SOME STUDIES ON CONDENSATION HEAT TRANSFER IN DIVERGING–CONVERGING TUBE

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C E R T I F I C A T E

This is to certify that the thesis entitled "Some Studies on Condensation Heat Transfer in Diverging-Converging Tube" submitted by Shri T.K. Chakravarty in fulfilment of the requirements of the degree of Doctor of Philosophy in Chemical Engineering, is a bonafide record of the investigations carried out by him in the Department of Chemical Engineering, Indian Institute of Technology, Kharagpur, under my supervision and guidance. In my opinion this thesis has reached the standard fulfilling the requirements of the Ph.D. degree as prescribed in the regulations of this Institute.

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(TAPAS KUMAR CHAKRAVARTY)

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INTRODUCTION

CHAPTER - 1.

1. INTRODUCTION

1.1 Advances In Heat Transfer And Importance of Condensation In Process Industries.

The fast growth of chemical industry both in complexity and number made it extremely necessary to work for the rapid development of process planning and equipment with a strict control over the cost involved. Heat exchangers are one of the omnipresent equipment found in chemical industries. As a natural consequence this particular chemical plant equipment plays a vital role in the economics of various process industries. Keeping pace with the tremendous technological advancement in all spheres of science and technology, researchers, in the different parts of the globe are trying to improve the operational efficiency of various types of heat exchangers during the last few decades, and the need to augment or intensify heat transfer has inspired engineers in searching for new methods and techniques. Motivation has not only come from the economic pressures but also from developing technology requiring more efficient equipment smaller in size and lesser in weight.

Condensation is a process of phase transformation from the vapour to the liquid state. The importance of condensation lies in the fact that this phase transition forms an integral part of every Rankine power generation cycle, production of liquid oxygen and nitrogen, chemical process plants and many other cooling devices of industrial use. In different industrial applications involving condensation heat transfer, a very common approach to escalate the efficiency of the process, is to increase the heat transfer coefficient. Only recently theoretical modelling and comprehensive experiments have been reported in order to define more clearly the conditions of augmentative techniques with improved heat transfer.

The various augmentative techniques adopted are, in brief, to have flow separation or flow injection with the help of ejectors, simulating rotational or vibrational motions, application of hydrophobic coatings, extending the effective heat transfer surface or by the promotion of turbulence, to name a few. These techniques have been discussed in Chapter - 2.

1.2 Objective Of The Present Investigation.

The main objective of this work is to study the heat transfer characteristics in a process of condensation of pure vapours in periodically constricted tubes, precisely, diverging-converging type of tube system. It is also intended to examine the enhancement of heat transfer coefficient in the proposed system compared to that in the straight tube. Relevant literature survey reveals that practically nothing has been reported on the condensation of pure (or mixed) vapours on a surface with varying cross sectional area like diverging-converging system, although sizeable amount of theoretical and experimental work has been performed and reported in the literature for condensation of vapour on inside and outside surfaces of straight tubes at horizontal or vertical positions. Chapter-2 gives detail literature survey.

In view of the objectives stated, mathematical analysis for condensation of pure vapours in diverging-converging tubes has been attempted. Chapter - 3 contains mathematical analysis in detail. Experiments have been performed to verify the validity of the theoretical values computed from mathematical models. Chapter - 4 and Chapter - 5 are dealing with experimental procedure and results and discussion respectively.

1.3 Scope For Future Development.

Since the present work has been confined essentially to the filmwise condensation of pure vapour characterising Newtonian, incompressible and laminar, there is excellent scope for future extension of this work to -

a) condensation of mixed vapours (both Newtonian and non-Newtonian);

b) dropwise condensation (both pure and mixed vapours);

c) condensation of vapour mixed with non-condensible;

d) compressible flow, and

e) unsteady state transfer.

The heat transfer study could be extended further to -

i) tubes in horizontal position;

ii) heat transfer with constant heat flux but variable wall temperatures;

iii) fluids with density and viscosity highly dependent on temperature, and

iv) heat transfer with chemical reaction.

Studies in these directions will be attractive as the system promises excellent prospect for improved and efficient means to augment heat transfer in process equipment.

CHAPTER - 2

LITERATURE REVIEW

CHAPTER - 2

LITERATURE REVIEW

The present state of art in condensation heat transfer is most completely covered in a publication by Isachenko [1] and in well known books on heat transfer theory by Kutateladge [2], Eckert and Drake [3] Collier [4] .

The literature survey in the context of the present work is primarily concerned with condensation of single component system i.e. pure liquid and its vapour in vertically mounted surfaces. The literature on filmwise and dropwise condensation as well as various methods of improving the condensation heat transfer has been reviewed. However, certain areas have been omitted because they are either not directly relevant to the present work or the present state of knowledge does not allow a coherent statement which might have wide spread acceptance.

2.1 Film Condensation.

2.1.1 The Nusselt Analysis For A Laminar Film.

As early as 1961, Nusselt [5] first analysed filmwise condensation of saturated vapour on vertical surface kept at uniform temperature. The analysis considered cases of stagnant vapour without interphase shear and second of a moving vapour with interphase shear. In the second case, the interphase shear was related to the velocity of vapour flow and interphase friction factor. Assumptions made in the analysis are as follows:

(1) The flow of condensate in the film is laminar.

(2) Condensate fluid properties are constant.

(3) Momentum changes of condensate are neglected (i.e. there is essentially a static balance of force).

(4) Liquid film is incompressible.

(5) Interfacial resistance to heat transfer is negligible.

(6) Axial conduction in the film and viscous dissipation is neglected.

(7) Effect of Convection in the film is neglected.

(8) The vapour is stationary and exerts no drag on the downward motion of the condensate.

(9) Interphase friction factor is constant and based on the wall shear for any flow of vapour forced along the vertical surface.

(10) Velocity of vapour flow in the computation of interphase shear is taken as the value far away from the wall.

2.1.2. Improvements To The Original Nusselt Theory.

During 1930's Jacob and his co-workers set forth to investigate experimentally the range of validity of Nusselt's solution, when applied to the condensation inside tubes. This work resulted in papers by Jacob, Erk and Eck [6] and Jacob [7]. Experimental work involved condensation of saturated and superheated stead in vertical tubes near atmospheric pressure. In these experiments the tube surface temperature was kept uniform and the inlet vapour flow was fully developed. For very low inlet steam velocities, Nusselt's simple theory on the assumption that no interphase shear occurs leads to results in close agreement with these heat transfer experiments. For higher inlet vapour velocities agreement was observed with extended (non-zero interphase shear) Nusselt theory only in the entry region of the condensation process. This prompted Jacob et.al., to relax assumption no.(10) in Nusselt's treatment. The resulting equations were solved numerically for particular data at hand and good agreement in heat transfer with the experiment was obtained for a moderate range of steam velocities and temperature differences. With high vapour velocities and high values of temperature differences, consistant under-prediction in heat transfer rate resulted.

The original analysis was then extended by Bromley [8] who considered the effect of subcooling of the condensate and by Rohsenow [9] who also allowed for the non-linear distribution of temperature through the film due to energy convection. Rohsenow showed that the latent heat of vaporisation term in Nusselt's equation for average heat transfer coefficient should be replaced by,

$$\lambda' = \lambda \left[1 + 0.68 \frac{C_p . \Delta T}{\lambda} \right] \quad \ldots \quad \ldots \quad \ldots \quad (2.1)$$

where, $\Delta T = T_{gi} - T_w$

and λ = Latent heat of vapourisation

T_{gi} = Interfacial temperature

T_w = Wall temperature

C_p = Specific heat of liquid phase.

Sparrow and Gregg [10] using boundary layer treatment removed assumptions no.(3) and considered momentum change in the film. For common fluids with Prandtl number around unity the result obtained shows that the momentum effects are indeed negligible, but for liquid metals with very low Prandtl number the heat transfer coefficients fall below the Nusselt prediction with increasing ($C_p . \Delta T / \lambda$).

More recently Chen [11] , Koh, Sparrow and Hartnett [12] and others have considered the influence of the drag exerted by the vapour on the liquid film. There again the assumption made by Nusselt, no.(8), appears justified at Prandtl numbers around unity. For the condensation of liquid metals, however, the inclusion of the interfacial shear effect does cause a further reduction of heat transfer substantially.

A number of workers [13] have considered the effect on inclusion of variations in physical properties across the condensate film.

An analytical study [14] of filmwise condensation of saturated vapour in forced flow in a vertical tube has been conducted recently for fully developed velocity profile at the inlet and at constant tube wall temperature. For a wide range of conditions of practical interest it is found that the condensation process is governed by five parameters. These are the ratio of vapour Froude to Reynolds number, Buoyancy number, vapour to liquid viscosity ratio, liquid Prondtl number and Subcooling number, $\left[C_p . (T_{sat} - T_w)/h_{fg} \right]$; where, h_{fg} = enthalpy of evaporation and C_p = specific heat of liquid. Comparison of the results with Nusselt's analytic solution of constant interphase shear is also made and it is found that at high pressures, high Prandtl numbers and high ratios of Froude to Reynolds number, Nusselt's solution underpredicts the condensation length and film thickness and overpredicts the interphase mass and heat transfer.

2.1.3 Influence of Turbulence.

Even at relatively low film Reynolds numbers, the assumption that the condensate layer is in viscous flow is open to some question. Experiments aimed at measuring the average thickness of liquid films flowing down vertical surfaces do confirm the Nusselt equation, but examination of the surface structure of the film indicates considerable waviness. This waviness may account for the observed differences [15] between theoretical and experimental values. For long vertical surfaces it is possible to obtain condensation rates such that the film Reynolds number exceeds the critical value at which turbulence begins.

Experiments on condensation in vertical tube have been initiated primarily to clarify the effect of inlet vapour velocity and condenser tube length on the rate of heat transfer [16-19]. These studies indicate that long tubes and high inlet vapour velocities cause a substantial part of tube surface to be covered by a turbulent liquid film, and transition from laminar to turbulent liquid film occurs at a very low value of film Reynold Number [18,19,20] . Analytic modelling of Carpenter and Colburn [18,19] considers turbulent liquid film with non-zero interphase shear stress. In the analysis the resistance to heat flow is considered to be only in the laminar sublayer of the turbulent film and interphase shear is calculated using correlations for adiabatic, co-current, gas-liquid system. Nusselt's assumptions (2) to (7) are again invoked. The equation for the local heat transfer coefficient resulting from this treatment is,

$$\left[\frac{h_1 \quad \mu_1}{k_1 \quad \rho_1}\right] = 0.045 \left[\frac{C_p \cdot \mu}{k}\right]_1^{\frac{1}{2}} \cdot \tau_w^{\frac{1}{2}} \quad \ldots \quad \ldots (2.2)$$

where, τ_w is the shear stress of the outer edge of the laminar sublayer. (i.e. the wall shear stress)

Refinement on the liquid film structure was initiated by Dukler [21] who used Deisster's and Von Kerman's expressions for eddy viscosity. Kunz and Yerazunis [22] went a step further by including interfacial resistance effects and variation of the ratio of the eddy diffusivity of heat to the eddy diffusivity of momentum with Prandtl and Reynolds number. The usefulness of these theoretical predictions confirmed by the good agreement between values of local heat transfer coefficient along the length of the tube calculated from Dukler's analysis and measured by Carpenter [19] .

With the help of further experimental data, Soliman, Schuster and Berenson [23] have improved the Carpenter and Calburn treatment and modified equation (2.2) as,

$$\left[\frac{h_1\,\mu_1}{k_1\,\rho_1}\right] = 0.036\left[\frac{C_p\,\mu}{k}\right]^{0.65} \cdot \quad \tau_w^{\frac{1}{2}} \qquad \ldots \qquad \ldots(2.3)$$

An analysis based on the conditions of constant heat flux and uniform inlet vapour velocity is presented by Shekriladze and Mestvirishvili [24] . In their analysis the shear stress at the liquid-vapour interface is determined from single phase boundary layer equations. In comparison with their experimental data with steam the vapour velocity was set equal to the tube inlet velocity. Good agreement was obtained, probably due to low value of condensation rates.

A paper by Isachenko et .al. [25] studied the effect of a laminar and turbulent vapour core on laminar liquid film. Their experiments with steam at atmospheric pressure, tube inner-diameter of 10 m.m., condensing length of 390 m.m. and tube inlet steam velocities to 50 m/s show that,

1) Liquid film was always laminar.
2) In laminar vapour flow for $Re_{steam} < 3000$, the steam velocity has no effect on heat transfer i.e. Nusselt's simple theory can be used.
3) For $Re_{steam} > 3000$, interphase friction increases heat transfer.

They have given correlations for local and average values of heat transfer coefficients in terms of vapour to liquid density ratio, vapour to liquid viscosity ratio, tube inlet steam Reynold's number and condensate film Reynolds number.

2.1.4 Heat Transfer In Film Condensation Of Quiescent Vapour On Vertical Surface.

It is known that Nusselt formula [5] has a very limited region of applicability, since the condensate film fall in a purely laminar flow is realised at a very small Reynolds number.

At Re $\sim$ 5, formation of waves is observed in a falling film that enhances the heat transfer rate. For the film Reynolds number, which characterises the onset of rippling Kapitsa [26] suggested a relation assuming a capillary nature of waves. Recent studies [27] have discovered gravitational waves on the surface of the falling film which are accompanied by capillary waves. To calculate heat transfer over the range of the film Reynolds number $5 < Re < 100$, the authors [1,28] have recommended to add an emperical correlation term, as given below, in the Nusselt formula to allow for the effect of the waves on enhancement of heat transfer.

$$h_v/h_o = Re^{0.04} \qquad \ldots \quad \ldots \quad \ldots \quad (2.4)$$

where, h_v and h_o are the coefficients of heat transfer calculated using the Nusselt formula and formula [1] respectively.

With further growth of the film Reynolds number the wave mode of the condensate flow is replaced by the turbulent mode. The process of heat transfer under turbulent condition is reported in [2, 28-32]. The relationships suggested in these publications are having the following functional characteristics,

$$\frac{h}{k} \left(\frac{\nu^2}{g} \right)^{1/3} = f \, (R_e, P_r) \qquad \ldots \qquad \ldots \qquad \ldots (2.5)$$

where, h = Coefficient of heat transfer (determined experimentally).

k = Thermal conductivity of the liquid.

