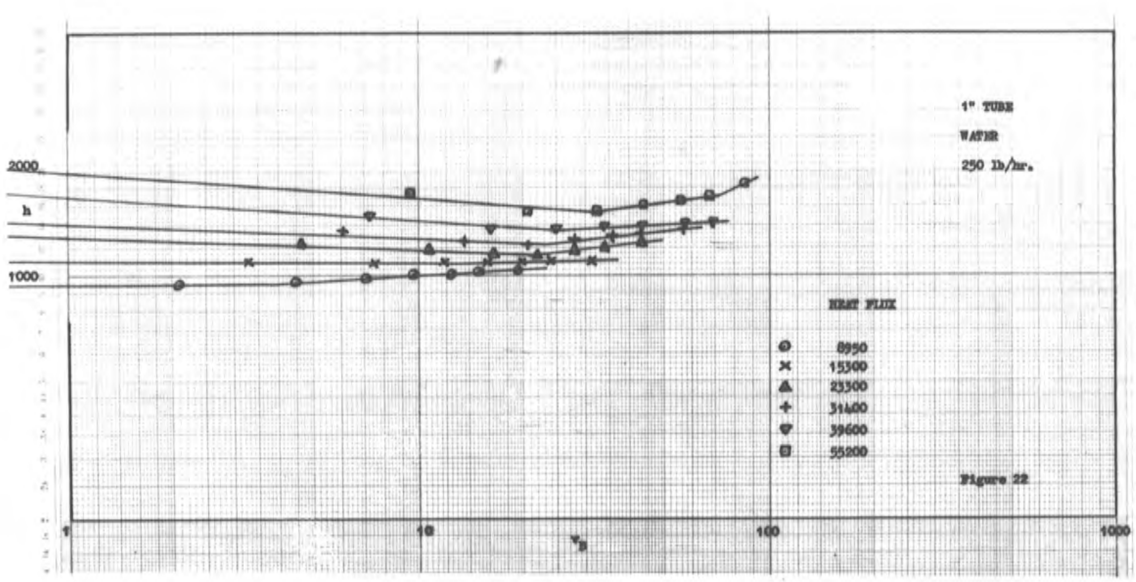


Figure 22

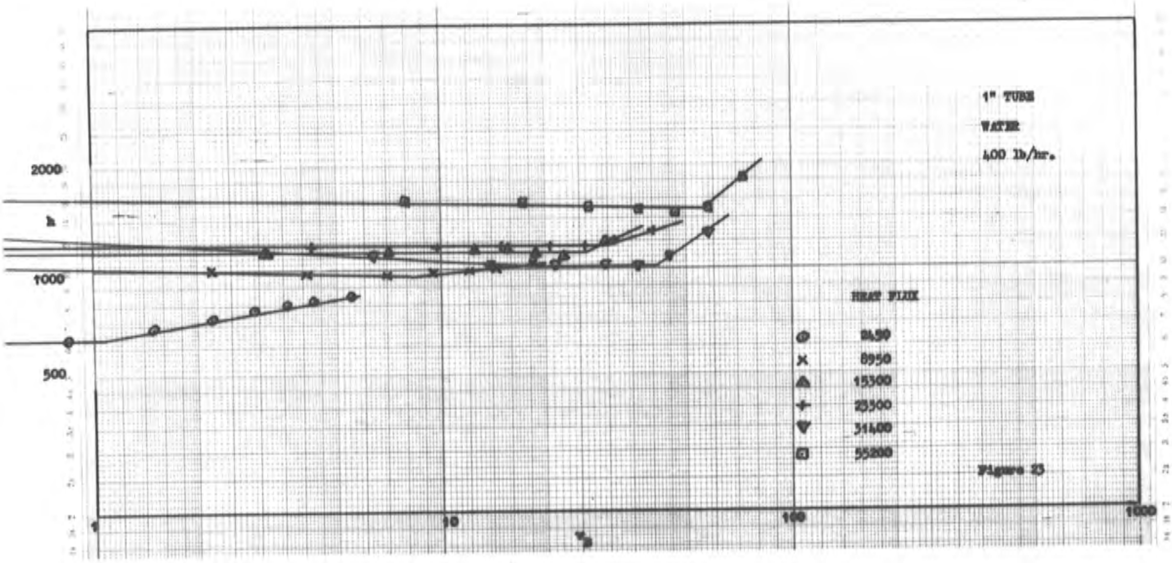


Figure 23

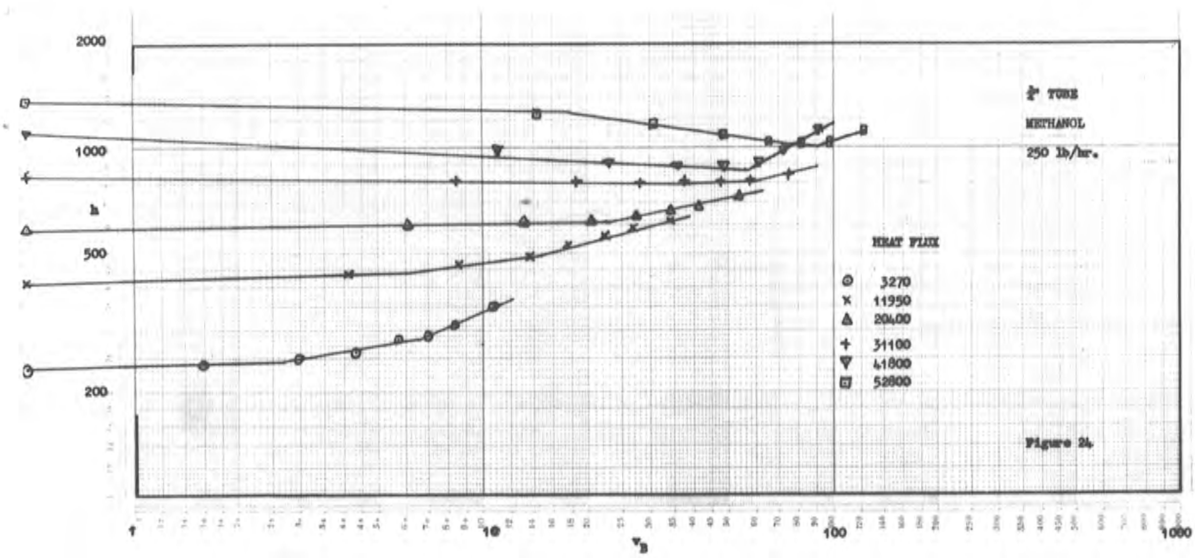


Figure 24

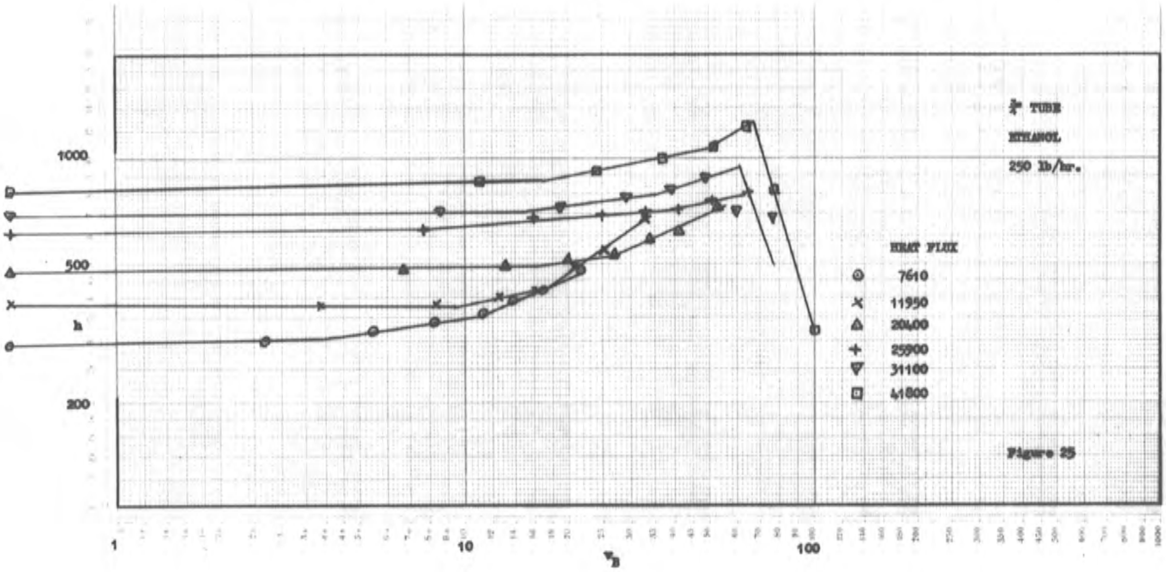


Figure 25

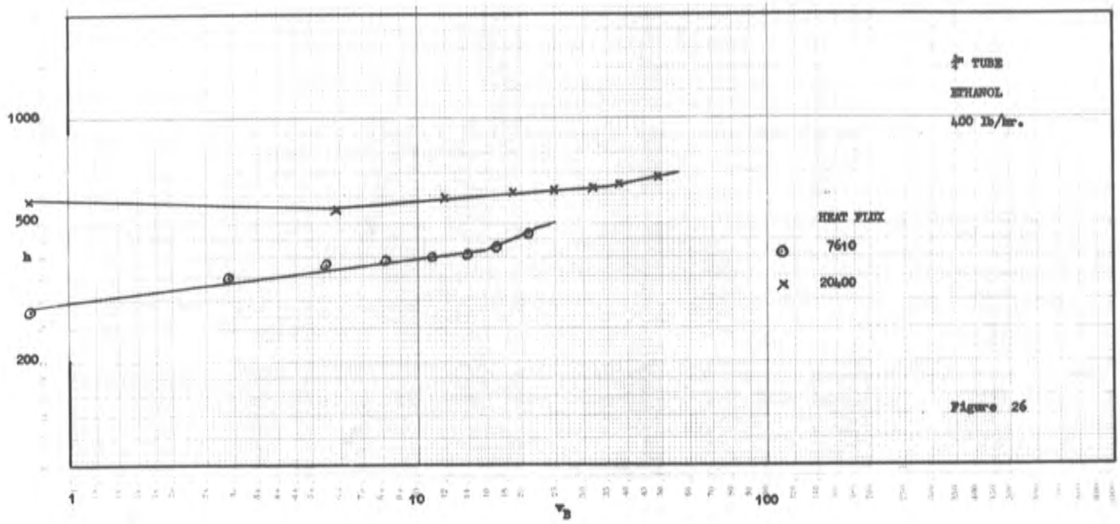


Figure 26

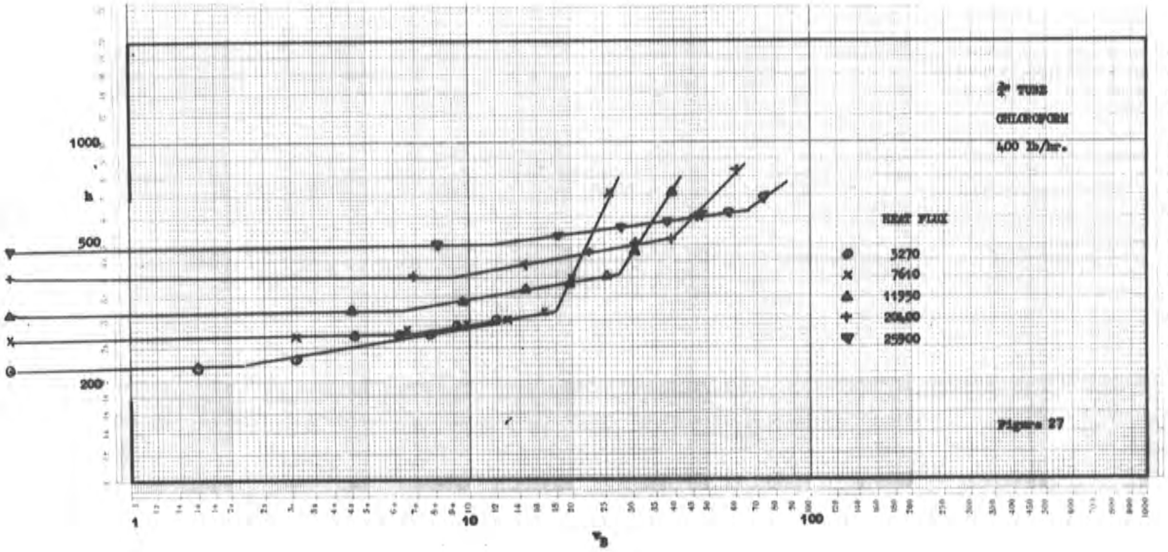


Figure 27

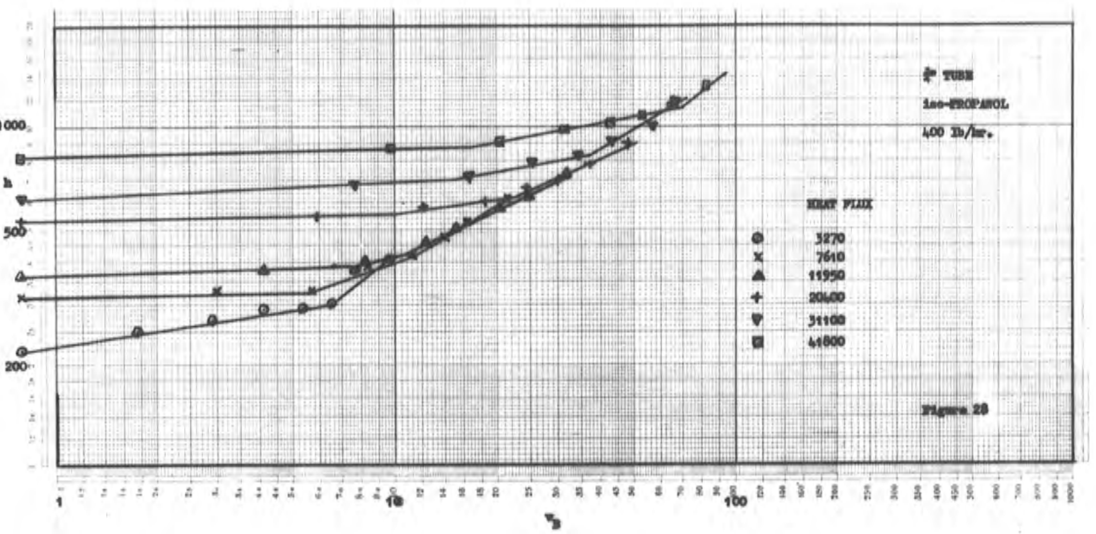
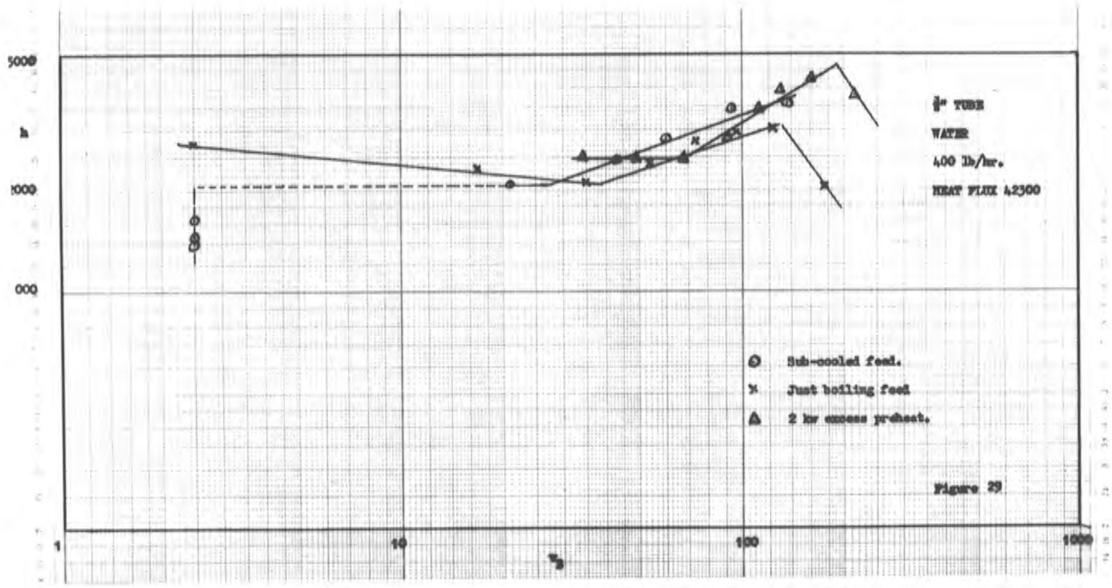
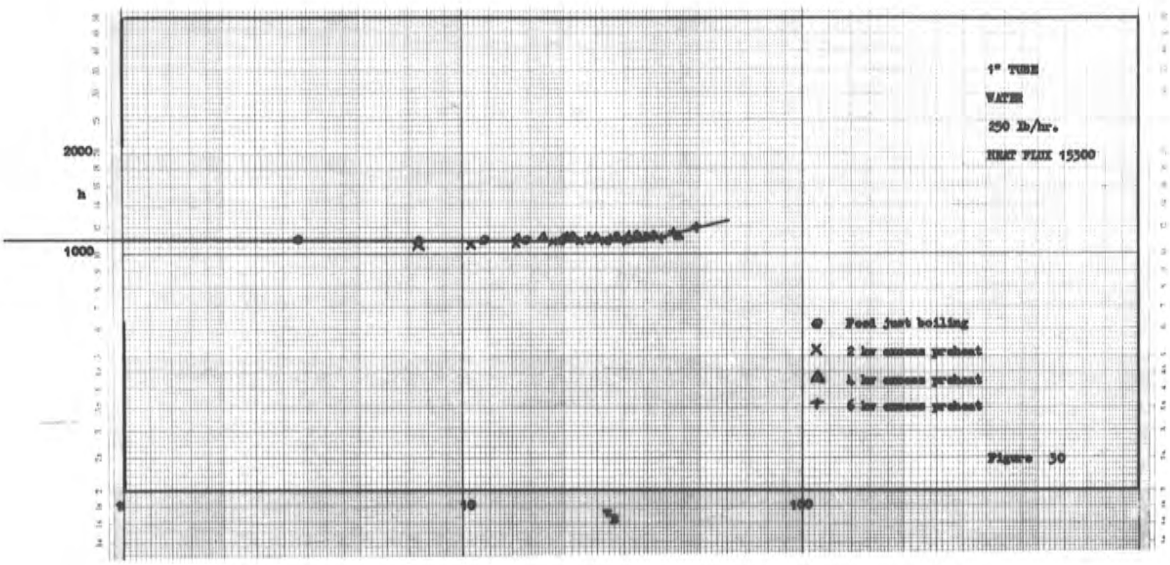


Figure 28



5000
h
2000
000
1
10
100
1000
¾" TUBE
WATER
400 lb/hr.
HEAT FLUX 42300
Sub-cooled feed.
Just boiling feed
2 kw excess preheat.
Figure 29



2000
h
1000
1
10
100
1" TUBE
WATER
250 lb/hr.
HEAT FLUX 15300
Feed just boiling
2 kw excess preheat
4 kw excess preheat
6 kw excess preheat
Figure 30

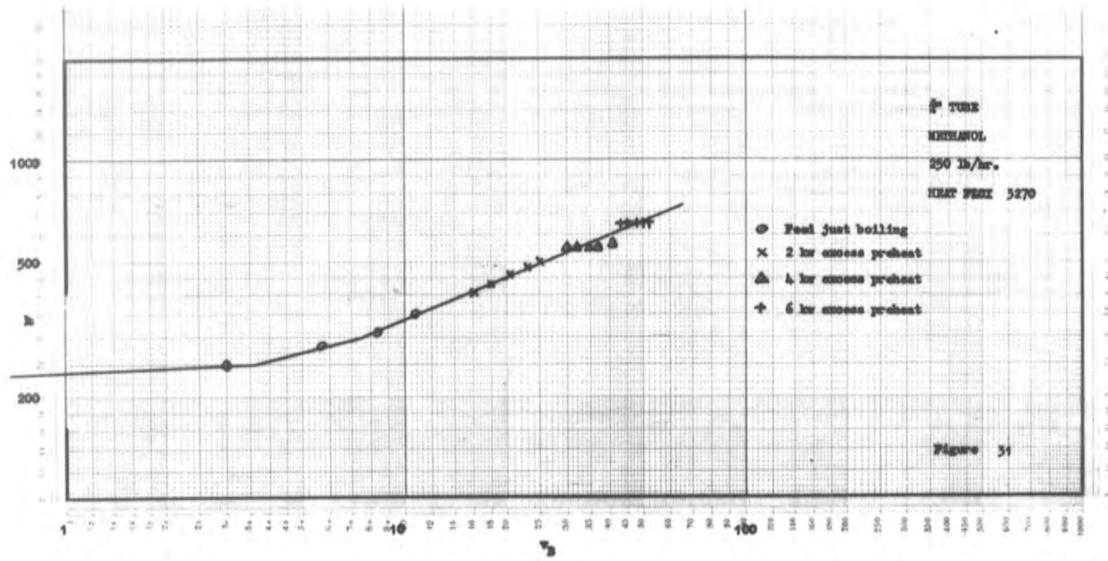


Figure 31

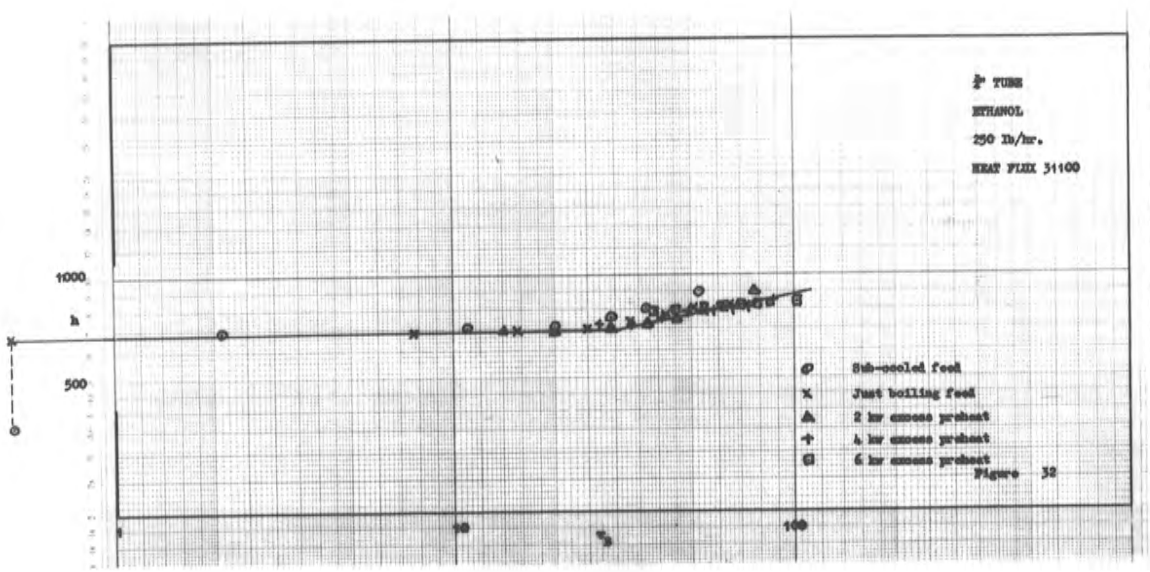


Figure 32

The results of the experiments to find the film thickness are given graphically in Appendix 5 and the film thicknesses calculated from these graphs are given in the following table. The values of the thicknesses are given in inches.

¾" Tube. Material - water.

Heat flux Btu/(hr)(sq ft).	Feed rate, lb/hr 150	250	400
7610	0.144	0.140	0.155
11950	0.119	0.138	0.104
20400	0.073	0.100	0.063
41800	0.045	0.056	0.037
52800	0.034	0.041	0.041
63500	-	0.035	-
74500	-	0.028	0.029

1" Tube. Material - water.

Heat flux Btu/(hr)(sq ft)	Feed rate, lb/hr 150	250	400
2450	*	*	-
5800	*	*	*
8950	-	0.18	*
12100	-	0.16	-
15300	0.089	0.113	0.129
23300	0.079	0.086	0.079
31400	0.074	0.083	-
39600	0.052	0.038	0.059
55200	0.038	-	-

$\frac{3}{4}$" Tube. Material - methanol.

Heat flux Btu/(hr)(sq ft)	Feed rate, lb/hr. 150	250	400
7610	0.104	0.109	0.162
11950	0.070	0.106	0.105
20400	0.041	0.055	0.065
31100	0.024	0.034	0.039
41800	-	0.022	0.023
52800	-	0.019	0.020
74500	-	-	0.014

$\frac{3}{4}$" Tube. Material - ethanol.

Heat flux Btu/(hr)(sq ft)	Feed rate, lb/hr. 150	250	400
7610	0.130	0.114	0.113
11950	0.080	0.089	0.096
20400	0.044	0.045	0.059
25900	0.041	-	-
31100	0.027	0.025	0.044
41800	-	0.014	0.026

The asterisk shows those readings which were found to have a concentration in the centre of the tube of more than 75%. It has been assumed that in these cases no meaning can be applied to film thickness.

Estimation of experimental accuracy:

There are two main methods which can be used to assess the reproducibility of the results. Either all possible errors can be mathematically summed to arrive at the maximum error possible, or the range of values obtained for any particular run can be compared with the values obtained when the run is repeated. Both will be discussed here.

Let us consider 3 runs using low, medium and high heat fluxes, say the runs made with water at 150 lb/hr in the $\frac{3}{4}$" tube. The heat transfer coefficient, h, is given by $h = \frac{q}{A\Delta T}$ and to enable us to assess the accuracy of h, the three terms on the right-hand side will be considered separately. Now the heat input to the column was obtained from a calibration chart of heat against wattmeter readings. The wattmeter is claimed to be accurate to $\pm$ 0.25% of full scale deflection and by careful calibration the chart should be as accurate. However, in reading the wattmeter it is possible for this error to be repeated, so q, the heat input, will be assumed to be accurate to $\pm$ 0.5% of the full scale deflection of the wattmeter. Hence with only 10% of full scale deflection the accuracy should be $\pm$ 5%. As a multiplying switch was available on the instrument, 10% was the minimum deflection used. The tube, area A, should be accurate to ± 0.1% but as the winding at each end does not terminate abruptly it is possible that further error could be introduced say to ± ½%. This does not, of course, vary with any one tube but is of importance when comparing the results with work done on other tubes. The most serious errors can be introduced in measuring temperatures. Because of calibration errors, interference of heating windings with the isothermal zones, inaccuracy of placing, but most of all because of thermal fluctuations of the fluid stream and the tube wall, it is thought that each temperature reading is only accurate to ± 1 F°, that is ΔT is accurate to ± 2 F°. Any errors introduced by the potentiometer or ancillary equipment will cancel out, being common to fluid and wall readings. Hence for the following runs the errors will be as shown:-

Heat flux 3270 Btu/(hr)(sq ft)	Error	= ± 5 %
ΔT = 4.2 ± 2 F°		= ± 50 %
Area		= ± ½%

These errors can be summed to find the total error by the usual mathematical methods (31).

Hence total error = ± 56 %

Heat flux 41,800 Btu/(hr)(sq ft)	Error	= ± 1 %
ΔT = 25.1 ± 2 F°		= ± 8 %
Area		= ± ½%
	Total error	= ± 9½%
Heat flux 74,500 Btu/(hr)(sq ft)	Error	= ± ½%
ΔT = 33.0 ± 2 F°		= ± 6 %
Area		= ± ½%
	Total error	= ± 7%

Examination of the graphs presented in Figures 9 to 27 will show that there are several instances of the curves for the lower heat fluxes crossing the curves for higher heat fluxes and it must be concluded that the lower heat fluxes can only be used as corroboratory evidence within their margin of accuracy.

To give some indication of the actual reproducibility, comparisons are presented of runs made under similar conditions at different times. The run shown in Figure 33 made with a low heat flux shows the biggest difference, but this falls within the expected ± 42% for this heat flux. The other pairs of graphs in Figures 34, 35 and 36 differ by no more than 5% of each other and this is within the accuracy expected.

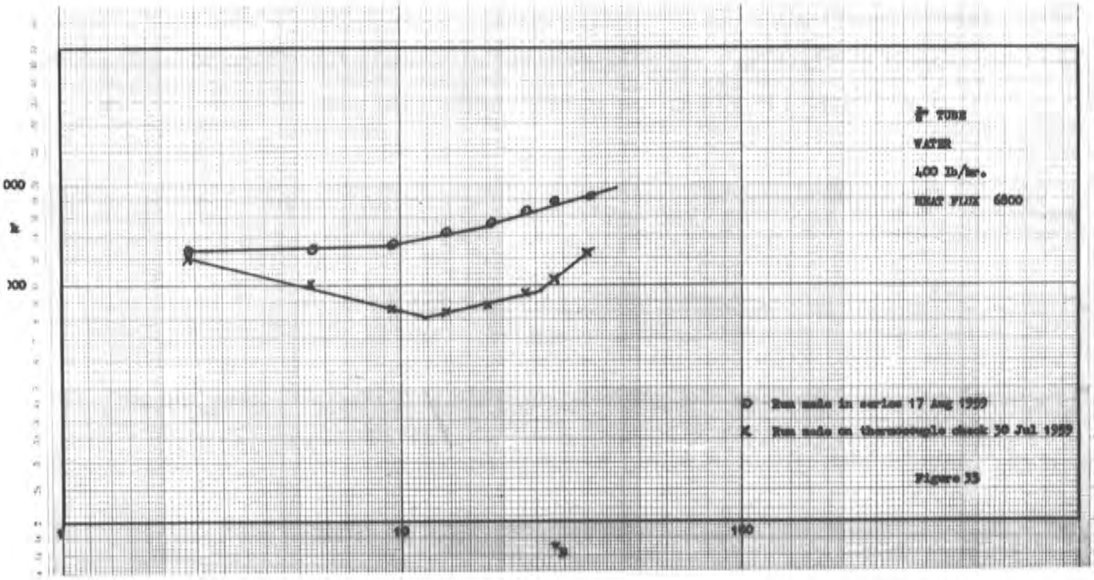


Figure 33

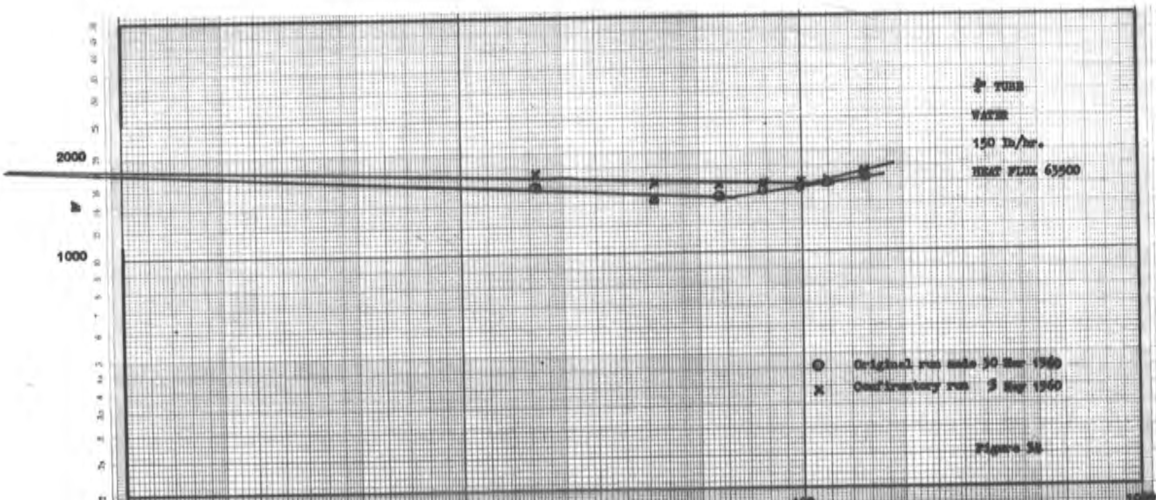


Figure 34

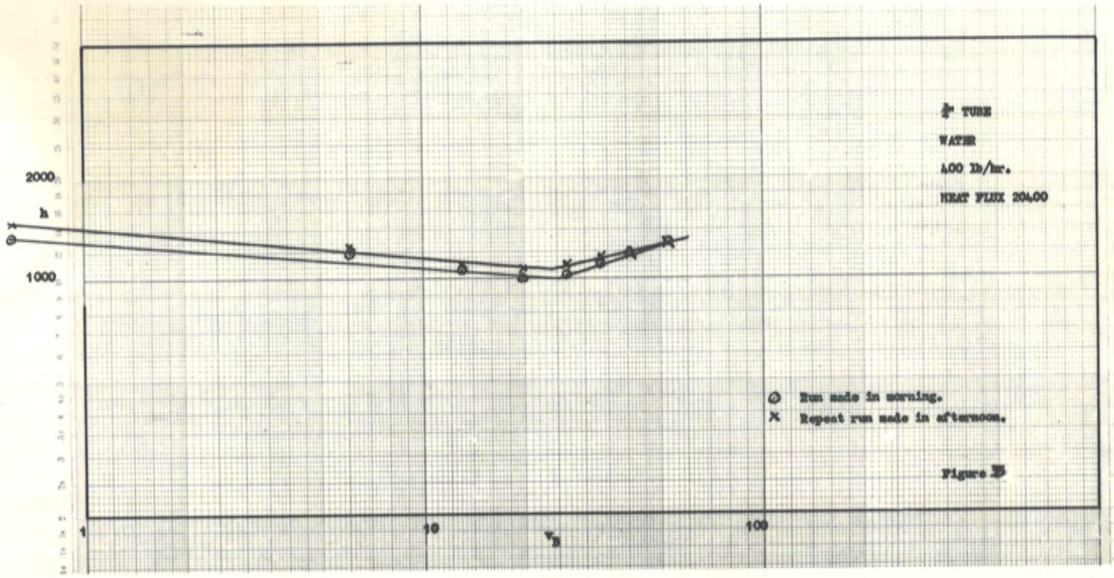


Figure 35

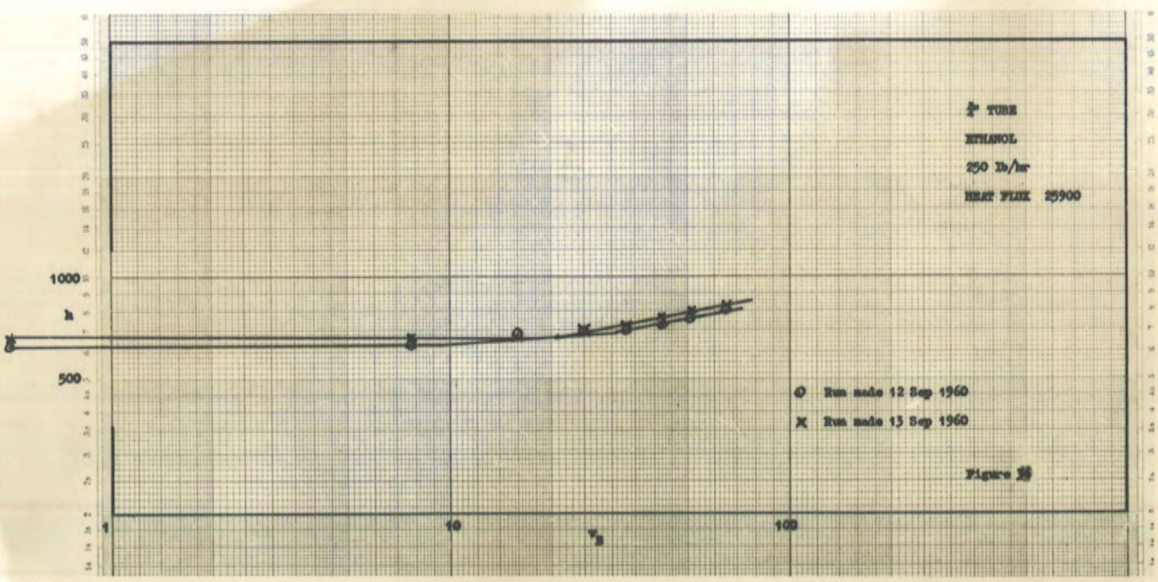


Figure 36

CHAPTER 5.

CORRELATION OF RESULTS.

Examination of the graphs of heat transfer coefficient against bulk velocity presented in Figures 9 to 28 shows that they fall into similar patterns. Firstly, for any series of runs with a particular feed rate, there is a region where the heat transfer coefficient is either constant, falling, or rising depending on the heat flux. Next is an intermediate zone where the graph has a gently rising slope, although this is not apparent on all curves. Then there is a line which is common to all runs made at a particular flow rate and which has a slope of approximately 0.5. Finally there is a region which is reached only in some runs at high heat fluxes where the heat transfer coefficient is falling or remains constant. This is presumably because of dry wall conditions, because it occurs when a large part of the feed has been vaporized.

An examination made of the results obtained when measuring the film thickness (see Appendix 5) shows some connection between the zones described and the concentration of liquid in the centre of the tube. Two assumptions will be made:-

1) That where the concentration of the liquid in the centre of the tube is greater than 75%, then slug boiling is not yet predominant and the tube has mostly a homogeneous mixture of vapour and liquid.

2) That where the concentration of liquid in the centre of the tube is less than 25%, climbing film conditions prevail.

Between these two sets of conditions slug type boiling exists.

Using these criteria, the following table is constructed. The columns marked "Homogeneous", "Slug" and "Climbing Film" contain the heat fluxes under which the tube was working when the conditions at the top were measured. The final column represents the observations made from the graphs of h v. v_B of the lowest heat flux which reached the line of slope 0.5.

Tube	Feed rate	Homogeneous	Slug	Climbing film	From graphs
$\frac{3}{4}$"	Water				
	150		7610 11950 20400	41800 and above	41800
	250		7610 11950 20400	41800 and above	20400
	400		7610 11950 20400	41800 and above	20400
1"	Water				
	150	2450 5800		15300 and above	23300
	250	2450 5800	8950 12100 15300	23300 and above	23300
	400	5800	8950 15300 23300	39600	15300

Tube	Feed rate	Homogeneous	Slug	Climbing film	From graphs
$\frac{3}{4}$"	Methanol				
	250			7610 and above	3270
$\frac{3}{4}$"	Ethanol				
	250			7610 and above	7610
	400			7610 and above	7610

From this table it is possible to deduce that the point where most graphs reach the line of slope 0.5 is the beginning of the climbing film action. Agreement is not found from every case as it is not easy to define where every curve reaches the line of slope 0.5.