ν = Kinematic Viscosity.

g = Gravitational acceleration

Condensation of quiescent vapours of different liquids on vertical tubes has been studied experimentally by many authors [33-39] . Figure 2.1, shows the data for the water vapour. Attention is directed to the large spread of experimental points, especially in the range $10^2 < Re < 10^3$, which corresponds to the transition regime. A similar figure could be presented for vapours of other substances. The results of most of the studies were obtained in a narrow range of film Reynolds number. Figure 2.1, also presents a comparison of the experimental data with the relationships given in [2, 28-30] .

It can further be seen from figure 2.1, that in the region of the laminar and laminar-wave regimes the experimental data are fairly well described by the Nusselt formula with the correction factor given in equation (2.4), for the wave flow of the film. Formulae suggested for the turbulent regime [2,28-30] differ significantly in the region which corresponds to transition

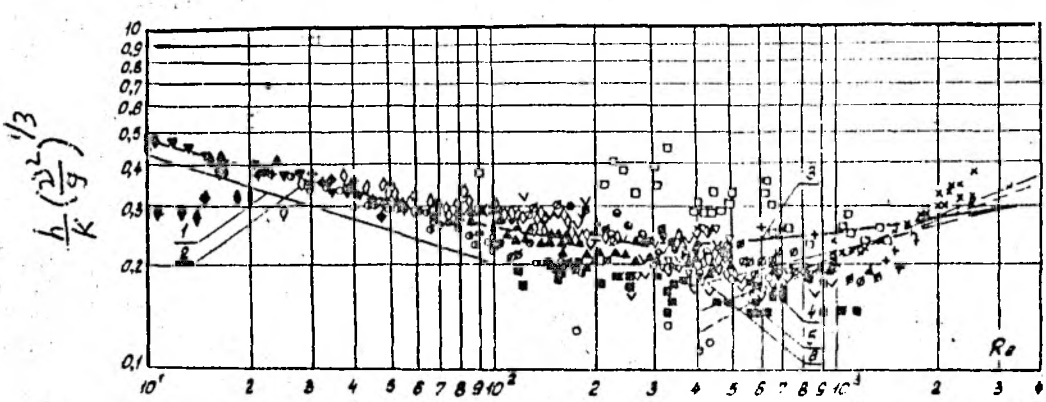

Fig.2.1. Water vapour condensation on vertical tubes
ϕ - [33]; ▲ - [34] ; x - [35] ; ◐ - [36] ; ◇ - [37]; ▼ - [38] ; ⊖ - [39] . The data of following authors are taken from [29, 37] ;
● - Callendar and Nicolson, 0-Jordan;
V-Hebbard and Badger; ▽ - Lozhkin and Kanaev;
▣ - Baker and Stroebe; ϕ - Shea and Krase;
◆ - English and Donkin; □ - Fragen.

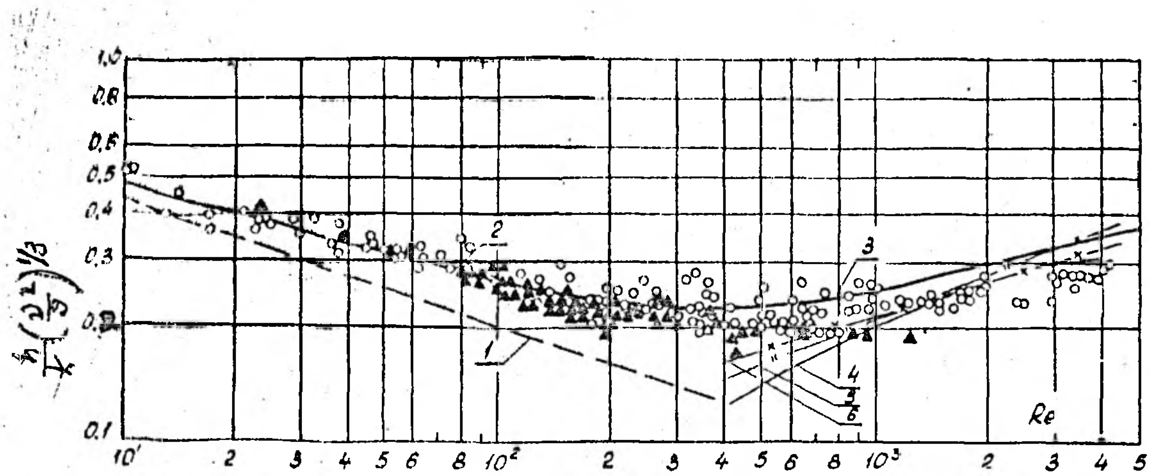

Fig.2.2 Water vapour and freon - 21, Condensation according to data [40] . ▲ - Water vapour [34] , 0 - Freon - 21 [40] .

from the laminar wave to the turbulent flow of the film. The validity of these formulae cannot be fully assessed on the basis of the experimental data presented due to the large spready of the experimental points.

Figure 2.2, shows the results of the investigations with water vapour [34] and freon-21 [40] . The Reynold number in these experiments, with water vapour, varied from 25 to 1200, while with freon-21, it was in the range of $10 < Re < 4300$, thus converging the laminar-wave, transition and turbulent film flow regimes. Comparison with the theoretical relations shows that upto $Re \sim 100$, the experimental results agree very closely with those computed from Nusselt formula with the correction for the wave motion [equation (2.4)]. As seen from figure 2.2, there is a vast range of the Reynolds numbers, $100 \leqslant Re \leqslant 1000$, in which heat transfer rate is practically constant, this appears to be clearly discovered for the first time in [41] .

2.2 Dropwise Condensation.

In view of the large heat fluxes observed in dropwise condensation this subject has received a considerable amount of attention. The mechanism of dropwise consideration is still a mistery and quite contradictory statements are common in the literature. Due to complexity, an exact analysis is virtually impracticable. However, several models have been proposed to approximate the condensation process but over simplifications invalidate some of these. Jacob [42] first postulated that condensation initially occurs in a filmwise manner on a thin unstable liquid film covering all or part of the surface. On reaching a critical thickness the film ruptures and then transforms into droplets by surface tension process. This process then repeats itself. This mechanism has been reiterated in a number of modified forms by Kast [43] and Silver [44] amongst others. The model has been supported by the findings of a number of studies including Baer and McKelvey [45] , Welch and West Water [46] and Sugawara and Katsuta [47] , which show condensation occuring entirely between drops in a very thin film. Welch and West Water examined the process by taking high speed photograph through a microscope. They concluded that droplets large enough to be visible (0.01 mm) grew mainly by coalescense leaving a 'lustrous bare area'. This lusture quickly faded and they explained this in terms of the build up of the thin film which fractured at a thickness of 0.5 to 1 μm. Eucken [48] proposed a concept in which it is assumed that droplet formation is fundamentally a heterogeneous nucleation process. Works of Umar and Griffith [49] and of Erb and Thelen [50] support the nucleation mechanism. Using an optical method with polarised light to detect changes in the thickness of liquid films of molecular dimensions, Umar and Griffith were able to establish that, for low temperature differences the area between drops has no liquid film greater than a monolayer in thickness, and that no net condensation takes place in that area.

McCormick and Baer [51] have suggested that innumerable submicroscopic droplets are randomly nucleated at active sites on the condenser surface. These active sites are wetted pits and grooves in the surface which are continually being exposed by numerous drop coalescences and by large drops falling from the surface.

Gose, Mucciardi and Baer [52] have developed a model for dropwise condensation which accounts for nucleation and growth, coalescence with neighbours, removal and nucleation on sites exposed by the removal and coalescence mechanisms. The model was simulated on a computer. One feature not simulated by this model was the sweeping action of a large drop running down a surface absorbing all other droplets in its path.

Fatica and Katz [53] , Sugawara and Michiyoshi [54] and Nijaguna [55] have considered steady state heat conduction in a single droplet with a discontinuity in the temperature along the edge. The discrepancy due to this discontinuity was recognised by Ahrendts [56] , Hurst and Olson [57] and it was later shown [58] that such a model is inadmissible bcause it predicts an infinite amount of heat flow across the droplet. Other models [49, 56, 58] are incomplete in the sense that the condenser material properties could not be considered and their validity is, therefore, restricted to cases in which the thermal resistance between the droplet and the vapour is dominant. However, there has been some interest in understanding the effect of the condenser material properties and in one of the first analysis Mikic [59] suggested that the effect was due to large droplets behaving as inactive areas constricting the heat flow. Recently this idea was further pursued by Hannemann and Mikic [60] with some modifications of the original model. Earlier, Hurst and Olson [57] considered the condenser material properties by numerically solving the heat equation for a hemispherical droplet on a flat disc-shaped condenser. Sadhal and Plesset [61] studied both evaporation and condensation of droplets and the effect of condenser (evaporator) material, by solving the steady heat conduction equation for a geometry consisting of a droplet in the form of a spherical segment on a semi-infinite solid.

A very considerable amount of work remains to be carried out to establish satisfactory explanations of all the published experimental results on dropwise condensation. While the droplet nucleation mechanism is certainly the more likely at low condensation rates (i.e. temperature difference upto 5°C) it is possible that at higher condensation rates there may be a film disruption mechanism as an intermediate stage before establishing fully developed filmwise condensation. At low temperature differences the dropwise heat transfer coefficient increases slightly with increasing temperature difference [62] . At higher temperature difference heat transfer coefficient decreases with increase in temperature difference [46] . This may be due to a change in mechanism or equally well to the presence of non-condensable gases in the vapour phase.

The basic assumption in the theory of dropwise condensation [62] is that the mean heat flux for the condensing surface may be obtained from a calculation of the steady heat transfer rate for a drop of given size and a steady distribution of drop sizes. In view of the highly non-steady nature of the actual process, where, about a million coalescences can occurin one second on a square centimetre of the condensing surface, this procedure may seem somewhat dubious. Rose [63] examined more carefully the validity of the basic assumption by adopting more rigorous theoretical solutions for heat transfer through a single drop.

The influence of pressure on dropwise condensation of steam at a fixed heat flux was studied by O'Bara et. al. [64] . They reported that for pressure above 3 atmosphere, dropwise condensation gradually replaced by combination of drop and film condensation and finally by filmwise condensation. This is probably brought about by the reduction in the surface tension of water at higher temperatures. They also studied the influence of vapour velocity. For a given temperature difference, the heat transfer coefficient increased upto a maximum value with increased vapour velocity upto 2 m/s and then started to decrease. This may be due to the increased coalescence between droplets blanketing the surface with a film of condensate.

2.3 Methods Of Improving The Heat Transfer Coefficient In Condensation.

An excellent review has been made by Williams et.al. [65]

on the methods of augmenting condensation heat transfer. The methods used fall into following categories,

a) change of geometry of the surface to increase the available area or to promote more rapid removal of condensate.

b) treatment of the surface to promote dropwise rather than filmwise condensation, and

c) use of force fields.

2.3.1 The Influence Of Surface Geometry On Condensation.

The influence of artificially roughened surface on filmwise condensation has been examined by Spencer and Ibele [66] and by Medwell and Nicol [67] . At low film Reynolds number coefficient lies below that in the smooth tube because the roughened surface appears to retain condensate due to surface tension forces. At higher Reynolds number (> 140) the heat transfer coefficients are found larger than for smooth surfaces.

Beatty et.al. [68,69] have examined the performance of integral finned tubes in horizontal and vertical orientations. Although the overall heat transfer coefficient for the finned surface was about 15 percent lower than that for the smooth surface, the increase in the surface area per unit length of the tube was such that a net increase in the heat transfer rate was observed. It should be noted that condensate may be held within the gap between narrow spaced fins.

There are many studies on condensation on vertical surfaces with an array of fins. Among them references [70, 71, 72, 73] report that a heat transfer surface with small sharp edge fins has a very high condensation heat transfer coefficient in comparison with a smooth surface. The enhancement is found to be brought by the following mechanisms: On sharp fin tips with a very small radius of curvature a strong surface tension aids the removal of condensate from the tips thereby producing a very thin liquid film. In addition to this the liquid layers on the side surfaces of the fins are also expected to become locally very thin because of the strong action of surface tension to bring the liquid into the grooves between the fins. Condensed liquid near the tip of the fins is driven almost horizontally towards the grooves due to the surface tension, while the liquid flows vertically down along the groove under gravity [72] .

Edwards et.al. [74] proposed a model on condensation for heat transfer surface with triangular fins. This was done on the assumption of liquid film being attached to the tip of fins with a contact angle. The effect of locally thin condensate film on the side of fins was not considered in their calculations. Fuji, et.al. [75] analysed condensation on a vertical surface with sinuous fins, but also did not take into consideration the effect of locally thin film on fin surfaces. Panchal et.al. [76] also analysed a sinuous film, where the surface temperature of the fin was assumed to be constant. Webb [77] studied the optimisation of a fluted surface, where the temperature of the fluted surface was assumed to be constant. The conduction effect of fin materials could not be investigated even though it is very important when the fin has an enhanced performance and high heat flux.

Variation of the condensate film thickness and variation of local heat transfer coefficient in the vertical direction have been analysed by Mori, et.al. [78]. They have also carried out theoretical analysis and experiments to investigate the optimum performance and geometry of vertical tubular condensers with small longitudinally parallel fins. The guiding principle is to keep the film as thin as possible over the possibly widest surface due to the effect of surface tension. The optimised condensing surface gives heat transfer coefficient close to that of dropwise condensation. The optimising study reveals that a vertical tube provided with fins should be such that the side surface curvature gradually varies from fin tip to the root. It should have a sharp leading edge, a flat groove and circular discs to remove condensate.

Recently, Walezyk [79] reported increase in the heat transfer coefficient upto 20 times, on the gas side, by additional injection of water into air stream. By injection of water into the air stream two phase flow occurs and heat transfer is coupled with mass transfer. Condensation and evaporation take place on finned surface and in the gas phase core. This complex simultaneous heat and mass transfer has been explained unsatisfactorily till now. The theoretical considerations reported in the literature [80,81,82] refer to the conditions in which the effect of condensation and evaporation on apparent heat transfer has been neglected.

Gregoric [83] first introduced the concept of fluted tube and obtained condensation coefficient many times more than those for plain tubes. Thereafter many experimental studies [84,85] on fluted surfaces were made to confirm his findings. The enhancement comes about from the fact that the condensate is drawn into and runs down within the grooves leaving a very thin film on the ridge of each flute of the tube. The movement of condensate into the groove is brought about by surface tension forces, resulting from changes in the radii of curvature of the condensate film surface. The local heat transfer coefficient is thus high at the ridge and low at the groove with average coefficient being much greater than that for a plain tube. When the condensate flow becomes too great to be carried entirely within the grooves, flooding occurs and a sharp drop in the heat transfer coefficient results. The increase in heat transfer and the onset of flooding are clearly functions of the design of a particular tube. Typical values for heat transfer coefficients during the condensation of steam are 40,000 Kcal/hr.m^2 °C, for the non-flooded condition [84]. When the grooves are not filled, the condensing side heat transfer coefficient is essentially independent of tube length.

A comprehensive series of experiments on the performance of fluted tubes investigating such features as vapour velocity, tube orientation and the presence of non-condensables has been reported by Carnovos [86]. A similar enhancement as caused by fluting can be achieved by placing wires along a vertical condenser tube. Again, surface tension force causes the condensate to flow towards the wires and to drain as rivulets alongside the wire. Thomas [87] made theoretical and experimental studies to optimise the number of wires around the tube circumference. He found that the highest heat transfer coefficient was achieved when the number of wires (n) was given by,

$$n = \frac{0.18 \pi D}{d} \qquad \ldots \qquad \ldots (2.6)$$

where, D = tube diameter, and

d = wire diameter

Some further improvements may be achieved by spirally winding the wires around the tube.

2.3.2 Promotion Of Dropwise Condensation.

For the promotion of dropwise condensation it is necessary to treat the surface in some way so as to make it non-wetting. Several methods have been tried, e.g.

a) Chemically coated surface,

b) Polymer coated surface,

c) Electroplated surfaces, etc.

Much of the early work on dropwise condensation was concerned with the identification of various chemicals to act as efficient promoters. An excellent review of the work on chemical promoters has been given by Osment and Tanner [88]. Chemical promoters tend to be removed from the surface within a short period of time and in general, the use of chemical coated surface has not been particularly successful.

In an attempt to find more permanent hydrophobic coating attention has been given to the use of polymer coated surfaces. In particular, considerable attention has been directed at the use of Teflon coatings. Much importance has been paid to the development of the process for a very thin (0.25 to 1 μm) uniform coating. For many years it was thought that only steam could be made to condense in a dropwise manner. However, many liquid including a number of organics condense in a dropwise manner on Teflon coatings due to its very low surface energy.

Electroplated coatings promote dropwise condensation. These coatings are ideal in that they are highly hydrophobic, have high thermal conductivity, and can be diffusion bonded to the base metal. Erb and Thelen [89] have reported excellent behaviour of a 1.25 μm gold deposit on to a Cupro-Nickel alloy tube which had been precoated by a 7.5 μm film of nickel. However, the coatings are expensive and this limits the use of such surfaces.

2.3.3 Use Of Force Fields To Enhance Condensation.

In filmwise condensation the condensate film constitutes the major resistance to heat transfer and a variety of external forces e.g. centifugal, vibrational, electrostatic/electrohydrodynamic etc. have been used to reduce its thickness and to promote drainage.

Numerous investigations have been carried out when condensation is made to occur on a horizontal rotating disc. For a plain disc Nandapurkar and Beatty [90] derived the following equation assuming that there is no interfacial shear.

$$h = 0.904 \left[\frac{k^3 \rho^2 \omega^2 \lambda}{(T_i - T_w)} \right]^{0.25} \quad \dots\dots \quad (2.7)$$

where, ω = angular rotation (radians/s); and

T_i = interfacial temperature.

Further information is available on grooved discs [91] and on rotating horizontal tube [92] . Sparrow and Hartnett [93] obtained a numerical solution for condensation on the outside of rotating cones. Their results for the average Nusselt number can be approximated for ordinary fluids by,

$$Nu = 0.904 \left[\frac{\omega^2 \sin^2\phi \, L^4}{\nu^2} \cdot \frac{Pr}{C_p (T_{sat} - T_w)/\lambda} \right]^{\frac{1}{4}} \quad \dots \quad (2.8)$$

where, L = length of the condenser surface, and

W = angular velocity of the condenser surface.

Pr = Prandtl Number.

Dhir and Leinhard [94] studied laminar film condensation on axisymmetric bodies in a nonuniform gravity field. In case of rotating truncated cones, they showed that,

$$Nu = 0.904 \left[\frac{w^2 L^2 R_o^2}{2} \cdot \frac{Pr}{C_p \, (T_{sat} - T_w)/\lambda} \right]^{\frac{1}{4}} \cdot G(\beta) \quad \ldots(2.9)$$

$$\text{where, } G(\beta) = \frac{\left[(1+\beta)^{8/3} - 1\right]^{4/3}}{\sqrt{\beta} \; (2+\beta)}; \quad \text{and} \quad \beta = \frac{L \sin\phi}{R_o}$$

where, R_o = minimum radius of truncated cones.

Equation (2.9) predicts zero heat transfer as the cone angle approaches zero (i.e., condensation on the inside of a rotating cylinder). Leppert and Nimmo [95,96] after studying film condensation on finite horizontal surfaces, have shown that even when the body force is normal to the condensing surface, a finite amount of heat transfer can occur, if, overall drainage is permitted on the edges. The condensate film thickness and the resulting heat transfer is then governed by hydrostatic pressure changes within the film thickness. Their results are applied to condensation on the inside of a rotating cylinder and are approximated by,

$$Nu = 0.82 \left[\frac{\omega^2 L^2 R_o^2}{\nu^2} \cdot \frac{Pr}{C_p (T_{sat} - T_w)/\lambda} \right]^{1/5} \qquad \ldots \quad \ldots \quad (2.10)$$

Marto [97] established a solution for film condensation on the inside of a slender, rotated, truncated cones in the region where the half cone angle, ϕ, is close to zero and where theequation (2.9) is no longer valid.

The influence of vibration on condensation has also been studied. Mathewson and Smith [98] condensed organic vapours within a vertical tube under conditions where acoustic vibrations were induced in the vapour phase. The vibration caused an enhancement of about 50 per cent over that of tube with no vibration condition. Experiments in which the tube itself was oscillated either longitudinally [99] or transversely [100] have been shown to enhance the film coefficient.

A number of reports are available [101-106] which show that the electric field of different geometry and frequency may provide an effective means for enhancing the vapour-film condensation heat transfer. The investigations were carried out, however, in a narrow range of the process parameters and electrophysical properties of heat agents. Didkovsky and Bologa [107] carried out experiments on heat transfer and hydrodynamics in film condensation of pure vapour on vertical short surfaces in an electric field of different strength, frequency and uniformity

2.4 Heat Transfer To Falling Liquid Films And Film Breakdown.

Heat transfer to liquid films flowing on a heating surface and breakdown of the films associated with increasing heat flux are of great importance. A number of investigations [108] have, therefore, been conducted on these subjects and it has also been shown that film breakdown conditions are influenced by several different mechanisms. However, knowledge of the precise nature of film breakdown is still incomplete.

In the previous investigations on film breakdown conditions, termed as the "minimum wetting rate" of falling liquid films and the 'dryout' in annular two phase flow, theoretical considerations have usually been concerned with whether a dry patch formed on the surface will remain or be re-wetted [109-116]. Therefore, a force balance at the upstream stagnation point of a dry patch has been discussed on various models of flow. However, as these models have included a contact angle of the liquid at the point of break-down, lack of data on the contact angle [116-118] whose measurement is very difficult, has precluded any quantitative understanding of the film breakdown.

An alternative approach to predict the film breakdown conditions has been conducted from a view point of total mechanical (Kinetic and surface) energy flow by Hartley and Murgatroyd [112] and Bankoff [118]. On the other hand Hallett [119] correlated the minimum wetting rate during heat transfer with the difference in surface tension set up due to uneven heating caused by wave motion of the film, and Bankoff [120] analysed the effect of the vapour leaving normally to the film surface on the wave amplitude of a thin liquid film. Toshihiko and Tatsuhiro [121] carried out experiments on heat transfer coefficient and film breakdown for both conditions of local and permanent dry patch formations.

Zollars and Krantz [122] solved the linear stability problem for film flow down a straight circular cone via a perturbation expansion technique. This represents the first solution for the stability of a non-parallel flow of this type. The occurance of stable, relatively unstable and absolutely unstable waves have also been predicted [122] .

2.5 Heat Transfer In Wavy/Diverging-Converging System.

From the literature survey done on diverging-converging heat exchangers, it is observed that most of the study concerns either the hydrodynamic or only the liquid-liquid heat transfer and practically no significant work has been reported on condensation heat transfer. Most of the studies primarily concerns the flow of fluid through converging/diverging nozzles/ diffusers [123-127] or converging-diverging ducts [128,129] and a very few have worked on converging-diverging channels.

2.5.1 Hydrodynamics Of Wavy/Converging-Diverging Flow.

Batra [130] and Batra, et.al. [131] carried out experimental investigations on laminar flow through periodically convergent-divergent (wavy) tubes and channels. They reported that the value of friction factor increases with decreasing wavelength to diameter ratios (< 0.5) by as much as 120% of the uniform tube value (wave amplitude-to-diameter ratios were in the range 0.13 - 0.25). The value of the critical Reynolds number of transition to turbulence decreases with decreasing wavelength-to-diameter ratios.

Payatakes, et.al. [132] developed a model for porous media comprised of monosized or nearly monosized grains. In applying this model to a packed bed, the bed is assumed to consist of a series of statistically identical unit bed elements each of which in turn consists of a number of unit cells connected in parallel. Each unit cell resembles a piece of constricted (divergent-convergent) tube with dimensions which are random variables. The problem of flow through each unit cell is reduced, subject to reasonable assumptions, to the determination of the flow in an infinitely long periodically constricted tube. The model together with the solution [133] of the flow through it, can be used for modelling of processes which take place in the void space of a bed. The theoretical friction factor values were compared with the experimental ones, for two different beds, and found to be in good agreement even in the region of high Reynold numbers.

Earlier, Fulford [134] found in his experiments on wavy film flow in channels that friction factors were 90 per cent higher than the theoretical values for smooth films when the channel inclination was $7\frac{1}{2}$ degree to the horizontal (i.e., when the wave amplitude-to-film thickness ratio was small) and 125 per cent higher than the theoretical values when channel was vertical (i.e., when the wave amplitude-to-film thickness ratio was large).

Lekoudis et.al. [135] have made a linear analysis of compressible boundary layer flows over a wavy wall. Shankar and Sinha [136] have studied the effects of wall waviness on the well known Reyleigh problem. Lessen and Gangwani [137] have analysed the effect of small amplitude wall waviness upon the stability of the laminar boundary layer. In all these studies [135-137] the authors have taken the wavy wall to be oriented in a horizontal direction and studied the effect of the waviness on the flow field.

2.5.2 Heat Transfer in Wavy/Diverging-Converging Tubes.

Gosse and Sehiestel [138] carried out numerical prediction of the convective heat transfer in wavy tubes on the basis of mathematical models of turbulence [139, 140] and the three dimensional method of Patankar and Spalding [141] and the method developed by Amsden and Harlow [142] . The numerical prediction [138] is free of all restrictive hypothesis regarding the turbulent viscosity and can be applied to the study of three dimensional flows presenting complex geometries. The various applications dealt with uptil now range from classical turbulent flows [140,143,144] , to flows in annuli [145,146] and in zig-zags [147] and have shown that the proposed scheme, using same values of numerical constants in all cases makes it possible to describe flows having strong dissymetries. The calculated pressure drop agrees well with experimental values. They showed that the waviness of the tubes is of least interest when the values of the Prandtl number are very large. On the other hand, for moderate or very small values of Prandtl number, it seems possible to define the forms of undulations which will assure on optimal thermal convection of a given zone of Reynolds numbers.

Vajravelu and Sastri [148,149] have made a systematic analysis of free convective heat transfer in a viscous fluid confined between a long vertical wavy wall and a parallel flat wall and have established that the flow and heat transfer characteristics are significantly affected by wall waviness. The skin friction coefficient and the heat transfer coefficients have been obtained along with the pressure drop. These flow and heat transfer characteristics have been found to depend on the free convection parameter; the frequency parameter, λ ; the wall temperature ratio, m; the Prandtl number, Pr; the amplitude of the wavy wall, ϵ ; and the heat source/sink parameter, α .

Narayan [150] made a generalised mathematical analysis of the momentum and heat transfer characteristics in irregular geometries with special reference to diverging-converging tubes and also made experimental verification. A deviation parameter

r_w (Z), has been defined which accounts for the distance from the centre of the tube to the tube wall which for a tube with irregular geometry is a function of axial distance Z and depends on the type of geometry under consideration. Satisfactory agreement has been obtained between the experimental values and the theoretically predicted ones. Excellent enhancement in the transfer coefficient has been observed for the constricted tube as compared to the plain uniform tube, having the same surface per unit length, with reasonably low pressure drop penalty.

Ramachandran [151] and Bandyopadhyay [152] carried out experiments on turbulent and laminar heat transfer studies respectively in the annulus of diverging-converging tubes. They also reported an enhancement of heat transfer in this type of system over that of straight tube.

Recently Sahoo [153] , Sen [154] and Sen [155] carried out hydrodynamic and heat transfer studies with diverging-converging tubes. Their studies revealed that the velocity profile is parabolic at the throat section of the tube, gradually changed into turbulent at the intermediate region and again to laminar at the widest section of the tube. Onset of turbulence was observed at Re = 1800.

Condensation heat transfer on the inside of a rotating truncated cone [97] and axisymmetric bodies in non-uniform gravity [94] has already been discussed. Varatharajan [156] , however, reported some work on condensation in a periodically divergent-convergent tube. But his study was confined to only condensation of steam and evaluation of overall heat transfer coefficient. Excellent augmentation of overall heat transfer coefficient was supported by his experiments.