It will be noted that four of the above thickness measurement results are shown as slug type flow whereas the same heat fluxes are found from the graphs as climbing film. This indicates the difficulty of assessing the change from slug type to climbing film. It will be seen that these four results indicate well developed slug type conditions and it is possible that the beginning of the climbing film is an unstable point. However these results indicate clearly that climbing film conditions are fully developed when the line of slope of 0.5 is reached, and hence the line will be called the "climbing film line". It is possible, though, that some climbing film action has started before this.

As the only change in the mechanism which takes place between the bottom of the tube and the climbing film is the transition from homogeneous flow to slug type flow, it can be assumed that this is represented on the graphs by the intermediate change in slope reported for most graphs.

With the runs made on the $\frac{1}{2}$" tube with water (figures 12 to 17) it will be noted that despite a feed rate range from 75 lb/hr to 750 lb/hr, the position of the climbing film line varied little. Hence it can be assumed that in this part of the tube, the feed rate does not affect the way in which the heat transfer coefficient varies with bulk velocity. So it is probable that the changes in values which are noted for some of the other runs are caused by the temperature effect on the fluid physical properties.

It was then assumed that the heat transfer coefficient, h, may be a function of the following physical properties and variables:-

$$D,\ v_V,\ v_L,\ \mu_V,\ \mu_L,\ \Delta T,\ k_L,\ c,\ \sigma,\ \rho_V,\ \rho_L,\ \lambda,\ w_L$$

where the symbols are defined in a table in appendix 9. By dimensional analysis the following relationship was derived:-

$$\frac{hD}{k_L} = \phi\left(\frac{\rho_L D v_L}{\mu_L}\right)\left(\frac{\rho_V D v_V}{\mu_V}\right)\left(\frac{v_V \mu_V}{\sigma}\right)\left(\frac{\lambda}{c\Delta T}\right)\left(\frac{4 w_L}{\pi D \mu_L}\right)\left(\frac{\mu_V}{\mu_L}\right)$$

As has already been discussed in Chapter 4, the 5th term on the right hand side, which is another way of expressing a Reynolds number, would be of little value in assessing the results. The variation of the heat transfer coefficient up the tube takes place despite a constant value for this term, and the pattern of behaviour of the heat transfer coefficient does not change even when this group has a ten-fold variation.

The first group, the liquid Reynolds number, involves the velocity of the liquid film. The way that this velocity varies with conditions in the tube has not yet been reported in the literature, but several authors have stated, as described in Chapter 2, that the film travels at a slower velocity than the vapour. Attempts were made using the film thickness results, and making several assumptions, to assess the liquid film velocity. It was concluded that insufficient experimental work had been carried out to enable this velocity to be reported. Using the values thus obtained, no satisfactory correlation could be obtained. The correlation was also attempted using the vapour velocity. It was argued that the liquid film velocity might be proportional to the vapour velocity, and that the nature of any relationship might be seen. However no correlation was again obtained.

The second group, the vapour Reynolds number, was next examined. Values of this group were plotted using not only the Nusselt group, $\frac{hD}{k_L}$, as defined as ordinate, but also using values of $\frac{ht}{k_L}$, where t is the film thickness as found. Neither of these relationships gave any satisfactory correlation.

The third group when combined with the vapour Reynolds number gives $\frac{\rho_V D v_V^2}{\sigma}$, which is the Weber number. When graphs of this group using the bulk velocity instead of vapour velocity were plotted as abscissa, a reasonable correlation was obtained, and eventually the following equation was found to represent the climbing film part of the graphs to within an accuracy of $\pm$ 15% .

$$\frac{hD}{k_L} = 50\left(v_B\sqrt{\frac{\rho_V D}{\sigma g_c}}\right)^{\frac{1}{2}}\left(\frac{c_{L}\mu_L}{k_L}\right)^{\frac{1}{3}}$$

This equation can be broken into mechanisms as follows:-

$$\left(\frac{\text{Total heat transfered}}{\text{Heat transfered by Conditions}}\right) = \phi\left(\frac{\text{Inertial forces}}{\text{Surface forces}}\right)\left(\frac{\text{Momentum diffusivity}}{\text{Thermal diffusivity}}\right)$$

and it becomes apparent that there are five mechanisms but only three groups, which is inconsistent. As the main correlation has been made with the Nusselt and Weber numbers, it is assumed that this is the basic relationship. Hence any modifying group should consist of any two mechanisms already considered. It was thought unreasonable that terms containing h or v_B should re-appear and hence the third group would be the ratio

$$\frac{c_L \rho_L \sigma g_c D}{k_L}$$

Accordingly, this group was tried and a best fit was found by least squares with the equation

$$\frac{hD}{k_L} = A\left(v_B\sqrt{\frac{\rho D}{\sigma g_c}}\right)^{0.5}\left(\frac{c_L \rho_L \sigma g_c D}{k_L}\right)^{0.9}$$

The difference in fit with a power of 0.9 and a power of unity was so slight that it was decided that the power of unity should be examined and the equation was accordingly re-written as

$$\frac{h}{c_L}\sqrt{\frac{D}{\rho_L \sigma g_c}} = A' \left\{ v_B \sqrt{\frac{\rho_v D}{\sigma g_c}} \right\}^{0.5}$$

This can now be described as :-

$$\left(\frac{\text{Force to transfer heat}}{\text{Resisting force of surface}}\right) = \phi'' \left(\frac{\text{Inertial forces}}{\text{Surface forces}}\right)$$

The force to transfer heat must not be considered as a heat energy gradient but should be considered in terms of diffusion forces. This equation has three mechanisms and two groups and hence is consistent. Graphs showing this relationship are presented in figures 37 to 56. The physical data used for the calculations throughout this thesis are tabulated in appendix 6.

The constant A' has the value 0.012 and hence the equation has the form

$$\frac{h}{c_L}\sqrt{\frac{D}{\rho_L \sigma g_c}} = 0.012 \left\{ v_B \sqrt{\frac{\rho_v D}{\sigma g_c}} \right\}^{0.5}$$

Correlations have also been made for conditions at the bottom of the tube and for the variation of the film thickness with the bulk velocity. Both these correlations are presented in appendices 7 and 8 respectively.

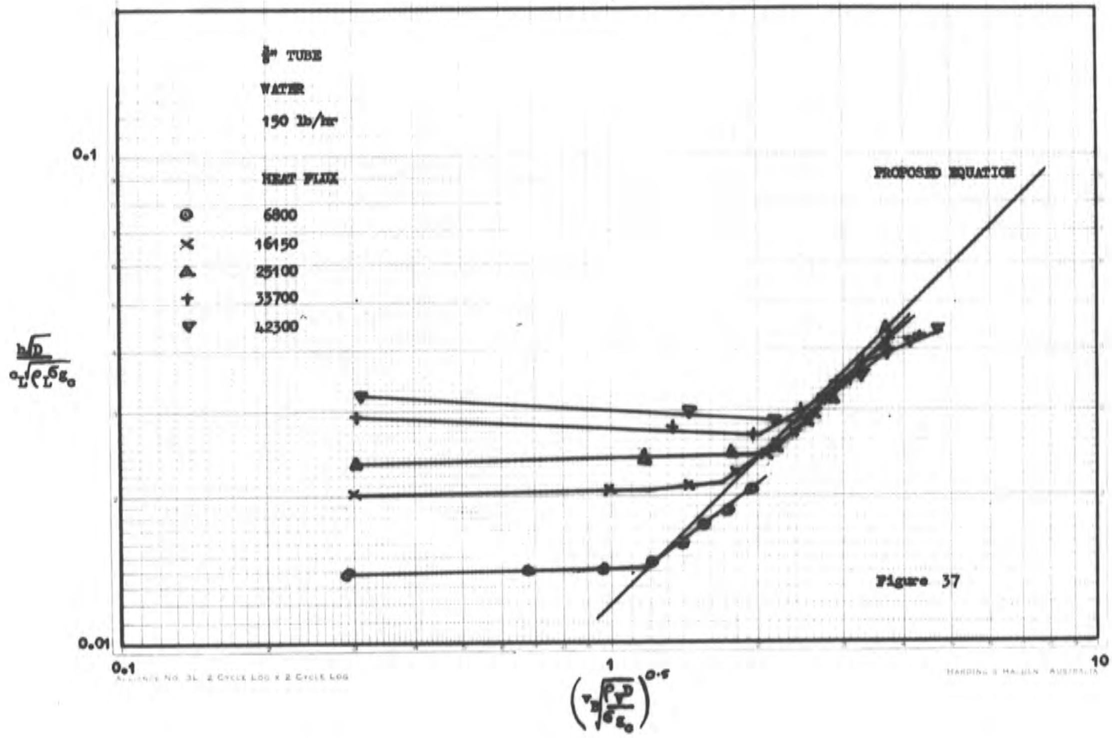


Figure 37

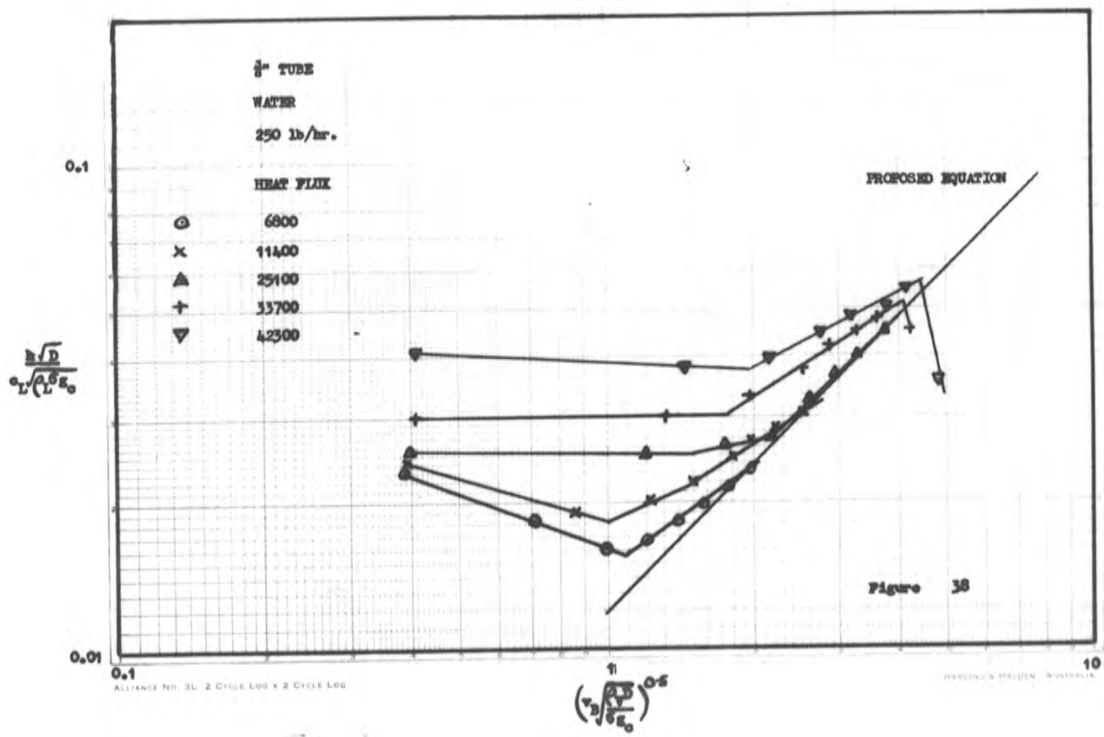
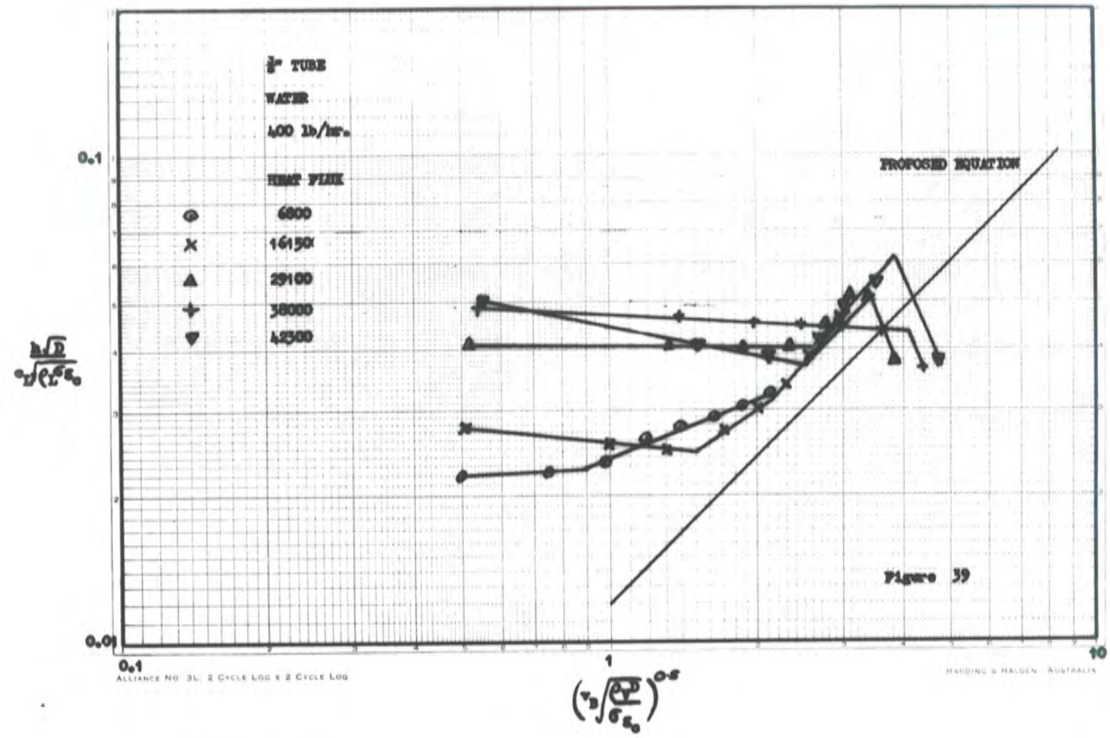
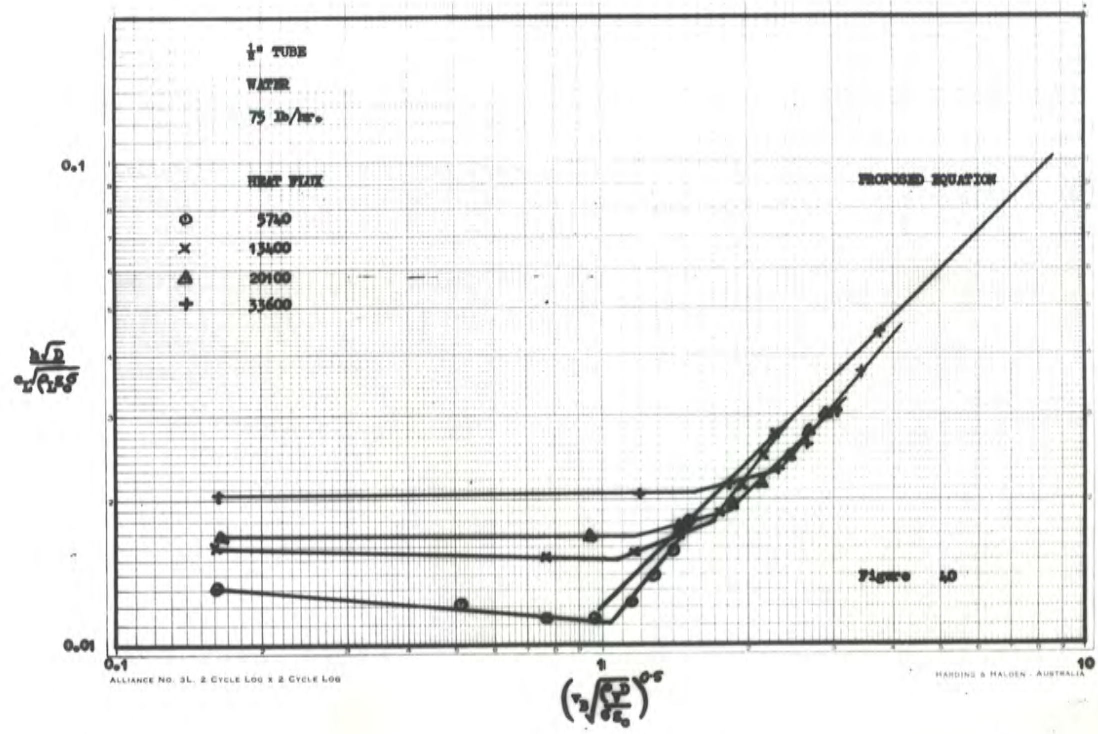


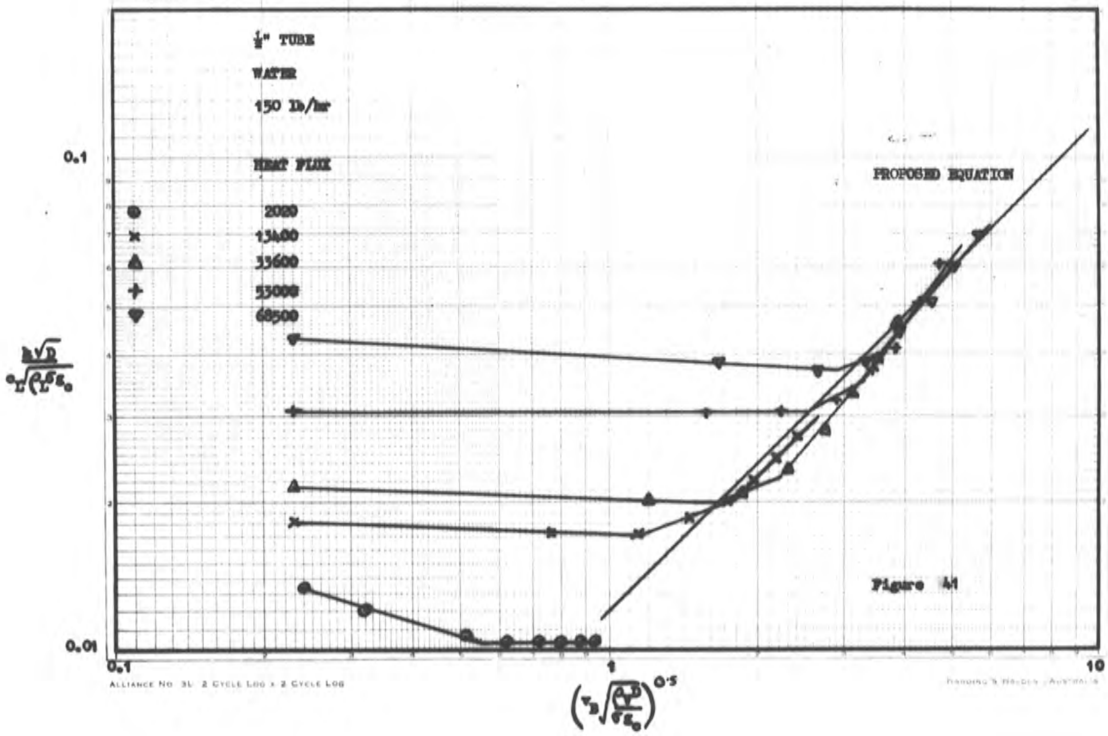
Figure 38



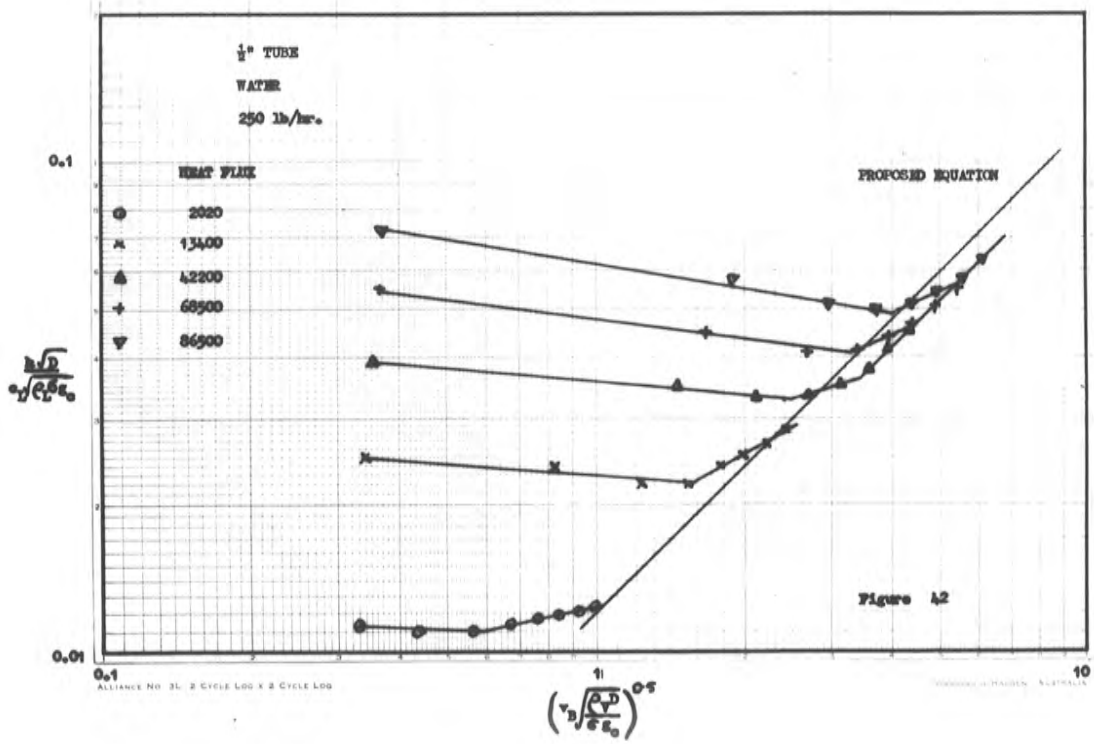
3/8" TUBE
WATER
400 lb/hr.
HEAT FLUX
6800
16150
29100
38000
42300
PROPOSED EQUATION
Figure 39
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN · AUSTRALIA



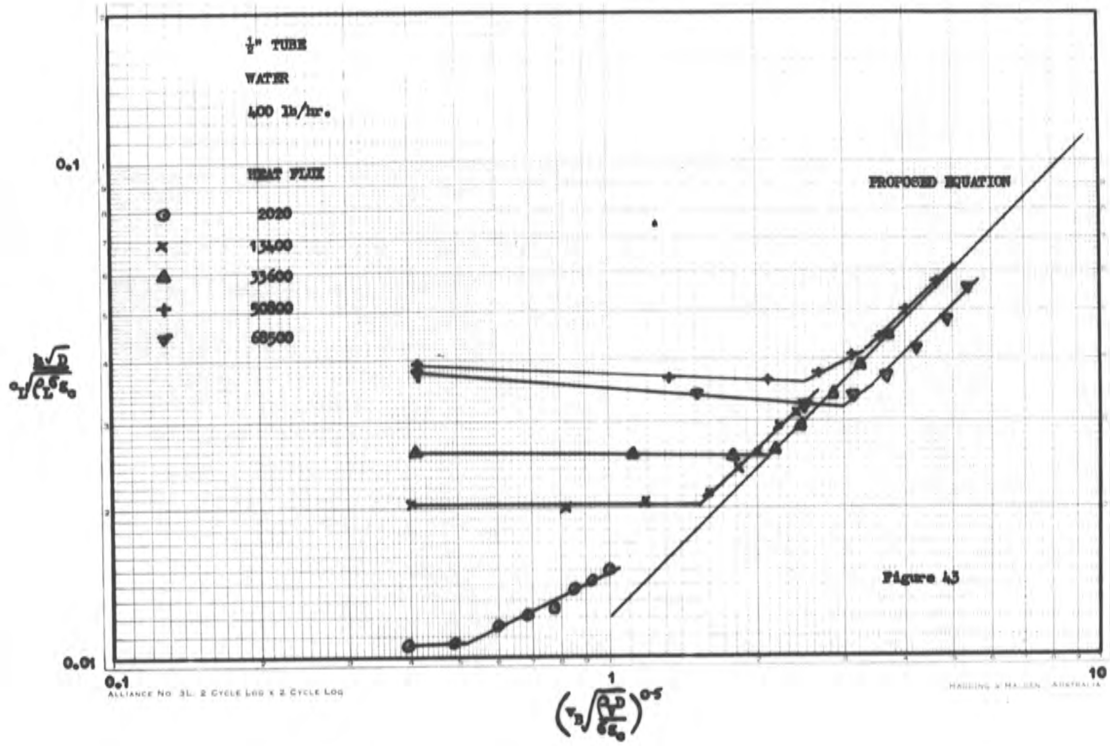
1/2" TUBE
WATER
75 lb/hr.
HEAT FLUX
5740
13400
20100
33600
PROPOSED EQUATION
Figure 40
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN · AUSTRALIA

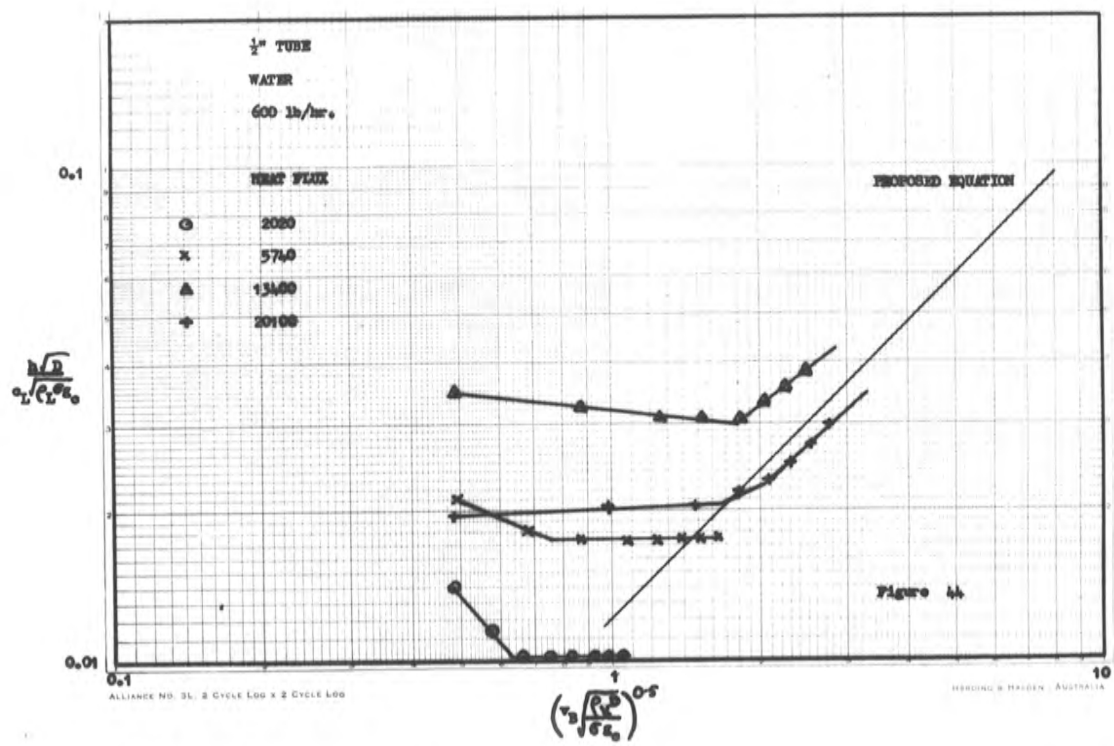


½" TUBE
WATER
150 lb/hr
HEAT FLUX
2020
13400
33600
53000
68500
PROPOSED EQUATION
Figure 41
0.1
0.01
0.1
1
10



½" TUBE
WATER
250 lb/hr.
HEAT FLUX
2020
13400
42200
68500
86500
PROPOSED EQUATION
Figure 42
0.1
0.01
0.1
1
10


½" TUBE
WATER
400 lb/hr.
HEAT FLUX
2020
13400
33600
50800
68500
PROPOSED EQUATION
Figure 43
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN . AUSTRALIA


½" TUBE
WATER
600 lb/hr.
HEAT FLUX
2020
5740
13400
20100
PROPOSED EQUATION
Figure 44
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN . AUSTRALIA

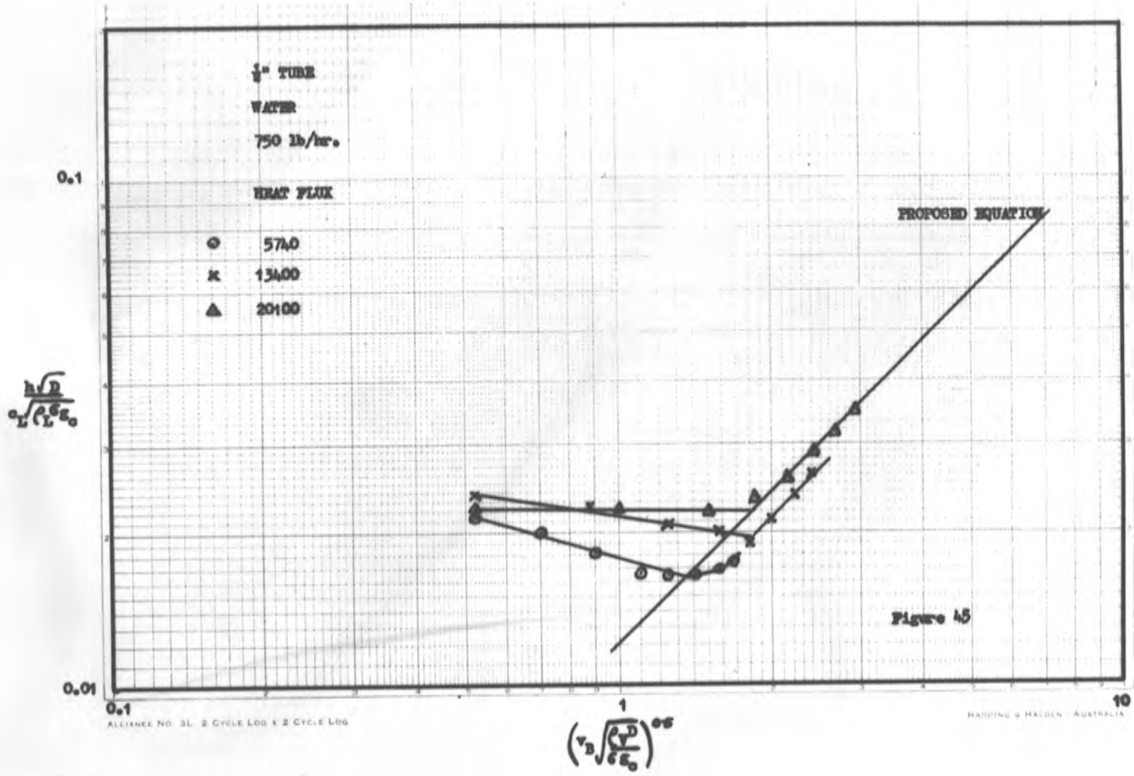


Figure 45

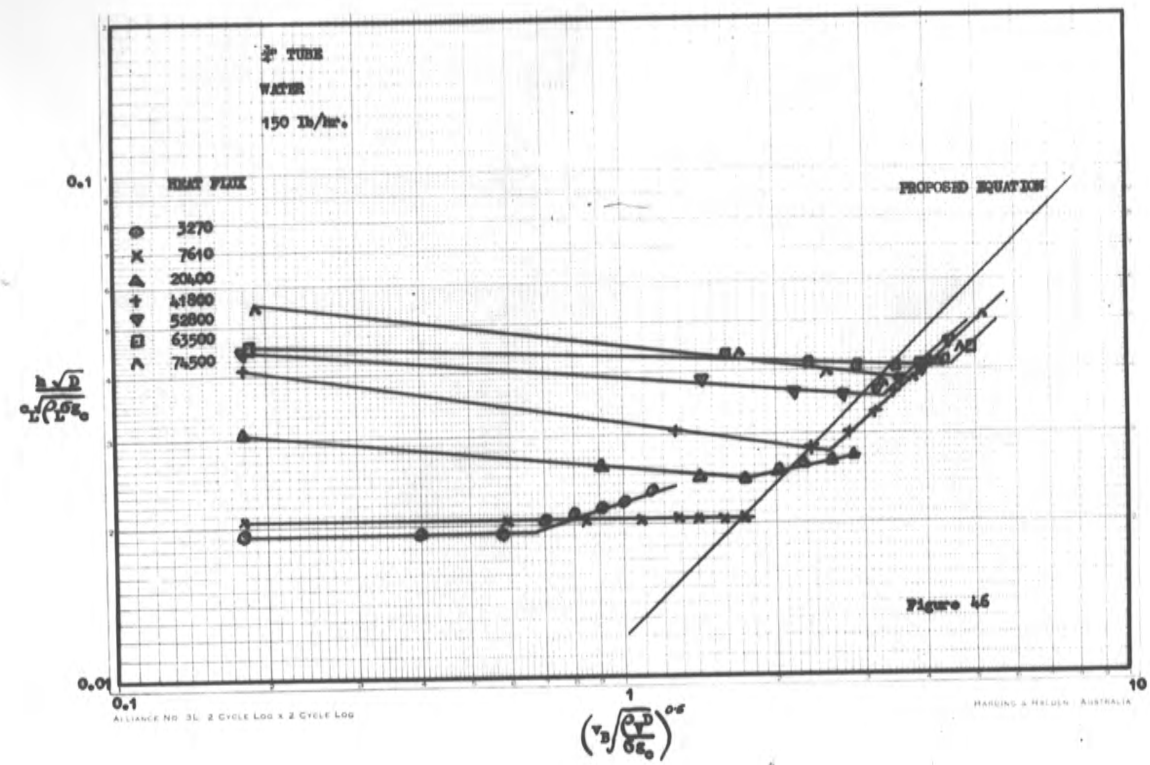
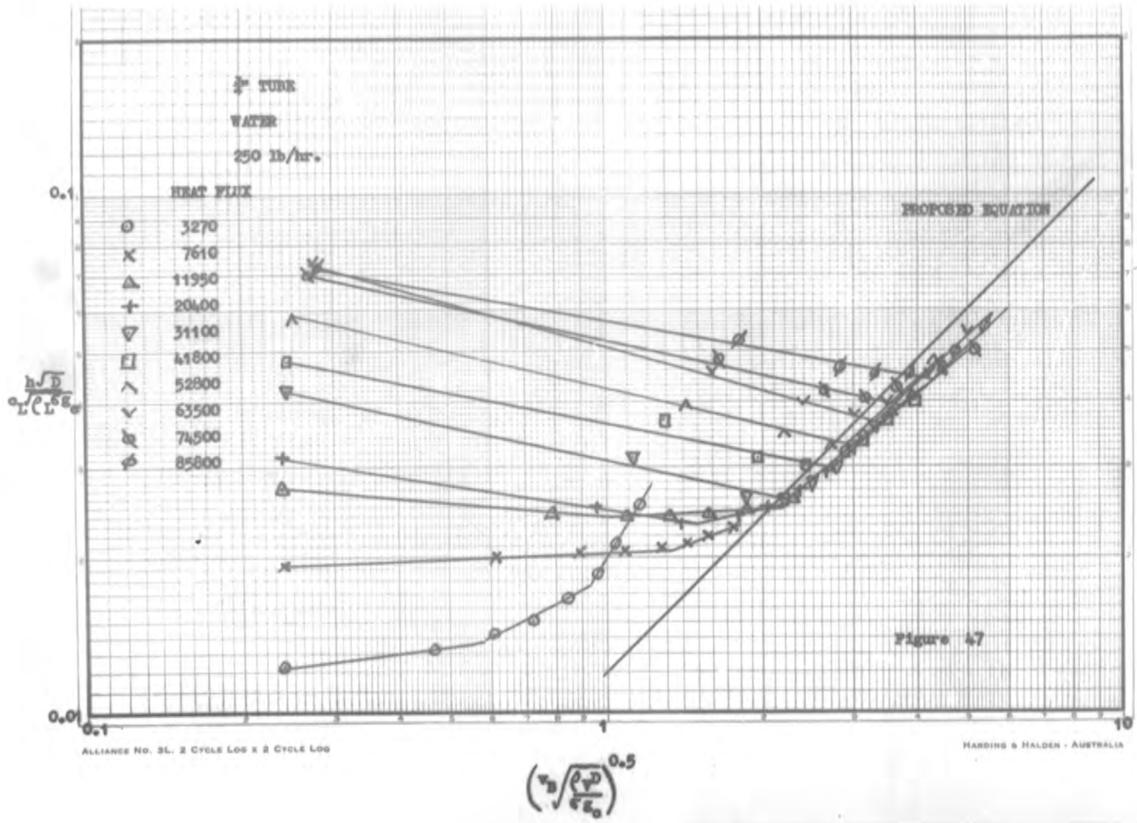
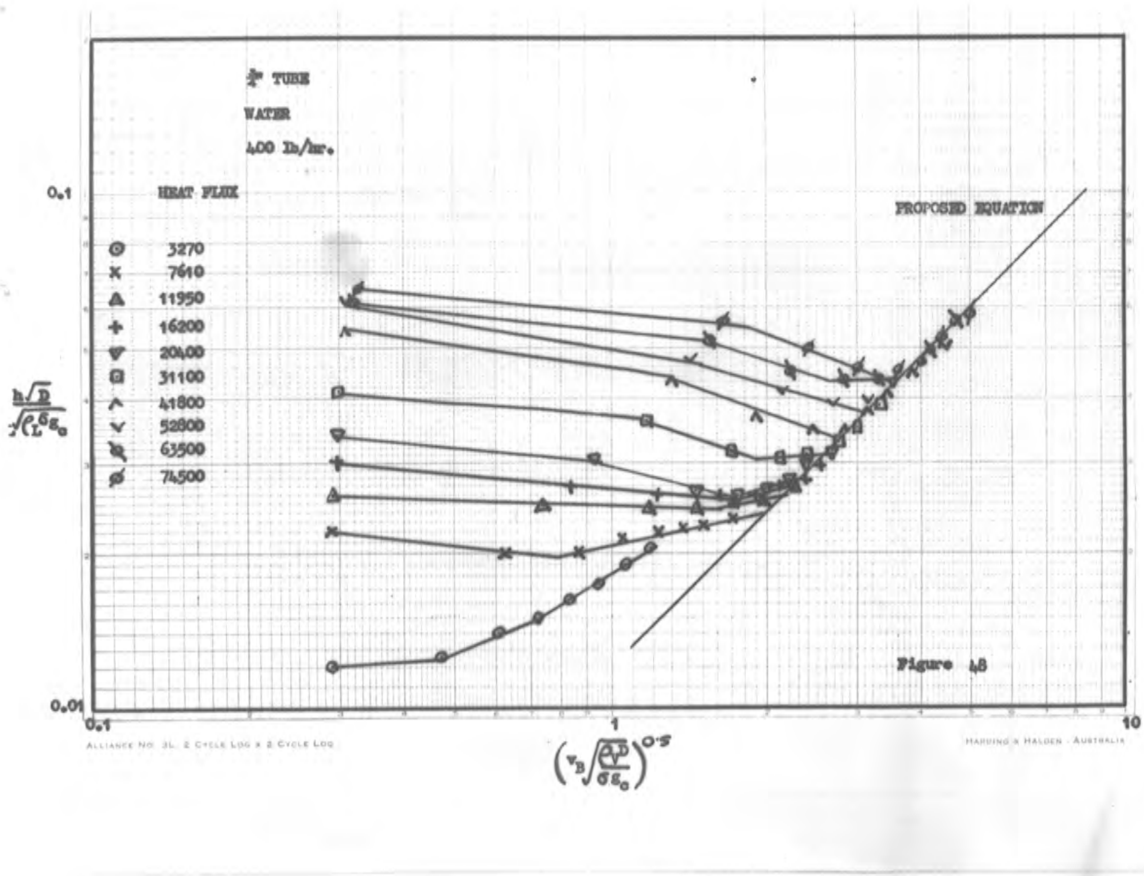