The preceding review of the state of art in the condensation of vapour shows that no significant work has been reported on condensation in diverging-converging tubes, inspite of its proven efficiency. Review on certain aspects of condensation in straight tube have been carried out to present in a logical and coherent manner the current state of art in estimating the important engineering parameters and their effects on condensation.

From the preceding survey it has been further established that the phenomena of flow separation and reattachment along with relaminarisation is expected in diverging-converging system. The zig-zag shape is favourable to heat transfer as a result of the combination of the following two effects. There is firstly increase in the heat transfer area over a plain tube for the same length. Secondly, there is increase in convection on both sides of the tube. On the inside the alternating changes in curvature favour the creation of secondary flows in the fluid, which are continuously restructured from one surface to the next. This is obviously accompanied by an increase in pressure drop and hence it can be expected that there will be improved internal convection. On the outside of the tubes the flow is much more complicated and must correspond at intervals to zones of separation with eddies.

CHAPTER - 3

MATHEMATICAL ANALYSIS

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MATHEMATICAL ANALYSIS

Due to the complex nature of irregular geometries mathematical modelling of the system becomes difficult. As a result, instead of mathematical analysis, more stress has to be given in the past for elaborate experimentation and empiricism. Condensation flow modelling is, as such, difficult due to the complex thermo-hydrodynamic coupling of the two phases. This coupling is significantly more pronounced in the internal flow of condensate than in the external flow.

In the present study mathematical analysis has been attempted for condensation of pure vapours inside diverging, converging and diverging-converging (combined) tube segments. It is to be noted that the Nusselt's classical approach has been adopted as the basis for the present study .

3.1 Assumptions.

The assumptions made in the analysis comprise,

(1) The drainage of the condensate film from the surface is laminar, unidimensional and is at steady state.

(2) The vapour is pure and exerts no drag on the downward motion of the condensate film.

(3) Heat transfer across the condensate layer is by conduction only and temperature distribution is linear

for a very thin film. Viscous dissipation of heat is neglected.

(4) The thickness of the film at any point is a function of the mean velocity of flow and of the amount of condensate passing at that point.

(5) The velocity of the individual layers of the film is a function of the relation between frictional shearing force and the weight of the film.

(6) The quantity of condensate is proportional to the quantity of heat transferred, which in turn is related to the thickness of the film and of the temperature difference between the vapour and the surface.

(7) Condensate fluid properties and condensing surface temperatures are constant.

(8) The surface is assumed to be relatively smooth and clean.

(9) Subcooling of the condensate is neglected.

(10) Liquid film is incompressible.

3.2 Diverging Cone Section.

Condensation of pure vapour inside a diverging tube section is considered and is shown in figure 3.1. The analysis is based on the equation of motion obtained by balancing the gravitational force and the viscous shear force acting on a differential volume element of the condensate.

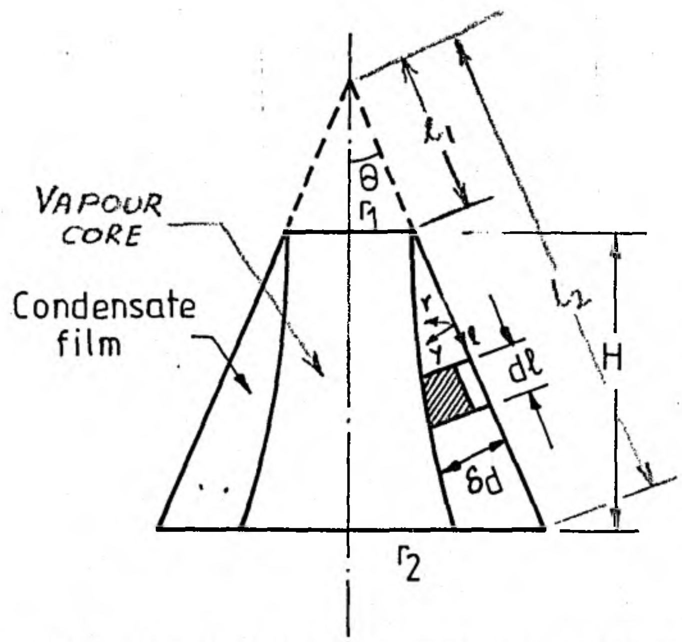


FIG.3.1. DIVERGING CONE SECTION

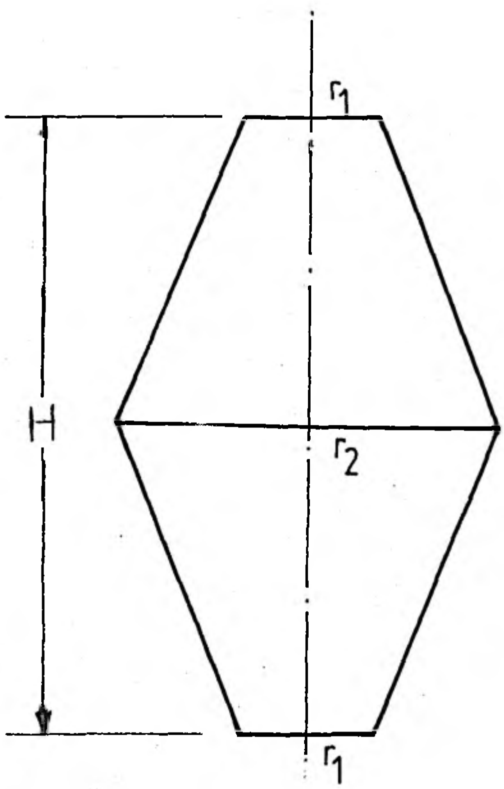


FIG.3.2. DIVERGING-CONVERGING CONE SECTION

A force balance over the shaded portion (figure 3.1) gives,

$$\mu\left(\frac{du}{dy}\right) dl = \rho(\delta_d - y) .\ g \cos\theta \ .\ dl \quad \ldots \quad \ldots (3.2.1)$$

or,

$$\frac{du}{dy} = \frac{\rho g \cos\theta}{\mu} (\delta_d - y) \quad \ldots \quad \ldots \quad \ldots (3.2.2)$$

Assuming no slip i.e., $u = 0$ at $y = 0$, the solution of equation (3.2.2) gives the following expression of velocity distribution across the condensate film at any distance along the wall,

$$u(y) = \frac{\rho g \cos\theta}{\mu} \left(\delta_d . y - \frac{y^2}{2}\right) \quad \ldots \quad \ldots (3.2.3)$$

At a slant distance 'l', the average downward velocity of condensate film is given by

$$\bar{u} = \frac{1}{\delta_d} \int_0^{\delta_d} u(y)\ dy \quad \ldots \quad \ldots (3.2.4)$$

Solution of equation (3.2.4) through equation (3.2.3) gives,

$$\bar{u} = \frac{\rho g \cos\theta}{3\mu} .\ \delta_d^2 \quad \ldots \quad \ldots (3.2.5)$$

Now the various expressions for the differential area dA of a cone for length dl, is obtained as follows,

$$dA = \pi(r+dr)(l+dl) - \pi rl$$

$$= \pi(rdl+ldr), \text{ (Neglecting (drdl) as it is very small)}$$

$$= 2\pi rdr.\ \mathrm{cosec}\,\theta \quad \ldots \quad \ldots \quad (3.2.6)$$

$$= 2\pi r.dl \quad \ldots \quad \ldots \quad (3.2.7)$$

$$= 2\pi l.\sin\theta.\ dl \quad \ldots \quad \ldots \quad (3.2.8)$$

[Because $r = l.\sin\theta$

therefore, $l = r.\mathrm{cosec}\,\theta$

or $dl = dr.\mathrm{cosec}\,\theta$]

At a distance 'l' the downward flow of the condensate across a horizontal plane of area $2\pi r\delta_d$ (i.e. $2\pi l.\sin\theta.\delta_d$)

$$= 2\pi l.\sin\theta.\delta_d.\ \bar{u}\rho \quad \ldots \quad (3.2.9)$$

At (l + dl) there is a gain in the amount of downward flow and is given by d ($2\pi l.\sin\theta.\delta_d.\ \bar{u}\rho$), which is further modified as follows with the help of equation (3.2.5).

$$d(2\pi l.\sin\theta.\delta_d.\bar{u}\rho) = \frac{2\pi\rho^2 g\ \sin\theta\ \cos\theta}{3\mu}\left[d(l\,\delta_d^3)\right] \ldots \quad (3.2.10)$$

If 'w' be the condensate mass flow rate out of vapour and 'normal' to the falling condensate layer per unit surface area, then,

$$d(2\pi l.\sin\theta.\delta_d.\bar{u}\rho) = W.dA = W.2\pi l.\sin\theta.dl \quad \ldots (3.2.11)$$

Again, a balance between the heat released in the condensation process and the energy conducted through the condensate film gives,

$$W = \frac{k\,(T_v - T_w)}{\lambda\,\delta_d} \qquad \ldots \quad \ldots \quad (3.2.12)$$

From equations (3.2.10), (3.2.11) and (3.2.12) we have,

$$\frac{k(T_v - T_w)}{\lambda\,\delta_d}\,(2\pi l\,dl \cdot \sin\theta) = \frac{2\pi\rho^2 g\,\sin\theta\,\cos\theta}{3\mu}\left[d\,(l\,\delta_d^3)\right] \qquad \ldots \quad (3.2.13)$$

$$\text{or} \quad \frac{l\,dl}{\delta_d} = \left[\frac{\rho^2 g\,\lambda\,\cos\theta}{3\mu k(T_v - T_w)}\right] \cdot \left[d(l\,\delta_d^3)\right] \qquad \ldots \quad \ldots \quad (3.2.14)$$

Solution of equation (3.2.14) [See Appendix -IV.1]

$$\delta_d = \left[\frac{24\mu k(T_v - T_w)}{7\rho^2 g\,\lambda\,\sin 2\theta}\right]^{\frac{1}{4}} \cdot \left[r\left\{1 - \left(\frac{r_1}{r}\right)^{7/3}\right\}\right]^{\frac{1}{4}} \qquad \ldots \quad (3.2.15)$$

The local heat transfer coefficient, h_{id}, from the vapour side to the tube surface is defined as,

$$h_{id} = \frac{k}{\delta_d} \qquad \ldots \quad \ldots (3.2.16)$$

From equations (3.2.15) and (3.2.16) we have,

$$h_{id} = \left[\frac{7\rho^2 g\lambda k^3 \sin 2\theta}{24\mu(T_v-T_w)}\right]^{\frac{1}{4}} \cdot \left[\frac{1}{r\left\{1-\left(\frac{r_1}{r}\right)^{7/3}\right\}}\right]^{\frac{1}{4}} \quad \ldots \quad (3.2.17)$$

$$\text{or,}\quad h_{id} = \left[\frac{7\rho^2 g\lambda k^3 \cos\theta}{12\mu(T_v-T_w)}\right]^{\frac{1}{4}} \cdot \left[\frac{1}{l\left\{1-\left(\frac{l_1}{l}\right)^{7/3}\right\}}\right]^{\frac{1}{4}} \quad \ldots \quad (3.2.18)$$

[Since, $r = l \cdot \sin\theta$] .

The total heat transfer through the condensate layer from l_1 to l_2 (or r_1 to r_2) is given by,

$$Q_d = \int_{l_1}^{l_2} h_{id} \cdot (T_v-T_w) \cdot 2\pi\, l\, dl \cdot \sin\theta \quad \ldots \quad \ldots \quad (3.2.19)$$

Solution of equation (3.2.19) [See Appendix-IV.2] gives,

$$Q_d\Big|_{r=r_2} = \frac{8}{7}\pi\,(T_v-T_w)^{\frac{3}{4}} \cdot \operatorname{cosec}\theta \cdot \left[\frac{7\rho^2 g\lambda k^3 \sin 2\theta}{24\mu}\right]^{\frac{1}{4}} \cdot (r_2)^{7/4} \cdot \left[1-\left(\frac{r_1}{r_2}\right)^{7/3}\right]^{\frac{3}{4}} \quad \ldots \quad (3.2.20)$$

The average condensation heat transfer coefficient is then obtained as,

$$\bar{h}_{id} = \frac{Q_d}{\Delta T.\pi(r_2 l_2 - r_1 l_1)}$$

$$= \frac{Q_d.\sin\theta}{\Delta T.\pi(r_2^2 - r_1^2)} \qquad \ldots \quad \ldots (3.2.21)$$

[because, $r = l.\sin\theta$]

Replacement of Q_d by equation (3.2.20) gives,

$$\bar{h}_{id} = 0.84 . \left[\frac{\rho_f^2\, g\lambda\, k_f^3 .\sin 2\theta}{\mu_f\, \Delta T_f\, r_2}\right]^{\frac{1}{4}} .. \left[\frac{1}{1-(\frac{r_1}{r_2})^2}\right] . \left[1 - (\frac{r_1}{r_2})^{7/3}\right]^{\frac{3}{4}} \qquad \ldots \quad \ldots (3.2.22)$$

where, k_f, ρ_f and μ_f are evaluated at the film temperature and the film temperature T_f is given as,

$$T_f = \tfrac{1}{2}(T_v + T_w)$$

and ΔT_f is the temperature difference between film and wall i.e.,

$$\Delta T_f = (T_f - T_w) .$$

3,3 Converging Cone Section.

Condensation of pure vapour inside a converging cone section is considered.

Following the similar approach as discussed in connection with diverging cone section, the average downward velocity of condensate is given by

$$\bar{u} = \frac{1}{\delta_c}\int_0^{\delta_c} u(y)\,dy = \frac{\rho g . \cos\theta}{3\mu} . \delta_c^2 \qquad \ldots (3.3.1)$$

At any section of radius r, the downward flow of condensate is $= 2\pi r.\delta_c . \bar{u}\rho$ ... ...(3.3.2)

For a differential area dA, the amount condensed on the film (of area dA) will be,

$$d(2\pi r.\delta_c . \bar{u}\rho) = \frac{2\pi\rho^2 g . \cos\theta}{3\mu} . \left[d(r\delta_c^3)\right] \qquad \ldots (3.3.3)$$

Putting $r = l . \sin\theta$, we have

$$d(2\pi l . \sin\theta . \delta_c \bar{u}\rho) = \frac{2\pi\rho^2 g . \sin\theta . \cos\theta}{3\mu} \left[d(l\delta_c^3)\right] \qquad \ldots (3.3.4)$$

Again, if 'w' be the condensate mass flow rate out of vapour and 'normal' to the falling condensate layer per unit surface area, then,

$$d(2\pi\, l \cdot \sin\theta . \delta_c\, \bar{u}\, \rho\ ) = W.dA = W.\ 2\pi l . \sin\theta . dl \qquad \ldots(3.3.5)$$

From energy balance, however,

$$W = \frac{k\ (T_v - T_w\ )}{\lambda\, \delta_c} \qquad \ldots(3.3.6)$$

Now from equations (3.3.4), (3.3.5) and (3.3.6), we have,

$$\frac{k(T_v - T_w)}{\lambda\, \delta_c}\ (2\pi\ l . \sin\theta . dl) = \frac{2\pi\, \rho^2 g . \sin\theta \cos\theta}{3\mu}\ \left[d(l\, \delta_c^3\ )\right] . \qquad (3.3.7)$$

It follows,

$$\frac{l dl}{\delta_c} = \left[\frac{\rho^2 g\, \lambda . \cos\theta}{3\mu\, k(T_v - T_w)}\right] . \left[\, d(l\, \delta_c^3\ )\, \right] \qquad \ldots(3.3.8a)$$

or

$$\frac{r dr}{\delta_c} = \left[\frac{\rho^2 g\, \lambda . \sin\ 2\theta}{6\mu\, k(T_v - T_w)}\right] . \left[d(r\, \delta_c^3\ )\right] \qquad \ldots(3.3.8b)$$

$\left[\text{since, } r = l \sin\theta\right]$.

Solution of equation (3.3.8b) [See Appendix-IV.3] gives,

$$\delta_c = \left[\frac{24\, \mu K(T_v - T_w)}{7\, \rho^2 g\, \lambda . \sin\ 2\theta}\right]^{\frac{1}{4}} . \left[r\left\{\left(\frac{r_2}{r}\right)^{7/3} - 1\right\}\right]^{\frac{1}{4}} \qquad \ldots(3.3.9)$$

The local heat transfer coefficient across the condensate layer per unit interfacial area is given by,

$$h_{ic} = \frac{k}{\delta_c} \qquad \ldots(3.3.10)$$

From equations (3.2.9) and (3.2.10), we have,

$$h_{ic} = \left[\frac{\gamma \rho^2 g \lambda k^3 \cdot \sin 2\theta}{24\, \mu (T_v - T_w)} \right]^{\frac{1}{4}} \cdot \left[\frac{1}{r \left\{ \left(\frac{r_2}{r}\right)^{7/3} - 1 \right\}} \right]^{\frac{1}{4}} \qquad \ldots(3.3.11)$$

For, $r = l.\sin\theta$, we can write

$$h_{ic} = \left[\frac{\gamma \rho^2 g \lambda k^3 \cdot \cos\theta}{12\, \mu (T_v - T_w)} \right]^{\frac{1}{4}} \cdot \left[\frac{1}{l \left\{ \left(\frac{l_2}{l}\right)^{7/3} - 1 \right\}} \right]^{\frac{1}{4}} \qquad \ldots(3.3.12)$$

The total heat transfer through the condensate layer from l_2 to l_1 or r_2 to r_1 is,

$$Q_c = - \int_{r_2}^{r_1} h_{ic} \cdot (T_v - T_w) \cdot 2\pi\, r dl \qquad \ldots(3.3.13a)$$

(Negative sign has been incorporated in the integration since for a converging cone local radius decreases from top to bottom of the cone).

$$= - \int_{l_2}^{l_1} h_{ic} \cdot (T_v - T_w) \cdot 2\pi l \, dl \, \sin\theta \qquad \ldots(3.3.13b)$$

Since, $r = l.\sin\theta$.

After substitution for h_{ic} and dA in equation (3.3.13) and by integration, we have, (See Appendix-IV-4)

$$Q_c \Big|_{r=r_1} = \frac{8}{7}\pi (T_v - T_w)^{\frac{3}{4}} \cdot \operatorname{cosec}\theta \cdot \left[\frac{7\rho^2 g \lambda k^3 . \sin 2\theta}{24\mu} \right]^{\frac{1}{4}} \cdot \left[r_1 \right]^{7/4} \cdot \left[\left(\frac{r_2}{r_1} \right)^{7/3} - 1 \right]^{\frac{3}{4}} \qquad \ldots(3.3.14)$$

The average condensation heat transfer coefficient for the convergent cone segment can be obtained from,

$$\bar{h}_{ic} = \frac{Q_c}{\Delta T . \pi (r_2 l_2 - r_1 l_1)} \qquad \ldots(3.3.15)$$

$$= \frac{Q_c . \sin\theta}{\Delta T . \pi (r_2^2 - r_1^2)} \qquad \ldots(3.3.16)$$

Replacement of Q_c by equation (3.3.14) gives,

$$\bar{h}_{ic} = 0.84 . \left[\frac{\rho_f^2 \; g \lambda K_f^3 . \sin 2\theta}{\mu_f \; \Delta T_f \; r_1} \right]^{\frac{1}{4}} . \left[\frac{1}{\left\{ \left(\frac{r_2}{r_1}\right)^2 - 1 \right\}} \right] . \left[\left(\frac{r_2}{r_1} \right)^{7/3} - 1 \right]^{\frac{3}{4}} \qquad \ldots(3.3.17)$$

where, k_f, ρ_f and μ_f are evaluated at the film temperature, T_f and the film temperature is,

$$T_f = \tfrac{1}{2} (T_v + T_w)$$

and $$\Delta T_f = (T_f - T_w)$$

3.4 Diverging-Converging Cone Section.

Here condensation of pure vapour inside a symmetrical diverging-converging cone section (apex angle, θ, being same) is considered.

In the mathematical analysis of diverging-converging cone further assumptions made are given below,

a) The condensation characteristics in the diverging cone section, remain unaffected, in conjunction with converging cone section. That means, initial and final conditions will be the same as that derived in section - 3.2.

b) Initial condition in the converging cone section will be the same as that obtained in the end of diverging cone section irrespective of the change of direction of film flow. This is justified as the film flow is assumed purely laminar.

Now following the similar approach as discussed in section-3.3, for converging cone section and from equation (3.3.8b) introducing the new boundary condition, the relationships obtained for the converging cone section in conjunction with diverging cone section are given below,

Rewriting equation (3.3.8b),

$$\frac{r\,dr}{\delta_c} = \left[\frac{\lambda \rho^2 g.\sin 2\theta}{6\mu k(T_V - T_W)} \right] . \left[d\,(r\delta_c^{\,3}) \right]$$

$$= \left[z \right] . \left[d(r\,\delta_c^{\,3}) \right] \qquad \ldots(3.3.8c)$$

where,

$$\left[z \right] = \left[\frac{\lambda \rho^2 g.\sin 2\theta}{6\mu k(T_V - T_W)} \right]$$

Solution of equation (3.3.8c) [See Appendix-IV.5] gives,

$$\delta_c = \left[\frac{24\mu k(T_V - T_W)}{7\rho^2 g\lambda \sin 2\theta} \cdot \frac{(r_2^{7/3} - r^{7/3})}{r^{4/3}} + \left(\frac{r_2 \delta_2^{\,3}}{r}\right)^{4/3} \right]^{\frac{1}{4}}$$

$$\ldots(3.4.1)$$

Therefore, the local heat transfer coefficient is given by,

$$h_{ic} = \frac{k}{\delta_c} \qquad \ldots(3.4.2)$$

From equations (3.4.1) and (3.4.2), we have,

$$h_{ic} = k\left[\frac{24\mu k(T_v-T_w)}{7\lambda\rho^2 g.\sin 2\theta} \cdot \frac{(r_2^{7/3}-r^{7/3})}{r^{4/3}} + \left(\frac{r_2\,\delta_2^{\,3}}{r}\right)^{4/3}\right]^{-\frac{1}{4}} \qquad \ldots(3.4.3)$$

The heat transfer through the condensate layer for the converging portion, of the diverging-converging system is,

$$Q_c = -\int_{r_2}^{r} h_{ic}.\ (T_v - T_w).\ 2\pi r dr.\ \ \mathrm{cosec}\,\theta \qquad \ldots(3.4.4)$$

(Negative sign has been incorporated in the integration because for a converging section 'r' decrease from top to bottom).

$$= 2\pi k.(T_v-T_w).\mathrm{cosec}\,\theta \ . \int_{r_2}^{r}\left[\frac{24\mu k(T_v-T_w)}{7\rho^2\lambda\ g.\sin 2\theta} \cdot \frac{(r_2^{7/3} - r^{7/3})}{r^{4/3}} + \left(\frac{r_2\,\delta_2^{\,3}}{r}\right)^{4/3}\right]^{-\frac{1}{4}} \cdot\ r.dr \qquad \ldots(3.4.5)$$

$$= -2\pi k.(T_v - T_w).\mathrm{cosec}\,\theta \cdot \int_{r_2}^{r} \left[\frac{4}{7(Z)}(r_2^{7/3} - r^{7/3}) + r_2^{4/3} \cdot \delta_2^{4}\right]^{-\frac{1}{4}} \cdot r^{4/3}.\,dr. \qquad \ldots(3.4.6)$$

Solution of equation (3.4.5) (See Appendix-IV-6) gives,

$$Q_c\Big|_{r=r_1} = 3.5937.\cos\theta \cdot \left[\frac{\rho^2 k^3 \lambda\, g(T_v - T_w)^3 .\ (r_2^{7/3} - r_1^{7/3})^3}{\mu .(\sin 2\theta)^3}\right]^{\frac{1}{4}} \qquad \ldots(3.4.7)$$

Again, heat transfer through the diverging portion is given by [from equation (3.2.20)]

$$Q_d\Big|_{r=r_2} = \frac{8}{7}\pi(T_v - T_w)^{\frac{3}{4}} . \frac{1}{\sin\theta} \cdot \left[\frac{7\rho^2 g\lambda k^3.\cos\theta}{12\mu}\right]^{\frac{1}{4}} . \left[\frac{r_2}{\sin\theta}\right]^{7/4} \cdot \left[1 - \left(\frac{r_1}{r_2}\right)^{7/3}\right]^{\frac{3}{4}} \qquad \ldots(3.2.20)$$

$$=5.279.\cos\theta \left[\frac{\lambda\rho^2 g k^3 (T_v - T_w)^3 .\ (r_2^{7/3} - r_1^{7/3})^3}{\mu(\sin 2\theta)^3}\right]^{\frac{1}{4}} \ldots(3.4.8)$$

Total heat transfer for the combined diverging -converging cone section, therefore, is,

$$Q_{dc} = Q_d + Q_c \qquad \ldots(3.4.9)$$

From equations (3.4.7), (3.4.8) and (3.4.9) we have,

$$Q_{dc} = 8.8727.\cos\theta.\left[\frac{\rho^2 k^3 \lambda g(T_v - T_w)^3.\ (r_2^{7/3} - r_1^{7/3})^3}{\mu.(\sin 2\theta)^3}\right]^{\frac{1}{4}}$$

$$\ldots(3.4.10)$$

Average heat transfer coefficient is given by

$$\bar{h}_{idc} = \frac{Q_{dc}}{\Delta T.\ 2\pi \operatorname{cosec}\theta\,(r_2^2 - r_1^2)} \qquad \ldots(3.4.11)$$

$$= 0.706\left[\frac{\lambda \rho_f^2 g k_f^3.\sin 2\theta}{\mu_f\ .\Delta T_f}\right]^{\frac{1}{4}}.\frac{(r_2^{7/3} - r_1^{7/3})^{\frac{3}{4}}}{r_2^2 - r_1^2} \qquad \ldots(3.4.12)$$

where, k_f, ρ_f and μ_f are evaluated at the film temperature and the film temperature, T_f, is given by,

$$T_f = \tfrac{1}{2}(T_v + T_w)$$

and $\quad \Delta T_f = (T_f - T_w)$

3.5 Equations Relating Heat Transfer Coefficient and Condensate Film Reynolds Number.

3.5.1 Diverging Core Section.

Rewriting equation (3.2.22),

$$\bar{h}_{id} = 0.84.\left[\frac{\rho_f^2 g \lambda k_f^3 . \sin 2\theta}{\mu_f . \Delta T_f . r_2}\right]^{\frac{1}{4}} . \left[\frac{1}{1 - (r_1/r_2)^2}\right] . \left[1 - (r_1/r_2)^{7/3}\right]^{\frac{3}{4}} \quad \text{...(3.2.22)}$$

Let, $r_1/r_2 = a$, then,

$$\bar{h}_{id} = 0.84 . \left[\frac{\rho_f^2 \; g \lambda k_f^3 \sin 2\theta}{\mu_f . \Delta T_f . r_2}\right]^{\frac{1}{4}} . \left[\frac{1}{1-a^2}\right] . \left[1-a^{7/3}\right]^{\frac{3}{4}} \quad \text{...(3.5.1)}$$

Again, $\bar{h}_{id} = \dfrac{Q_d}{\Delta T_f . A_d}$

$$= \frac{Q_d}{\Delta T_f . (\pi DeH)} = \frac{\lambda w'}{\Delta T_f . (\pi DeH)} = \frac{\lambda G'}{H . \Delta T_f} \quad \text{...(3.5.2)}$$

Now,

$$\frac{\lambda G'}{H . \Delta T_f} = \frac{\lambda}{H . \Delta T_f} . \frac{4G'}{4} . \frac{\mu_f}{\mu_f} = \frac{\lambda \mu_f}{4H . \Delta T_f} . (Re_f)$$

where, $Re_f = \dfrac{4G'}{\mu_f}$

So, $$\bar{h}_{id} = \frac{\lambda \mu_f}{4H . \Delta T_f} (Re_f) \quad \text{...(3.5.3)}$$

$$\text{or,} \quad \frac{1}{\Delta T_f} = \frac{\bar{h}_{id}}{Re_f} \cdot \left(\frac{4H}{\lambda \mu_f} \right) \qquad \ldots (3.5.4)$$