Figure 46



¾" TUBE
WATER
250 lb/hr.
HEAT FLUX
3270
7610
11950
20400
31100
41800
52800
63500
74500
85800
PROPOSED EQUATION
Figure 47
0.1
0.01
1
10
ALLIANCE No. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN - AUSTRALIA



¾" TUBE
WATER
400 lb/hr.
HEAT FLUX
3270
7610
11950
16200
20400
31100
41800
52800
63500
74500
PROPOSED EQUATION
Figure 48
0.1
0.01
1
10
ALLIANCE No. 3L. 2 CYCLE LOG x 2 CYCLE LOG
HARDING & HALDEN - AUSTRALIA

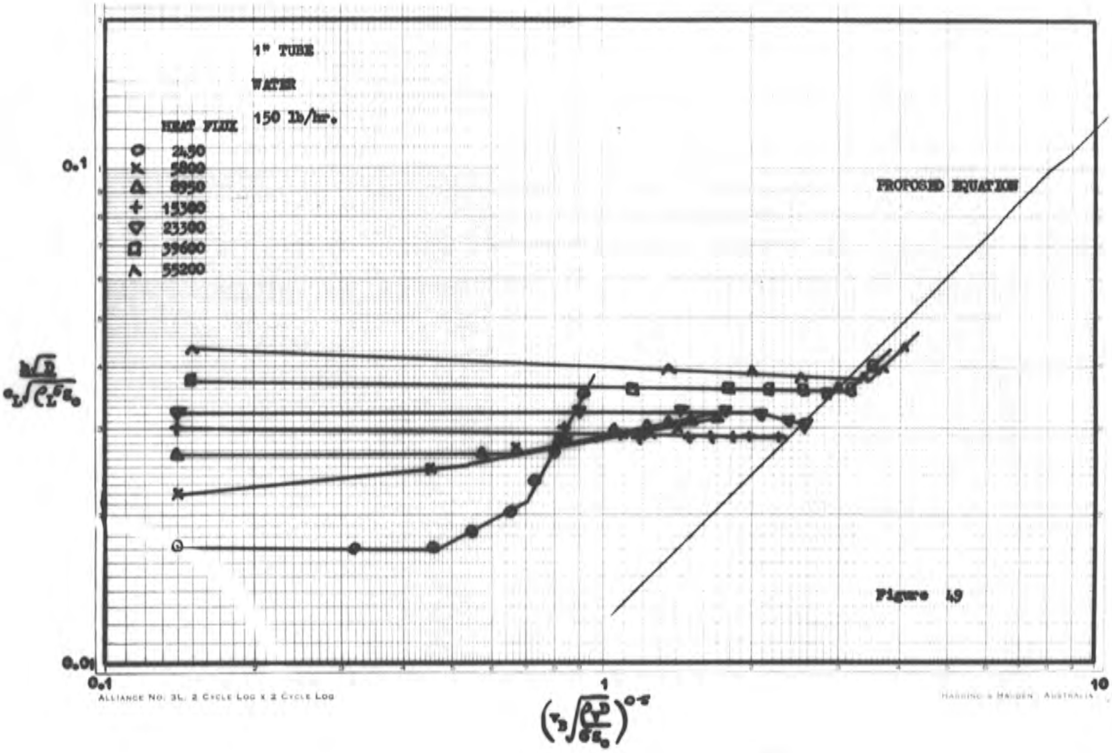


Figure 49

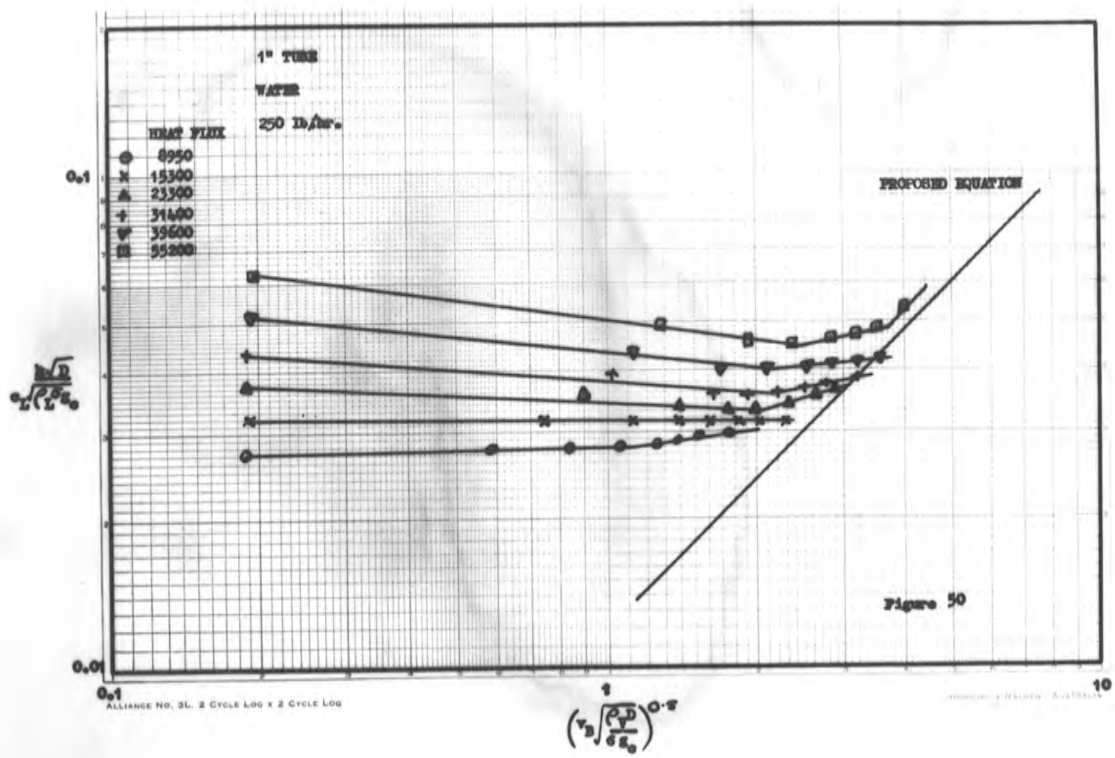
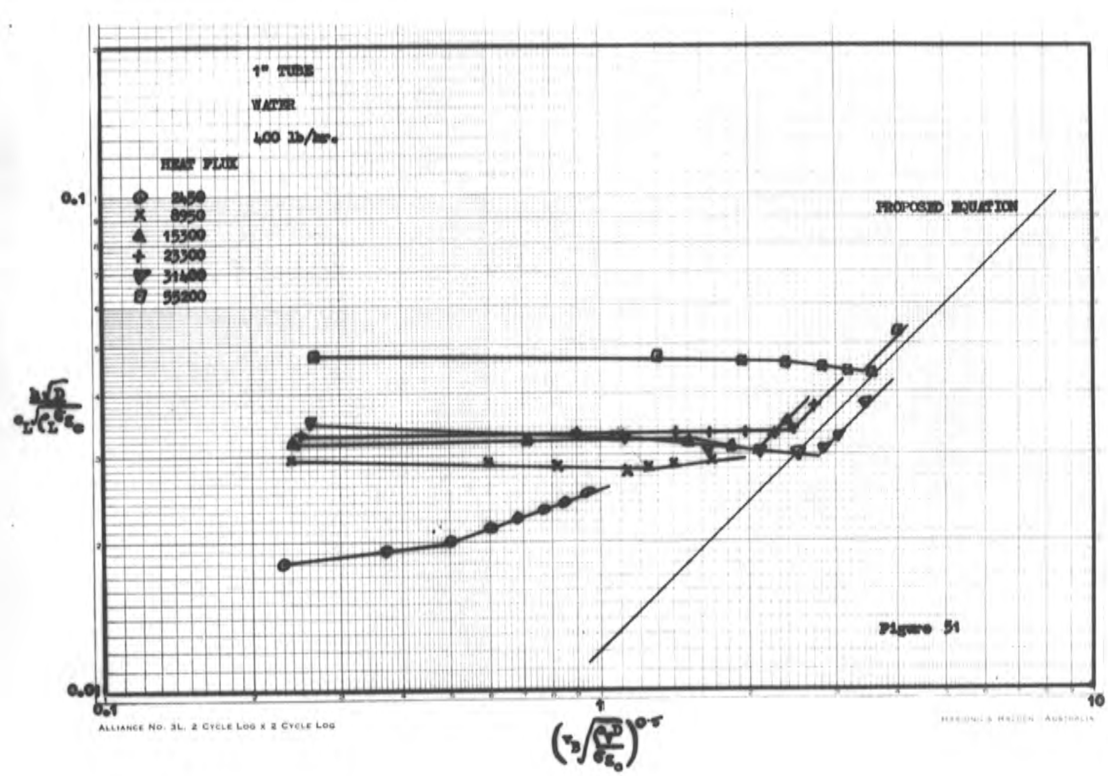
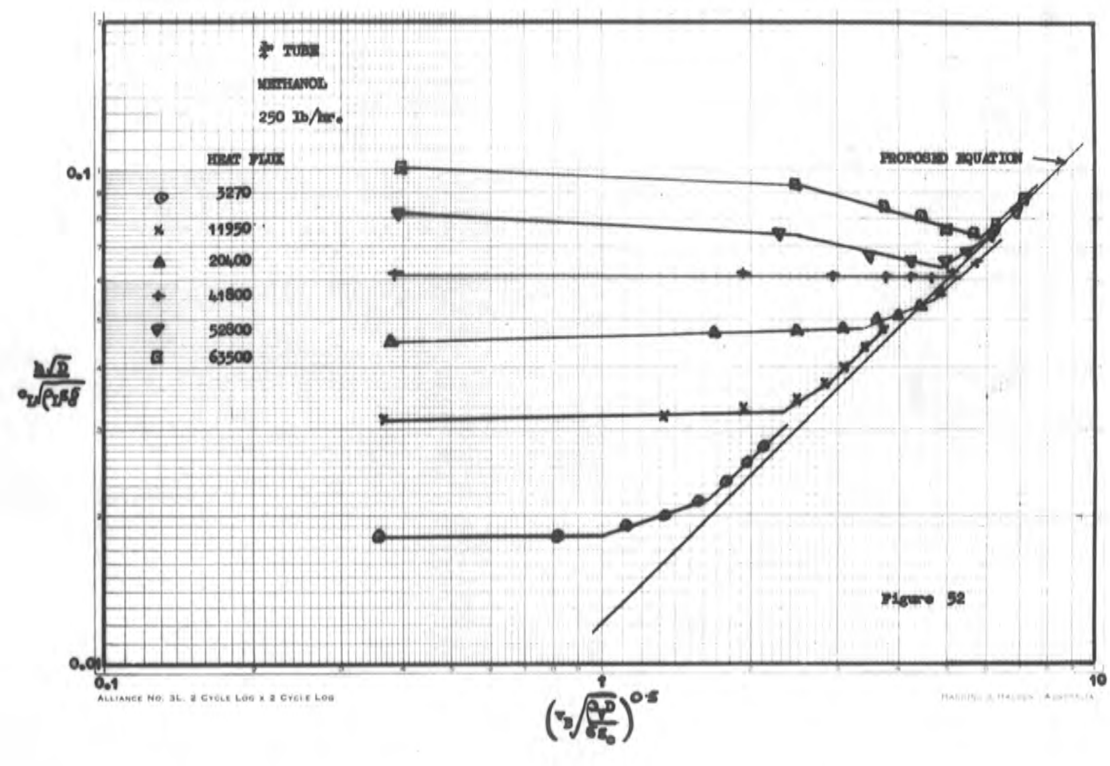


Figure 50


1" TUBE
WATER
400 lb/hr.
HEAT FLUX
2450
8950
15300
23300
31400
55200
PROPOSED EQUATION
Figure 51
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG


$\frac{3}{4}$" TUBE
METHANOL
250 lb/hr.
HEAT FLUX
3270
11950
20400
41800
52800
63500
PROPOSED EQUATION
Figure 52
0.1
0.01
0.1
1
10
ALLIANCE NO. 3L. 2 CYCLE LOG x 2 CYCLE LOG

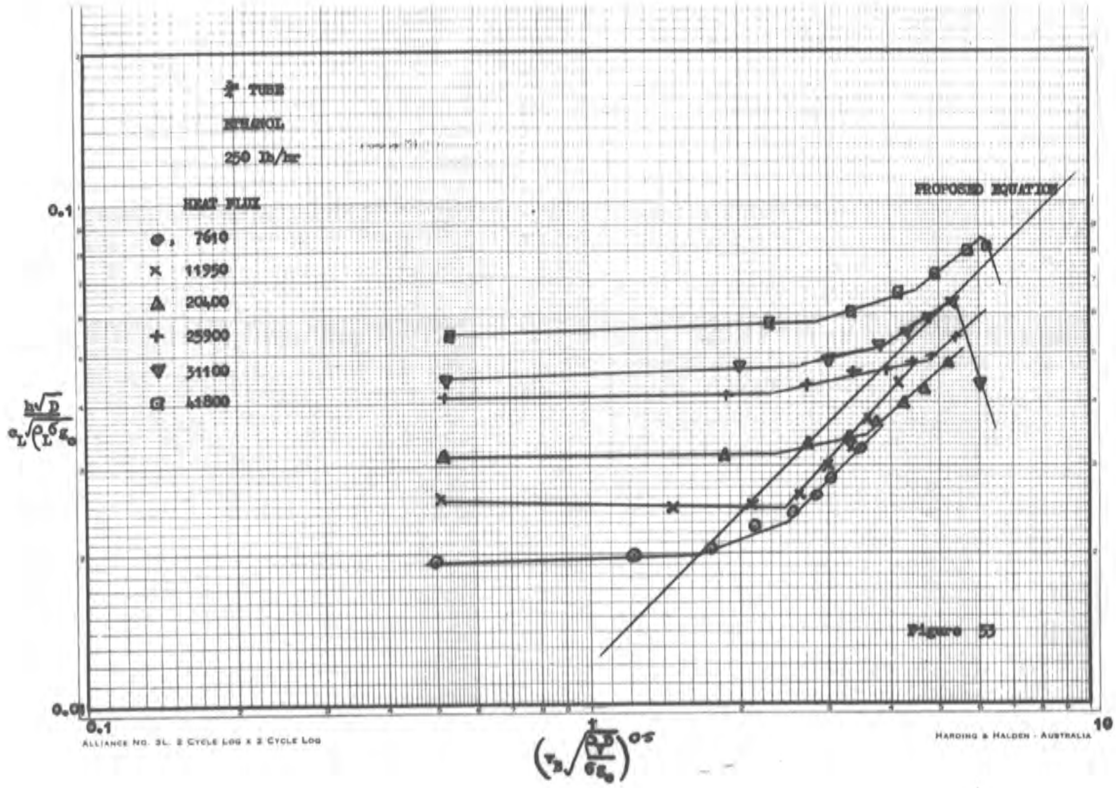


Figure 53

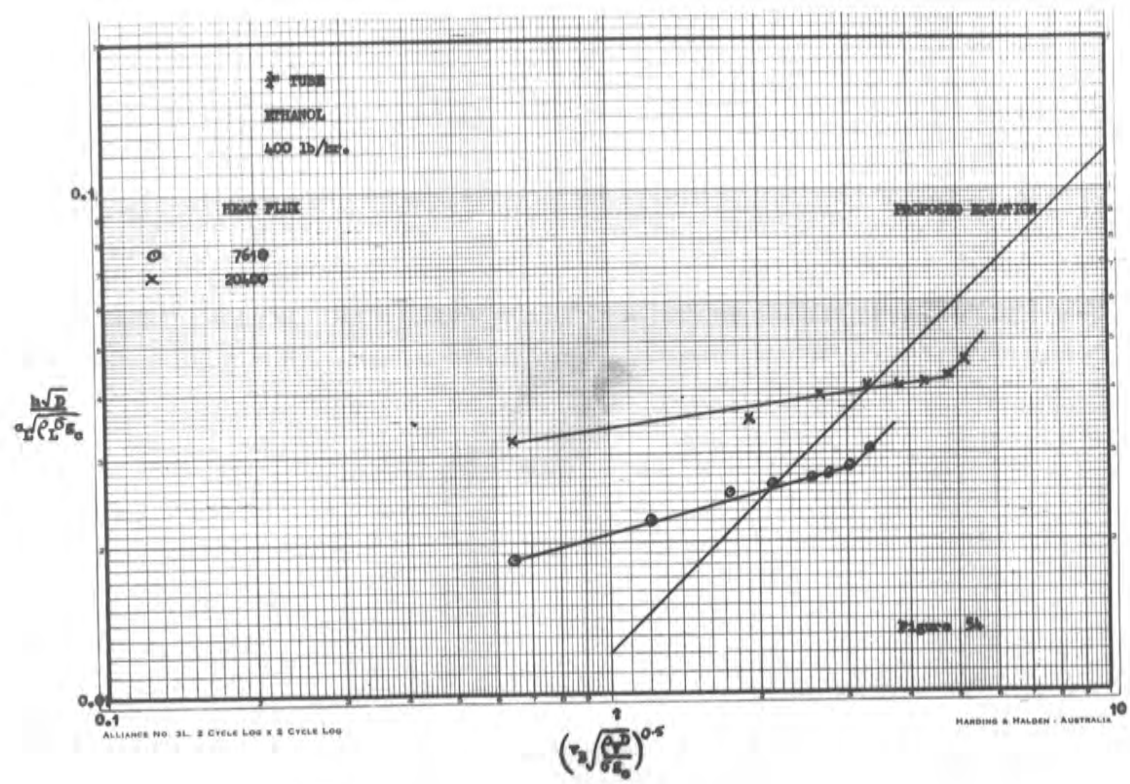


Figure 54

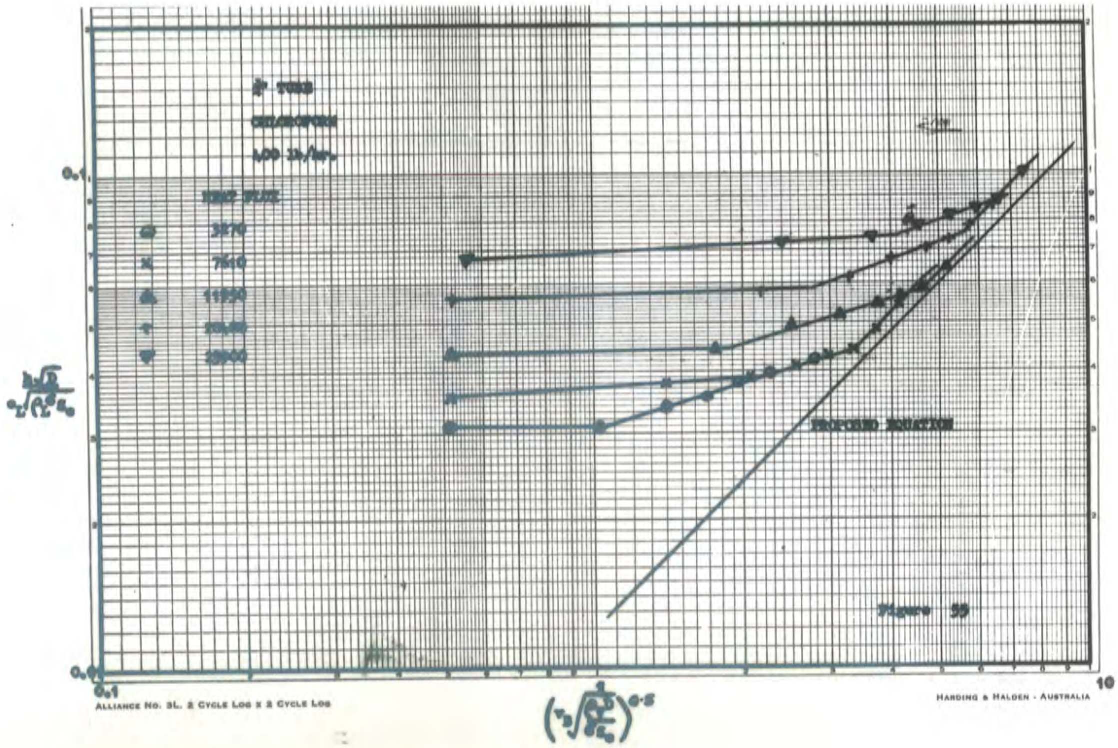


Figure 55

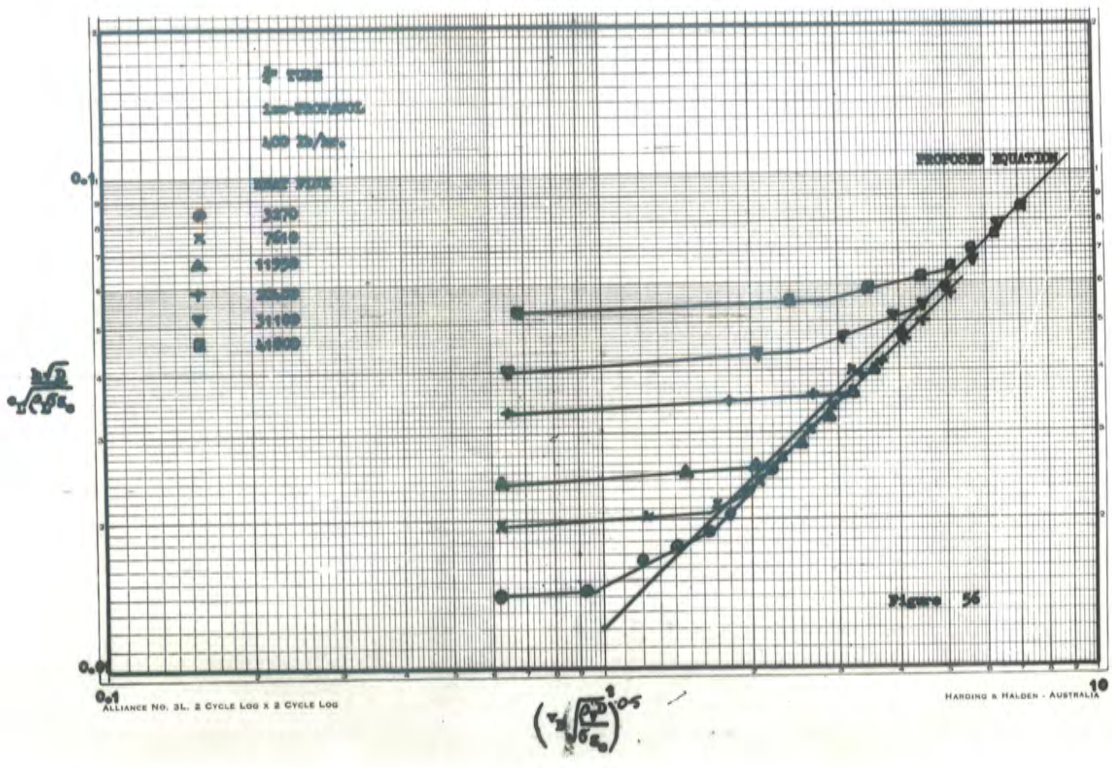


Figure 56

CHAPTER 6.

DISCUSSION OF RESULTS.

Variation of the heat transfer coefficient with tube length.

The graphs presented in figures 29 to 32 show the effect of feeding in vapour at the bottom of the tube. This experiment was done for the range of heat fluxes used throughout the work, for three of the four tube sizes and with three of the five liquids used.

All the graphs show that, within the experimental error, the value of the heat transfer coefficient depends solely on the value of the bulk velocity and is independent of the position in the tube where the readings were taken. Hence the length, or length-diameter ratio, is not a factor. Once climbing film conditions are reached, the point value of h can be calculated from the corresponding value of v_B and the physical properties of the fluid.

Comparison with other workers.

To measure the success of the equation derived for the climbing film, the data of other authors were examined to determine whether they fitted. Unfortunately very few papers have been published on the climbing film in sufficient detail to enable individual coefficients to be calculated up the tube.

The data from McNelly (23) were obtained from his graphs 38 to 44 where he was attempting to find the effect of length on the heat transfer coefficient. To do this he found the average value of "h" for the bottom 12" length of his tube, then for the bottom 24", the bottom 36", and so on. By working backwards it is possible to find the value of "h" for each 12" increment up the tube.

McNelly's experiments were carried out with water in a $\frac{1}{2}$" tube and the results are plotted in figure 57. It will be noted that in runs 41, 44, and possibly 43, dry wall conditions have occurred at the top, which McNelly would not have been able to see by his method of plotting. Presumably these and other such results have affected the value of his equation.

Good agreement has been found with these results as all McNelly's lines come within -20% of the proposed equation.

The only other workers who developed climbing film conditions and published enough data to enable their results to be used are Dengler and Addoms (52). They used a 20 ft. 1" diameter tube with water as the feed material. Steam condensing in jackets was the heating medium and the wall temperatures were read by thermocouples in the jackets. As has been discussed in Chapter 2, Turner (21) states that thermocouples thus placed cannot be expected to indicate correctly. Cameron (59) has found evidence that drop-wise condensation on the steam side may initiate nucleation on the boiling side. Thus even if nucleate

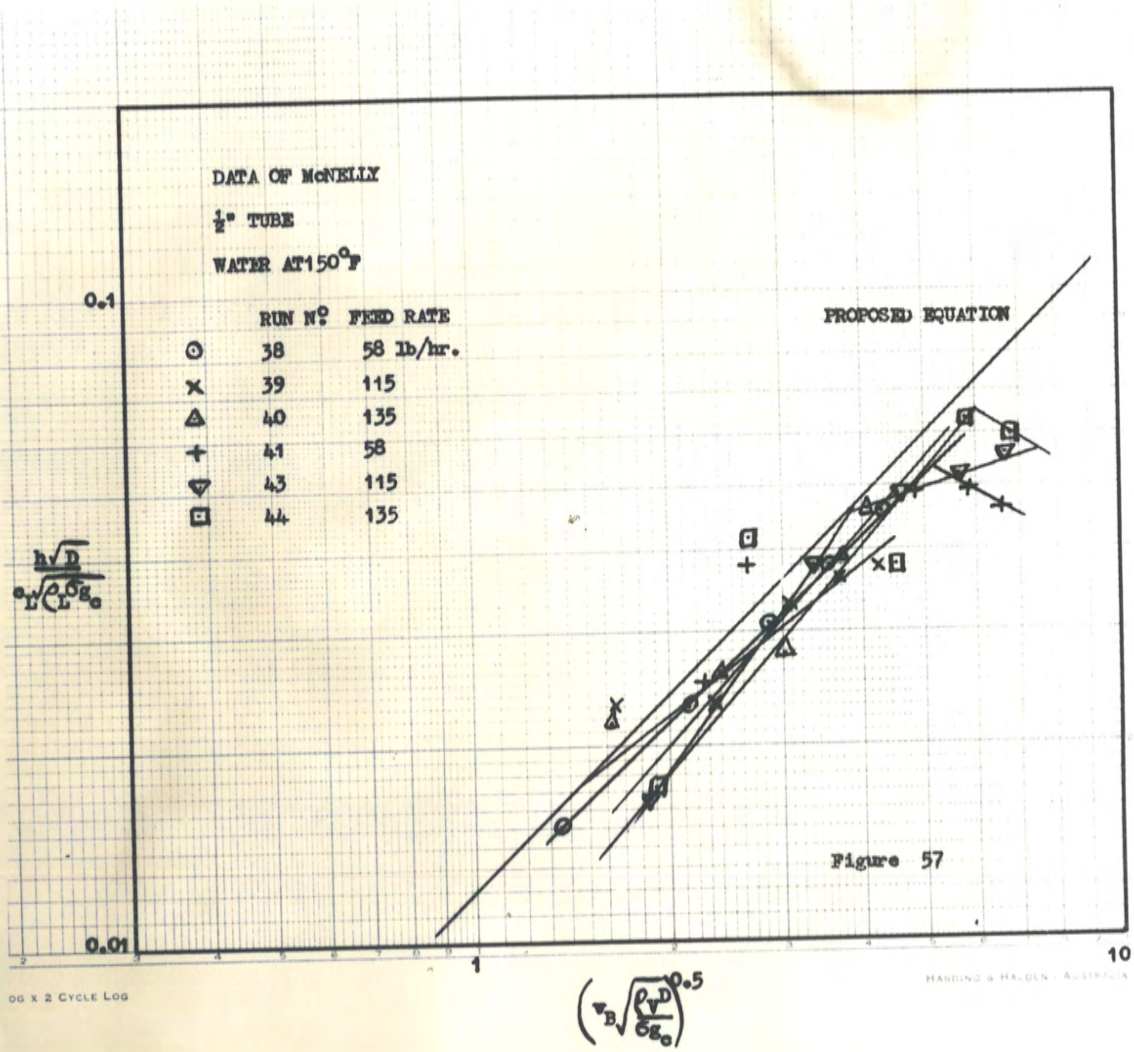

DATA OF McNELLY
½" TUBE
WATER AT 150°F
RUN Nº FEED RATE
38 58 lb/hr.
39 115
40 135
41 58
43 115
44 135
PROPOSED EQUATION
0.1
0.01
1
10
og x 2 Cycle Log
Harding & Halden · Australia

Figure 57

boiling is not the mechanism, it is possible that the heating may not be uniform up the tube. It will be noted from the graph in figure 58 that, although of the same shape and slope, the values obtained are approximately 100% high. The two sets of results have been obtained from small scale diagrams used to present other data on pages 97 and 98 of their paper, and the difficulties of interpretation from these may have added to the divergence.

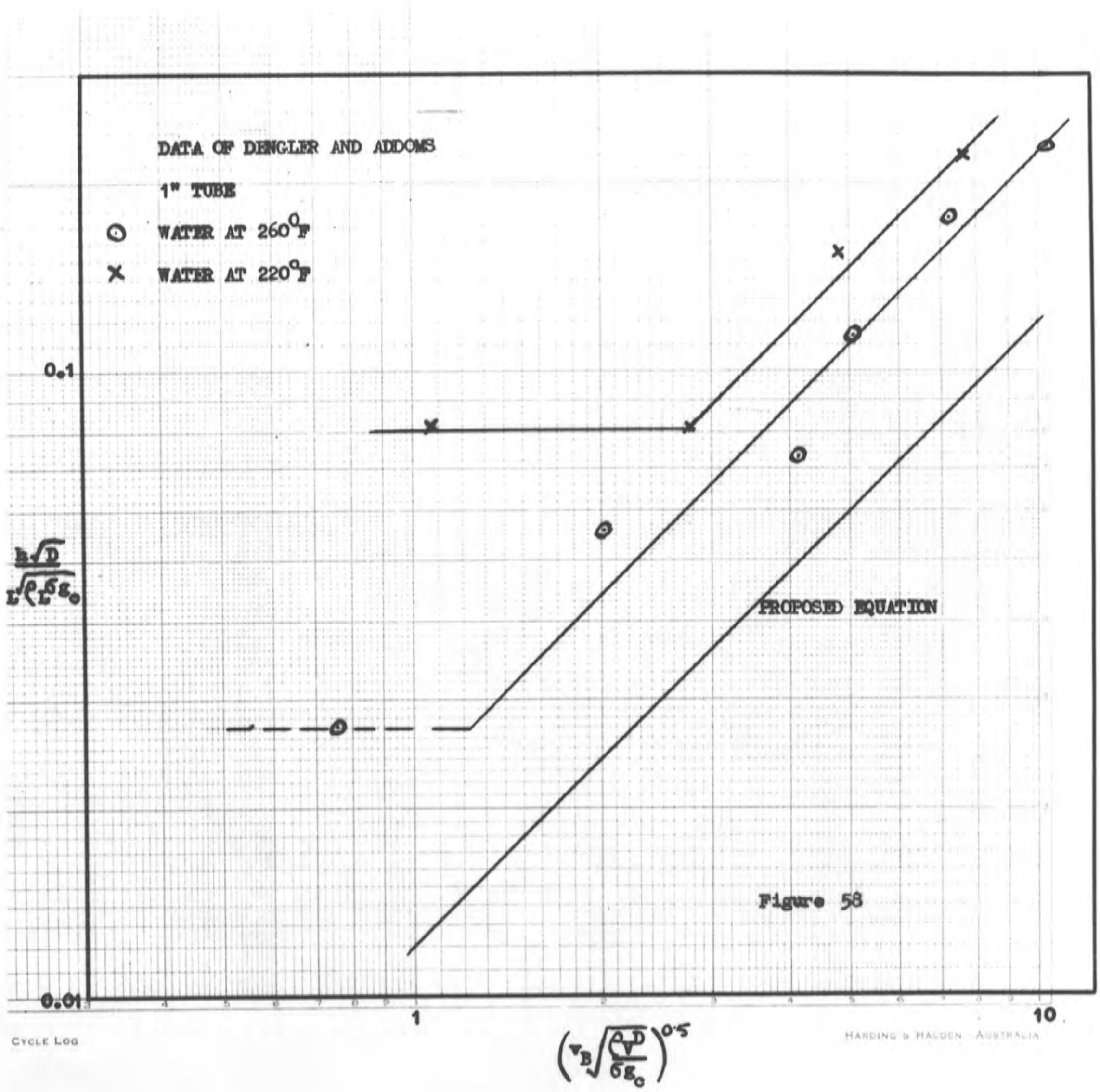
Comparison with the falling film evaporator.

It is apparent from the proposed equation that the heat transfer coefficient is dependent on the resistance of the liquid-vapour surface. All other effects, such as the diffusion of either heat or momentum in the liquid, or the effect of gravity, are so small as not to be noticed in comparison with the main resistance. In this case, once annular conditions have been established, it should matter little whether the tube is vertical, with the flow up or down, or horizontal, or at an angle.

Accordingly the data of Staker were examined (31) and the graphs in figures 59 and 60 were taken from his graphs on his pages 59 to 61. As can be seen, with the exception of one graph in each series, very close agreement is found.

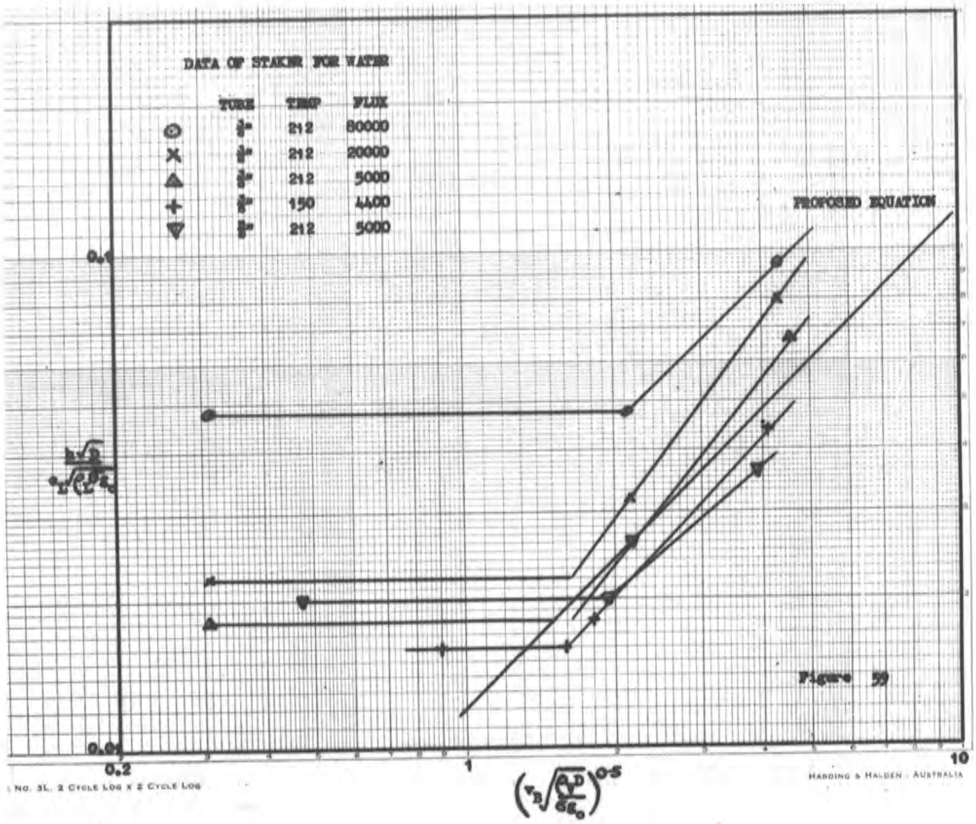
Comparison with horizontal tube evaporators.