From equations (3.5.1) and (3.5.4)

$$\bar{h}_{id} = 0.84 \cdot \left[\frac{\rho_f^2 \, g \lambda k_f^3 . \sin 2\theta}{\mu_f \cdot \; r_2} \right]^{\frac{1}{4}} \cdot \left[\frac{(1-a^{7/3})^{\frac{3}{4}}}{1 - a^2} \right] \cdot$$

$$\left[\frac{\bar{h}_{id}}{Re_f} \cdot \left(\frac{4H}{\mu_f \lambda} \right) \right]^{\frac{1}{4}} \qquad \ldots (3.5.5)$$

After rearranging, we have, [See Appendix-IV - 7]

$$\bar{h}_{id} \cdot \left[\frac{\mu_f^2}{\rho_f^2 \; g k_f^3} \right]^{1/3} = 1.585 . \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} . \cos\theta \right]^{2/3} . Re_f^{-1/3}$$

$$\ldots (3.5.6)$$

$$= 1.585 . f_c . Re_f^{-1/3} \qquad \ldots (3.5.7)$$

$$\text{where, } f_c = \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} . \cos\theta \right]^{2/3} \qquad \ldots (3.5.8)$$

3.5.2 Converging Cone Section.

From equation (3.3.17) and introducing, $a = r_1/r_2$, we have,

$$\bar{h}_{ic} = 0.84 \left[\frac{\rho_f^2 \lambda g k_f^3 . \sin 2\theta}{\mu_f . \Delta T_f . r_1} \right]^{\frac{1}{4}} . \left[\frac{1}{(1/a)^2 - 1} \right] .$$

$$\left[(1/a)^{7/3} - 1 \right]^{\frac{3}{4}} \qquad \ldots(3.5.9)$$

Again $\bar{h}_{ic} = \dfrac{Q_c}{\Delta T_f . A_c}$

$$= \frac{Q_c}{\Delta T_f . (\pi D_e H)} = \frac{\lambda w'}{\Delta T_f . (\pi D_e . H)} = \frac{\lambda G'}{H . \Delta T_f} \qquad \ldots(3.5.10)$$

Now, $\dfrac{\lambda G'}{H . \Delta T_f} = \dfrac{\lambda}{H . \Delta T_f} . \dfrac{4G'}{4} . \dfrac{\mu_f}{\mu_f} = \dfrac{\lambda \mu_f}{4H . \Delta T_f} (Re_f)$

where, $Re_f = \dfrac{4G'}{\mu_f}$.

So,

$$\bar{h}_{ic} = \frac{\lambda \mu_f}{4H . \Delta T_f} . (Re_f) \qquad \ldots(3.5.11)$$

or,

$$\frac{1}{\Delta T_f} = \frac{\bar{h}_{ic}}{Re_f} . \left(\frac{4H}{\lambda \mu_f} \right) \qquad \ldots(3.5.12)$$

From equations (3.5.9) and (3.5.12),

$$\bar{h}_{ic} = 0.84 \left[\frac{\rho_f^2 \, g \lambda k_f^3 \sin 2\theta}{\mu_f \cdot r_1} \right]^{\frac{1}{4}} \cdot \left[\frac{(1-a^{7/3})^{\frac{3}{4}}}{1 - a^2} \right] \cdot$$

$$\left[\frac{\bar{h}_{ic}}{Re_f} \left(\frac{4H}{\lambda \mu_f} \right) \right]^{\frac{1}{4}} \qquad \ldots (3.5.13)$$

After rearranging,

$$\bar{h}_{ic} \cdot \left[\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3} \right]^{1/3} = 1.585 \cdot \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} \cdot \cos \theta^{2/3} \right] Re_f^{-1/3} \qquad \ldots (3.5.14)$$

$$= 1.585 \cdot f_c \cdot Re_f^{-1/3} \qquad \ldots (3.5.15)$$

where,

$$f_c = \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} \cdot \cos \theta^{2/3} \right] \qquad \ldots (3.5.16)$$

3.5.3 Diverging-Converging Cone.

From equation (3.4.12) and putting, $a = r_1/r_2$, we have,

$$\bar{h}_{idc} = 0.706 \cdot \left[\frac{\rho_f^2 \, g \lambda k_f^3 \sin 2\theta}{\mu_f \, \Delta T_f \, r_2} \right]^{\frac{1}{4}} \cdot \frac{(1-a^{7/3})^{\frac{3}{4}}}{(1-a^2)} \qquad \ldots (3.5.17)$$

Again, $$\bar{h}_{idc} = \frac{Q_{dc}}{\Delta T_f \cdot A_{dc}}$$

$$= \frac{\lambda w'}{\Delta T_f \cdot (2\pi De . H)} = \frac{\lambda G'}{2H . \Delta T_f} \qquad \ldots (3.5.18)$$

Now,

$$\frac{\lambda G'}{2H \Delta T_f} = \frac{\lambda}{2H \Delta T_f} \cdot \frac{4G'}{4} \cdot \frac{\mu_f}{\mu_f} = \frac{\lambda \mu_f}{8H \Delta T_f} (Re_f).$$

where, $Re_f = \dfrac{4G'}{\mu_f}$

So, $$\bar{h}_{idc} = \frac{\lambda \mu_f}{8H \Delta T_f} (Re_f) \qquad \ldots (3.5.19)$$

or, $$\frac{1}{\Delta T_f} = \frac{\bar{h}_{idc}}{Re_f} \left(\frac{8H}{\lambda \mu_f} \right) \qquad \ldots (3.5.20)$$

From equations (3.5.17) and (3.5.20), we have,

$$\bar{h}_{idc} = 0.706 \left[\frac{\rho_f^2 \, g \, \lambda k_f^3 \sin 2\theta}{\mu_f \quad r_2} \right]^{\frac{1}{4}} \cdot \left[\frac{(1-a^{7/3})^{\frac{3}{4}}}{(1-a^2)} \right] \cdot$$

$$\left[\frac{\bar{h}_{idc}}{Re_f} \left(\frac{8H}{\lambda \mu_f} \right) \right]^{\frac{1}{4}} \qquad \ldots (3.5.21)$$

After rearranging, we have

$$\bar{h}_{idc} \left[\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3} \right]^{1/3} = 1.585 \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} . \mathrm{Cos}\,\theta \right]^{2/3} . \mathrm{Re}_f^{-1/3} \qquad \ldots(3.5.22)$$

$$= 1.585 .\ f_c .\ \mathrm{Re}^{-1/3} \qquad \ldots(3.5.23)$$

where,

$$f_c = \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} . \mathrm{Cos}\,\theta \right]^{2/3} \qquad \ldots(3.5.24)$$

3.6 Expression For Mean Nusselt Number.

From equation (3.2.22), for a diverging cone section, we have,

$$\bar{h}_{id} = 0.84 \left[\frac{\rho_f^2\, g \lambda k_3^3 \sin 2\theta}{\mu_f\ \Delta T_f\ r_2} \right] . \left[\frac{r_2^2}{r_2^2 - r_1^2} \right] . \left[\frac{r_2^{7/3} - r_1^{7/3}}{r_2^{7/3}} \right]^{3/4}$$

$$= 0.84 \left[\frac{\rho_f^2 g\, \lambda k_f^3\ \sin 2\theta (r_2^{7/3} - r_1^{7/3})^3}{\mu_f\ \ \Delta T_f} \right]^{1/4} . \left[\frac{1}{(r_2^2 - r_1^2)} \right] \ldots(3.6.1)$$

Multiplying both sides by $\left[(r_1 + r_2)/k_f . \cos\theta \right]$, we have,

$$\overline{Nu} = \frac{\bar{h}_{id}(r_1+r_2)}{k_f \cdot \cos\theta} = \frac{\bar{h}_{id} \cdot De}{k_f} = \frac{0.84}{\cos\theta} \cdot \left[\frac{\rho_f^2\, g\, \lambda \sin 2\theta (r_2^{7/3} - r_1^{7/3})^3}{\mu_f\, \Delta T_f\, k_f\, (r_2 - r_1)^4}\right]^{1/4}$$

...(3.6.2)

Similarly, for converging cone section, we have,

$$\overline{Nu} = \frac{\bar{h}_{ic}(r_1+r_2)}{k_f \cdot \cos\theta} = \frac{\bar{h}_{ic} \cdot De}{k_f} = \frac{0.84}{\cos\theta} \cdot \left[\frac{\rho_f^2\, g\, \lambda \sin 2\theta (r_2^{7/3} - r_1^{7/3})^3}{\mu_f\, \Delta T_f\, k_f\, (r_2 - r_1)^4}\right]^{1/4}$$

...(3.6.3)

Similarly, for diverging-converging cone section, we have,

$$\overline{Nu} = \frac{\bar{h}_{idc}(r_1+r_2)}{k_f \cdot \cos\theta} = \frac{\bar{h}_{idc} \cdot De}{k_f} = \frac{0.706}{\cos\theta} \left[\frac{\rho_f^2\, g\, \lambda \sin 2\theta (r_2^{7/3} - r_1^{7/3})^3}{\mu_f \cdot \Delta T_f \cdot k_f (r_2 - r_1)^4}\right]^{1/4}$$

...(3.6.4)

CHAPTER - 4

EXPERIMENTAL INVESTIGATION

CHAPTER - 4.

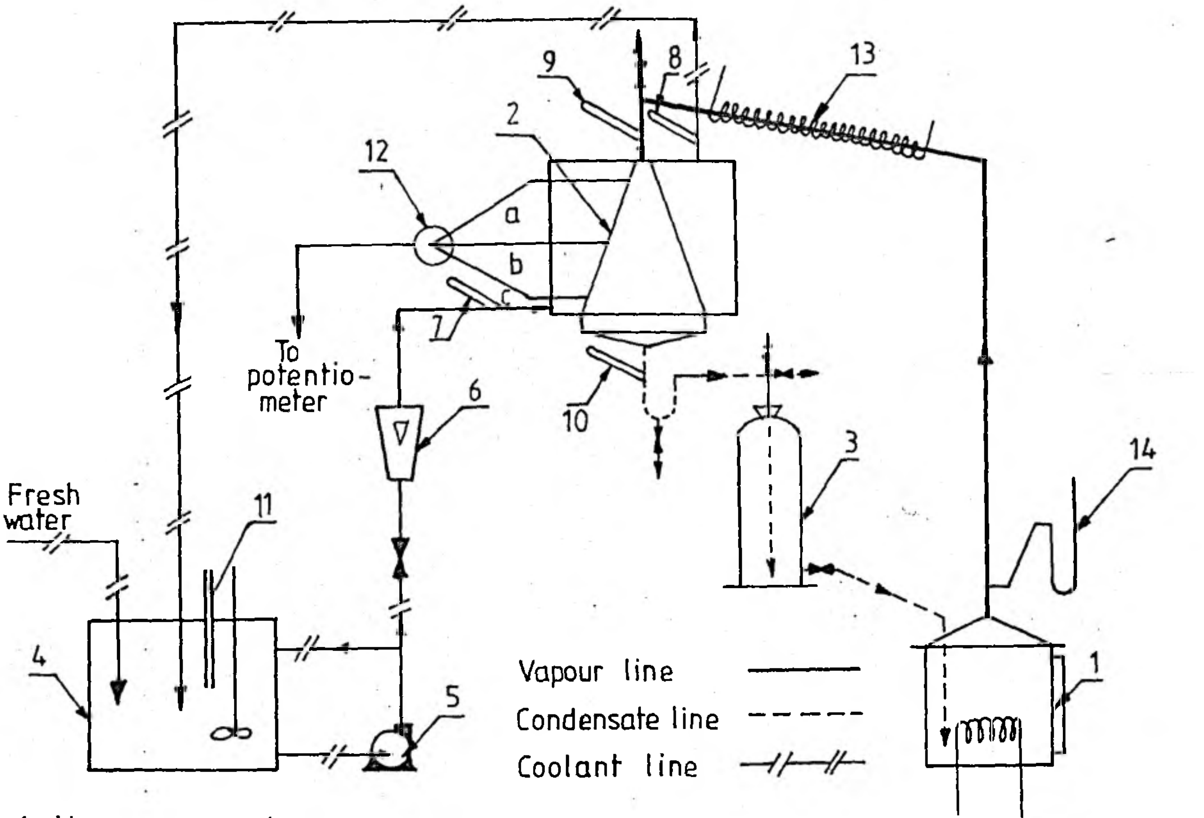
4. EXPERIMENTAL INVESTIGATION

4.1 Experimental Set-up.

Figure 4.1 shows the schematic diagram and Plate - 1 gives the overall view of the experimental set-up in the present investigation. The set-up may broadly be divided into three parts, (1) the cooling water recirculation; (2) the vapour generation and (3) test section.

4.1.1 Cooling Water Recirculation.

The constant temperature bath (4) [Type: NBE, No.23404, 220V, 50 Hz, 18KW, Made in West Germany] fitted with a stirrer, cooling coils, electrical heater with a regulator and a temperature controller was used to supply cooling water to the test condenser at constant temperature. A precision thermometer (11) having a least count of 0.1°C was inserted into the bath to record the temperature of the water. A Tulu pump (5) [Tulu-35, Set No. GBW 88704, Head 10.5 m, LPH 175, Drive 0.038 KW, 0.5A, 230V, AC/DC, manufactured by U.P. National Mfrs.P. Ltd., India] was used to supply the coolant, at constant temperature to the jacket of the condenser test section. A rotameter (6) [supplied by Chempipi Consultants, Calcutta, India] was used to record the coolant flow rate. The range of the rotameter was from 50 to 600 LPH. From the outlet of the pump (5) a by-pass line was connected back to the bath in order to control the coolant



1. Vapour generator
2. Condenser test section
3. Condensate collector
4. Constant temperature bath
5. Pump
6. Rotameter

7,8 Thermometers to measure water inlet and outlet temperatures

9,10. Thermometers to measure vapour and condensate temperatures

11. Thermometer to measure bath temperature
12. Selector switch
13. Heating coil
14. Manometer

a,b,c Thermocouples to measure condensing wall temperatures

FIG.4.1. SCHEME OF THE EXPERIMENTAL SET-UP (showing diverging test section)

flow rate effectively. The cooling water recirculation arrangement is shown in plate - 2.