By the same argument, the data of Bryan and Seigel (74) were examined. They evaporated Freon 11 (trichloromono-fluoromethane)

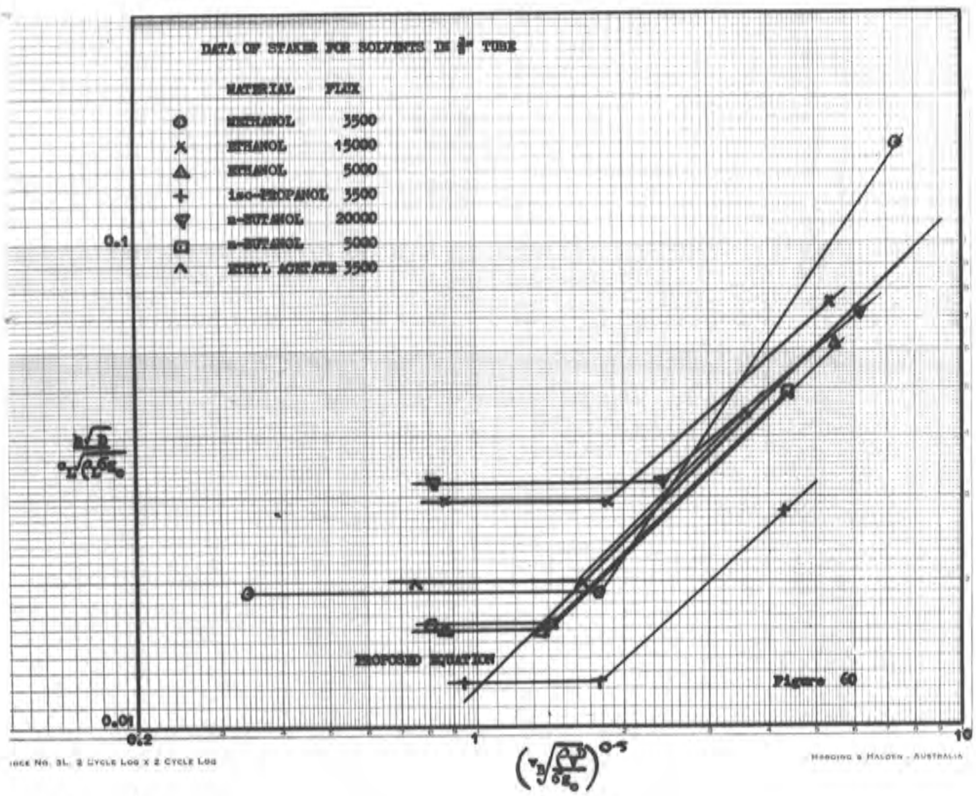


DATA OF DENGLER AND ADDOMS
1" TUBE
WATER AT 260°F
WATER AT 220°F
PROPOSED EQUATION
0.1
0.01
1
10
CYCLE LOG
HARDING & HALDEN AUSTRALIA

Figure 58



DATA OF STAKER FOR WATER
TUBE TEMP FLUX
212 80000
212 20000
212 5000
150 4400
212 5000
PROPOSED EQUATION
0.1
0.01
0.2
1
10
Figure 59
No. 3L. 2 Cycle Log x 2 Cycle Log
Harding & Halden - Australia



DATA OF STAKER FOR SOLVENTS IN 3/4" TUBE
MATERIAL FLUX
METHANOL 3500
ETHANOL 15000
ETHANOL 5000
iso-PROPANOL 3500
n-BUTANOL 20000
n-BUTANOL 5000
ETHYL ACETATE 3500
PROPOSED EQUATION
0.1
0.01
0.2
1
10
Figure 60
No. 3L. 2 Cycle Log x 2 Cycle Log
Harding & Halden - Australia

at a pressure of 22 psi in a 10 ft. tube 0.55" I.D. The results for their runs 1 and 14 have been recalculated and are plotted in figure 61. It will be noted that where annular flow has been established close agreement is found with the proposed equation.

These authors' data are also useful in that they give the values of "h" found for the bottom and the top of their horizontal tubes. The following table has been taken from run No. 14.

top h	Bottom h	% difference
400	355	11
360	340	5
350	320	9
350	290	17
360	290	19
380	340	11
410	360	12
460	400	13
		Average = 12%

It can be seen that the difference between the top and the bottom coefficients approximates to 12%. The greater heat transfer is at the top where presumably the force of gravity assists the action of the vapour in tearing the liquid from the surface, thus reducing the resistance of the surface to heat transfer. The change in heat transfer where gravity is a factor compared with the cases where it is not will be of the order of 6% which is not very significant.

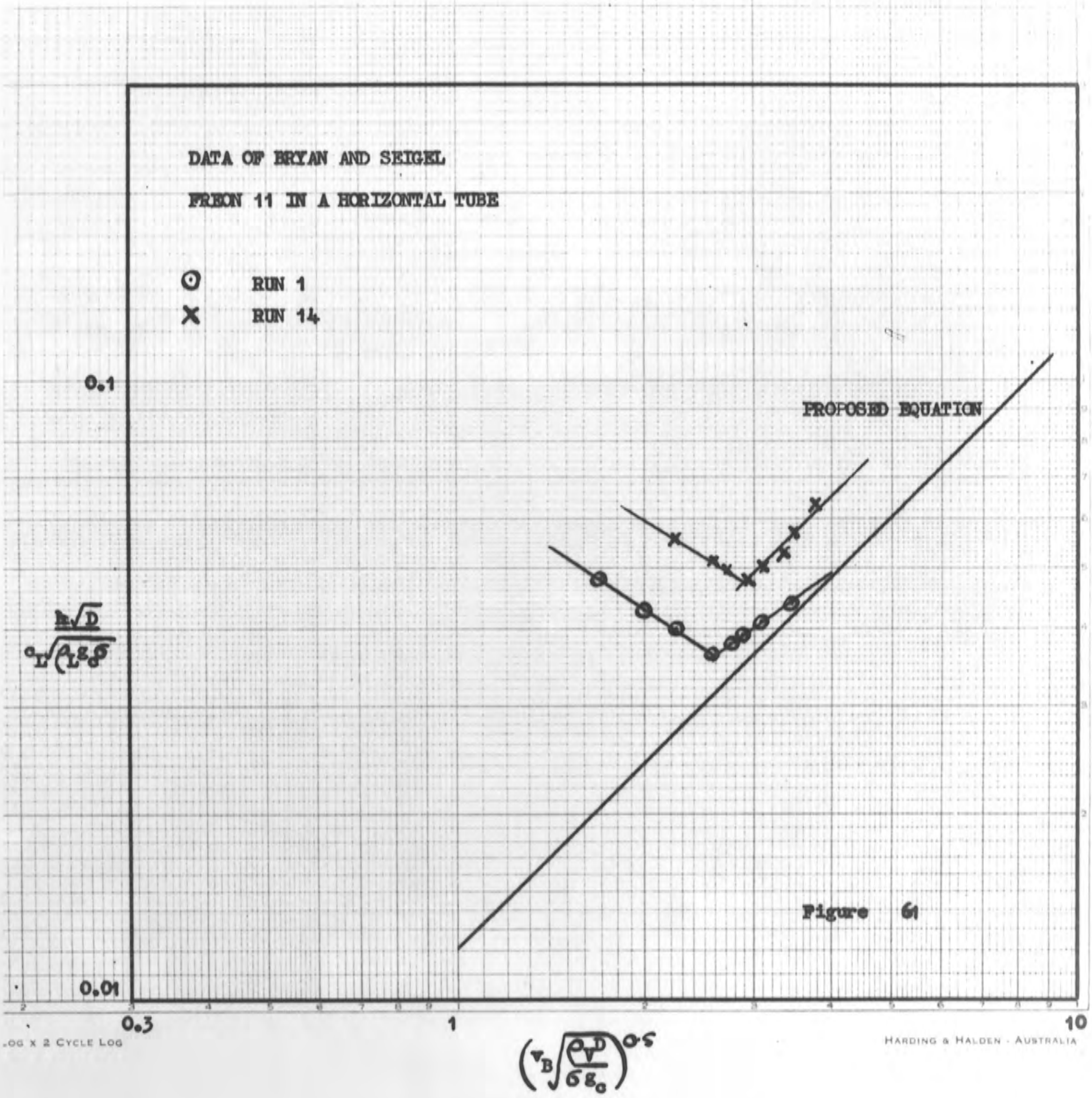

DATA OF BRYAN AND SEIGEL
FREON 11 IN A HORIZONTAL TUBE
RUN 1
RUN 14
PROPOSED EQUATION
0.1
0.01
0.3
1
10
LOG X 2 CYCLE LOG
HARDING & HALDEN - AUSTRALIA

Figure 61

Comparison with concentric tube evaporators.

The data of Bennett, Collier, Pratt and Thornton (48) were also examined. These authors used an electrically heated central tube inside an outer tube. When a liquid is vaporised inside an annulus it is probable that when discrete flow paths of liquid and vapour are established the mechanism will be the same as for the climbing film. The vapour-liquid interface will still be the limiting resistance and the dimension which must be examined for "D" in the proposed equation is that of the diameter of the inside tube. Now this diameter represents an outward curving surface instead of the inward curving one as found in the climbing film evaporator and used in the derivation of the formula. It is possible to imagine that this surface of the liquid would not be as stable as the surface which is formed in the climbing film evaporator. As mass transfer from the surface would thereby be facilitated it seems likely that the heat transfer from the inside tube of an annulus would be higher than found in the climbing film. To confirm this the following runs were examined from the paper of Bennett et al.

Run	Central tube O.D.	Outer tube I.D.	Heat flux
A 8	0.375	0.551	126800
A 9	0.375	0.551	126800
C 66	0.625	0.866	126800
C 59	0.625	0.866	121300
C 61	0.625	0.866	63400
C 63	0.625	0.866	31700

The results have been recalculated and are shown in figure 62.

As can be seen, the graphs have the same pattern as those for the climbing film, but they are on the average 50% higher than the proposed equation for the climbing film. It will also be noted that the divergence from the proposed equation increases as the heat flux increases, being 30% for run C63, 40% for run C61, and approximately 60% for the others. Bennett et al measured their wall temperatures by allowing a central thermocouple to come into radiation equilibrium with the inner surface of the inner tube. Although the precautions which the authors describe seem adequate, if any inaccuracies did occur in their method of temperature measurement, then this would explain the increasing divergence with increase in heat flux.

It seems clear that the same mechanism occurs in an annular evaporator as in a climbing film evaporator when both have developed discrete flow of liquid and vapour. Unfortunately there is insufficient evidence available to state whether the divergence from the proposed equation is because the equation should have a different constant or whether the same equation should be used and the difference is because of experimental error. From the argument at the beginning of this section it seems probable that the same equation should be used but with a different constant.

Comparison with spray evaporators.

When the climbing film is established, fully turbulent conditions are developed, not only in the vapour space but, we may assume, in the

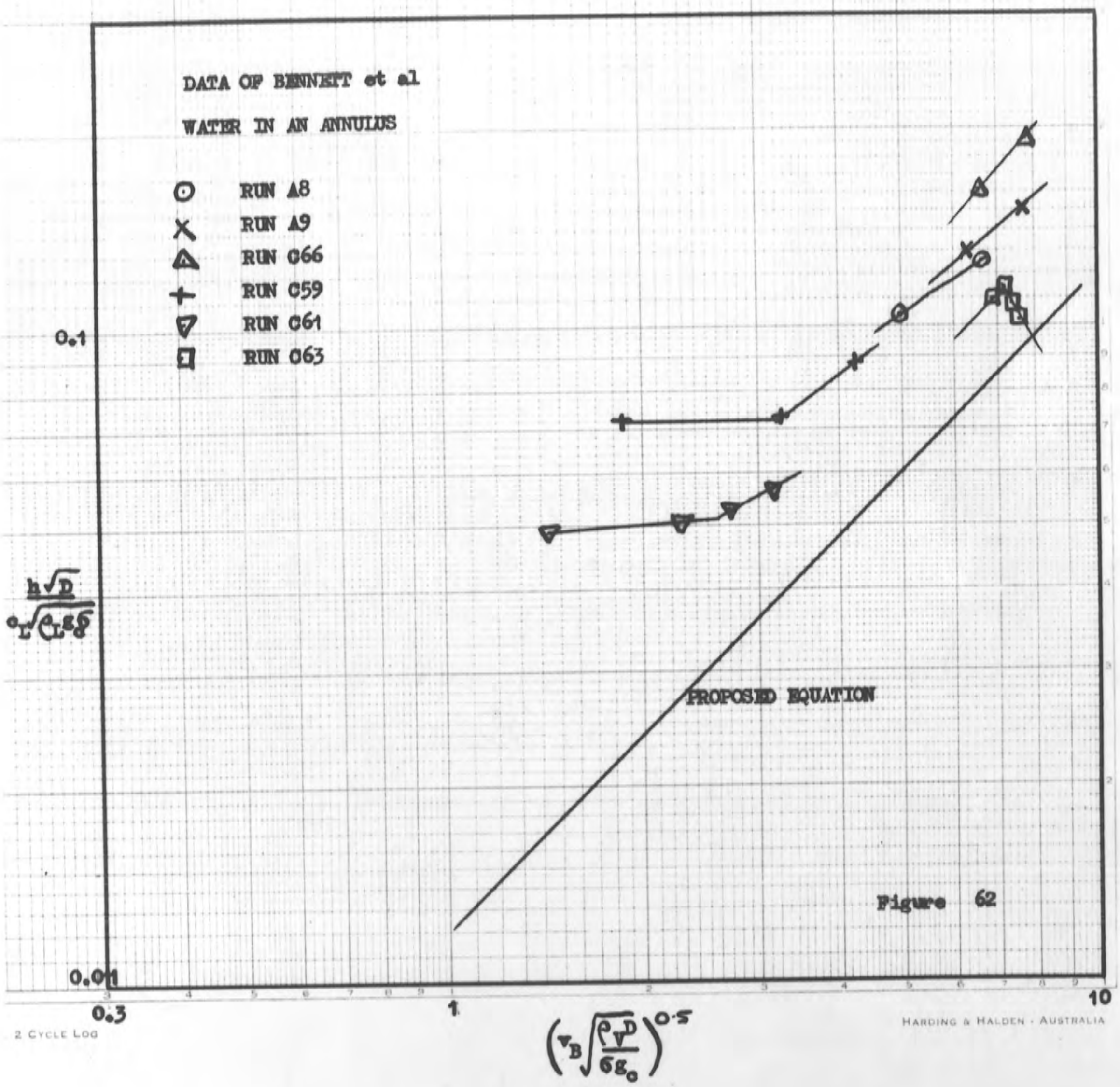

DATA OF BENNETT et al
WATER IN AN ANNULUS
RUN A8
RUN A9
RUN C66
RUN C59
RUN C61
RUN C63
0.1
0.01
PROPOSED EQUATION
0.3
1
10
$\frac{h\sqrt{D}}{c_L\sqrt{\rho_L g \delta}}$
$\left(v_B\sqrt{\frac{\rho_V D}{\sigma g_o}}\right)^{0.5}$
2 CYCLE LOG
HARDING & HALDEN - AUSTRALIA

Figure 62

liquid film as well. The photographic studies by Cathro (6) show the surface of the film with vigorous waves developed, and we can also deduce the presence of waves from the profiles found when measuring film thicknesses (appendix 8). It can also be observed from the graphs presented in chapter 4 that the film thickness is not a factor in the heat transferred. Hence it is not entirely unexpected when the final correlation does not include terms for thermal or momentum diffusion through the film. We can conclude that the major resistance to heat transfer is at the liquid-vapour interface.

The nature of the reaction between the vapour and the liquid film is then of some major interest. It can be seen from Appendix 8 that the film thickness increases by about 20% for each increase in feed rate, whereas the increase in feed rate is of the order of 60%. It was also noted in certain runs with the ½" tube that it was not possible to to find the point of exact boiling by the methods described in Chapter 4 which up till then had been satisfactory. It was found that feed rates greater than 750 lb/hr could not be used at all, but for the maximum heat fluxes a feed rate of only 440 lb/hr was possible. The only reason which can satisfy both these points is that as the film tends to become thicker because the feed rate is increased, so the vapour tears from the surface more liquid and carries it as drops suspended in the stream.

At the higher feed rates the vapour is carrying such a mass of liquid that any increase in the quantity of liquid in the system increases the pressure drop rapidly. This in turn causes a boiling point elevation. When the preheat increase causes a boiling point elevation which approaches the rise in temperature of the feed, then the tube has reached its limiting condition.

From this it can be suggested that the main effect of the vapour velocity is to entrain liquid from the surface, and it seems reasonable to suggest that this liquid carries heat from the surface into the vapour stream. It seems clear that the transfer of mass is not caused by evaporation entraining liquid into the vapour, for if this be the mechanism, it would be reasonable to expect that the heat transfer is dependent on the heat flux. However, it can be seen from the results that the heat transfer coefficient is solely a function of bulk velocity for any one tube and liquid, and not dependent on heat flux at all.

This limitation of the transfer of heat by the action of the vapour on the surface has been confirmed by the results which have been calculated from the data obtained in falling film evaporators, horizontal tube evaporators, and in the concentric tube evaporators. It would seem that any method by which the surface could be more disturbed would give high values of heat transfer coefficients.

Pratt and Thornton (75) have described how they sprayed water into a concentric tube evaporator through which steam was passing at atmospheric pressure. Examples of the results of their work which they presented graphically, have been recalculated and are shown in figure 63. As can be seen the points fall on a line which is almost identical to that obtained with the concentric tube evaporator without spraying.

Thus it can be concluded that the essential part of the mechanism is the removal of drops from the liquid film as a spray and not the presence of an introduced spray.

To give some indication of the spread of the results obtained from other workers, the graphs presented in figures 57 to 61 have been reproduced in figure 64. Only the sections showing the climbing film have been drawn. The graphs marked 1 and 2 are those of Dengler and Addoms which can be discounted in view of the doubtfulness of their method of measuring their temperatures, and those marked 3 and 4 are two of Staker's results. With the exception of these, it will be noted that close agreement has been obtained and that the results are distributed on either side of the line of the proposed equation.

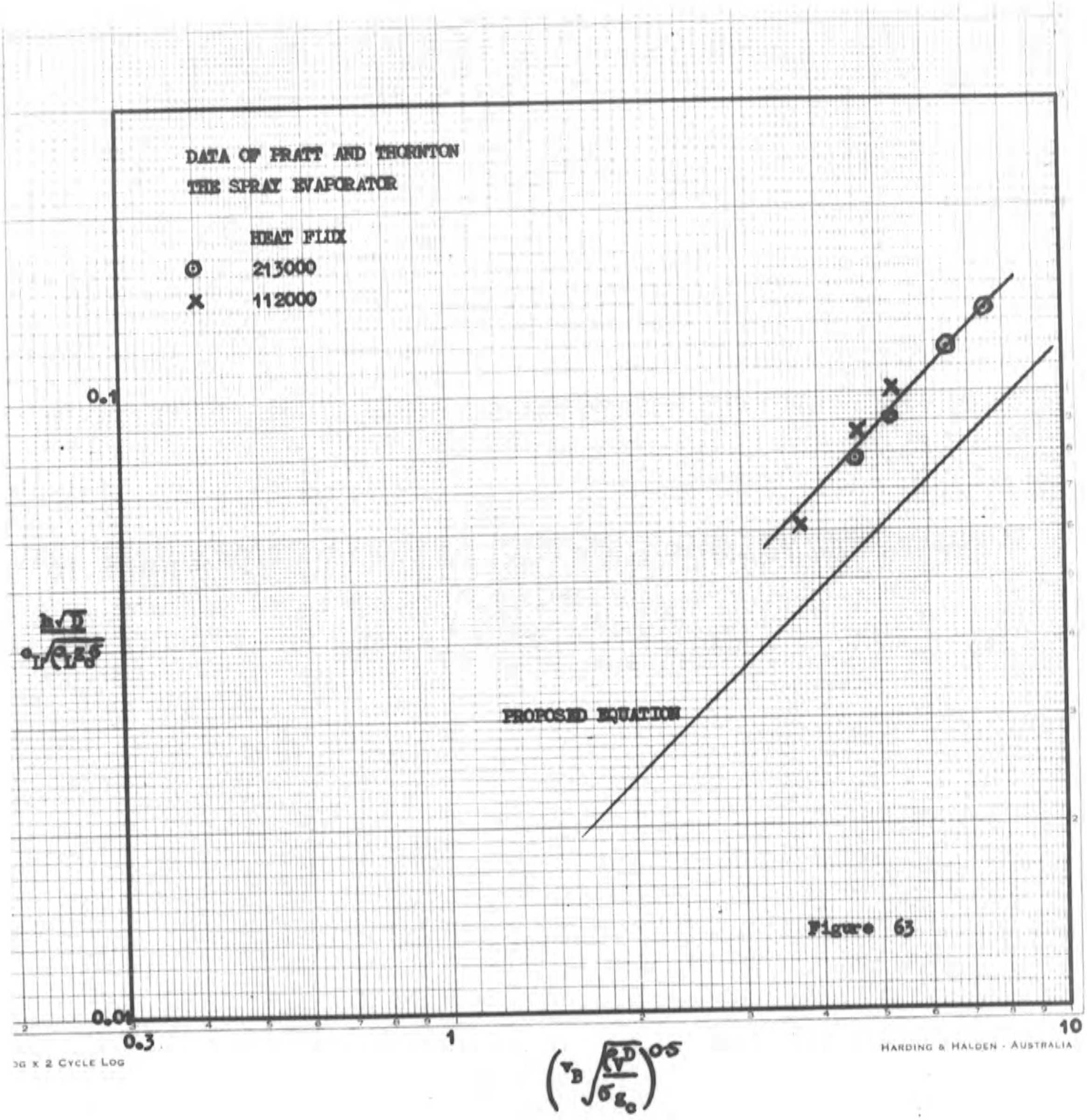
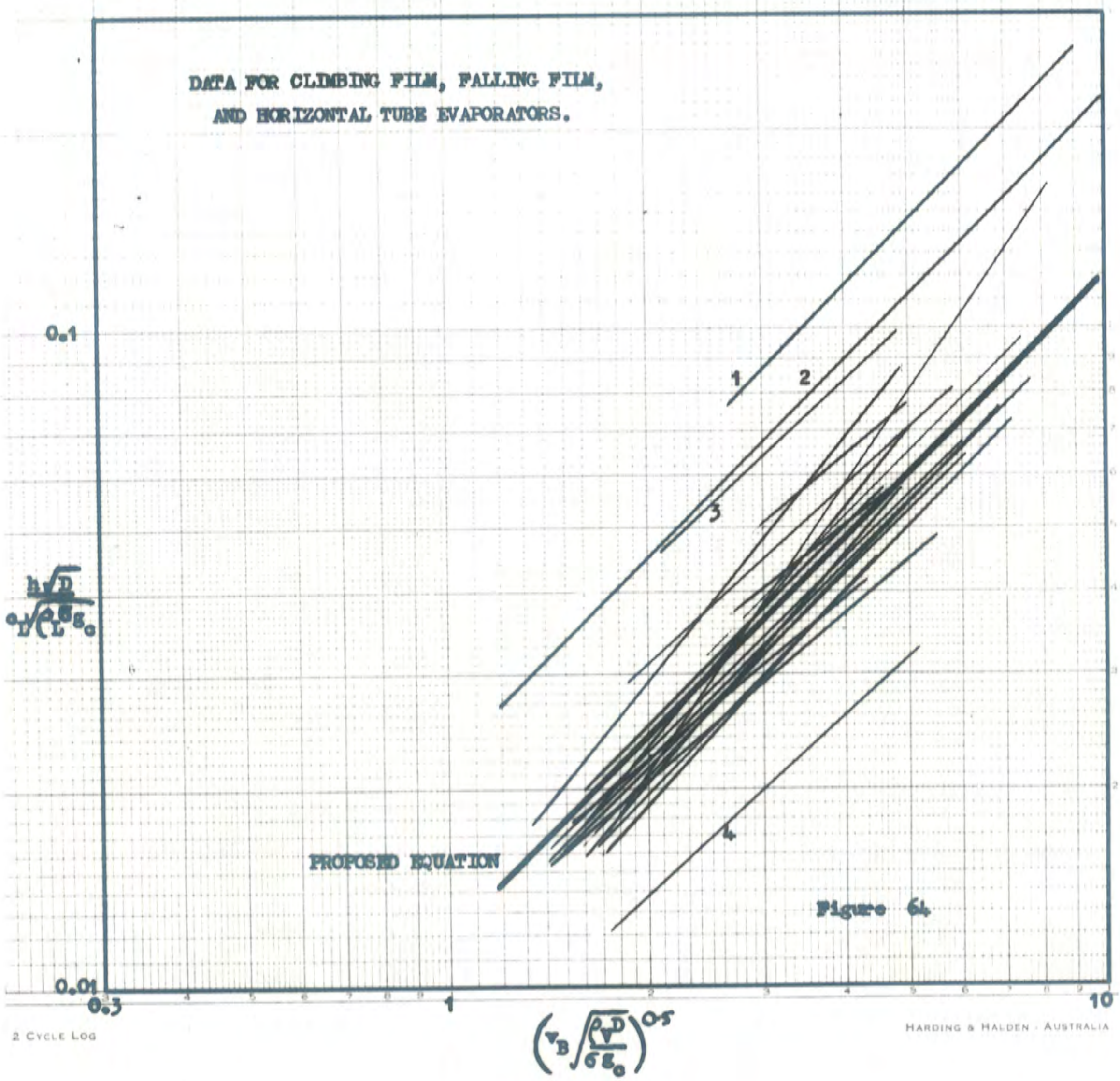

DATA OF PRATT AND THORNTON
THE SPRAY EVAPORATOR
HEAT FLUX
213000
112000
0.1
0.01
PROPOSED EQUATION
0.3
1
10
HARDING & HALDEN - AUSTRALIA

Figure 63



DATA FOR CLIMBING FILM, FALLING FILM,
AND HORIZONTAL TUBE EVAPORATORS.
1
2
3
4
PROPOSED EQUATION
0.1
0.01
0.3
1
10
$\frac{h\sqrt{D}}{c_L\sqrt{\frac{\sigma}{\rho_L g_c}}}$
$\left(v_B\sqrt{\frac{\rho_V D}{\sigma g_c}}\right)^{0.5}$
2 CYCLE LOG
HARDING & HALDEN - AUSTRALIA

Figure 64

CHAPTER 7

CONCLUSIONS.

The work which has been described in Chapter 6 has shown that when climbing film conditions have been developed the value of the heat transfer coefficient is dependent on the bulk velocity for any one tube and liquid. This value is independent of the point of measurement in the tube or the length of the tube itself. Hence the length or length-diameter ratio of the tube is not a factor and should not enter into any correlation for point conditions.

The mechanism of heat transfer in the climbing film has been described in Chapter 6. The main resistance occurs at the vapour-liquid interface and depends on the rate at which the vapour can transfer liquid from the surface into the fluid stream. Hence, once climbing film conditions have been developed, there is no significant difference between heat transfer coefficients which are obtained when the film is climbing or falling, or when the tube is vertical or horizontal. The order of the change which would be observed when gravity is a factor compared with when it is not has been shown to be about 6%.

The following equation was developed in Chapter 5 and correlates climbing film conditions:-

$$\frac{h\,D}{c_L \rho_L \sigma E_c} = 0.012\, v_B \left\{ \frac{\rho_V D}{\sigma E_c} \right\}^{0.5}$$

and it has been shown in chapter 6 that it represents conditions in the falling film evaporator and the horizontal tube evaporator as well. The work of several other authors has been recalculated and plotted against the proposed equation. With the exception of one group of workers very good agreement has been found.

By means of the equation it is possible to calculate the working conditions which will be obtained when a liquid is being evaporated in a tube where annular flow has been developed. If the heat flux is constant, then the vapour or bulk velocity can be easily calculated, the heat transfer coefficient found from the equation, and the temperature difference which will be required for each point along the tube evaluated. If the temperature difference is a constant, then a simple integration must be made to find the heat input for the length of the tube where annular flow is occurring.

It has also been shown in Chapter 6 that the same mechanism operates in a concentric tube evaporator, except that values of the heat transfer coefficient which will be obtained are 50% higher than those obtained in a simple tube. It was also shown that when the feed is introduced as a spray the same equation is obtained in the concentric tube evaporator as when the feed is introduced just boiling. It was concluded that the spray is of significance only when it is formed by the vapour transferring liquid from the heated film.

It was shown in appendix 7 that the conditions at the inlet to the tube, where boiling is just starting, can be correlated by two equations.

$$\frac{hD}{k} = 0.00021 \left\{\frac{\Delta T_c}{\lambda}\right\}^{0.5} \left\{\frac{\sqrt{\rho \sigma g_c D}}{\mu}\right\}^{0.4} \left\{\frac{D\rho\lambda J}{\sigma}\right\}^{0.5}$$

$$\frac{hD}{k} = 1.54 \times 10^{-18} \left\{\frac{\Delta T_c}{\lambda}\right\}^{2} \left\{\frac{\sqrt{\rho \sigma g_c D}}{\mu}\right\}^{1.6} \left\{\frac{D\rho\lambda J}{\sigma}\right\}^{2}$$

The first of these equations must be used when the value of the Nusselt group is less than 130 and the second when the calculated value exceeds 130. These two equations represent different types of boiling. Although the shape of the graphs indicates that the two lines represent forced convective boiling and nucleate boiling there was no strong evidence that there was any change in the convective boiling coefficients with change in feed rate. Hence it cannot be claimed that in fact these two graphs actually represent these types of boiling.

Insufficient data were available to enable any correlation to be made of the behaviour of the film thickness with any variable. It is shown in Appendix 8 and Chapter 6 that film thickness does not increase proportionally with the feed rate, which leads to the conclusion, supported by other evidence described in Chapter 6, that the vapour stream carries a greater part of any increase in feed rate.

It was noted that the graph of the film thickness plotted against bulk velocity generally had a slope of $-4/3$ for the climbing film and $-\frac{1}{2}$ for the region of slug type boiling.

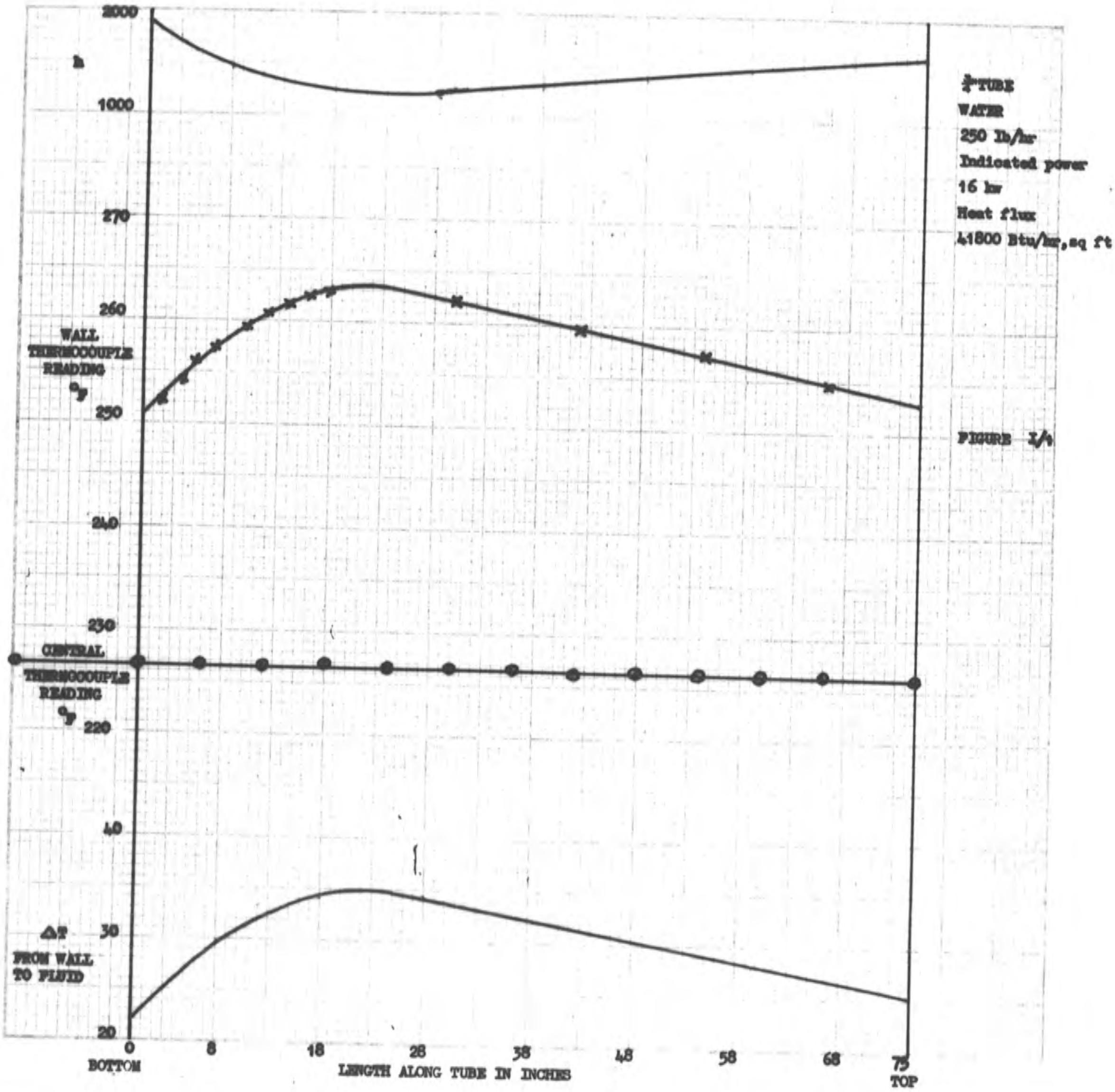
APPENDIX 1.

Typical calculations to find v_B and h.

$\frac{3}{4}$" tube. 250 lb/hr feed rate of water. 16 kw indicated power. The readings as taken are shown graphically in Figure 1/1. A point 18" from the bottom will be taken.

Calculation for v_B.

Heat input to apparatus (indicated)	= 16 kw
Heat to winding (from calibration curve)	= 51,400 Btu/hr
Ambient temperature	= 82°F
Average tube wall temperature	= 260°F
Temperature difference	= 178 F°
Area of lagging	= 4.87 sq ft
Combined coefficient	= 0.27 Btu/hr,sq ft,F°
Heat lost through lagging	= 0.27 x 4.87 x 178
	= 234 Btu/hr
Hence, heat to tube	= 51,400 - 234
	= 51,200 Btu/hr
Total winding length	= 75"
Heat input to 18"	= $\frac{18}{75}$ x 51,200



2000
h
1000
270
260
WALL THERMOCOUPLE READING °F
250
240
230
CENTRAL THERMOCOUPLE READING °F
220
40
30
ΔT FROM WALL TO FLUID
20
0
8
18
28
38
48
58
68
75
BOTTOM
TOP
LENGTH ALONG TUBE IN INCHES
¾" TUBE
WATER
250 lb/hr
Indicated power
16 kw
Heat flux
41800 Btu/hr,sq ft

FIGURE I/4

	$= 12{,}250$ Btu/hr
Temperature of fluid at this point	$= 226.8\ ^{\circ}F$
Latent heat at this temperature	$= 961$ Btu/lb (67)
Water evaporated	$= \frac{12{,}250}{961}$
	$= 12.75$ lb/hr
Specific volume of vapour at 226.8°F	$= 20.3$ cu ft/lb
Hence volume of vapour formed	$= 12.75 \times 20.3$
	$= 258$ cu ft/hr
Liquid left	$= 250 - 12.8$
	$= 237$ lb/hr
Density of water at 226.8°F	$= 59.4$ lb/cu ft (68)
Volume of water	$= \frac{237}{59.4}$
	$= 4$ cu ft/hr
Volume of vapour and liquid	$= 258 + 4$
	$= 262$ cu ft/hr
Internal diameter of the tube	$= 0.747''$
Cross sectional area of the tube	$= \frac{\pi \times (0.747)^2}{4 \times 144}$
	$= 0.00304$ sq ft
Hence bulk velocity, v_B	$= \frac{262}{3600} \times \frac{1}{0.00304}$
	$= 24.1$ ft/sec.

<u>Calculation for "h".</u>

Heat to tube	= 51,200 Btu/hr
Outside diameter of tube	= 1.006"
Inside diameter of tube	= 0.747"
Mean diameter	= 0.877"
Thickness of wall	= 0.130"
Thermal conductivity of copper at 260°F	= 217 Btu/(hr)(sq ft)(F°/ft) (69)
Temperature drop across the wall	$= \dfrac{51{,}200 \times 0.130 \times 12}{217 \times \pi \times 0.877 \times 75}$
	= 1.8 F° say.
Temperature of outside of wall	= 263.2°F
Temperature of inside of wall	= 263.2 - 1.8
	= 261.4°F
Temperature of fluid	= 226.8°F
Hence ΔT	= 34.6 F°
Heat flux based on inside area	$= \dfrac{51{,}200 \times 144}{\pi \times 0.747 \times 75}$
	= 41,800 Btu/hr, sq ft
Hence heat transfer coefficient, h	$= \dfrac{41{,}800}{34.6}$
	= 1210 Btu/(hr)(sq ft)(F°)

APPENDIX 2.

Typical calculation to find Film Thickness.

A typical result for Methanol in the $\frac{3}{4}$" tube with a feed rate of 400 lb/hr and a heat flux of 20400 Btu/(hr)(sq ft) on the tube is shown in graph 2/1.

Volume of rotation of concentration profile

$$= \pi(3^2-1.6^2)0.2$$
$$+(2.45-1.6)\frac{\pi}{3}(0.52-0.2)(1.6+2\times2.45)$$
$$+(2.88-2.45)\frac{\pi}{3}(2.0-0.52)(2.45+2\times2.88)$$
$$+\pi(2.88^2-2.45^2)(0.52-0.2)$$
$$+\pi(3^2-2.88^2)(2-0.2)$$
$$= 4.04+1.85+5.48+2.31+4.00$$
$$= 17.68$$

If the film thickness $= t$

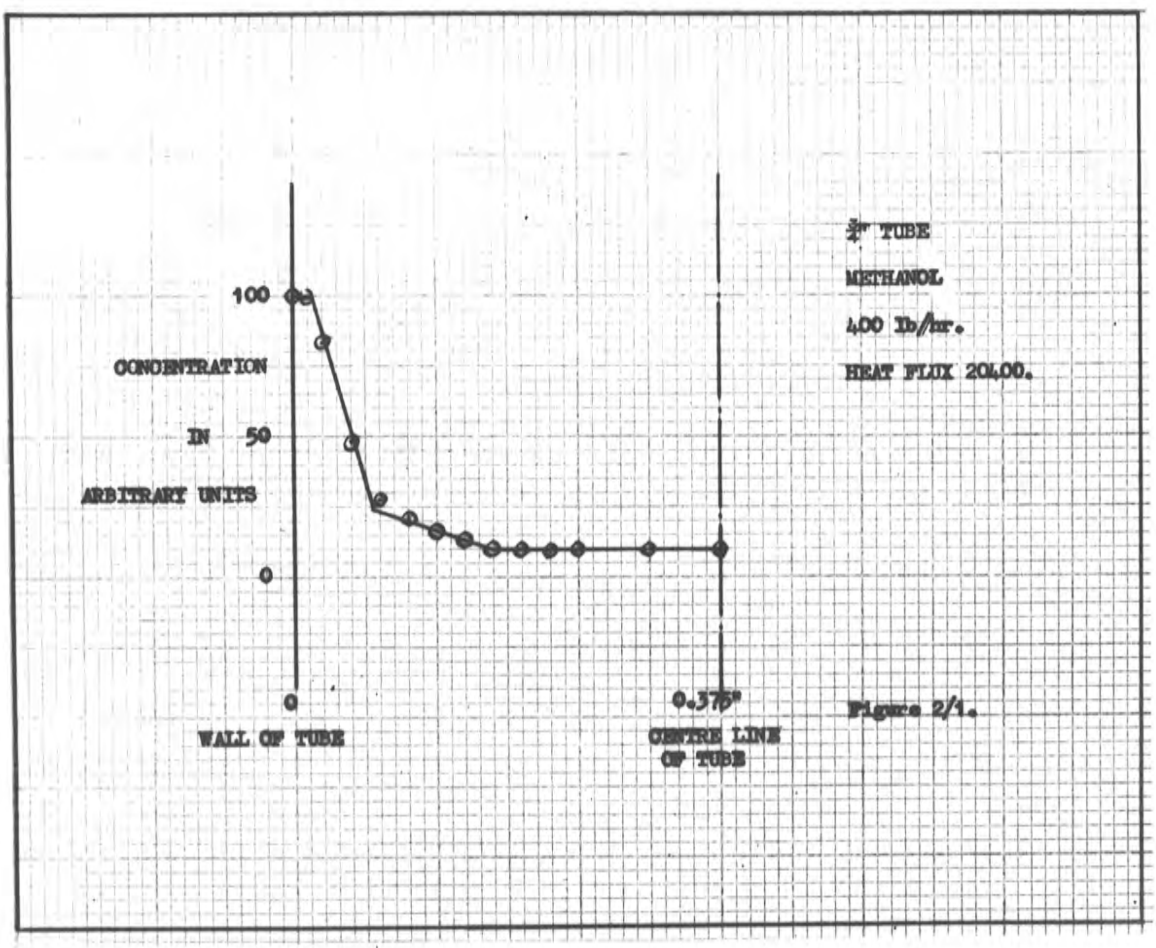
then $3\pi(9-(3-t)^2) = 17.68$

or $t = 3 - \sqrt{9-\frac{17.68}{2\pi}}$

$= 0.51$

But this is a scale dimension, where the scale 1" = 0.125" actual.

Hence actual thickness = 0.064"



¾" TUBE
METHANOL
400 lb/hr.
HEAT FLUX 20400.
100
CONCENTRATION
IN 50
ARBITRARY UNITS
0
0
0.375"
WALL OF TUBE
CENTRE LINE
OF TUBE

Figure 2/1.

APPENDIX 3.

Tabulation of results

The results given below under the heading "conditions up the tube" are in five rows as follows:

row 1: Temperature of the tube wall thermocouples, T_{w_o},

row 2: Temperature of the fluid, T_B,

row 3: Temperature difference between the tube surface and the fluid, ΔT.

This is the temperature read by the tube wall thermocouples minus the temperature drop across the wall, minus the fluid temperature,

row 4: Heat transfer coefficient, h,

row 5: Bulk velocity, v_B.

As many more readings were taken than would be practical to record, and as dissimilar increments were used for the tube wall and fluid temperature readings, arbitrary distances have been used and the values at these distances taken from a smooth graph drawn through the points. Because of this smoothing, it will be noted that in some cases below, ΔT as reported may be greater than $T_{w_o} - T_B$.

$\frac{3}{8}$" tube.   Feed rate 150 lb/hr.   Material - water.

Conditions up the tube.

Heat flux		0"	8"	18"	28"	38"	48"	58"	72"
6800	T_{W_O}	229.9	229.0	228.1	227.1	226.0	225.0	223.9	222.3
	T_B	220.5	219.9	219.3	218.8	218.2	217.8	217.0	216.2
	ΔT	9.2	9.0	8.9	8.2	7.8	7.2	6.8	6.1
	h	790	800	800	820	910	980	1060	1160
	v_B	0.9	4.85	9.2	14.8	20	25	31	39
16150		237.9	236.1	235.0	233.1	231.2	229.4	227.6	225.0
		223.2	222.5	221.6	220.7	219.9	218.8	217.9	216.3
		14.0	13.9	13.1	12.2	11.3	10.5	9.8	8.4
		1150	1180	1200	1280	1390	1520	1690	1960
		0.9	9.6	21	32	43	55	68	91
25100		245.1	243.8	241.8	239.2	236.5	234.0	231.2	228.7
		225.6	224.7	223.3	222.2	221.0	219.8	218.7	217.0
		19.0	18.9	17.8	16.2	14.9	13.3	12.0	10.0
		1320	1350	1400	1500	1620	1820	2080	2510
		0.9	13.5	29.6	46	63	81	102	133
33700		251.8	252.3	252.2	248.2	244.2	240.7	237.6	234.8
		229.9	229.9	228.8	227.8	226.2	224.3	222.5	219.9
		21.0	21.7	22.5	25.0	17.3	15.6	14.3	14.1
		1620	1550	1500	1700	1920	2150	2380	2400
		0.9	16.9	37.6	58	82.5	107	133	173

Heat flux.		0"	8"	18"	28"	38"	48"	58"	72"
42300	T_{W_o}	257.1	259.2	259.1	254.8	250.2	246.5	243.2	240.0
	T_B	232.9	232.8	232.8	230.4	228.8	227.0	225.7	221.7
	ΔT	23.2	25.6	28.0	23.8	20.7	18.3	17.1	17.1
	h	1820	1690	1550	1780	1990	2200	2320	2320
	v_B	0.9	20.1	45	71	99.5	130	162	210

$\frac{3}{8}$" Tube. Feed rate 250 lb/hr. Material - water.

6800	228.7	229.9	229.9	228.4	227.0	225.8	224.5	223.0
	223.7	223.0	222.2	221.7	220.8	219.9	218.9	217.6
	5.2	6.8	7.2	6.8	6.2	6.0	5.7	5.2
	1310	1060	930	970	1060	1130	1190	1280
	1.52	5.1	9.8	14.9	19.5	24.6	30	37.7
11400	234.2	236.8	234.9	233.0	231.0	229.2	227.6	225.2
	226.1	225.8	225.2	224.3	223.3	222.2	220.8	218.5
	8.0	10.8	9.5	8.6	7.7	7.2	7.0	6.8
	1420	1100	1190	1320	1440	1530	1620	1720
	1.52	7.4	14.7	22.3	30.8	37.7	47.7	61.5
25100	246.8	246.9	245.5	242.6	239.8	236.9	234.1	231.0
	229.2	229.0	228.8	227.1	225.8	224.2	222.8	220.6
	16.8	17.0	16.5	14.9	13.3	12.2	10.9	10.0
	1470	1410	1470	1620	1830	2010	2210	2490
	1.52	13.4	28.6	44.5	60.5	79	98.5	128

Heat flux.		0"	8"	18"	28"	38"	48"	58"	72"
33700	T_{w_o}	258.1	256.8	255.0	251.6	248.2	245.5	243.4	241.7
	T_B	237.8	237.0	236.5	235.2	233.5	231.8	230.0	227.8
	ΔT	19.8	18.9	17.8	16.0	14.0	13.2	12.9	13.2
	h	1710	1720	1890	2120	2400	2580	2640	2590
	v_B	1.52	15.6	33.7	53.2	73.5	95	121	164
42300		263.2	263.8	261.5	258.2	255.0	252.0	249.9	247.9
		241.1	240.8	240.2	239.0	237.9	235.0	231.9	226.8
		21.0	21.6	20.5	18.3	16.3	15.0	17.0	21.5
		2270	2160	2230	2500	2700	2790	2480	1980
		1.52	17.8	39.8	61	85.5	121	141	200

$\frac{3}{8}$" tube. Feed rate 400 lb/hr. Material - water.

6800	238.0	237.3	236.6	234.7	232.3	230.2	228.2	225.9
	232.2	231.9	231.1	230.0	228.0	226.1	224.2	221.6
	5.6	5.2	5.0	4.7	4.3	4.0	3.8	3.7
	1250	1280	1320	1420	1530	1640	1780	1820
	2.44	5.5	9.5	13.7	18.3	23.2	28.2	36.2
16150	245.8	246.3	246.0	243.0	239.7	236.5	233.2	228.9
	234.5	234.1	233.7	232.5	230.5	228.4	226.2	222.8
	10.7	13.1	15.2	14.2	13.1	12.0	11.0	9.8
	1490	1380	1320	1530	1700	1900	2200	2700
	2.45	9.2	18	27	37	48	58	79

Heat flux.		0"	8"	18"	28"	38"	48"	58"	72"
29100	T_{W_O}	258.4	258.8	259.0	255.9	252.7	249.7	247.0	244.1
	T_B	245.5	245.4	244.8	242.8	241.1	239.1	236.1	229.3
	ΔT	12.1	12.9	13.5	12.6	11.4	10.3	10.3	14.5
	h	2320	2220	2200	2280	2520	2880	2800	2050
	v_B	2.46	13.3	27.5	40	57	74	93	129
38000		264.2	265.0	266.0	263.2	260.2	257.3	254.4	250.4
		250.0	249.7	249.0	246.5	244.0	241.1	238.2	230.3
		13.7	14.3	15.5	15.6	15.5	15.3	15.2	19.0
		2720	2600	2460	2460	2460	2460	2500	2000
		2.46	15.7	32.5	50	71	93	118	165
42300		269.7	270.2	271.3	266.3	261.2	256.2	251.0	244.0
		253.1	252.2	251.2	250.2	248.2	244.6	241.0	232.3
		15.7	17.9	30.5	17.5	15.3	14.9	14.2	20.5
		2720	2300	2100	2380	2780	2970	3000	2050
		2.48	16.8	34.5	53.5	74	97	124	178

½" tube. Feed rate 75 lb/hr. Material - water.

Heat flux.	0"	7"	17"	27"	37"	47"	57"	68"
5740	226.0	226.1	226.2	226.2	226.0	225.3	224.0	222.3
	216.8	216.7	216.5	216.3	216.1	216.0	215.9	215.8
	8.8	9.2	9.8	10.0	9.8	8.8	7.5	6.5
	660	550	550	560	600	680	770	870
	0.25	2.5	5.8	9.1	12.4	15.5	18.8	21.6

Heat flux.		0"	7"	17"	27"	37"	47"	57"	68"
13400	T_{w_o}	235.0	235.1	235.4	233.4	231.6	229.8	228.0	226.2
	T_B	217.5	217.2	216.9	216.5	216.4	216.2	216.0	215.9
	ΔT	16.8	17.1	17.8	16.0	14.7	13.0	11.5	10.0
	h	800	780	780	820	920	1040	1200	1320
	v_B	0.25	5.5	12.9	20.2	28.5	36.2	43.8	50
20100		243.1	243.1	241.2	239.0	236.8	234.4	232.2	230.2
		218.0	217.8	217.3	217.1	216.9	216.7	216.2	216.0
		24.2	24.4	23.0	21.0	19.0	17.1	15.1	13.5
		820	820	880	930	1050	1180	1320	1490
		0.25	8.1	19.5	30.8	42	54	64	74
33600		254.0	253.0	251.8	249.7	245.5	241.3	237.2	233.9
		219.2	219.0	218.7	218.2	217.9	217.6	217.1	216.9
		33.0	32.2	31.5	30.0	26.1	22.2	18.6	15.5
		1010	1020	1060	1130	1270	1500	1850	2190
		0.25	12.8	30.6	49.6	68	88	106	123

½" tube. Feed rate 150 lb/hr. Material - water.

2020	223.0	223.0	222.9	222.9	222.7	222.2	221.9	221.3
	219.5	219.2	218.9	218.4	218.0	217.8	217.2	217.0
	3.2	3.4	3.8	4.0	4.2	4.2	4.2	4.2
	630	580	520	500	500	500	500	500
	0.49	0.96	2.4	3.5	4.7	5.9	7.1	8.1

Heat flux		0"	7"	17"	27"	37"	47"	57"	68"
13400	T_{w_o}	235.8	235.8	234.8	233.3	232.0	230.8	229.2	228.0
	T_B	219.7	219.3	219.0	218.7	218.2	218.0	217.7	217.2
	ΔT	15.2	15.7	15.1	14.2	13.2	12.2	11.1	10.3
	h	880	850	840	900	980	1080	1200	1300
	v_B	0.49	5.5	12.4	19.8	27.5	35	42.4	51.2
33600		254.9	254.9	253.9	249.9	245.7	241.4	237.2	234.0
		222.1	221.9	221.0	220.2	219.8	219.0	218.2	217.8
		31.1	31.8	31.5	28.0	24.7	21.1	18.0	15.0
		1030	990	1000	1120	1350	1600	1920	2250
		0.49	12.3	30	47.5	66	84	106	121
53000		261.0	261.2	262.0	258.0	253.2	248.0	242.9	238.8
		225.0	224.2	223.3	222.3	221.6	220.6	219.8	219.0
		33.8	34.8	36.0	33.2	29.2	25.0	21.0	17.5
		1550	1490	1480	1550	1800	2120	2590	3010
		0.49	18.3	42	73	100	127	159	185
68500		264.8	266.5	269.2	264.2	258.9	253.2	247.9	243.1
		227.5	226.6	225.2	224.0	222.8	221.3	220.1	219.1
		34.1	37.0	41.0	37.6	33.2	29.0	24.8	21.1
		2070	1900	1780	1850	2180	2520	2970	3340
		0.49	24	59	94	132	172	214	248

$\frac{1}{2}$" tube. Feed rate 250 lb/hr. Material - water.

Heat flux.		0"	7"	17"	27"	37"	47"	57"	68"
2020	T_{w_o}	222.2	222.7	222.5	222.0	221.7	221.2	220.8	220.2
	T_B	218.2	218.0	217.8	217.2	217.0	216.8	216.7	216.2
	ΔT	4.1	4.7	4.7	4.5	4.3	4.2	4.1	4.0
	h	470	470	450	480	480	480	490	500
	V_B	0.85	1.7	2.8	5.9	5.1	6.2	7.4	8.7
13400		231.9	232.3	232.2	231.0	229.8	228.7	227.2	225.9
		220.7	220.1	219.5	218.9	218.2	217.7	217.0	216.2
		10.4	11.6	12.2	11.6	11.0	10.3	9.8	9.1
		1220	1120	1070	1080	1120	1200	1300	1420
		0.85	5.7	12.7	19.6	26.8	34.3	41.7	51.1
42200		252.1	254.2	255.2	252.2	249.2	246.0	243.1	239.8
		228.1	227.2	225.8	224.4	223.0	221.7	220.2	218.8
		22.1	25.3	27.8	26.1	24.4	22.8	21.1	19.2
		1910	1680	1520	1530	1650	1800	1980	2180
		0.85	14.6	35	56.1	79.5	102	127	156
68500		261.5	265.4	268.8	265.6	261.7	257.6	253.6	249.1
		233.1	231.9	229.9	228.0	226.1	224.1	222.3	220.0
		26.1	31.1	35.9	35.2	33.0	30.8	28.7	26.1
		2600	2150	1890	1920	2050	2210	2390	2620
		0.85	21.6	53.3	88.2	121	158	198	246

Heat flux		0"	7"	17"	27"	37"	47"	57"	68"
86500	T_{W_o}	269.1	272.6	275.1	273.1	268.8	264.1	259.3	254.2
	T_B	239.6	237.8	235.0	232.2	229.7	227.0	224.3	221.6
	ΔT	26.1	31.8	36.9	37.9	35.8	33.8	31.7	29.6
	h	3310	2710	2360	2290	2380	2520	2710	2920
	v_B	0.85	24.9	61.8	97.6	144	190	243	305

$\frac{1}{2}$" tube. Feed rate 400 lb/hr. Material - water.

2020	224.1	223.7	222.9	222.1	221.5	220.9	220.1	219.3
	220.0	219.7	219.0	218.4	218.0	217.3	216.9	216.1
	3.9	3.9	3.8	3.4	3.2	3.1	3.0	2.8
	520	530	570	600	620	680	710	740
	1.31	2.2	3.3	4.4	5.5	6.7	7.8	9.3
13400	235.2	235.2	234.1	232.2	231.6	228.9	227.1	225.1
	222.1	221.6	220.8	219.9	219.1	218.3	217.5	216.8
	12.9	13.2	12.8	11.8	11.0	10.0	9.1	8.0
	1010	990	1000	1080	1170	1290	1430	1600
	1.31	6.0	12.7	19.8	26.9	34.7	42.4	52.1
33600	252.8	252.2	252.2	251.0	246.9	242.9	238.8	234.2
	229.6	229.6	229.6	229.4	227.2	225.0	222.5	220.1
	25.2	25.1	25.1	24.9	22.2	19.8	17.2	14.5
	1280	1220	1230	1290	1420	1650	1900	2200
	1.31	11.3	26.7	39.8	56.8	74.8	91.5	117

Heat flux		0"	7"	17"	27"	37"	47"	57"	68"
50800	T_{w_o}	259.8	260.4	261.0	256.7	252.2	248.0	243.7	238.9
	T_B	232.2	231.0	229.1	227.1	225.4	223.7	221.9	219.9
	ΔT	25.8	27.8	30.0	27.7	25.1	22.9	20.2	17.8
	h	1920	1800	1730	1810	1980	2200	2490	2810
	v_B	1.31	16.3	37.7	58.8	80.3	104	135	185
68500		272.3	274.1	276.5	272.1	267.0	261.3	255.2	248.2
		233.0	231.9	230.0	228.1	226.3	224.8	222.9	223.3
		36.6	39.6	44.1	41.8	38.2	34.3	30.1	25.2
		1850	1650	1570	1620	1780	2010	2320	2720
		1.31	21.7	51.4	83	118	154	196	241

½" tube. Feed rate 600 lb/hr. Material - water.

2020	229.8	229.6	228.9	227.9	226.9	225.8	224.8	223.8
	226.8	225.9	224.8	223.7	222.6	221.4	220.2	219.2
	3.0	3.8	4.2	4.2	4.2	4.2	4.2	4.2
	700	560	510	500	500	500	500	500
	1.97	2.7	3.8	4.9	5.9	7.2	8.3	9.3
5740	233.9	233.8	233.0	231.9	230.3	229.1	227.9	226.9
	228.2	227.2	226.0	225.9	223.5	222.1	221.0	219.9
	5.2	6.1	6.7	6.7	6.7	6.7	6.7	6.7
	1050	890	840	840	840	840	840	840
	1.97	4.0	6.9	10.2	13.2	16.3	20.2	22.8

Heat flux		0"	7"	17"	27"	37"	47"	57"	68"
13400	T_{w_o}	242.1	242.0	241.9	240.1	237.9	235.7	233.3	231.5
	T_B	231.0	229.8	228.1	226.6	225.0	223.3	221.9	220.3
	ΔT	10.9	12.0	13.5	13.2	12.7	11.9	11.2	10.8
	h	1220	1090	1000	990	1010	1090	1180	1270
	v_B	1.97	6.4	13.3	19.9	27.8	34.5	41.6	47.7
20100		255.9	253.9	250.8	247.8	244.9	241.9	238.9	236.4
		234.0	232.8	230.8	228.9	227.0	225.1	223.2	221.8
		21.2	20.7	19.4	18.3	17.3	16.2	15.0	14.0
		940	970	1010	1080	1120	1220	1320	1440
		1.97	8.5	18.8	28.7	40.2	50.8	60.9	73.4

½" tube. Feed rate 750 lb/hr. Material - water.

5740	223.2	224.0	225.1	226.1	226.2	226.0	225.7	226.1
	217.1	217.3	218.0	218.2	218.1	218.0	217.9	217.8
	5.5	5.9	6.3	7.0	7.2	7.0	7.0	6.8
	1070	990	890	820	800	800	810	820
	2.26	4.2	7.1	10.4	13.4	17.0	20.7	23.1
13400	231.0	232.0	234.0	234.2	234.8	232.8	231.0	229.4
	217.2	218.2	219.4	219.1	219.0	218.8	218.2	218.0
	12.6	12.9	13.3	13.9	14.4	13.1	11.8	10.4
	1160	1110	1020	990	920	1020	1140	1250
	2.26	6.7	13.4	19.8	27.9	34.5	41.5	51.2

Heat flux.		0"	7"	17"	27"	37"	47"	57"	68"
20100	T_{W_O}	237.9	239.2	241.2	241.1	241.0	238.9	236.0	233.6
	T_B	217.8	219.2	221.8	224.1	223.5	222.2	221.0	220.0
	ΔT	18.8	18.7	18.3	17.0	15.8	14.4	13.3	12.2
	h	1080	1080	1080	1150	1250	1380	1500	1630
	V_B	2.26	8.9	19.1	29	40.6	51.2	61.4	73.5

$\frac{3}{4}$" tube. Feed rate 150 lb/hr. Material - water.

	0"	8"	18"	28"	38"	48"	58"	75"
3270	223.3	223.0	222.9	222.2	222.0	221.8	221.2	220.4
	218.8	218.5	218.3	218.1	218.0	217.8	217.5	217.2
	4.2	4.2	4.1	4.0	4.0	3.9	3.8	3.3
	780	780	790	810	820	850	890	960
	0.23	1.18	2.4	3.6	4.8	6.1	7.3	9.5
7610	227.1	227.0	226.8	226.5	226.2	226.0	225.9	225.2
	217.9	217.9	217.7	217.2	217.0	216.9	216.8	216.2
	9.0	9.0	9.0	9.0	9.0	9.0	9.0	9.0
	830	830	830	830	830	830	830	830
	0.23	2.5	5.3	8.2	11.1	14.0	16.9	21.9
20400	236.5	239.9	239.9	239.2	238.9	238.4	237.9	237.0
	219.2	219.1	219.0	218.8	218.6	218.2	218.0	217.5
	16.5	19.8	19.9	19.8	19.6	19.3	19.1	18.7
	1250	1050	1000	1020	1020	1040	1050	1080
	0.23	6.2	13.6	21.2	30.6	36.6	45.7	57.4

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
41800	T_{W_O}	250.1	259.0	263.9	262.3	259.4	256.8	254.0	249.2
	T_B	223.5	223.4	223.2	223.1	223.0	222.9	222.8	222.3
	ΔT	25.1	33.8	38.9	37.8	35.0	32.3	29.5	25.0
	h	1660	1230	1070	1130	1190	1310	1420	1650
	v_B	0.23	11.4	26.6	40.0	54.3	68.8	82.7	117
52800		255.1	259.9	262.3	262.9	262.1	260.5	258.2	253.6
		223.2	223.1	223.1	223.0	222.9	222.8	222.7	222.2
		29.4	34.5	36.8	37.5	37.0	35.5	33.3	29.0
		1770	1580	1450	1400	1420	1500	1600	1810
		0.23	14.5	32.4	50.5	68.3	86.5	104	135
63500		259.8	267.9	272.1	271.3	270.2	269.0	267.6	265.3
		225.4	225.4	225.4	225.5	225.6	225.7	225.7	225.5
		32.0	39.8	44.0	43.1	42.0	40.9	39.7	37.8
		1970	1600	1440	1480	1510	1540	1600	1670
		0.23	16.4	37	57.2	78	99	119	155
74500		264.8	273.8	276.9	276.3	276.1	275.9	275.5	275.0
		228.3	228.3	228.2	228.1	228.0	228.0	227.9	227.5
		33.0	42.0	45.8	45.7	45.3	45.1	45.0	44.4
		2220	1720	1580	1530	1530	1620	1750	2050
		0.23	18.6	41.7	64.8	88	111	134	175

$\frac{3}{4}$" tube. Feed rate 250 lb/hr. Material - water.

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
3270	T_W °	225.0	224.4	224.0	223.5	223.0	222.3	221.9	220.9
	T_B	218.2	218.2	218.1	218.1	218.0	217.9	217.8	217.2
	ΔT	6.5	6.1	5.8	5.2	4.9	4.3	3.9	3.2
	h	500	520	580	600	660	740	840	1000
	v_B	0.38	1.5	2.6	3.7	5.0	6.2	7.4	9.4
7610		229.2	229.0	228.6	228.1	227.8	227.2	226.9	226.0
		219.1	219.1	219.1	218.9	218.7	218.3	218.1	217.8
		9.4	9.3	9.1	9.0	8.8	8.7	8.2	8.0
		800	800	810	830	850	880	900	930
		0.38	2.6	5.4	8.1	11.0	13.8	16.6	21.5
11950		231.1	232.2	233.4	233.2	231.9	231.5	231.0	230.2
		219.8	219.6	219.2	219.1	219.0	218.8	218.4	218.0
		10.6	12.0	12.2	12.8	12.3	12.1	12.0	11.4
		1090	1000	930	930	970	990	1010	1030
		0.38	3.8	8.1	12.6	17.0	22.3	25.8	33.4
20400		238.0	241.0	242.9	242.3	241.3	240.2	239.1	237.2
		221.2	221.0	220.8	220.4	220.1	220.0	219.8	219.4
		15.8	18.9	21.7	21.1	20.3	19.4	18.5	16.9
		1280	1020	930	980	1020	1080	1110	1200
		0.38	6.1	13.3	20.5	27.9	35.3	42.6	55.6

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
31100	T_{W_O}	243.3	250.9	255.5	255.0	253.0	251.2	249.3	245.9
	T_B	224.2	224.1	224.0	224.0	223.9	223.8	223.7	223.2
	ΔT	18.0	25.1	30.5	29.7	28.0	26.1	24.2	21.3
	h	1680	1260	1060	1030	1100	1200	1290	1420
	v_B	0.38	8.4	18.5	28.7	39.0	49.3	59.5	77.0
41800		250.5	258.2	263.2	262.8	260.7	258.7	256.5	252.9
		226.8	226.8	226.8	226.7	226.5	226.4	226.2	226.1
		22.0	29.3	34.6	34.5	32.6	30.6	28.7	25.2
		1890	1450	1210	1180	1230	1380	1470	1620
		0.38	10.9	24.1	37.3	50.5	63.8	77.4	101
52800		250.1	264.6	270.5	270.8	268.1	265.3	262.9	258.2
		230.2	230.0	229.9	229.7	229.4	229.2	229.2	229.0
		23.0	32.9	38.8	39.0	36.5	33.9	31.2	27.0
		2330	1600	1330	1330	1420	1530	1690	1920
		0.38	13.0	28.9	45.0	60.6	77.0	93.5	122
63500		256.9	269.1	276.6	277.2	274.1	271.1	268.1	262.9
		231.9	231.8	231.7	231.5	231.2	231.1	231.1	231.0
		22.8	35.0	42.6	43.8	40.8	37.8	34.7	29.2
		2820	1840	1500	1480	1530	1680	1830	2150
		0.38	15.2	34.0	53	71	89.5	108	143

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
74500	T_{w_o}	263.9	275.2	283.8	282.8	280.8	279.0	277.0	273.8
	T_B	233.0	233.0	233.0	233.0	233.0	233.0	233.0	233.0
	ΔT	27.8	39.9	47.2	46.4	44.7	42.8	40.9	37.8
	h	2650	1910	1600	1580	1650	1730	1810	1930
	v_B	0.38	17.3	38.4	59.5	80	92.5	123	159
85800		269.9	281.2	289.3	288.8	286.7	284.6	282.5	279.0
		236.1	236.1	236.0	235.9	235.8	235.7	235.5	235.0
		30.0	42.1	50.0	49.7	47.7	45.8	43.8	40.3
		2850	2050	1760	1730	1780	1870	1970	2100
		0.39	18.9	41.8	65	88	112	136	178

¾" tube. Feed rate 400 lb/hr. Material - water.

3270	226.1	225.8	225.0	224.0	223.0	222.1	221.2	219.9
	219.8	219.4	219.1	218.9	218.7	218.2	218.0	217.3
	6.5	6.2	5.9	5.5	5.1	4.8	4.2	3.7
	480	510	570	610	670	710	780	830
	0.61	1.52	2.7	3.8	5.2	6.5	7.6	9.7
7610	230.2	229.9	229.7	229.0	228.7	228.0	227.4	226.6
	220.5	220.3	220.2	220.0	219.9	219.6	219.2	218.2
	8.2	9.1	8.9	8.8	8.5	8.2	8.1	7.8
	920	830	850	870	890	900	920	970
	0.61	2.8	5.5	8.2	11.0	13.8	16.7	21.5

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
11950	T_{w_o}	233.2	233.2	233.2	232.9	232.2	231.9	231.2	230.2
	T_B	221.1	221.0	220.7	220.2	219.9	219.7	219.2	218.5
	ΔT	10.2	11.8	12.0	11.9	11.8	11.6	11.4	11.0
	h	1050	1000	980	990	1000	1010	1020	1060
	v_B	0.61	3.95	8.2	12.5	16.7	21.1	25.6	33.4
16200		235.9	237.2	238.0	238.0	237.4	236.8	236.0	234.8
		222.2	222.1	222.0	221.9	221.7	221.2	221.0	220.5
		13.0	14.8	15.6	15.4	15.2	14.8	14.3	13.9
		1220	1100	1020	1020	1070	1090	1100	1170
		0.61	5.05	10.7	16.2	21.9	28.0	33.6	43.5
20400		238.9	241.5	243.2	243.5	242.9	242.0	241.0	239.2
		223.4	223.3	223.2	223.1	223.0	222.9	222.8	222.2
		14.6	17.3	19.0	19.5	19.0	18.3	17.4	16.0
		1370	1190	1060	1000	1020	1100	1180	1280
		0.61	6.1	13.0	19.8	26.7	33.7	40.7	53.0
31100		242.5	248.6	251.6	252.1	251.2	249.9	248.3	246.0
		225.4	225.4	225.4	225.4	225.2	225.0	224.8	224.5
		15.4	22.3	24.3	25.1	24.6	23.2	22.0	20.0
		2030	1490	1350	1200	1220	1320	1400	1530
		0.61	8.7	18.7	29.8	39.0	54.1	59.5	74.9

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
41800	T_{w_o}	249.9	255.2	260.2	261.1	259.5	157.8	156.0	153.0
	T_B	229.0	229.0	229.1	229.0	228.9	228.8	228.6	228.0
	ΔT	18.8	24.5	29.2	30.3	28.9	27.2	25.7	23.0
	h	2200	1700	1450	1400	1430	1550	1650	1820
	v_B	0.61	10.9	23.6	36.2	49.0	62.1	74.6	98.0
52800		256.0	261.6	266.8	269.0	267.3	265.2	263.2	259.9
		231.8	231.7	231.5	231.2	231.0	230.9	230.7	230.1
		22.2	28.0	33.1	35.2	33.8	32.0	30.4	37.6
		2380	1850	1580	1500	1550	1660	1760	1920
		0.61	12.8	28.3	44.0	59.6	75.3	91.0	119
63500		269.7	274.0	279.1	281.2	279.0	276.9	274.7	270.8
		240.2	240.1	240.0	239.9	239.8	239.5	239.2	239.0
		26.8	31.6	37.0	39.0	37.1	35.0	32.8	29.1
		2360	2020	1730	1680	1720	1820	1930	2180
		0.61	13.3	29.5	45.5	61.8	78.2	94.5	123
74500		277.2	281.9	287.6	290.8	289.5	286.9	284.1	279.9
		245.2	245.1	245.0	244.8	244.7	244.4	244.2	243.9
		29.0	33.4	39.2	42.8	41.9	39.3	37.0	33.0
		2540	2240	1940	1780	1790	1880	2000	2260
		0.61	14.6	31.8	49.6	67.3	85.6	103	134

1" tube Feed rate 150 lb/hr. Material - water

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
2450	T_{W_O}	223.2	222.9	222.0	221.2	220.6	219.8	219.1	218.0
	T_B	219.1	219.0	218.8	218.2	217.9	217.5	217.1	216.5
	ΔT	4.0	3.9	3.4	3.0	2.8	2.3	2.0	1.5
	h	590	600	600	660	720	820	930	1200
	v_B	0.13	0.66	1.33	1.95	2.7	3.4	4.2	5.4
5800		227.5	226.2	225.0	224.6	224.2	223.8	223.3	222.9
		219.9	219.6	219.1	218.8	218.3	218.0	217.8	217.0
		7.3	6.6	5.8	5.8	5.8	5.8	5.7	5.5
		780	880	990	1000	1000	1010	1020	1050
		0.13	1.35	2.9	4.5	6.1	7.7	9.4	12.2
8950		229.7	229.2	228.8	228.1	227.6	227.0	226.4	225.5
		219.8	219.4	219.1	218.9	218.6	218.2	217.9	217.3
		9.3	9.1	9.0	8.9	8.7	8.2	7.9	7.3
		940	980	1000	1030	1070	1090	1100	1130
		0.13	2.1	4.3	7.0	9.5	12.0	14.5	19.0
15300		235.8	235.6	235.3	235.1	235.0	234.8	234.7	234.1
		220.2	220.0	219.7	219.4	219.1	218.8	218.6	218.1
		14.8	14.8	14.8	14.9	15.0	15.0	15.1	15.2
		1020	1020	1020	1020	1020	1010	1010	1000
		0.13	3.4	7.5	11.7	16.0	20.3	24.5	32.0

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
23300	T_{w_o}	243.3	243.6	243.8	244.1	244.7	244.9	244.5	242.0
	T_B	220.0	221.8	221.5	221.2	221.2	221.0	220.9	220.5
	ΔT	20.8	21.0	21.5	21.9	22.2	22.9	22.8	20.2
	h	1110	1100	1080	1050	1020	1000	1100	1180
	v_B	0.13	4.9	11.0	17.2	23.4	29.7	35.8	46.5
39600		259.0	259.3	259.8	260.2	260.8	261.1	258.4	253.8
		225.9	225.8	225.5	225.2	225.1	225.0	224.9	224.7
		31.5	31.9	32.3	33.1	33.8	34.0	31.6	27.5
		1290	1250	1220	1200	1170	1160	1220	1380
		0.13	7.7	17.3	27.0	36.7	46.6	56.5	73
55200		269.2	270.7	272.0	273.7	273.8	273.9	273.0	267.0
		228.9	228.8	228.7	228.6	228.4	228.2	228.1	228.0
		38.0	39.2	40.9	42.5	43.0	43.1	42.1	36.6
		1470	1400	1370	1300	1280	1290	1300	1500
		0.13	10.2	23.1	35.9	48.6	61	74	97

1" tube Feed rate 250 lb/hr. Material - water.

8950	229.7	229.3	229.0	228.8	228.5	228.2	228.0	227.4
	219.8	219.5	219.4	219.0	219.0	219.0	218.8	218.6
	9.2	9.2	9.1	9.0	9.0	8.9	8.8	8.4
	950	950	960	980	1000	1000	1020	1030
	0.21	2.1	4.5	7.1	9.7	12.2	15.0	19.4

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
15300	T_{w_o}	235.6	235.5	235.4	235.3	235.1	234.8	234.2	233.8
	T_B	221.0	220.8	220.6	220.2	220.0	219.8	219.5	219.0
	ΔT	13.8	13.9	14.1	14.2	14.2	14.1	14.0	13.9
	h	1110	1100	1090	1090	1090	1090	1100	1100
	v_B	0.21	3.3	7.5	11.6	15.7	19.9	24.1	31.2
23300		245.3	245.8	246.1	246.5	245.9	245.1	244.3	243.1
		225.6	225.2	225.0	224.8	224.5	224.1	224.0	223.3
		18.5	19.0	19.8	20.5	20.2	19.9	19.4	18.8
		1250	1220	1180	1130	1140	1180	1200	1220
		0.21	4.7	10.4	16.3	22.1	28.0	34	44
31400		249.7	251.9	254.2	253.7	253.0	252.3	251.9	250.9
		227.9	227.5	227.2	227.1	226.9	226.8	226.7	226.2
		20.3	22.9	25.3	25.0	24.8	24.2	24.0	23.3
		1520	1370	1230	1210	1270	1290	1300	1350
		0.21	6.1	13.5	20.7	28.2	35.7	43.2	57.6
39600		253.9	258.2	261.0	260.2	259.9	259.2	258.9	257.9
		230.3	230.2	230.1	230.0	229.8	229.2	228.7	227.9
		22.0	26.5	29.0	28.9	28.7	28.4	28.2	28.1
		1810	1480	1380	1370	1380	1390	1400	1410
		0.21	7.3	16.2	25.0	33.8	43.4	56.8	69.4
55200		262.9	268.0	273.8	272.3	271.2	270.0	268.8	266.8
		234.8	234.7	234.5	234.3	234.2	234.1	234.0	234.0
		25.5	31.0	36.5	35.3	34.2	33.1	32.0	30.2
		2160	1740	1530	1530	1600	1630	1690	1820
		0.21	9.5	20.9	32.4	44.2	55.5	67.4	87.4

1" tube Feed rate 400 lb/hr. Material - water.