4.1.2 Vapour Generation.

The vapour generator (1) was a cylindrical container with flat bottom and conical top having the capacity around five litres and consisted of two immersion heaters [Manufactured by Rex Industries, New Delhi] of 1.5 KW and 2 KW respectively. The two immersion heaters were connected to the mains through two separate variacs to ensure steady vapour generation. The vapour generator was provided with a mercury manometer (14) and a level indicator. The outlet from the top of the generator was connected to the inlet part of the test section through a 19 mm. i.d. glass tube. In order to prevent entrainment in the vapour the wall of the tube was wound up with heating coil (13). The vapour generator and the connecting pipe lines were thoroughly lagged with asbestos rope, to prevent heat losses. Arrangement was also made to recycle the condensate, formed in the test section, into the vapour generator.

4.1.3 Test Section.

This particular section of the experimental set-up comprised of the test condenser (2) itself, condensate collector (3) and a potentiometer. The test condenser consisted of either diverging, converging and diverging-converging (combined) tube sections, made from brass sheet, enclosed in a cylindrical cooling jacket or shell, also made of brass. Detailed dimensions of these tube sections and cooling jacket are given in Table.AII-1 to TableAII-4 of Appendix - II. The same diverging test condensers were used as converging test condensers simply by inverting. The detailed connections of a test condenser with the shell have been shown in figure 4.2 to figure 4.5.

Copper-constantan thermocouples (22 BWG) were fixed on the test condenser, uniformly spaced along the height, in order to record the inside condensing surface temperature at different positions. The arrangement of fixing these thermocouples has been shown in figure 4.6. These thermocouples were connected to a potentiometer [Cat. No. PL54 , manufactured by Toshniwal Industries Pvt. Ltd., India] through selector switch (12). The cold junction was kept at 0°C by using ice in a thermoflask and dipping the junction into it. The thermocouple wires were inserted into plastic sleeves to prevent any short-circuiting.

A graduated condensate collector (3) [Capacity - 500 ml, made of glass] was used to collect the condensate formed in the test condenser. The collector was connected to the test condenser through a U-tube. Four precision glass thermometers (7,8,9 and 10) were provided to measure the inlet and outlet temperatures of coolant, vapour and condensate temperatures respectively. The jacket of the test condenser was properly lagged with asbestos rope to prevent heat losses. The vapouriser used in the investigation and the test section are shown in Plate - 3.

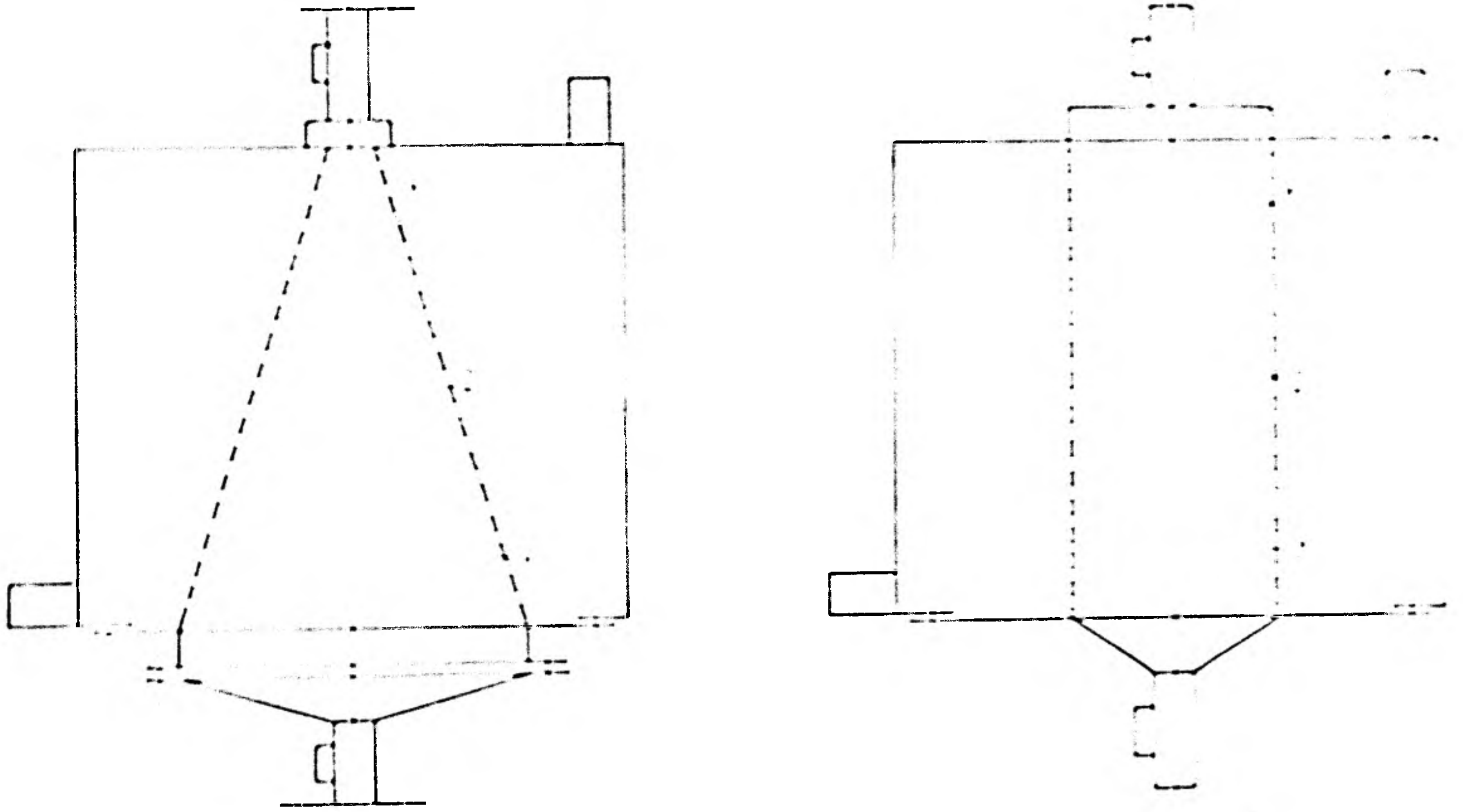


FIG.4.2.(a) DIVERGING / CONVERGING TUBE TEST-SECTION, (b) UNIFORM CYLINDRICAL TUBE HAVING SAME HEAT TRANSFER AREA AND LENGTH

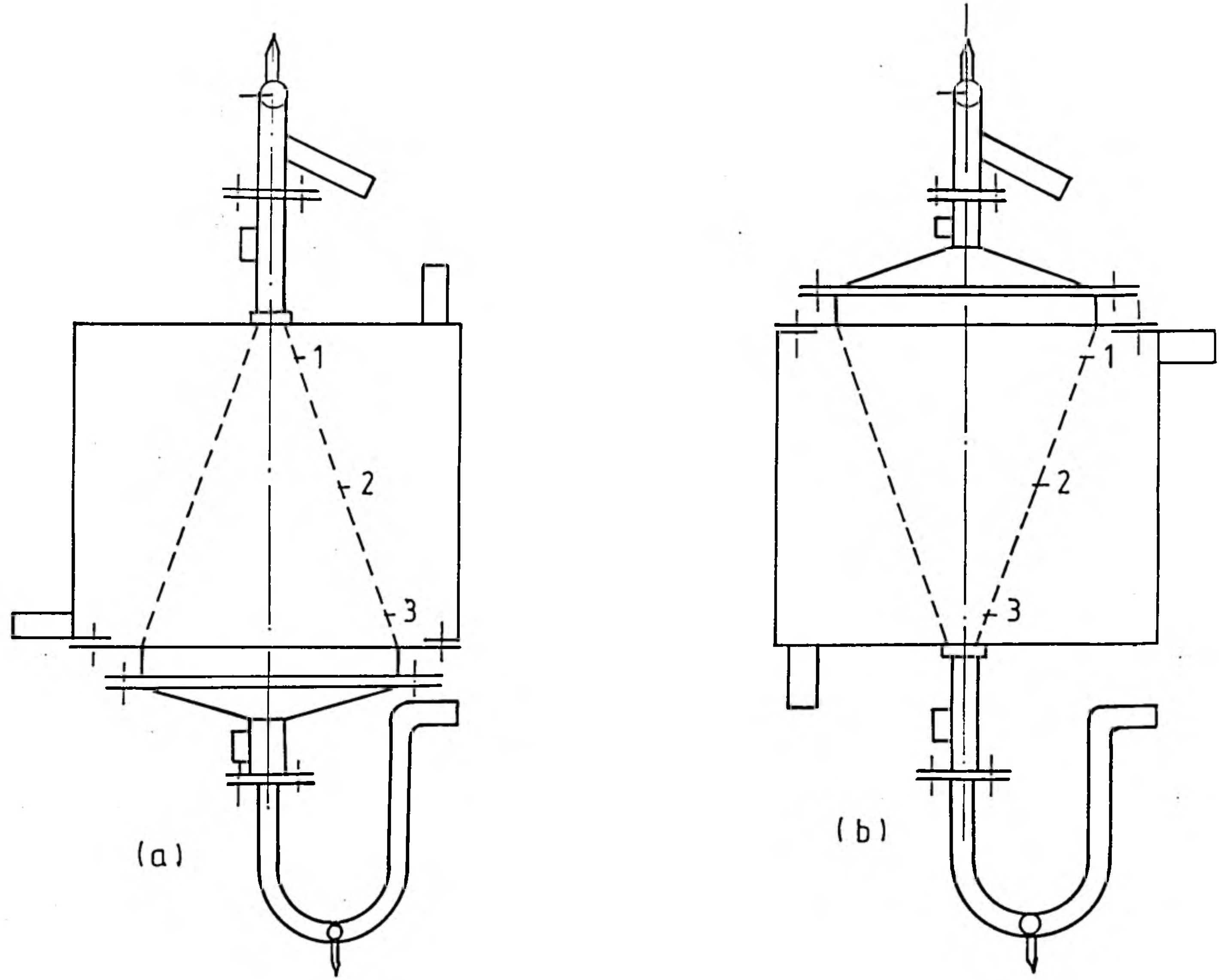
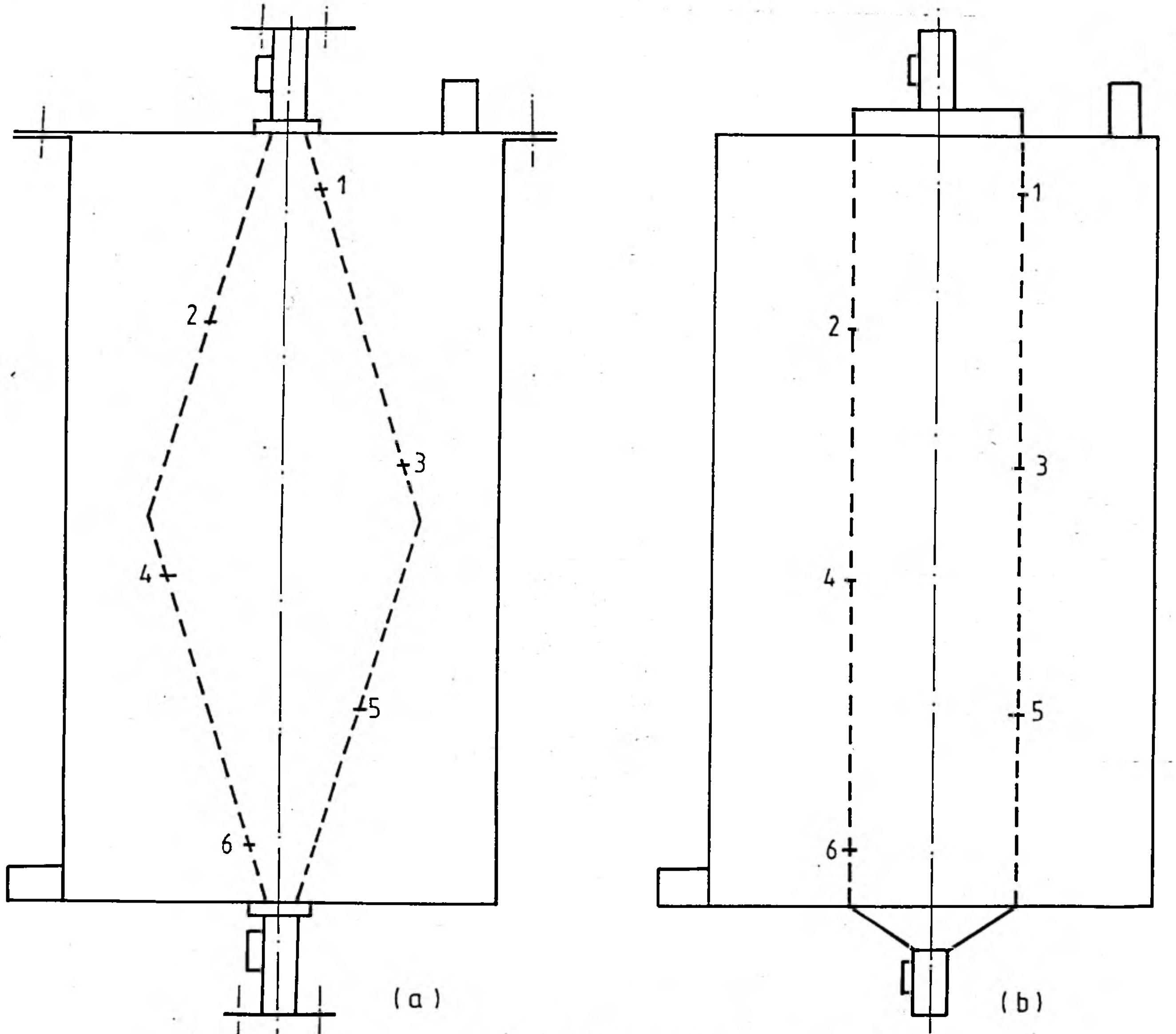
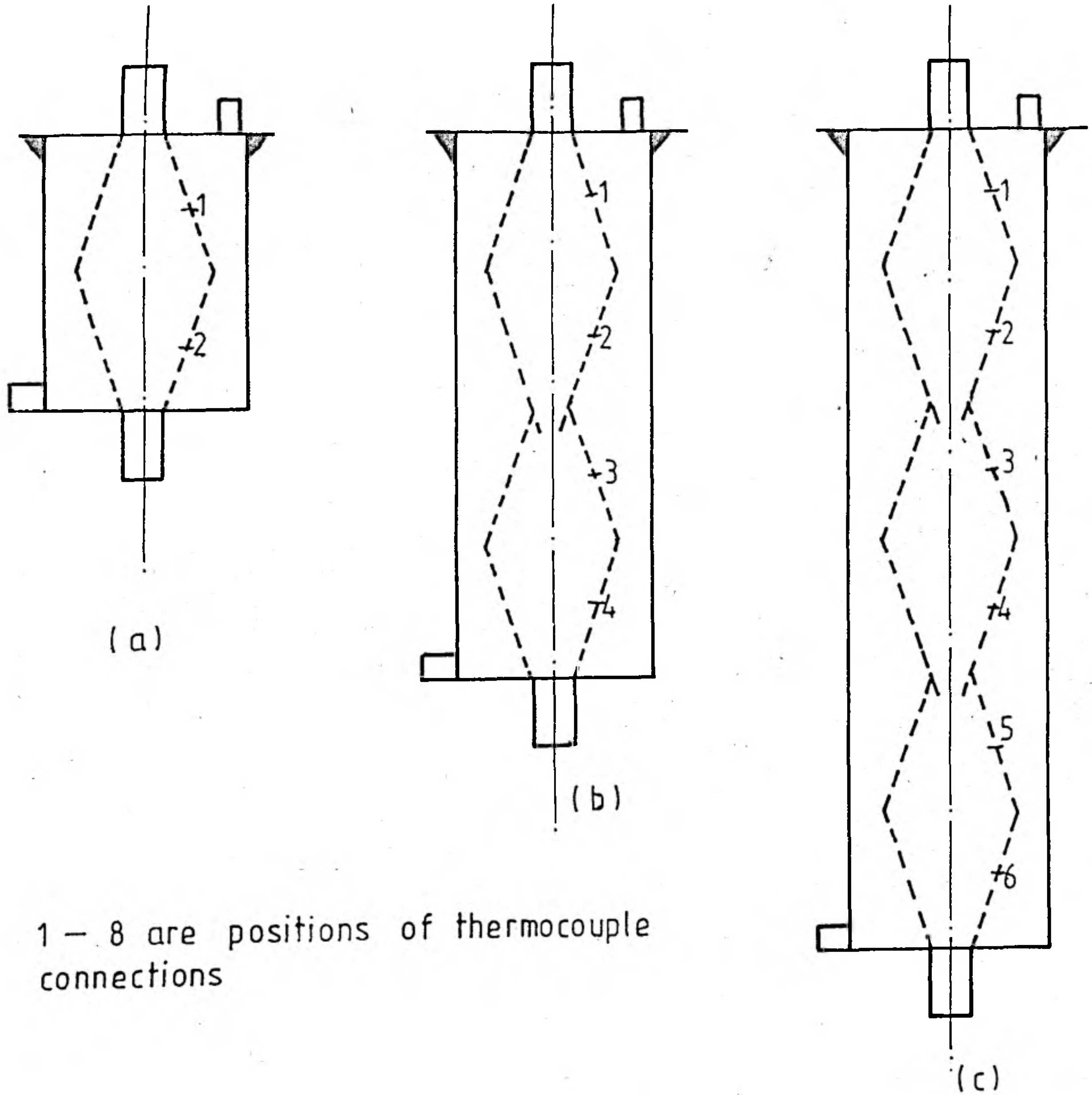