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
2450	T_{w_o}	224.0	223.5	223.0	222.5	222.0	221.5	221.0	220.2
	T_B	220.0	219.6	219.3	219.1	218.6	218.4	218.0	217.8
	ΔT	3.9	3.6	3.5	3.3	3.1	3.0	2.9	2.5
	h	630	670	700	730	780	800	820	850
	v_B	0.34	0.87	1.5	2.2	2.9	3.6	4.3	5.5
8950		231.0	230.8	230.5	230.2	229.8	229.3	228.8	228.0
		221.3	221.0	220.8	220.3	219.9	219.5	219.1	218.3
		8.9	9.0	9.0	9.2	9.1	9.0	8.9	8.8
		1020	1000	980	970	980	990	1000	1030
		0.34	2.2	4.1	7.0	9.5	12.0	14.5	18.6
15300		240.2	240.0	239.6	239.3	239.2	239.1	238.7	237.3
		225.7	225.2	224.8	224.4	224.0	223.7	223.2	222.5
		13.8	13.9	13.9	14.0	14.2	14.5	14.6	13.0
		1120	1120	1120	1120	1100	1050	1050	1180
		0.34	3.3	7.1	12.4	14.7	18.7	22.7	29.5
23300		251.8	251.5	251.2	251.0	250.8	250.7	250.0	248.1
		230.0	229.9	229.7	229.3	229.1	229.0	228.9	228.4
		20.2	20.1	20.0	20.0	20.0	20.0	18.9	18.2
		1130	1130	1130	1140	1160	1160	1200	1290
		0.34	4.5	9.9	15.2	20.7	25.9	31.5	41.0

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
31400	T_{W_o}	268.2	270.0	271.8	271.7	271.7	271.7	270.0	264.2
	T_B	239.8	239.7	239.3	239.1	239.0	238.8	238.4	238.1
	ΔT	26.9	28.9	30.9	31.0	31.1	31.2	30.5	25.0
	h	1170	1090	1000	1000	1000	1000	1050	1260
	V_B	0.34	6.4	14.0	21.6	29.5	37.3	45.2	58.6
55200		282.2	283.1	284.1	284.5	284.9	285.0	283.1	276.8
		245.2	245.1	245.0	244.9	244.8	244.5	244.2	244.0
		33.2	34.1	35.0	35.8	36.5	37.2	36.5	29.9
		1650	1600	1580	1520	1490	1430	1500	1820
		0.34	8.0	17.7	27.4	37.0	46.8	57.4	73.5

$\frac{3}{4}$" tube. Feed rate 250 lb/hr. Material - methanol.

3270	165.2	165.0	164.5	163.9	163.2	162.8	162.1	161.1
	151.4	151.4	151.4	151.4	151.4	151.4	151.4	151.4
	14.0	13.6	12.9	12.2	11.8	11.1	10.5	9.3
	235	240	250	260	280	290	310	350
	0.49	1.6	3.0	4.3	5.7	7.0	8.4	10.7
11950	186.2	185.3	184.0	183.6	181.1	179.8	176.9	175.9
	156.8	156.8	156.8	156.8	156.6	156.5	156.4	156.2
	29.0	28.0	26.8	25.3	24.1	22.8	21.4	19.1
	410	440	470	490	530	560	590	620
	0.49	4.1	8.6	13.7	17.7	22.2	26.8	34.7

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
20400	T_{W_o}	196.2	195.7	194.8	193.9	193.0	192.0	191.1	189.4
	T_B	161.0	160.9	160.9	160.8	160.8	160.7	160.6	160.2
	ΔT	34.9	34.1	33.2	32.5	31.6	30.8	29.9	28.3
	h	590	600	610	620	640	660	680	725
	v_B	0.49	6.1	13.1	20.1	27.3	34.4	41.5	53.6
31100		204.8	205.3	206.2	206.9	206.3	206.0	205.8	205.0
		166.2	166.2	166.2	166.3	166.3	166.3	166.3	166.3
		37.3	38.0	38.6	38.9	38.5	38.1	37.8	37.2
		830	820	805	800	805	810	820	840
		0.49	8.5	18.4	28.3	38	48	58	75
41800		210.2	212.0	214.0	216.1	218.1	216.2	213.5	209.0
		170.2	170.2	170.1	170.1	170.1	170.0	170.0	170.0
		38.1	39.9	42.1	44.3	46.8	44.7	41.9	37.2
		1100	990	900	880	890	910	980	1100
		0.49	10.4	23.0	35.7	48	60.5	73	94
52800		215.4	217.8	220.7	223.4	226.2	226.5	225.1	224.1
		173.2	173.2	173.2	173.2	173.2	173.2	173.2	173.2
		39.9	42.1	45.9	47.9	50.6	50.8	49.9	48.3
		1350	1250	1160	1080	1030	1020	1030	1090
		0.49	14.2	31.3	48.5	65.5	82.5	99.5	127

$\frac{3}{4}$" tube. Feed rate 250 lb/hr. Material - Ethanol.

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
7610	T_{W_O}	205.2	204.0	202.4	200.3	199.2	197.7	196.0	193.1
	T_B	178.2	178.2	178.2	178.1	178.0	177.8	177.5	177.0
	ΔT	26.3	25.2	23.8	22.3	21.0	19.6	18.1	15.6
	h	290	300	320	340	360	390	420	480
	v_B	0.50	2.7	5.5	8.3	11.2	13.9	16.8	21.9
11950		213.1	211.5	209.2	207.1	205.0	202.8	200.6	196.9
		179.2	179.2	179.2	179.2	179.2	179.1	179.1	179.0
		33.0	31.3	29.5	27.2	25.1	21.0	18.9	17.2
		390	380	380	400	430	490	550	690
		0.50	3.9	8.2	12.5	16.7	21.1	25.3	32.8
20400		224.1	222.9	221.1	219.3	217.8	216.0	214.2	211.1
		181.5	181.5	181.8	181.8	181.7	181.6	181.5	181.2
		42.5	40.7	38.7	37.0	35.2	33.5	31.8	28.8
		480	480	490	510	530	580	620	730
		0.50	6.7	13.1	20	27	34	41	53
25900		224.7	223.8	222.7	221.5	220.3	219.3	218.2	216.2
		182.1	182.7	183.1	183.0	183.0	183.0	182.9	182.8
		41.5	40.5	39.3	38.2	37.0	35.9	34.7	32.7
		620	630	680	690	700	720	760	800
		0.50	7.7	16	25	33	42	51	65

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
31100	T_{W_o}	229.1	227.8	226.1	224.2	222.6	221.1	225.0	231.1
	T_B	184.0	184.0	184.0	184.0	184.0	184.0	184.0	184.0
	ΔT	44.0	42.3	40.6	38.8	37.0	35.9	39.8	45.9
	h	700	710	730	780	820	890	710	680
	v_B	0.50	8.6	19	29	39	49	60	77
41800		239.2	236.8	233.1	229.6	226.0	222.2	241.1	276.8
		186.0	186.0	186.0	186.0	186.0	186.0	186.0	186.0
		51.4	48.7	45.0	41.6	38.0	34.3	53.0	88.5
		820	870	920	1000	1090	1220	810	320
		0.50	11	24	37	51	64	77	100

$\frac{3}{4}$" tube. Feed rate 400 lb/hr. Material - Ethanol.

7610	204.2	202.5	200.2	199.0	198.5	198.1	197.8	196.9
	178.8	179.5	180.5	180.3	180.1	179.9	179.7	179.2
	25.5	22.9	19.5	18.8	18.3	18.0	17.8	17.2
	280	350	380	390	400	410	430	470
	0.8	2.9	5.5	8.2	11	14	17	21
20400	227.5	226.4	223.0	220.8	219.8	218.9	218.0	216.3
	185.8	185.9	185.9	185.9	185.9	185.9	185.9	185.8
	40.9	39.0	36.4	34.1	33.2	32.3	31.4	29.9
	590	550	590	610	620	630	650	680
	0.8	5.9	12	19	25	32	38	49

$\frac{3}{4}$" tube. Feed rate 400 lb/hr. Material - Chloroform.

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
3270	T_{w_o}	163.1	162.3	161.2	160.2	160.0	159.7	159.2	158.6
	T_B	149.2	149.1	148.9	148.7	148.2	148.0	147.7	147.0
	ΔT	13.9	13.2	12.5	11.8	11.7	11.6	11.5	11.2
	h	220	220	230	270	270	270	290	300
	v_B	0.42	1.6	3.1	4.2	6.2	7.7	9.2	12.0
7610		180.2	178.9	177.0	175.8	174.3	173.2	170.0	160.3
		150.9	150.9	150.9	150.9	150.9	150.9	150.7	150.3
		29.0	28.0	26.8	25.5	24.2	22.8	19.8	10.2
		270	270	280	290	300	320	400	720
		0.42	3.1	6.5	9.9	13.3	16.6	20.0	25.8
11950		193.1	190.9	188.1	185.2	183.8	182.7	179.8	169.0
		153.2	153.1	153.1	153.0	153.0	152.9	152.8	152.5
		39.5	37.2	34.5	31.8	30.3	29.2	26.1	16.0
		310	320	340	370	390	400	480	720
		0.42	4.5	9.6	14.7	19.9	25.5	30.3	39
20400		210.6	208.3	206.7	203.0	200.2	197.4	193.2	182.9
		158.4	158.4	158.4	158.4	158.4	158.3	158.2	158.1
		51.1	49.0	46.3	43.7	40.9	38.1	34.0	23.4
		400	410	440	480	500	520	600	830
		0.42	6.9	14.8	22.8	30.7	39	47	61

Heat flux		0"	8"	18"	28"	38"	48"	58"	75"
25900	T_{W_O}	215.5	213.8	211.7	209.3	207.2	205.0	202.8	198.9
	T_B	161.2	161.2	161.2	161.2	161.1	161.1	161.1	161.1
	ΔT	53.2	51.5	49.3	47.2	44.9	42.7	40.3	36.8
	h	490	500	530	560	580	600	620	690
	v_B	0.42	8.2	18.1	28	38	47.5	57.5	74

$\frac{3}{4}$" tube. Feed rate 400 lb/hr. Material - Iso-Propanol

3270	200.0	199.3	198.4	197.7	196.8	195.9	195.0	193.3
	185.6	185.6	185.6	185.6	185.6	185.3	185.2	185.1
	14.1	13.3	12.8	11.9	11.1	10.2	9.3	8.0
	220	250	270	290	290	300	380	410
	0.8	1.75	2.9	4.1	5.4	6.5	7.7	9.7
7610	212.0	210.7	208.9	207.0	205.0	203.1	201.2	198.0
	186.5	186.5	186.5	186.3	186.2	186.2	186.1	186.0
	25.1	23.8	21.9	20.1	18.2	16.5	14.8	11.7
	320	330	330	390	420	480	530	620
	0.8	3.0	5.7	8.4	11.2	14.0	16.7	21.4
11950	221.9	220.0	217.9	216.4	213.1	210.9	208.5	204.5
	188.0	188.4	189.0	188.9	188.8	188.7	188.5	188.2
	32.9	30.8	28.1	25.9	23.8	21.6	19.2	15.8
	370	380	400	460	500	570	620	740
	0.8	4.1	8.2	12.3	15.4	20.6	24.7	31.9

Heat Flux		0"	8"	18"	28"	38"	48"	58"	75"
20400	T_{w_o}	234.9	233.1	230.8	228.3	226.1	223.8	221.3	217.2
	T_B	193.9	194.0	194.0	194.0	194.0	194.0	194.0	193.9
	ΔT	39.1	37.4	35.2	33.0	30.9	28.6	26.3	22.5
	h	530	550	580	600	650	700	780	900
	v_B	0.8	5.9	12.2	18.4	24.7	31.2	37.6	48.5
31100		247.1	245.1	242.2	239.4	236.8	233.8	231.0	226.0
		198.8	198.8	198.8	198.8	198.8	198.8	198.8	198.8
		47.6	45.2	42.4	39.7	36.8	33.8	30.8	25.8
		620	680	720	790	830	910	1000	1190
		0.8	7.7	16.7	25.6	34.5	43.2	57.2	67.1
41800		254.9	252.9	250.2	247.8	245.1	242.6	240.0	235.6
		201.8	202.1	202.8	202.9	202.9	202.9	202.9	202.5
		51.2	49.0	46.4	44.0	41.2	39.7	36.0	31.6
		820	870	900	980	1010	1080	1140	1320
		0.8	9.8	20.7	31.7	42.8	53.5	64.9	83.4

APPENDIX 4.

Effect of tube length on the heat transfer coefficient.

The results for this series of experiments are arranged in a similar manner to those given in Appendix 3.

$\frac{3}{8}$" tube. Feed rate 400 lb/hr. of water.
Heat flux 42300 Btu/(hr)(sq.ft.)

Feed subcooled by the equivalent of 2 kw preheat.

	0"	8"	18"	28"	38"	48"	58"	72"
T_{w_o}	259.0	266.0	267.5	263.5	257.8	253.0	248.3	241.9
T_B	229.2	234.8	241.7	241.8	239.8	237.8	235.7	230.0
ΔT	28.5	30.0	25.1	20.2	17.7	15.0	12.2	11.8
h	1490	1400	1650	2080	2450	2810	3450	3600
v_B	2.5	2.5	2.5	21	43	60	94	135

Feed just boiling.

	269.7	270.2	271.3	266.3	261.2	256.2	251.0	244.0
	253.1	252.2	251.2	250.2	248.2	244.6	241.0	232.3
	15.7	17.9	20.5	17.5	15.3	14.9	14.2	20.5
	2720	2300	2100	2380	2780	2970	3000	2050
	2.5	17	35	54	74	97	124	178

	0"	8"	18"	28"	38"	48"	58"	72"

2 kw excess preheat.

T_{W_o}	274.2	272.9	271.0	265.8	261.1	256.9	252.9	246.4
T_B	257.5	255.8	253.5	252.3	249.1	247.0	243.0	235.1
ΔT	16.9	17.0	17.2	14.8	12.3	10.8	10.0	11.1
h	2500	2490	2450	2850	3450	3900	4250	3800
v_B	34	49	68	91	111	131	161	215

1" tube. Feed rate 250 lb/hr. of water.
Heat flux 15300 Btu/(hr)(sq.ft).

0"	8"	18"	28"	38"	48"	58"	75"

Feed just boiling.

235.6	235.5	235.4	235.3	235.1	234.8	234.2	233.8
221.0	220.8	220.6	220.2	220.0	219.8	219.5	219.0
13.8	13.9	14.1	14.2	14.2	14.1	14.0	13.9
1110	1100	1090	1090	1090	1090	1100	1100
0.21	3.3	7.5	11.6	15.7	19.9	24.1	31.2

2 kw excess preheat.

237.8	237.5	237.1	236.9	236.8	236.3	236.1	235.9
222.2	222.1	222.1	222.0	221.9	221.7	221.6	221.3
14.5	14.4	14.2	14.1	14.1	14.0	14.0	13.8
1060	1060	1080	1090	1090	1090	1090	1110
7.5	10.5	14.5	18.4	22.3	26.3	30.1	36.0

4 kw excess preheat.

	0"	8"	18"	28"	38"	48"	58"	75"
T_{W_O}	239.1	239.0	238.8	238.7	238.4	238.2	238.1	237.8
T_B	224.7	224.4	224.1	224.0	223.8	223.5	223.2	222.9
ΔT	13.8	13.8	13.8	13.8	13.8	13.8	13.8	13.7
h	1110	1110	1110	1110	1110	1110	1110	1120
V_B	14.3	17.3	21.2	24.8	28.6	32.5	36.3	43.0

6 kw excess preheat.

241.0	240.9	240.7	240.4	240.2	240.0	239.2	238.8
226.0	225.8	225.7	225.5	225.4	225.2	225.1	225.0
13.8	13.8	13.8	13.8	13.8	13.8	13.2	12.8
1110	1110	1110	1110	1110	1110	1160	1200
20.9	23.8	27.5	31.1	35.1	38.6	42.3	48.8

$\frac{3}{4}$" tube. Feed rate 250 lb/hr. of methanol.
Heat flux 3270 Btu/(hr)(sq.ft).

Feed just boiling.

0"	18"	38"	58"	75"
165.2	164.5	163.2	162.1	161.1
151.4	151.4	151.4	151.4	151.4
14.0	12.9	11.8	10.5	9.3
235	250	280	310	350
0.49	3.0	5.7	8.4	10.7

	0"	18"	38"	58"	75"
	2 kw excess preheat.				
T_{W_O}	165.5	165.2	164.7	164.1	163.7
T_B	157.8	157.8	157.8	157.5	157.3
ΔT	8.0	7.7	7.2	6.8	6.2
h	410	430	460	480	500
v_B	16.1	18.0	20.5	23.0	25.0
	4 kw excess preheat.				
	166.6	166.2	165.8	165.2	164.7
	160.5	160.1	159.7	159.1	158.8
	6.0	6.0	6.0	5.9	5.8
	550	550	550	550	560
	30	32	35	37	41
	6 kw excess preheat.				
	168.3	168.0	167.6	167.1	166.6
	162.9	162.5	162.1	161.7	161.1
	5.5	5.5	5.5	5.5	5.5
	650	650	650	650	650
	43	45	48	50	52

$\frac{3}{4}$" tube. Feed rate 250 lb/hr. of ethanol.
Heat flux 31100 Btu/(hr)(sq.ft).

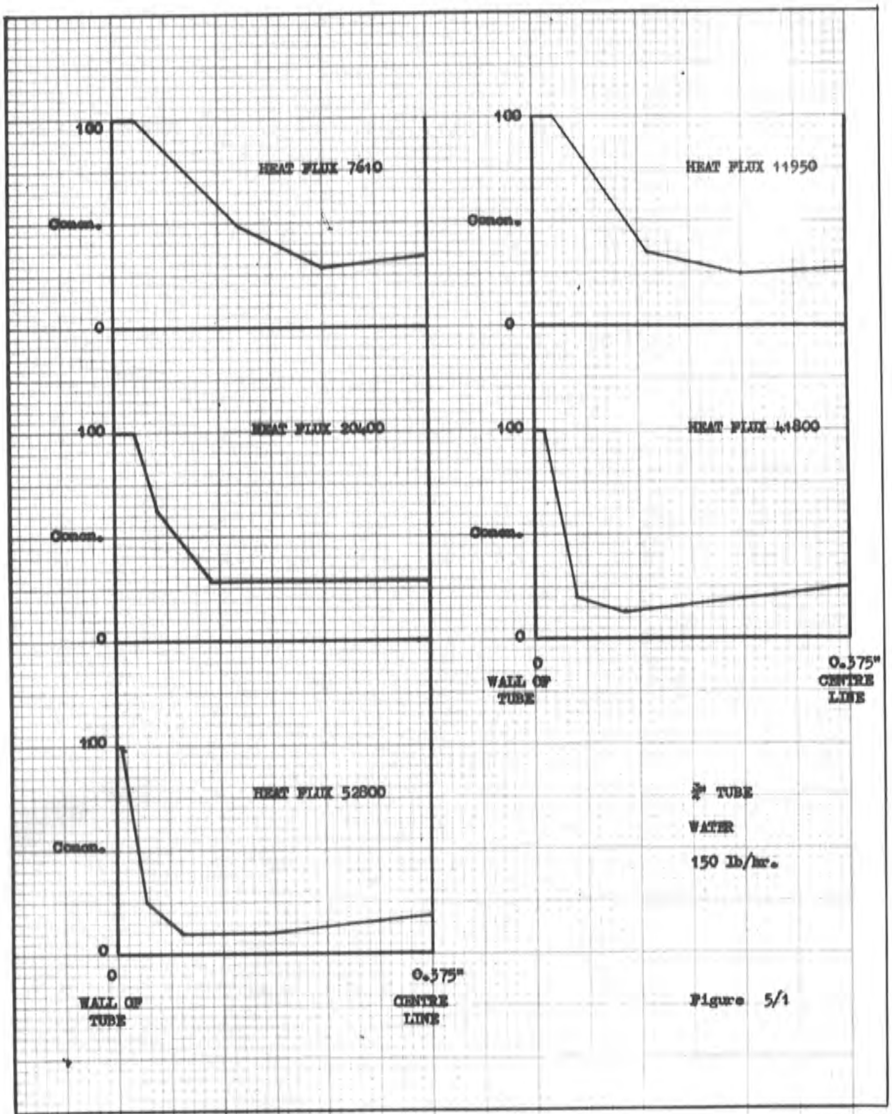
	0"	8"	18"	28"	38"	48"	58"	75"
	Feed sub-cooled by 2 kw.							
T_{w_o}	147.2	169.1	182.0	182.0	182.0	181.9	181.8	181.4
T_B	220.2	225.5	220.5	221.9	217.5	216.0	214.3	211.7
ΔT	72.1	67.5	37.2	35.8	34.3	32.8	31.2	28.8
h	360	360	690	700	710	760	800	900
v_B	0.5	0.5	2.1	11	20	29	37	53
	Boiling feed.							
	224.0	223.1	221.8	220.4	219.1	217.8	216.4	214.1
	183.2	183.2	183.2	183.2	183.2	183.2	183.2	183.2
	39.8	38.8	37.4	36.1	34.8	33.5	32.1	29.9
	680	680	690	700	720	770	800	830
	0.5	7.7	16	25	33	42	51	65
	2 kw excess preheat.							
	224.0	223.0	221.8	220.8	219.3	218.1	216.9	214.8
	185.7	185.6	185.4	185.2	185.0	184.8	184.7	184.1
	37.3	36.8	35.5	34.4	33.2	32.2	31.1	29.2
	690	690	700	720	750	790	810	900
	14	20	29	37	46	54	63	77

	0"	8"	18"	28"	38"	48"	58"	75"
	4 kw excess preheat.							
T_{W_o}	222.2	221.9	221.1	220.4	219.8	219.0	218.3	217.1
T_B	186.8	186.8	186.8	186.8	186.7	186.3	186.0	185.4
ΔT	34.2	34.0	33.5	33.0	32.5	31.9	31.3	30.3
h	730	730	760	780	790	800	810	850
v_B	27	33	41	49	58	66	74	89
	6 kw excess preheat.							
	221.9	221.8	221.2	220.0	219.8	219.2	219.0	218.2
	187.8	187.8	187.7	187.3	187.0	186.9	186.8	186.2
	32.9	32.8	32.7	31.6	31.3	31.2	31.1	30.9
	790	790	800	810	820	830	830	840
	39	45	54	61	70	78	86	101

APPENDIX 5

Results of experiments to find the film thickness.

The results of these experiments are presented in the following graphs, 5/1 to 5/12. The values of the film thickness which were calculated from these graphs are presented in the table in Chapter 4 of the report.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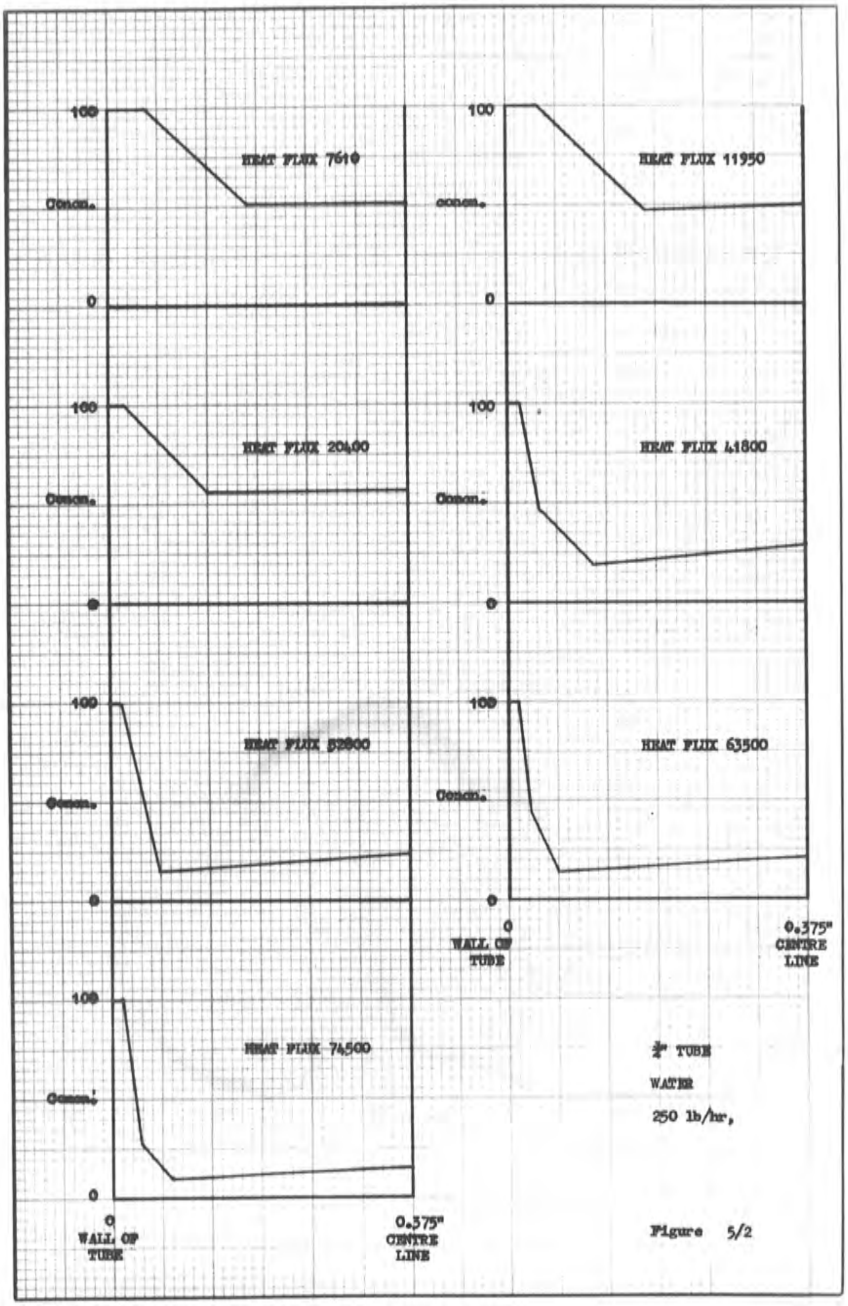
HEAT FLUX 7610
Concn.
0
100
HEAT FLUX 11950
Concn.
0
100
HEAT FLUX 20400
Concn.
0
100
HEAT FLUX 41800
Concn.
0
0
WALL OF TUBE
0.375"
CENTRE LINE
100
HEAT FLUX 52800
Concn.
0
0
WALL OF TUBE
0.375"
CENTRE LINE
¾" TUBE
WATER
150 lb/hr.

Figure 5/1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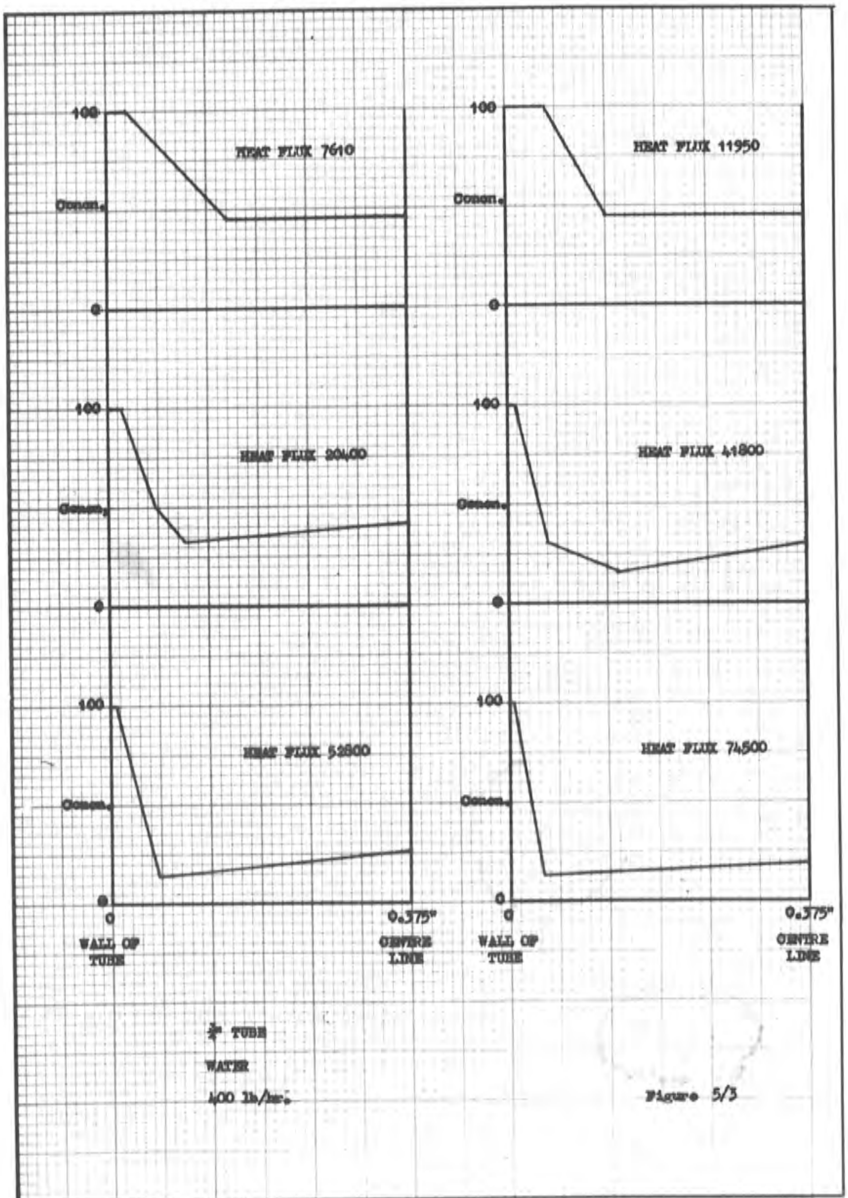
Concn.
0
HEAT FLUX 7610
HEAT FLUX 11950
HEAT FLUX 20400
HEAT FLUX 41800
HEAT FLUX 52800
HEAT FLUX 63500
HEAT FLUX 74500
WALL OF TUBE
0.375"
CENTRE LINE
¾" TUBE
WATER
250 lb/hr.

Figure 5/2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HEAT FLUX 7610
Concn.
0
100
HEAT FLUX 11950
Concn.
0
100
HEAT FLUX 20400
Concn.
0
100
HEAT FLUX 41800
Concn.
0
100
HEAT FLUX 52800
Concn.
0
100
HEAT FLUX 74500
Concn.
0
0
0.375"
WALL OF TUBE
CENTRE LINE
0
0.375"
WALL OF TUBE
CENTRE LINE
¾" TUBE
WATER
400 lb/hr.

Figure 5/3

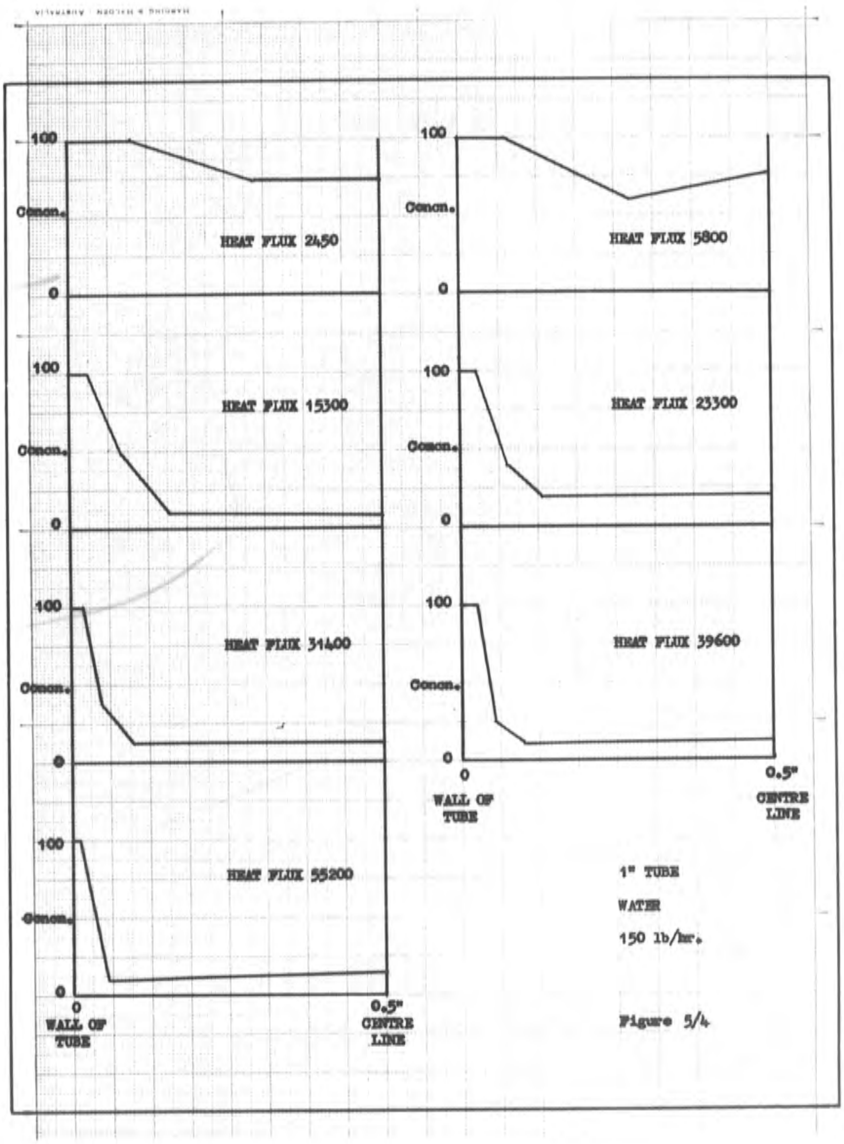

100
Concn.
HEAT FLUX 2450
0
100
Concn.
HEAT FLUX 5800
0
100
HEAT FLUX 15300
Concn.
0
100
HEAT FLUX 23300
Concn.
0
100
HEAT FLUX 31400
Concn.
0
100
HEAT FLUX 39600
Concn.
0
0
0.5"
WALL OF TUBE
CENTRE LINE
100
HEAT FLUX 55200
Concn.
0
0
0.5"
WALL OF TUBE
CENTRE LINE
1" TUBE
WATER
150 lb/hr.

Figure 5/4

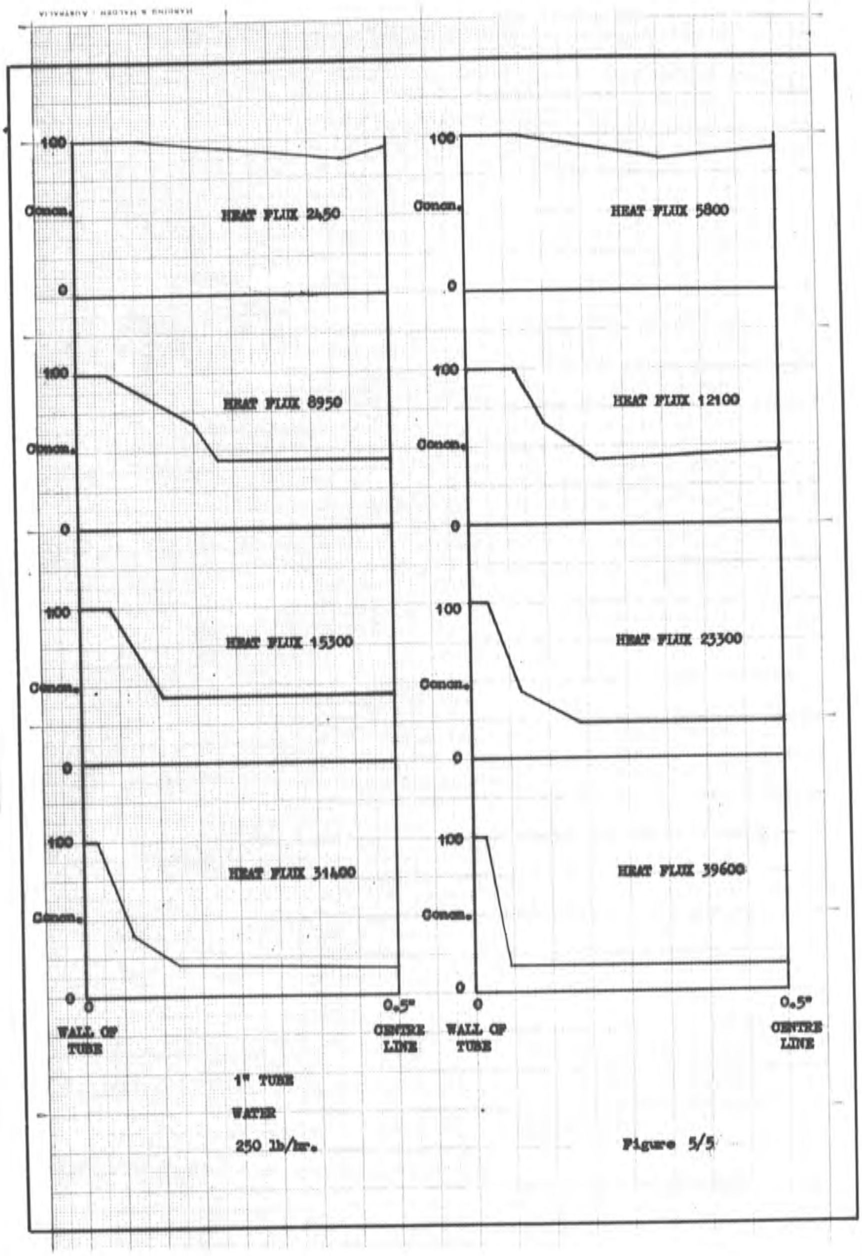
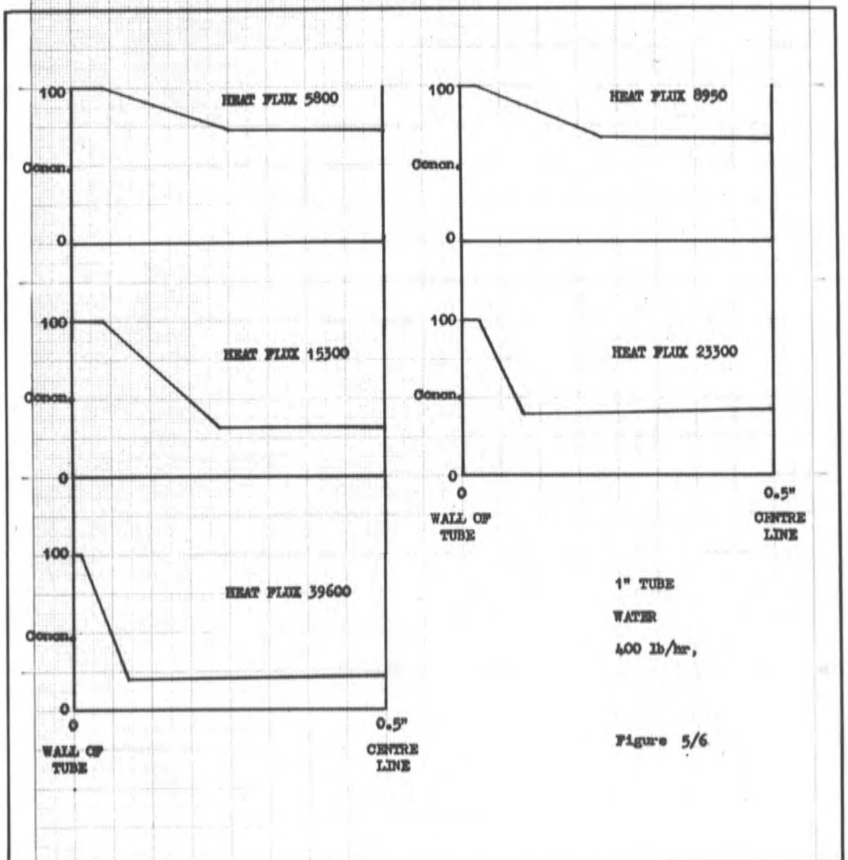
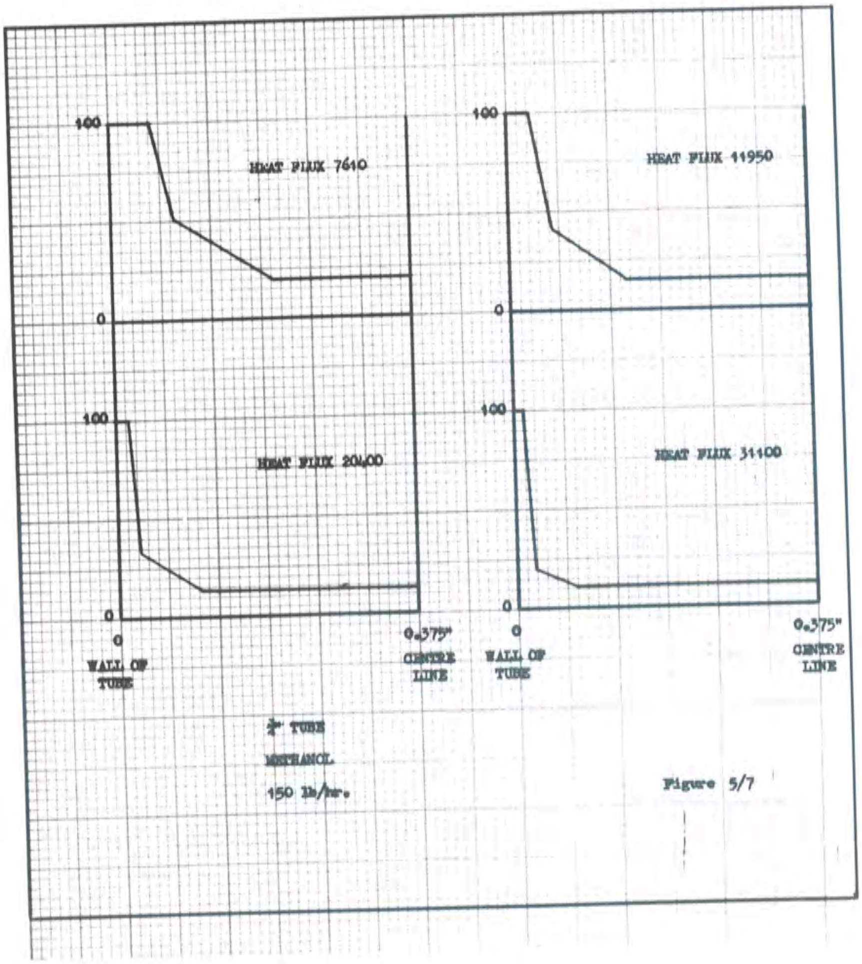

100
Concn.
HEAT FLUX 2450
0
100
HEAT FLUX 5800
Concn.
0
100
HEAT FLUX 8950
Concn.
0
100
HEAT FLUX 12100
Concn.
0
100
HEAT FLUX 15300
Concn.
0
100
HEAT FLUX 23300
Concn.
0
100
HEAT FLUX 31400
Concn.
0
100
HEAT FLUX 39600
Concn.
0
0
WALL OF TUBE
0.5"
CENTRE LINE
1" TUBE
WATER
250 lb/hr.

Figure 5/5



HEAT FLUX 5800
HEAT FLUX 8950
HEAT FLUX 15300
HEAT FLUX 23300
HEAT FLUX 39600
100
Concn.
0
0.5"
WALL OF TUBE
CENTRE LINE
1" TUBE
WATER
400 lb/hr.

Figure 5/6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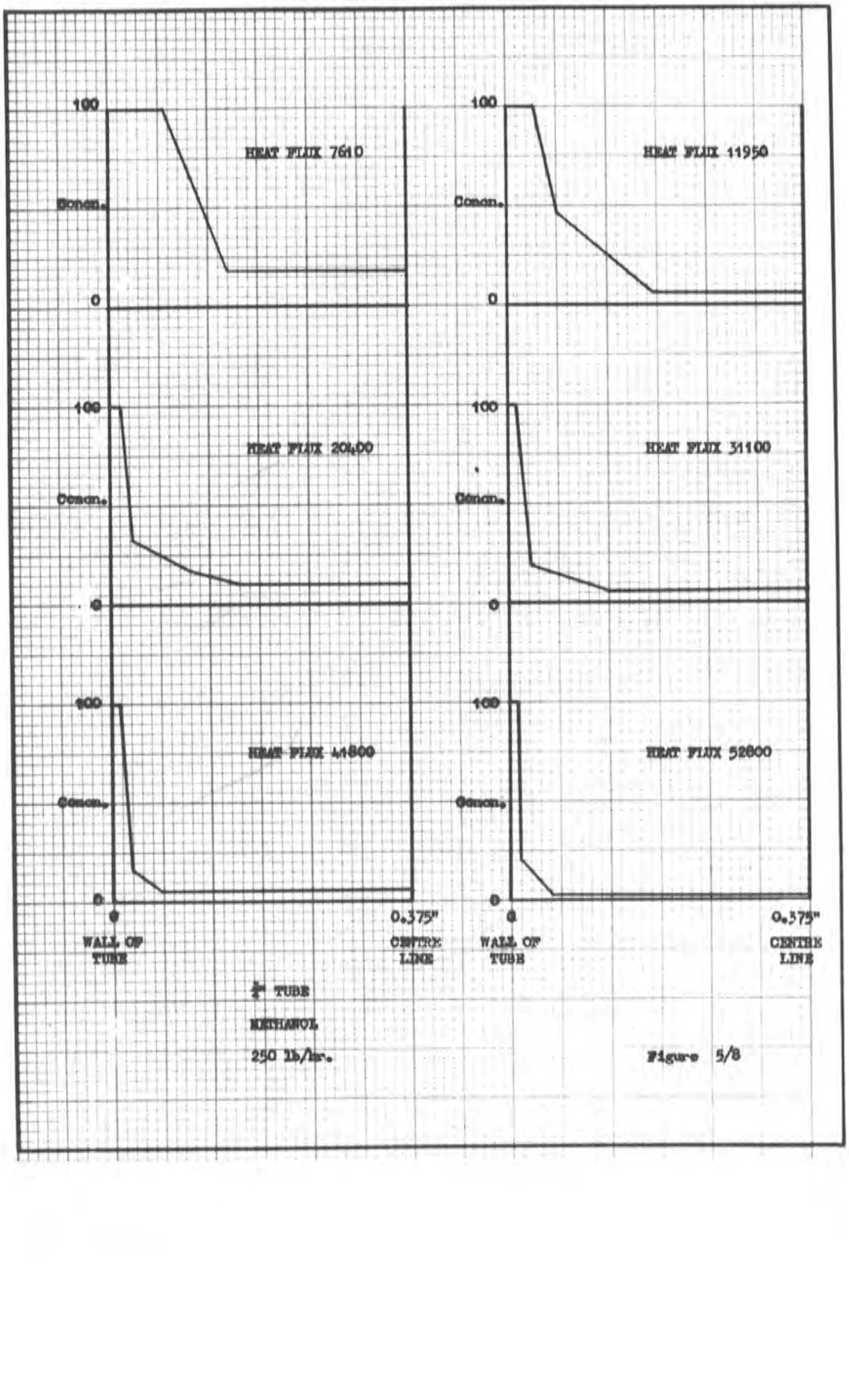
HEAT FLUX 7610
0
100
HEAT FLUX 20400
0
0
WALL OF TUBE
0.375"
CENTRE LINE
100
HEAT FLUX 11950
0
100
HEAT FLUX 31100
0
0
WALL OF TUBE
0.375"
CENTRE LINE
¾" TUBE
METHANOL
150 lb/hr.

Figure 5/7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HEAT FLUX 7610
Concn.
0
100
HEAT FLUX 11950
Concn.
0
100
HEAT FLUX 20400
Concn.
0
100
HEAT FLUX 31100
Concn.
0
100
HEAT FLUX 44800
Concn.
0
100
HEAT FLUX 52800
Concn.
0
0
0.375"
WALL OF TUBE
CENTRE LINE
0
0.375"
WALL OF TUBE
CENTRE LINE
¾" TUBE
METHANOL
250 lb/hr.

Figure 5/8

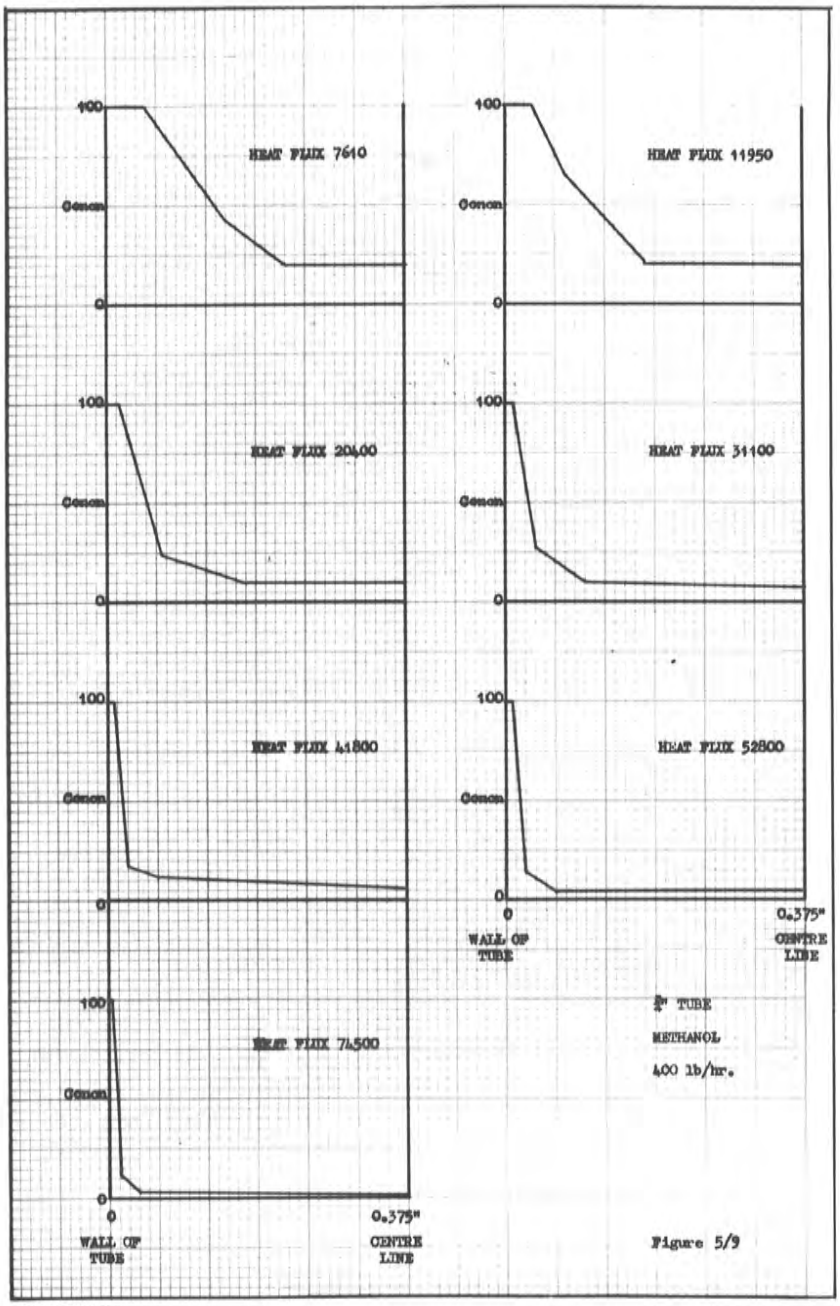

100
Concn
0
HEAT FLUX 7610
HEAT FLUX 11950
HEAT FLUX 20400
HEAT FLUX 31100
HEAT FLUX 41800
HEAT FLUX 52800
HEAT FLUX 74500
WALL OF TUBE
0.375"
CENTRE LINE
¾" TUBE
METHANOL
400 lb/hr.

Figure 5/9

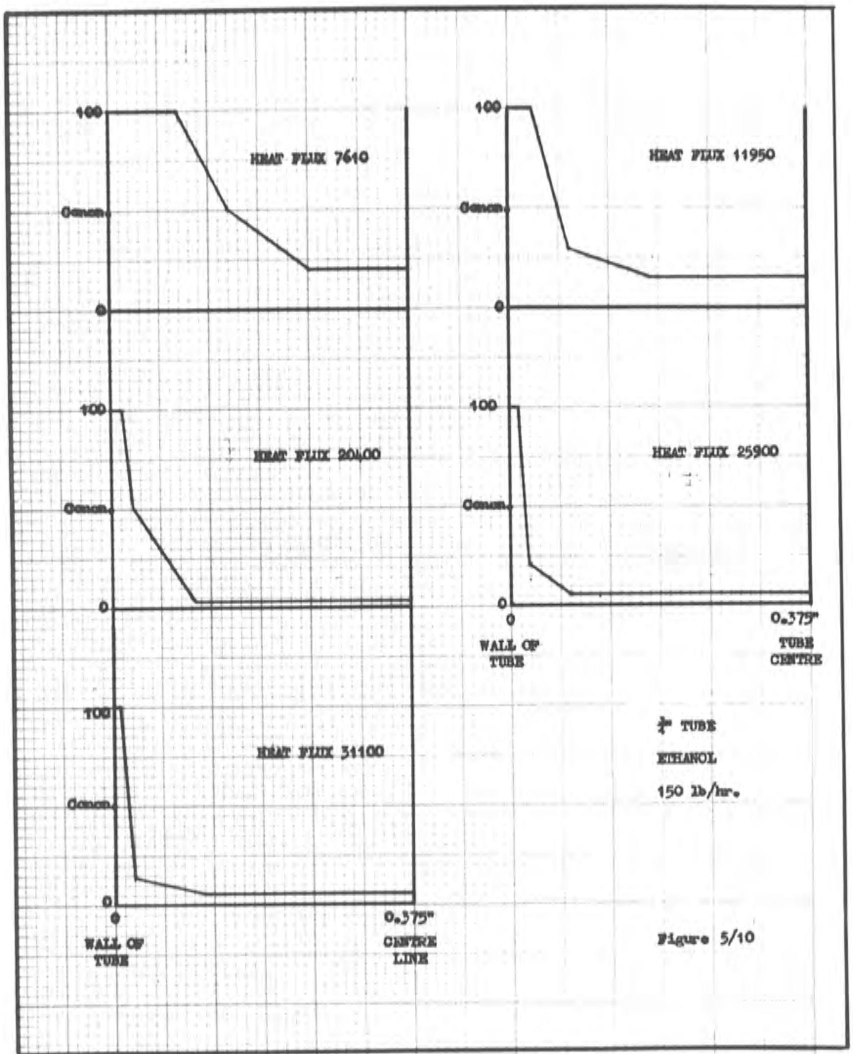

100
HEAT FLUX 7610
Concn.
0
100
HEAT FLUX 11950
Concn.
0
100
HEAT FLUX 20400
Concn.
0
100
HEAT FLUX 25900
Concn.
0
0
0.375"
WALL OF TUBE
TUBE CENTRE
100
HEAT FLUX 31100
Concn.
0
0
0.375"
WALL OF TUBE
CENTRE LINE
¾" TUBE
ETHANOL
150 lb/hr.

Figure 5/10

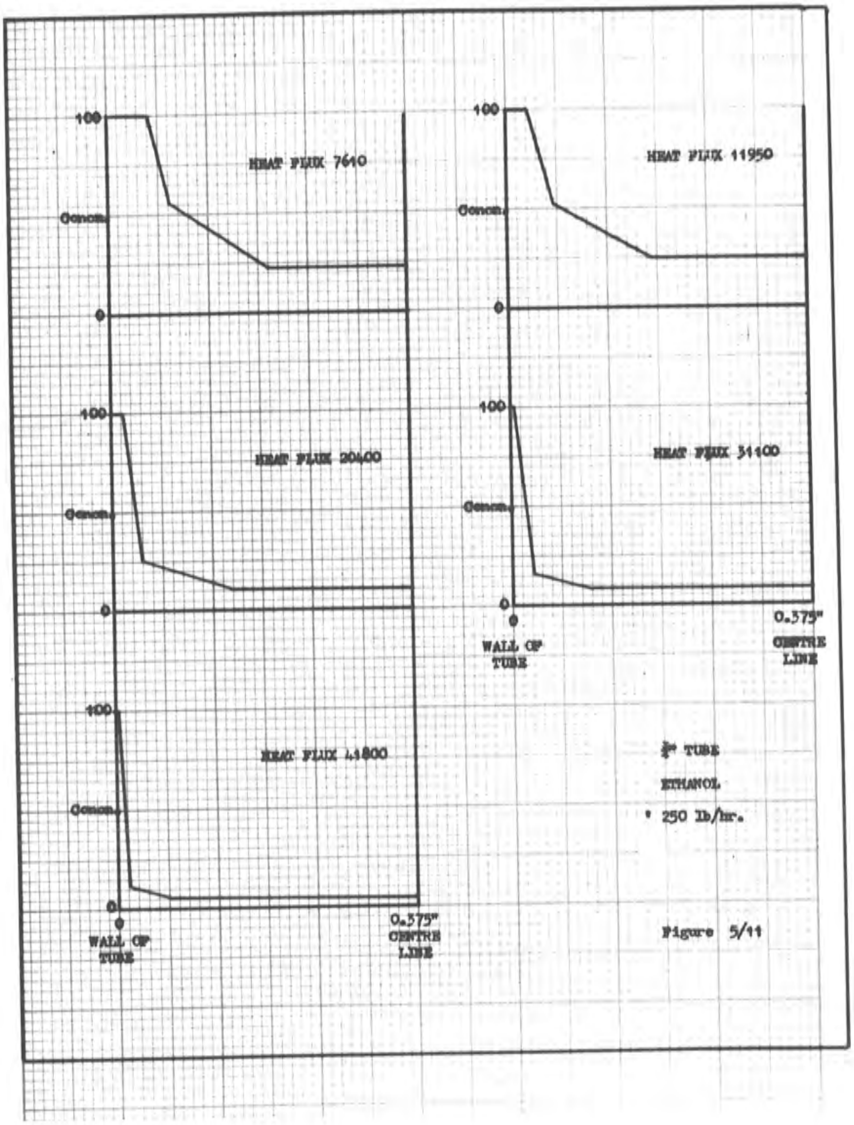
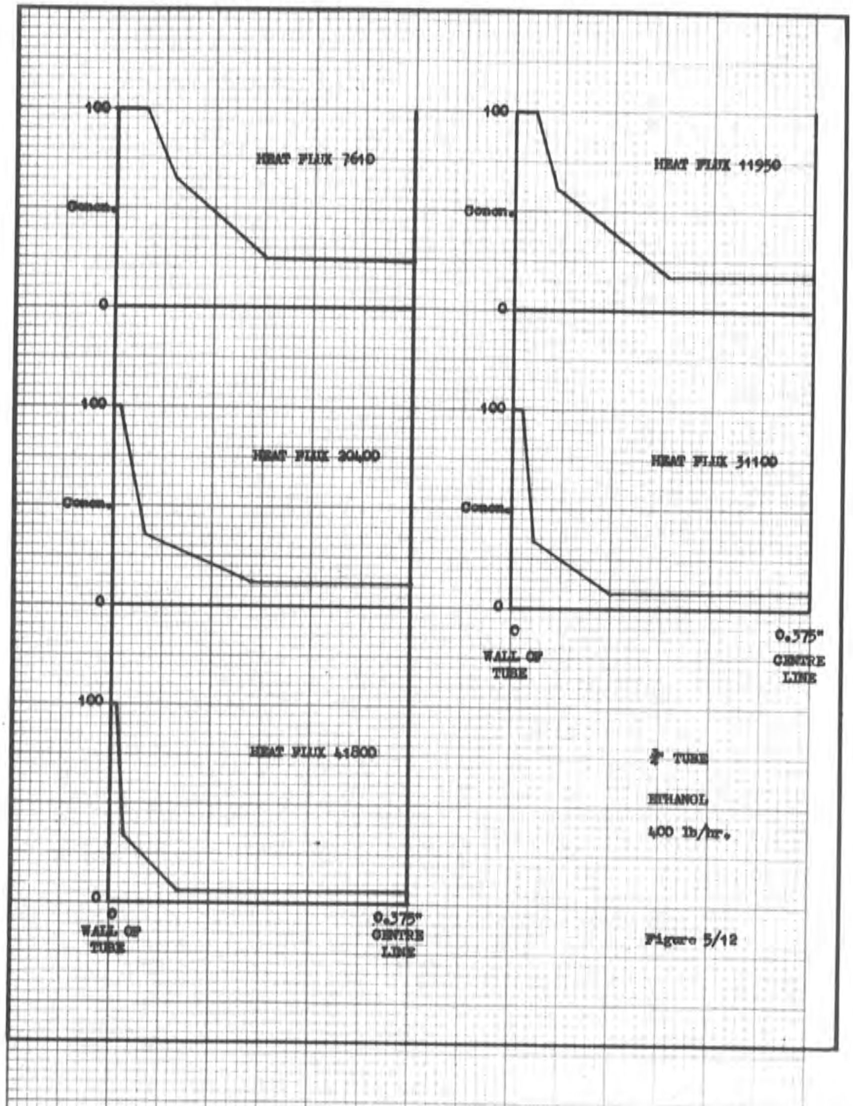

HEAT FLUX 7610
HEAT FLUX 11950
HEAT FLUX 20400
HEAT FLUX 31100
HEAT FLUX 41800
100
Concn.
0
WALL OF TUBE
0.375"
CENTRE LINE
¾" TUBE
ETHANOL
250 lb/hr.

Figure 5/11



100
Concn.
0
HEAT FLUX 7610
HEAT FLUX 11950
HEAT FLUX 20400
HEAT FLUX 31100
HEAT FLUX 41800
0
WALL OF
TUBE
0.375"
CENTRE
LINE
¾" TUBE
ETHANOL
400 lb/hr.

Figure 5/12

APPENDIX 6

PHYSICAL DATA.

Material	Temp °F	λ Btu/lb	ρ_L lb/cu ft	ρ_V lb/cu ft	c_L Btu/lb F°	σ lbf/ft	μ_L lb/ft	k_L Btu/hr sqft F°/ft.
ref		(1)	(2)	(1)	(3)	(4)	(5)	(6)
Water	210	972	59.9	0.0357	1.01	0.00406	0.000178	0.393
	220	966	59.6	0.0433	1.01	0.00397	0.000162	0.394
	230	959	59.4	0.0521	1.01	0.00389	0.000149	0.395
	240	952	59.1	0.0621	1.01	0.00381	0.000136	0.396
	250	946	58.8	0.0724	1.01	0.00373	0.000125	0.397
	260	939	58.5	0.0840	1.01	0.00366	0.000114	0.397
		(7)	(8)	(9)	(3)	(4)	(5)	(10)
Methanol	150	472	47.5	0.0602	0.648	0.00129	0.000235	0.112
	160	467	47.2	0.0735	0.653	0.00126	0.000221	0.111
	170	461	47.0	0.1085	0.659	0.00123	0.000208	0.110
	180	454	46.7	0.1298	0.663	0.00120	0.000195	0.109
		(7)	(11)	(9)	(3)	(12)	(5)	(10)
Ethanol	170	361	46.1	0.099	0.79	0.001175	0.000316	0.072
	180	357	45.7	0.118	0.81	0.00114	0.000282	0.069
	190	353	45.3	0.144	0.83	0.001115	0.000262	0.067
		(7)	(11)	(9)	(3)	(13)	(5)	(14)
Chloroform	140	106.3	88.1	0.263	0.243	0.00149	0.000276	0.080
	150	105.3	87.4	0.313	0.245	0.00144	0.000255	0.080
	160	104.4	86.7	0.369	0.247	0.00139	0.000245	0.080
	170	103.4	85.9	0.433	0.249	0.00133	0.000237	0.080

Material	Temp °F	λ Btu/lb	ρ_L lb/cu ft	ρ_V	c_L Btu/lb F°	σ lbf/ft	μ_L lb/ft	k_L Btu/hr sqft F°/ft
ref		(19)	(15)	(16)	(3)	(17)	(5)	(18)
isopropanol								
	180	281	45.8	0.1315	0.835	0.001155	0.000356	0.090
	190	281	45.5	0.158	0.85	0.00112	0.000322	0.090
	200	281	45.2	0.191	0.87	0.00110	0.000289	0.090
	210	281	44.8	0.228	0.89	0.001075	0.000252	0.090

<u>References for table of physical data:-</u>

1) Callendar, H. L., "Abridged Steam Tables", 4th Edition (1939), Edward Arnold.

2) Badger, W.L., and Banchero, J.T., "Introduction to Chemical Engineering" (1955), p.733, McGraw-Hill.

3) McAdams, W. H., "Heat Transmission", 3rd Edition (1954), p.462, McGraw-Hill.

4) "International Critical Tables" IV p.447 (1928) McGraw-Hill.

5) Perry, J. H., "Chemical Engineers' Handbook", 3rd Edition (1953), p.373, McGraw-Hill.

6) Perry, J. H., ibid, p.459.

7) Perry, J. H., ibid, p.218.

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19) " " " " V p.137.

APPENDIX 7

CONDITIONS AT THE BOTTOM OF THE TUBE

From the equation presented in Chapter 5, it is now possible to predict the manner in which the heat transfer coefficient varies with the bulk velocity when climbing film conditions predominate. It is the purpose of this section to investigate the conditions at the bottom of the tube where the vapour is just beginning to be formed and hence to find where along the climbing film graph the line representing any particular set of conditions will join.

Graphs of the heat flux against the temperature difference have been plotted for the bottom point of the tube, and these are shown in figures 7/1 to 7/8.

As can be seen, the graphs have two slopes. The lesser of the two has a slope of from 1.2 to 1.8 and an average of 1.5, while the steeper varies from 3 to 11 with an average of 6. These slopes correspond to those presented in the literature for forced convection and nucleate boiling (2, 23, 24, 25, 26, 27, 71).

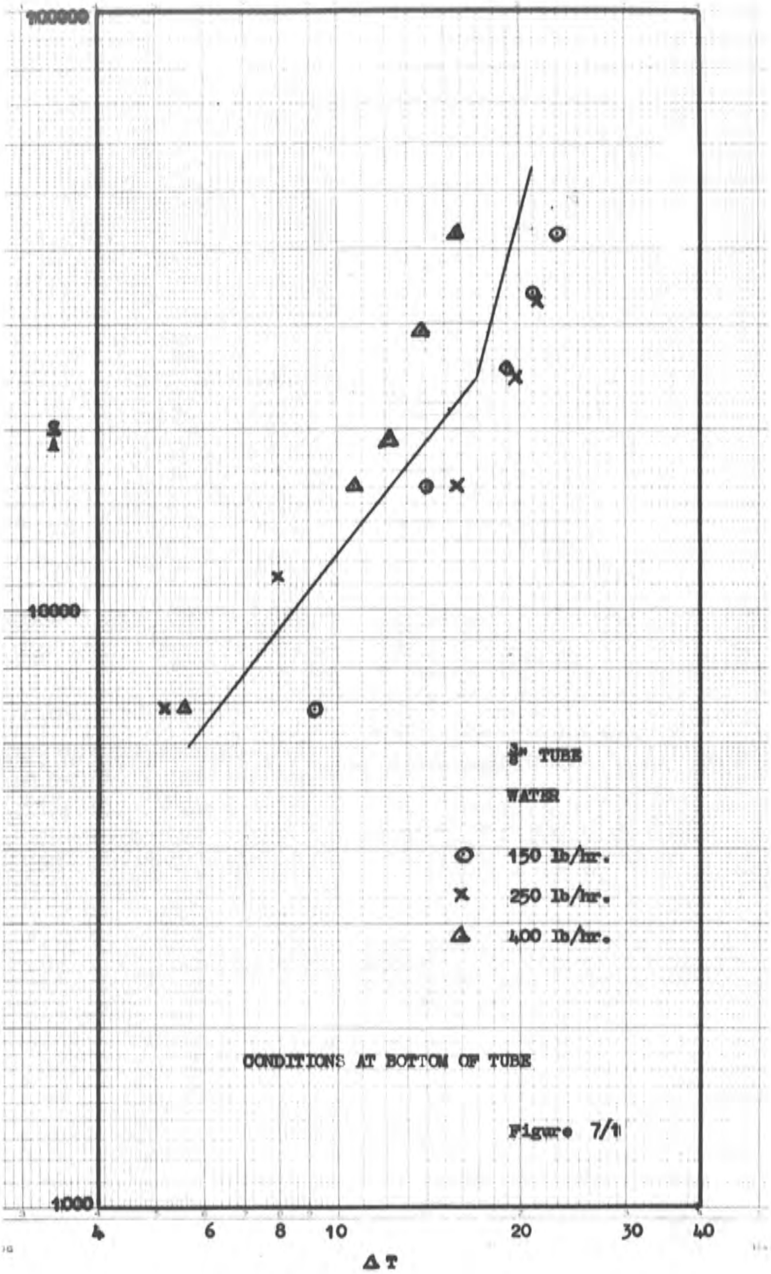
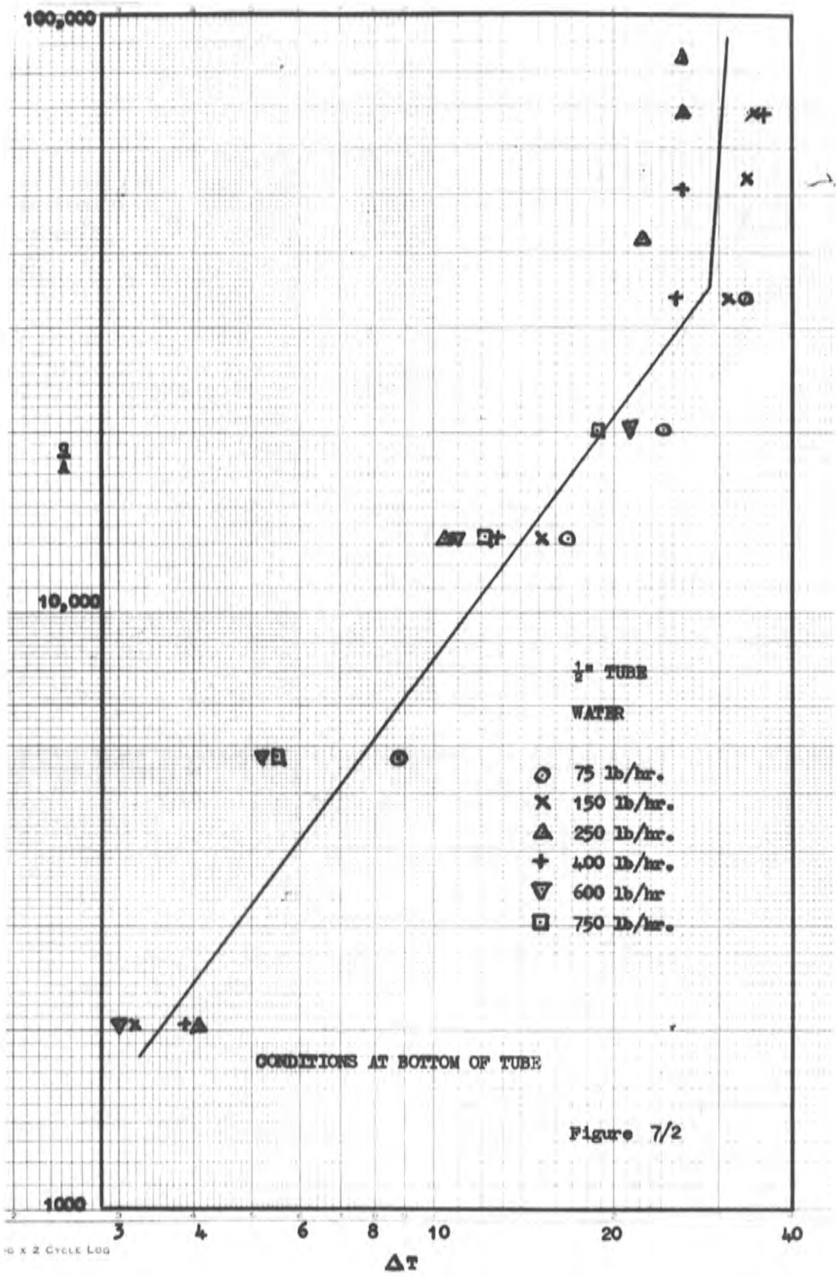

100000
10000
1000
Q/A
4
6
8
10
20
30
40
Δ T
3/8" TUBE
WATER
150 lb/hr.
250 lb/hr.
400 lb/hr.
CONDITIONS AT BOTTOM OF TUBE

Figure 7/1



100,000
10,000
1000
Q/A
3
4
6
8
10
20
40
ΔT
½" TUBE
WATER
75 lb/hr.
150 lb/hr.
250 lb/hr.
400 lb/hr.
600 lb/hr
750 lb/hr.
CONDITIONS AT BOTTOM OF TUBE

Figure 7/2

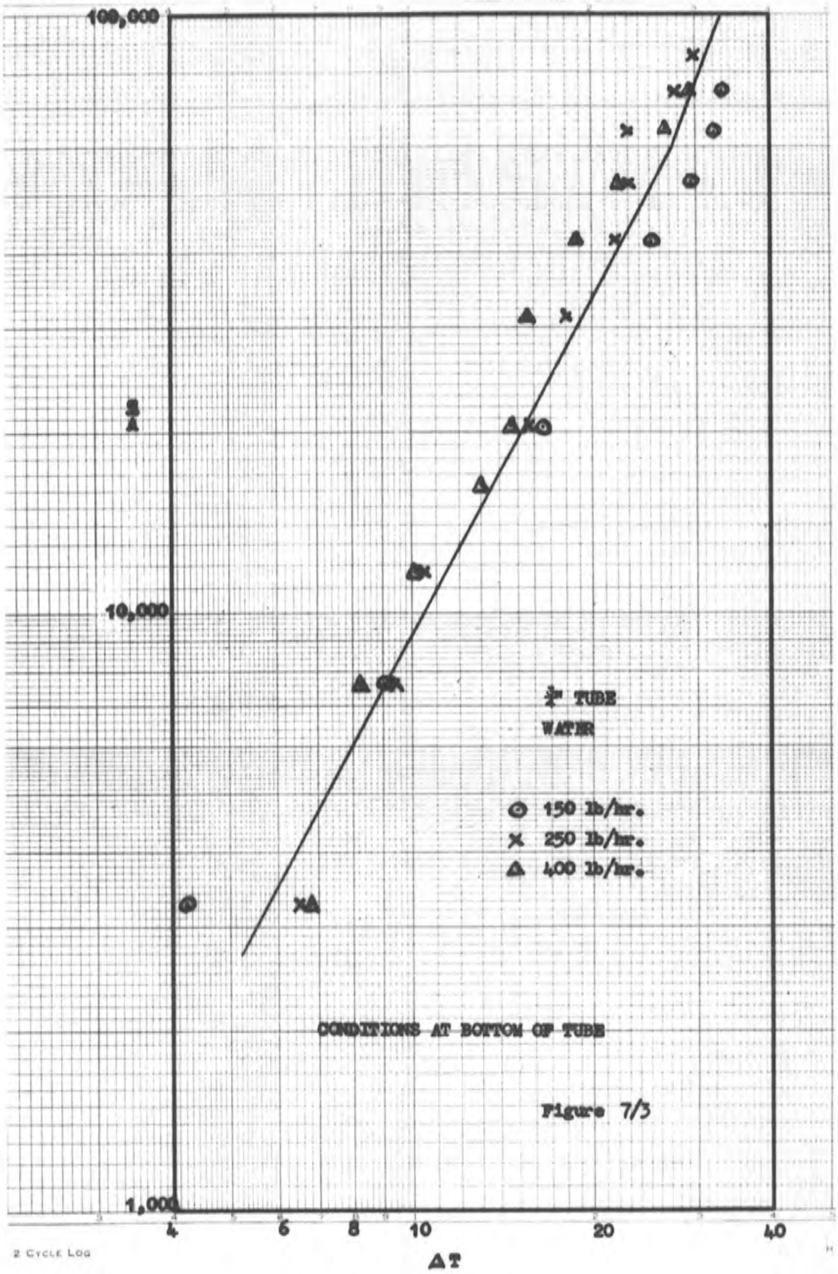
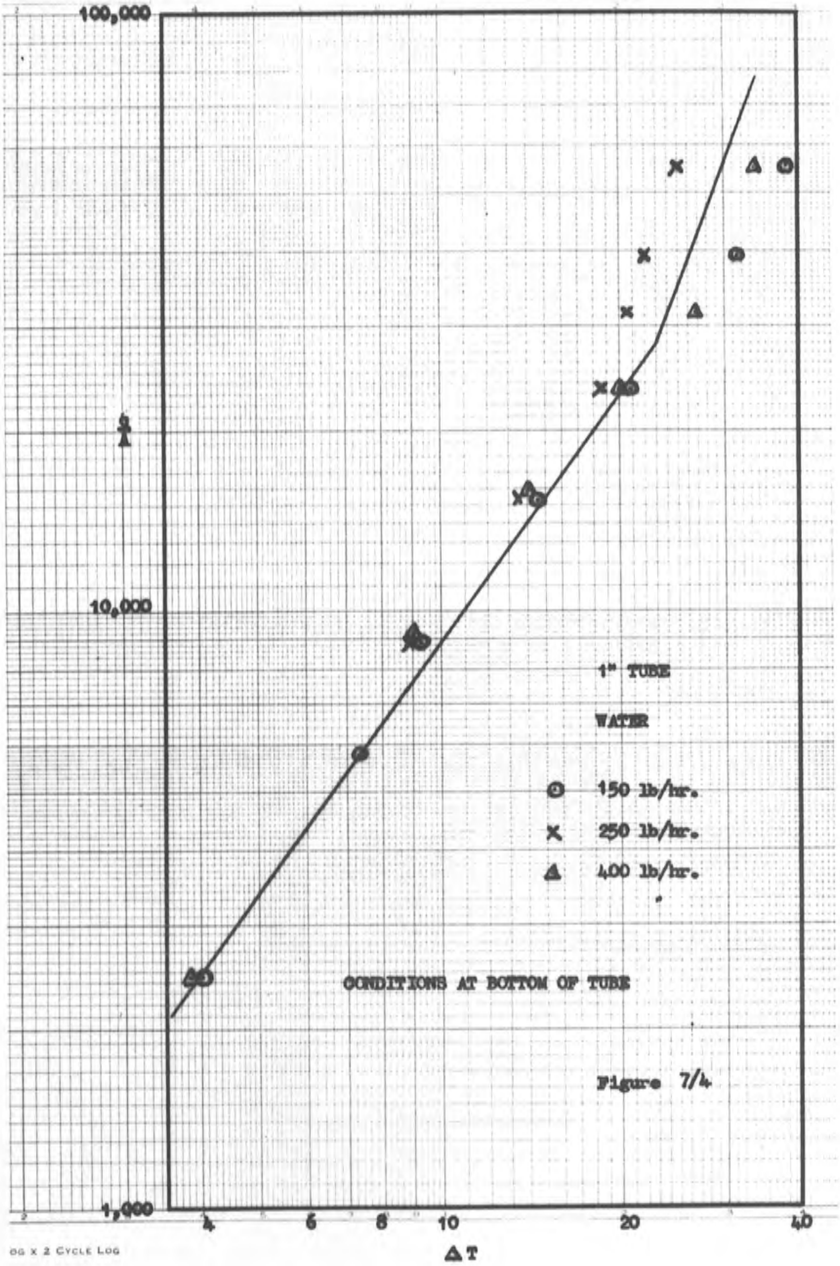

100,000
10,000
1,000
Q/A
4
6
8
10
20
40
ΔT
2 CYCLE LOG
¾" TUBE
WATER
⊙ 150 lb/hr.
× 250 lb/hr.
△ 400 lb/hr.
CONDITIONS AT BOTTOM OF TUBE

Figure 7/3


100,000
10,000
1,000
q/A
4
6
8
10
20
40
ΔT
1" TUBE
WATER
150 lb/hr.
250 lb/hr.
400 lb/hr.
CONDITIONS AT BOTTOM OF TUBE
Figure 7/4

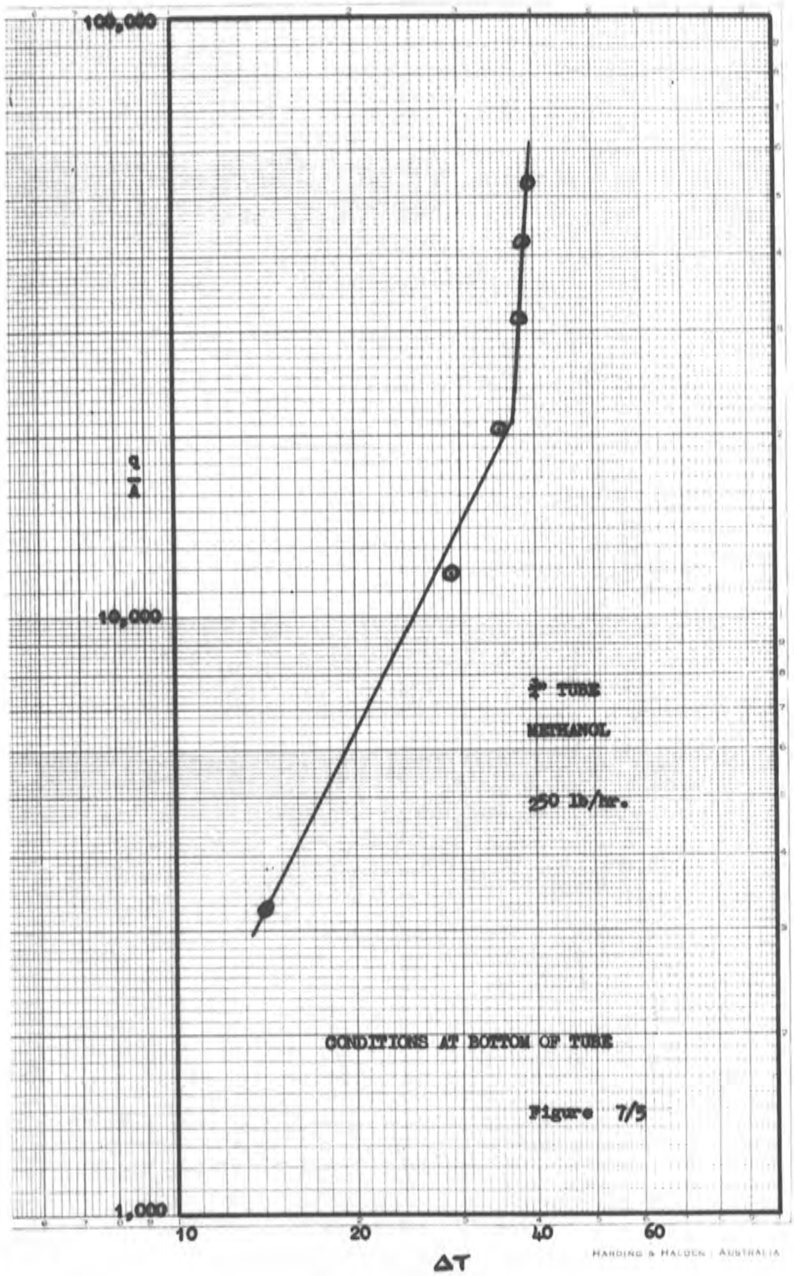
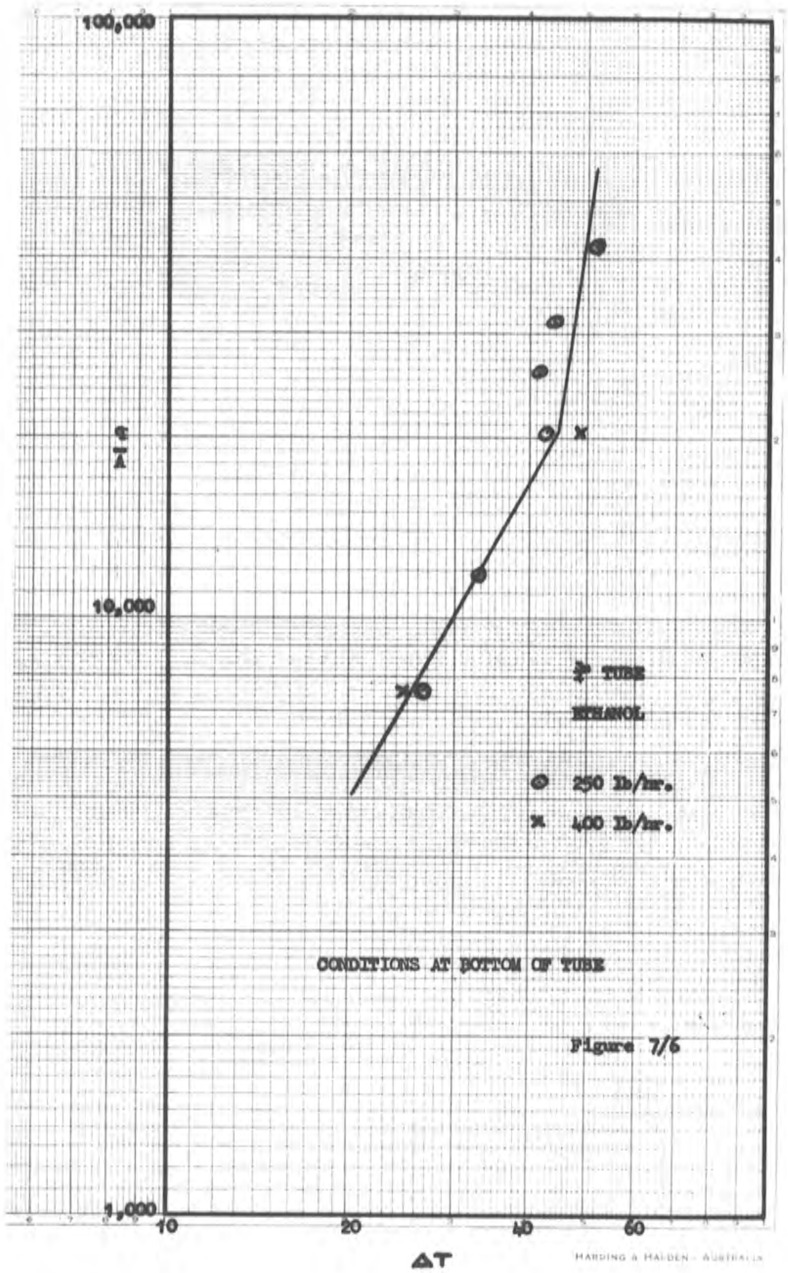

100,000
10,000
1,000
q/A
10
20
40
60
ΔT
¾" TUBE
METHANOL
250 lb/hr.
CONDITIONS AT BOTTOM OF TUBE
HARDING & HALDEN : AUSTRALIA

Figure 7/5



100,000
10,000
1,000
q/A
10
20
40
60
ΔT
¾" TUBE
ETHANOL
250 lb/hr.
400 lb/hr.
CONDITIONS AT BOTTOM OF TUBE
HARDING & HALDEN - AUSTRALIA

Figure 7/6

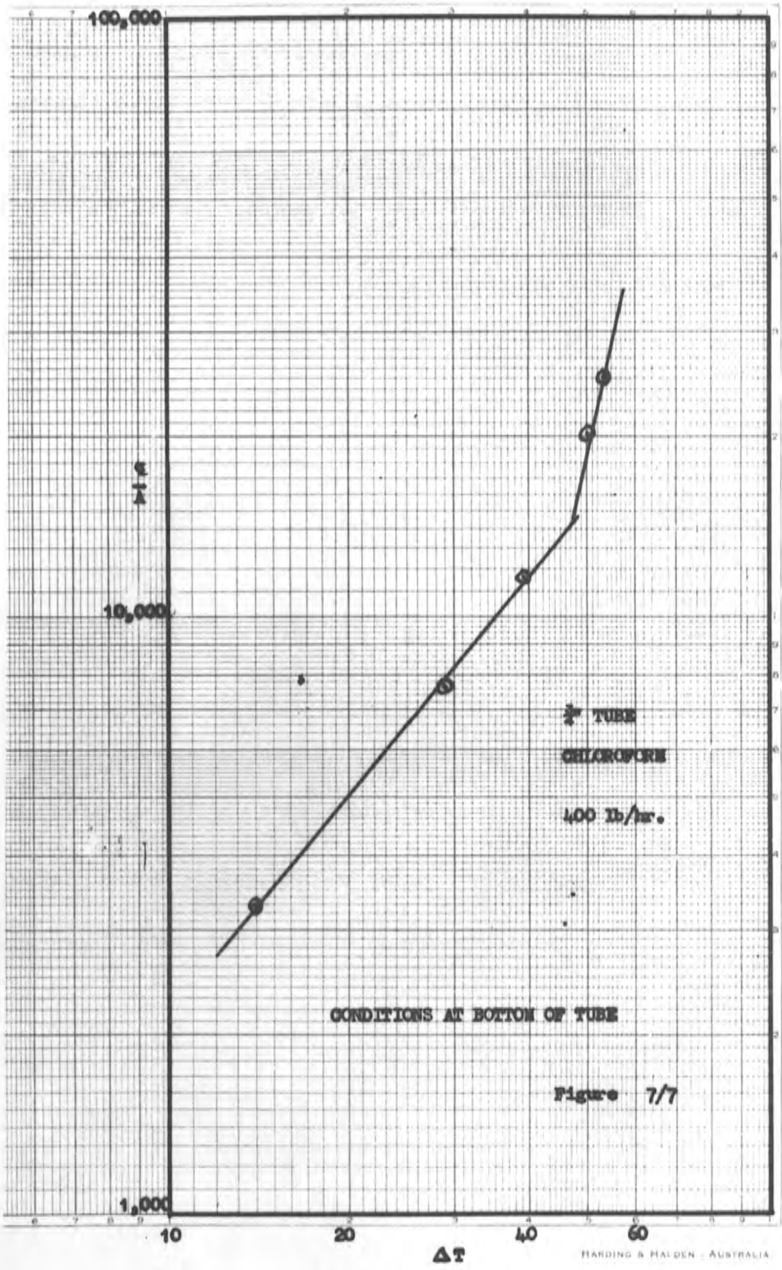

100,000
10,000
1,000
q/A
10
20
40
60
ΔT
3/4" TUBE
CHLOROFORM
400 lb/hr.
CONDITIONS AT BOTTOM OF TUBE
HARDING & HALDEN : AUSTRALIA

Figure 7/7

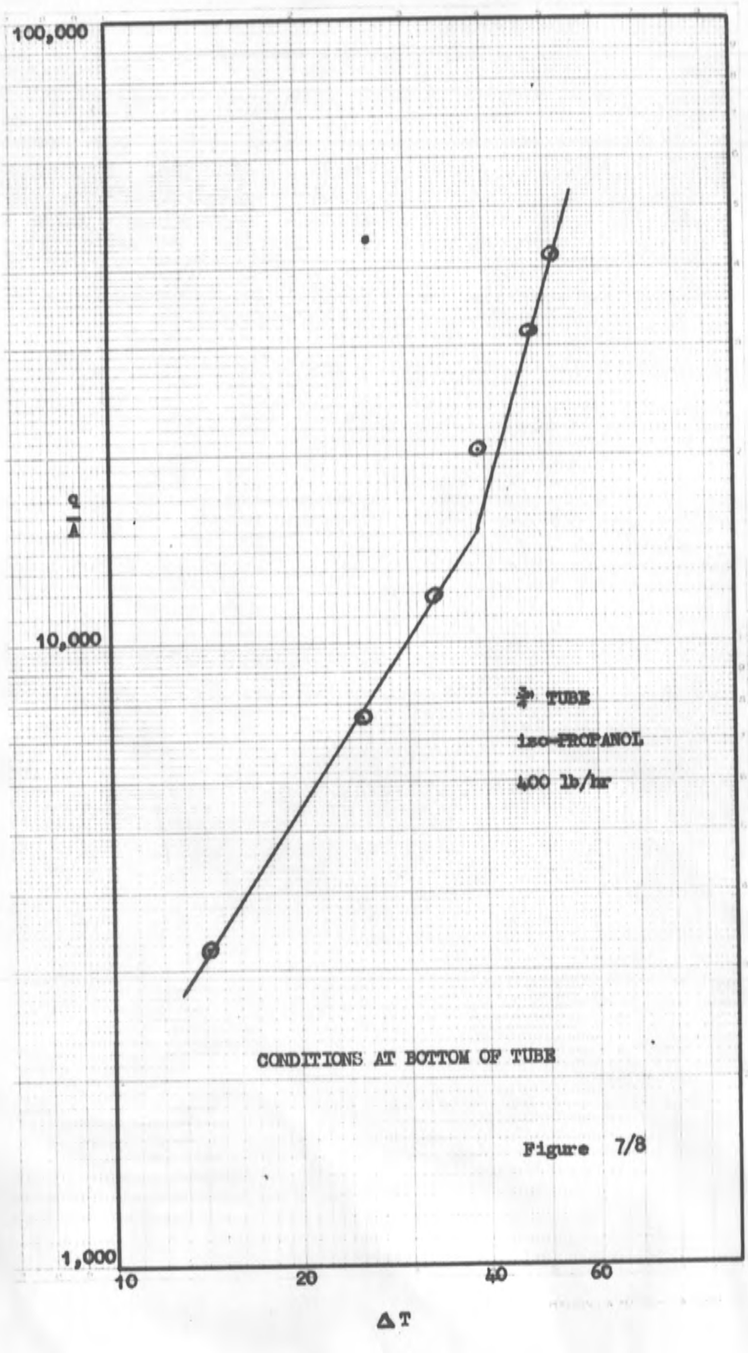

100,000
10,000
1,000
q/A
10
20
40
60
ΔT
¾" TUBE
iso-PROPANOL
400 lb/hr
CONDITIONS AT BOTTOM OF TUBE

Figure 7/8

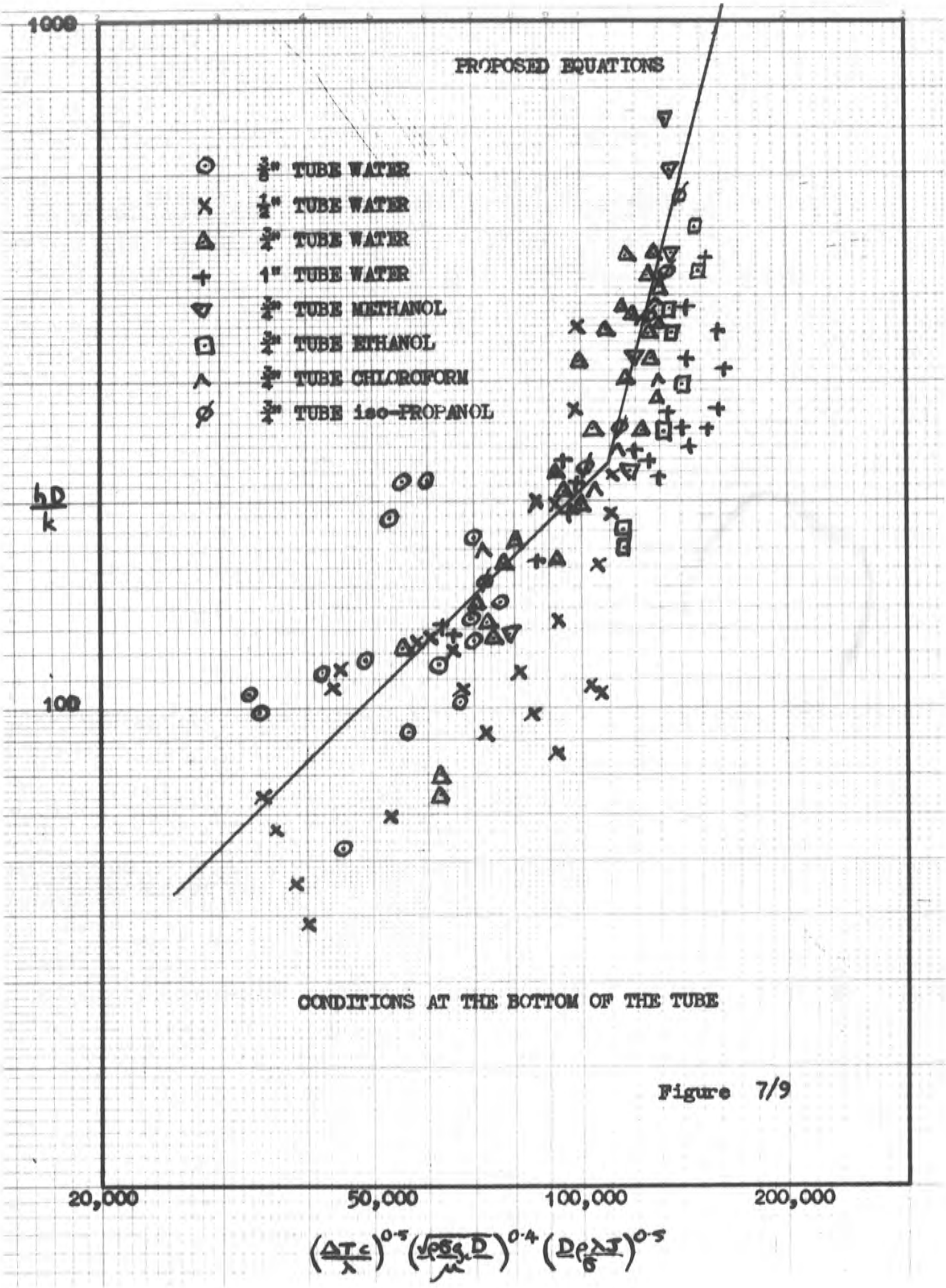
However, most of these references show that as the velocity of the fluid increased so the forced convective boiling line intersected the nucleate boiling line at higher values. This cannot be observed from the results presented in the graphs. Even with the results for the $\frac{1}{2}$" tubs, where the feed rate is increased from 75 lb/hr to 750 lb/hr, only a very slight trend can be seen. However, the scatter of points about the nucleate boiling section, which should be a common line, is as great as this trend. Hence it is concluded that over this range of experimental conditions there is insufficient evidence to include the feed rate or velocity as a factor in the investigations.

Using dimensional analysis, the following equation was found to give the best fit for the line of lesser slope:-

$$\frac{hD}{k} = 0.00021 \left(\frac{\Delta T c}{\lambda}\right)^{0.5} \left(\frac{\rho \sigma g_c D}{\mu}\right)^{0.4} \left(\frac{D \rho N}{\sigma}\right)^{0.5}$$

The graph shown in figure 7/9 represents the values which are presented in figures 7/1 to 7/8.

It was also found that the best fit for the nucleate boiling line was given by an equation on the same ordinates as for the first equation, but with a slope four times as great.



PROPOSED EQUATIONS
3/8" TUBE WATER
1/2" TUBE WATER
3/4" TUBE WATER
1" TUBE WATER
3/4" TUBE METHANOL
3/4" TUBE ETHANOL
3/4" TUBE CHLOROFORM
3/4" TUBE iso-PROPANOL
1000
100
hD/k
20,000
50,000
100,000
200,000
$\left(\frac{\Delta T c}{\lambda}\right)^{0.5}\left(\frac{\sqrt{\rho \sigma g_c} D}{\mu}\right)^{0.4}\left(\frac{D \rho \lambda J}{\sigma}\right)^{0.5}$
CONDITIONS AT THE BOTTOM OF THE TUBE

Figure 7/9

The equation for this line is then :-

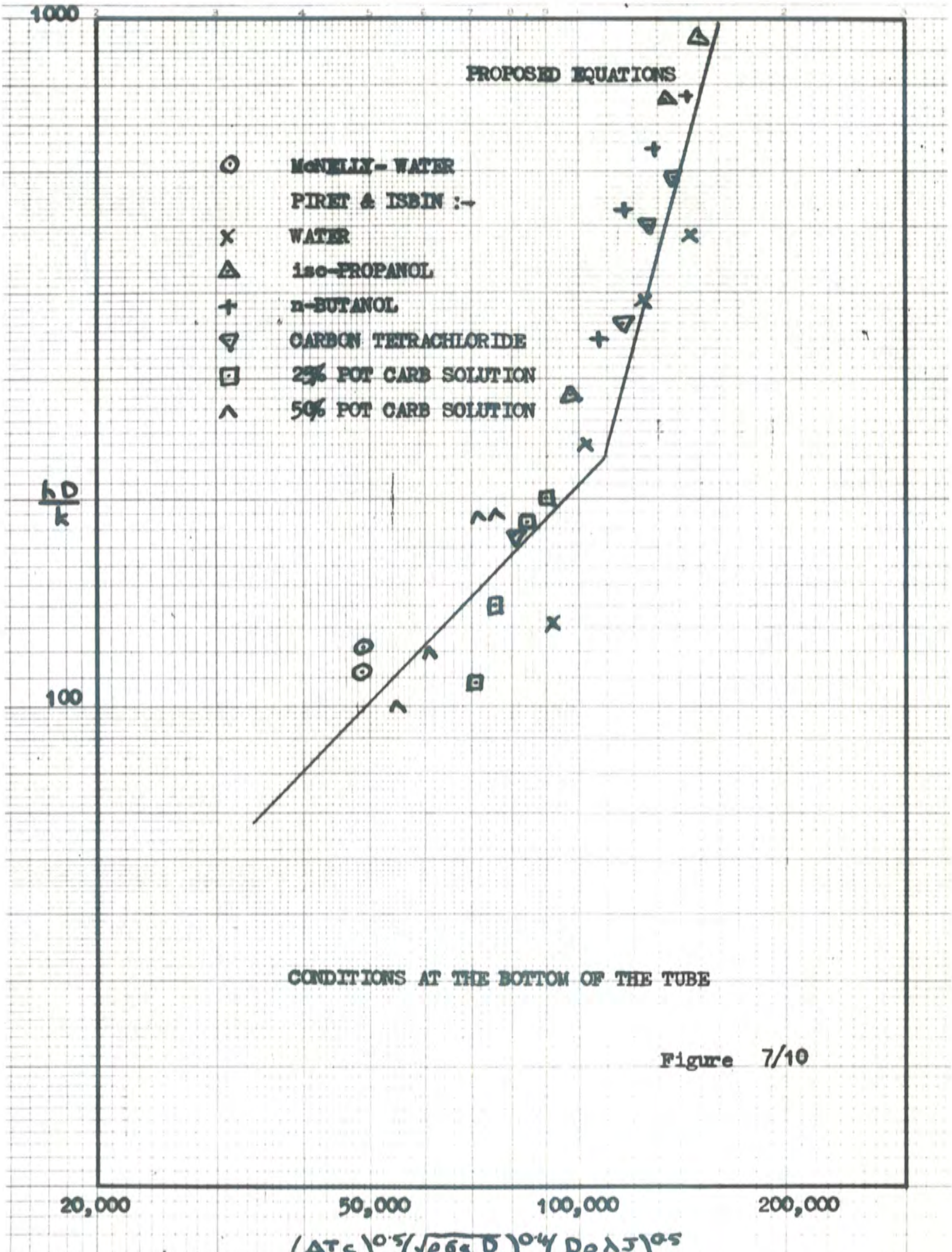
$$\frac{hD}{k} = 1.54 \times 10^{-18}\left(\frac{\Delta T c}{\lambda}\right)^{2}\left(\frac{\rho\sigma g_c D}{\mu}\right)^{1.6}\left(\frac{D\rho\lambda J}{\sigma}\right)^{2}$$

and it is also presented in figure 7/9.

In figure 7/10 are shown data from McNelly (23), for water and Piret and Isbin (57) for water, solvents and potassium carbonate solutions. These results were the only ones found in the literature presented by authors, who, having worked under similar conditions, published enough information to enable a comparison to be made. As can be seen, good agreement has been found.

Discussion of equations.

The equations were found by fitting the individual lines shown in figures 7-1 to 7-8. It will be noted that there is a considerable scatter of points from the line of the equations, some points falling outside the ± 50% range. Although at low values of ΔT this is within the accepted limits, it is outside these limits for some of the higher values. However in this correlation ΔT occurs on both sides of the equation. Consequently an inaccuracy of say +30% will now be shown on the graph as +48% and it will then be seen that the scatter is within acceptable limits.



$\left(\frac{\Delta T c}{\lambda}\right)^{0.5}\left(\frac{\sqrt{\rho \sigma g_c} D}{\mu}\right)^{0.4}\left(\frac{D \rho \Delta J}{\sigma}\right)^{0.5}$

Figure 7/10

The left hand side of the expression is the Nusselt number and this is the ratio of total heat transferred to the heat transferred by diffusion. $\frac{\Delta T c}{\lambda}$ has been described (72) as the bubble pumping number and it has been shown (73) that it is a modification of the Stanton number, the ratio of total heat transferred to momentum transferred by turbulence. It has also been described (36) as the ratio of the superheat in the liquid film to the total heat transferred. $\frac{\rho \sigma g_c D}{\mu}$ is the ratio of the resistance to the creation of new surface to momentum diffusivity while $\frac{D \rho \lambda J}{\sigma}$ is the ratio of the mechanical to the thermal energy required to create new surface.

If, as seems likely from the shape of the graph, nucleate boiling occurs in the lower part of the tube at the higher heat fluxes, we can expect that the manner in which a particular run is done, as discussed in Chapter 2, will effect the heat transfer. It is unfortunate that small changes of the value of the ordinate cause big changes in the corresponding Nusselt number.

APPENDIX 8

VARIATION OF FILM THICKNESS WITH BULK VELOCITY.

In figures 8-1 to 8-4 are presented graphs showing the variation of the film thickness with the bulk velocity for the results presented in Chapter 4.

It cannot be claimed that a clearly defined relationship has been established. The results for methanol give a strong indication that the film thickness varies with the feed rate, and except for one point so do the results for ethanol. A clear line for water at 150 lb/hr in the ¾" tube can be seen but the lines for the other two feed rates are conjectural. No variation can be seen with the 1" tube.

All graphs suggest that a change in slope occurs between 30 and 50 ft/sec. The heat fluxes corresponding to these velocities approximate to the runs shown in figures 18-26 which just reach the climbing film line. Hence it is probable that this change in slope occurs at the transition from slug type flow to climbing film.

The slope of all the climbing film lines is -4/3 except for the water in the ¾" tube which has a slope of -1, and the slope for the slug type boiling lines is -½ except for the methanol which is -1.

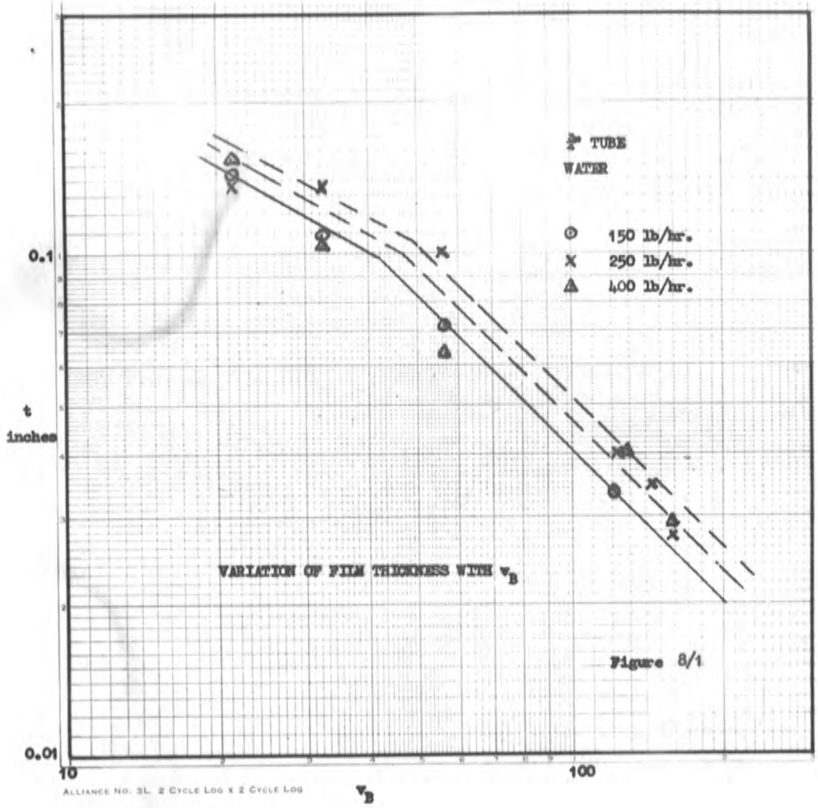


Figure 8/1

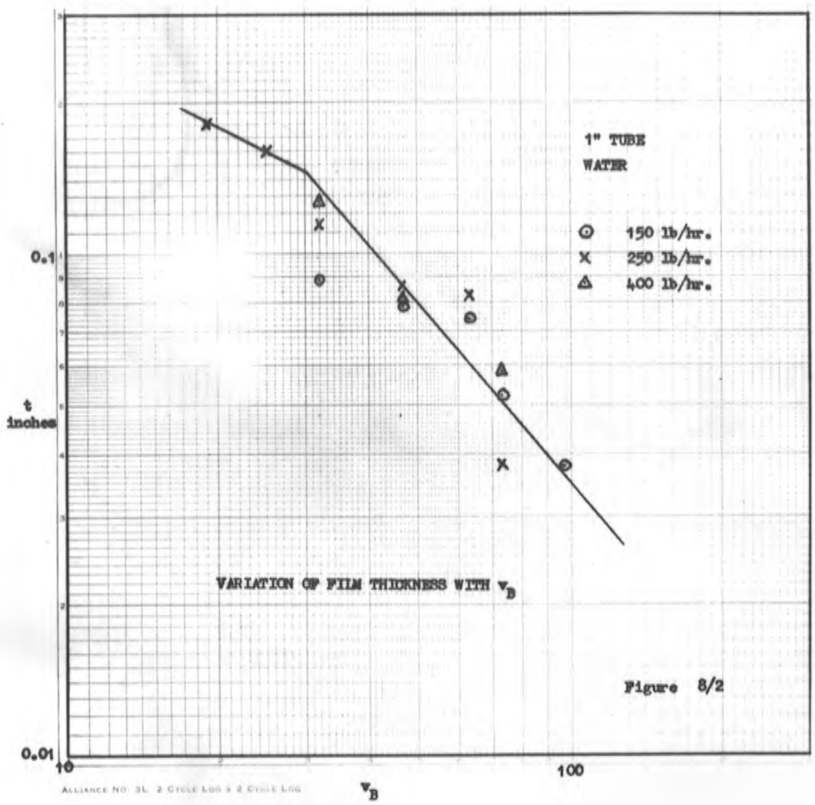


Figure 8/2

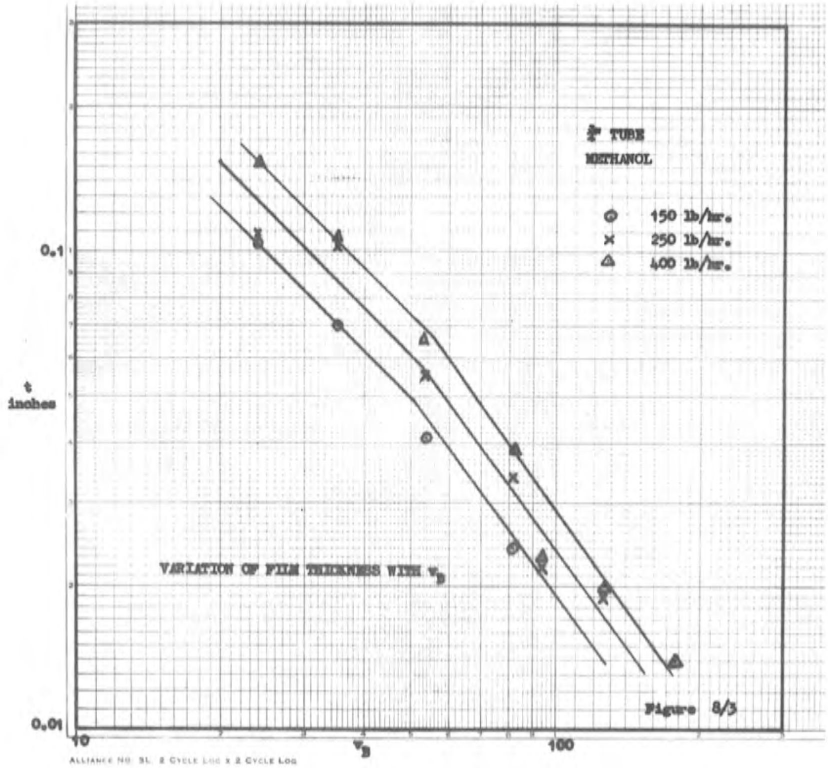


Figure 8/3

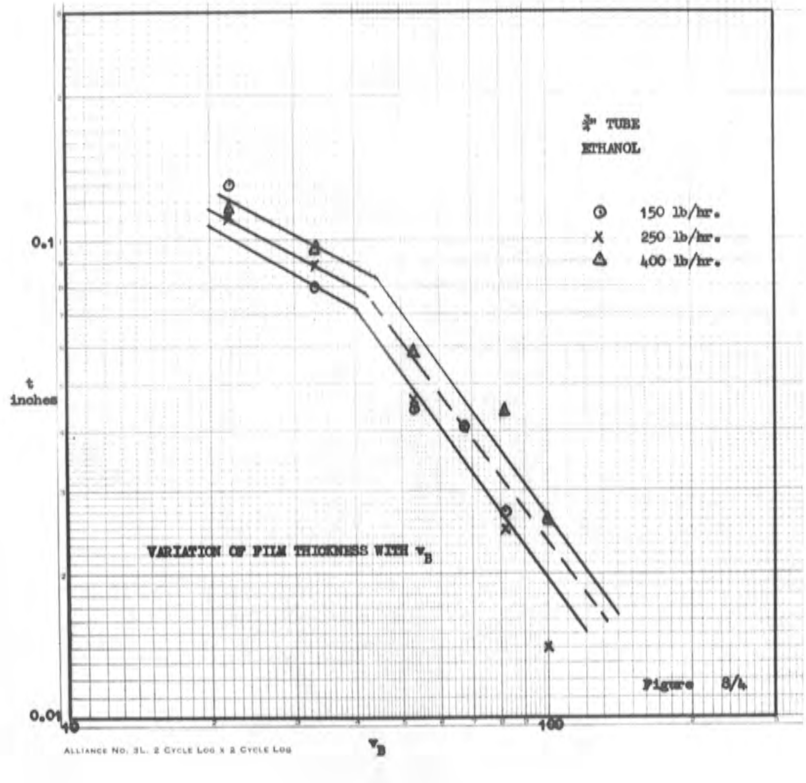


Figure 8/4

APPENDIX 9.

NOMENCLATURE.

A, A'	Constants.
A	Area of heat transfer, (sq.ft).
B	Steam content, (mass % evaporated).
c	Specific heat, (BTU/hr, F^{o}).
D	Diameter, (ft).
g	Acceleration due to gravity, (ft/sec^2).
g_c	Conversion factor in Newton's law of motion, (lb mass ft/lb force sec^2).
h	Coefficient of heat transfer, (BTU/hr sq.ft F^{o}).
k	Thermal conductivity, (BTU/hr sq.ft F^{o}/ft).
q	Heat flow, (BTU/hr).
R	Electrical resistance, (Ohms).
T	Temperature, (^{o}F or ^{o}C).
t	Film thickness, (feet or inch).
v	Velocity, (ft/sec).
V	Specific volume, (cu.ft/lb).
w	Feed rate, (lb/hr).
x	Steam quality, (wt %).

X_c	Inductive reactance, (Ohms).
X_L	Capacitive reactance, (Ohms).
X_{tt}	Martinelli parameter $= \left(\frac{x}{1-x}\right)^{0.9}\left(\frac{V_V}{V_L}\right)^{0.5}\left(\frac{\mu_V}{\mu_L}\right)^{0.1}$
α	Temperature coefficient of electrical resistance, $(1/C^{o})$.
ΔT	Temperature difference, (F^{o}).
λ	Latent heat, (BTU/lb).
μ	Absolute viscosity, (lb mass/ft hr).
ρ	Density, (lb/cu.ft).
σ	Surface tension, (lb force/ft).
ϕ, ϕ', ϕ''	Prefixes meaning "a function of".

Suffixes.

B	Bulk, Applied to velocity as defined in Chapter 4.
L	Liquid.
V	Vapour.

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