FIG.4.3. TEST SECTION ASSEMBLY FOR (a) DIVERGING AND (b) CONVERGING CONE SECTIONS USING THE SAME TEST PIECES.



1 – 6 are positions of thermocouple connections

FIG.4.4.(a) DIVERGING – CONVERGING TUBE TEST-SECTION, (b) UNIFORM CYLINDRICAL TUBE HAVING SAME HEAT TRANSFER AREA AND LENGTH



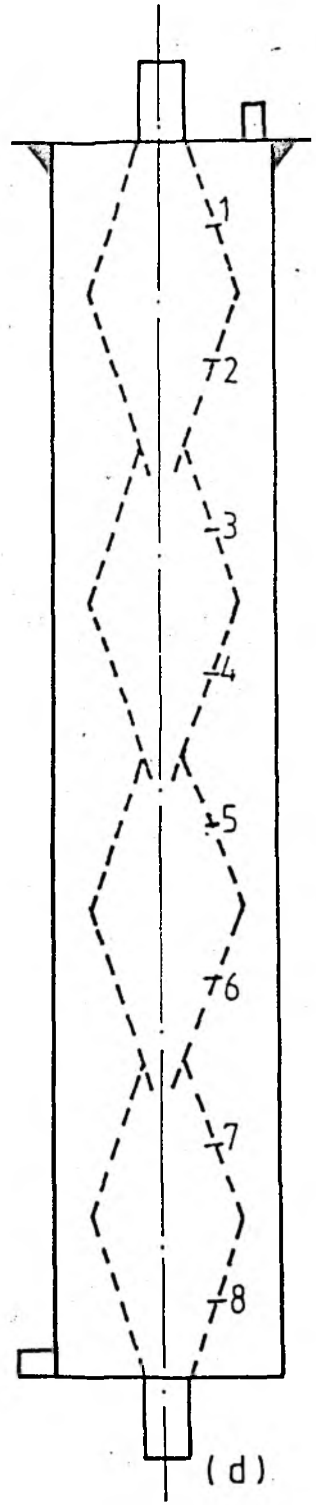


FIG.4.5. DIVERGING - CONVERGING TUBE AND SHELL CONTAINING (a) SINGLE UNIT, (b) TWO UNITS, (c) THREE UNITS, (d) FOUR UNITS OF DIVERGING - CONVERGING TUBE

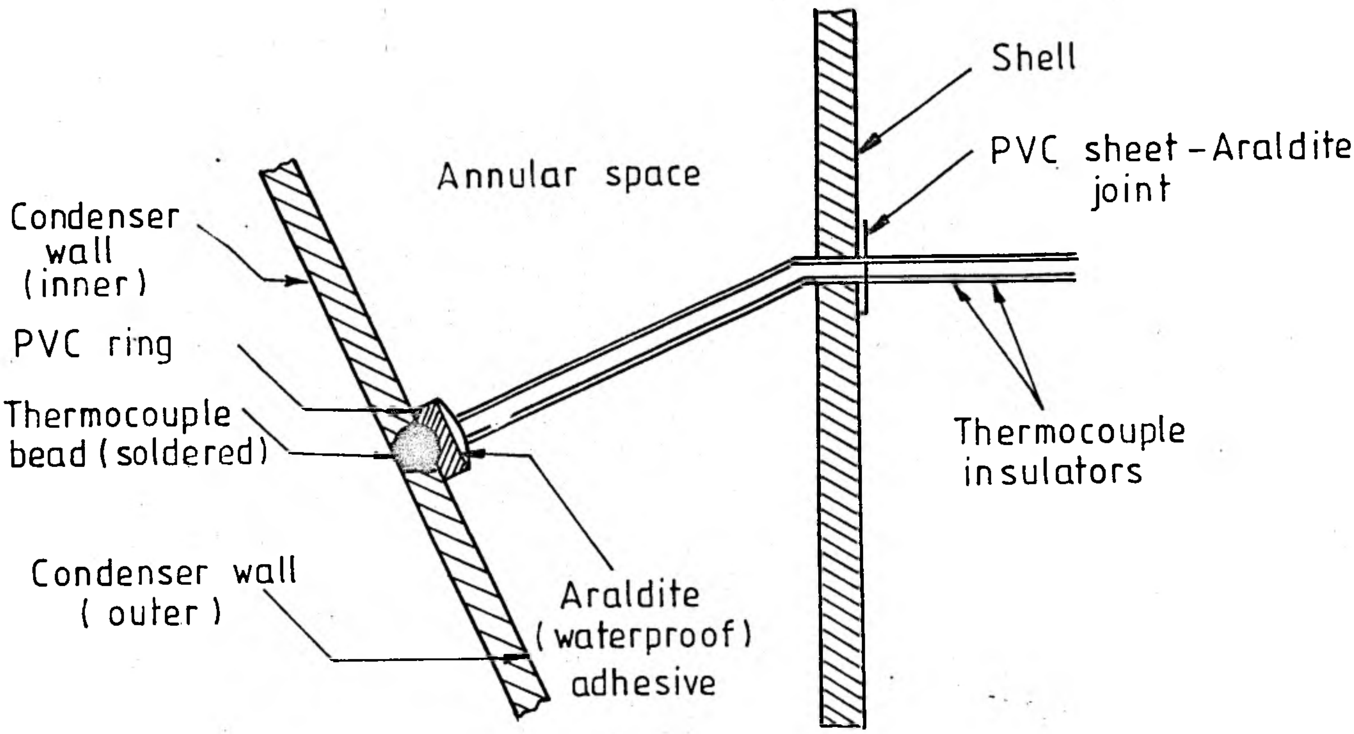


FIG.4.6. ARRANGEMENT OF THERMOCOUPLE CONNECTION ON CONDENSING SURFACE.

4.2 Experimental Procedure.

Prior to actual experimentation, the rotameter and thermocouples, used during the experiment, were calibrated and the respective calibration charts were made for ready use.

4.2.1 Calibration of Rotameter.

The rotameter was calibrated against known measured flow rates. During the calibration the rotameter float was set at a particular marking position say at 60, 100, 200 LPH. The flow was controlled with the help of a by-pass line valve. When the float was found steady, water was collected in a bucket for some stipulated time period, which was recorded by means of a stop watch. The water thus collected was then measured in a measuring cylinder and the flow rate was computed in litres/hr. Measurements were repeated for various rotameter readings and for three water temperatures viz. around 30°, 40° and 50°C. The measured volumetric flow rates were plotted against the rotameter readings to get the calibration curve. These calibration curves have been employed for all the subsequent calculations.

4.2.2 Calibration of Thermocouple.

The thermocouples were inserted in a copper tube closed at the bottom end and fitted with sand. A precision glass thermometer with 0.1°C accuracy was also inserted along with the thermocouples. Care was taken to place the thermometer bulb as well as the thermocouple junctions at the same depth. It was then immersed in a constant temperature water bath fitted with stirrer and a temperature controller. Thermocouple readings were recorded from the potentiometer in millivolts under steady state conditions and the corresponding temperatures were recorded from the thermometer. Following this procedure thermocouple readings were noted from 30°C to 100°C at every 5°C intervals. During calibration the cold junction was kept in an ice bath to maintain it at 0°C. It was found that the readings were same for all the thermocouples for a particular temperature. The readings in millivolts were then plotted against corresponding degree centigrades to get the calibration curve. This calibration curve has been used for all the subsequent calculations.

4.2.3 Heat Transfer Study.

The experimental investigation on heat transfer study can broadly be divided into following sections depending upon the different system configurations. These are,

i) diverging cone sections;

ii) converging cone sections;

iii) diverging-converging (combined) cone sections, both cones being symmetrical in shape;

iv) corresponding uniform cylindrical tubes having same heat transfer area and length/height; and

v) diverging-converging tube containing one unit and more than one unit,

4.2.3.1 Liquids Used In The Investigation.

Following liquids were used for the generation of vapour,

i) Distilled water;

ii) Ethyl alcohol;

iii) Ethyl acetate, and

iv) Carbon-tetra-chloride.

The chemicals were of A.R. grade. The physical, thermodynamic and transport properties of test fluids have been given in Table AIII-1 of Appendix-III.

4.2.3.2 Initiation Of Steady State Condition For Data Recording.

The vapouriser was first dried and charged with the test fluid, ¾th to its capacity, keeping a reasonable vapour space above the liquid. The immersion heaters of the vapouriser were then switched on. Dimmerstats connected to the heaters were regulated to supply sufficient energy to vapourise the liquid. Before passing the coolant into the jacket of the test condenser, the vapour from the vapouriser was allowed to pass through the test condenser to drive out any air/noncondensables present in the system. This was carried out by opening the top and bottom stopcocks of the test condenser. Cooling water at a particular constant temperature was then pumped into the jacket of the condenser.

The first and foremost thing of the initiation process was to maintain a constant coolant temperature in the bath. This was done by regulating the heater and by controlling the flow of cold water through the cooling coil fitted with the bath. When the temperature of the coolant in the bath became constant and steady, as recorded by the thermometer (11), attention was focussed on to the control of vapour generation at a constant rate, at atmospheric pressure. A condition of stagnant vapour inside the test condenser was maintained throughout the experiment, by not allowing vapour to pass through the bottom U-tube by maintaining a liquid level in it. The pressure was checked from the level of mercury manometer provided with the vapouriser. While observing for the steady state the condensate was recirculated to the vapouriser as it was formed in the test condenser. Steady state condition was ensured by checking the constancy of condensing wall temperatures, coolant inlet and outlet temperatures, vapour inlet and condensate outlet temperatures of the system.

4.2.3.3 Data Recording.

After reaching steady state condition the system was ready for data recording. The stop-cock at the outlet of the condensate collector was closed simultaneously the stopwatch was made on. At a constant coolant temperature and flow rate a measured amount [200 cc. for diverging/converging and 100 c.c. for diverging-converging (combined)] of condensate was collected in the collector and the corresponding time for collection was recorded from the stopwatch. While condensate was getting collected, the inlet and outlet coolant temperatures, vapour and condensate temperatures and also the temperatures of the condensing wall were recorded. The stop-cock of the condensate collector was then opened and the hold up condensate in the collector was allowed to recycle back to the vaporiser. To ensure the correctness of the data recorded the experiment was repeated atleast for thrice by maintaing the same conditions.

Following the above procedure runs were taken for four cooling water flow rates, e.g., 100, 200, 300 and 400 litres/hr. In a similar way experiments were repeated for three different coolant inlet temperature conditions e.g. 30°, 40° and 50°C. Thus for a particular liquid vapour and for a particular test condenser system, twelve sets of runs were taken, each set being repeated thrice.

The collected experimental data, their graphical manifestations and analysis have been presented in the next chapter.

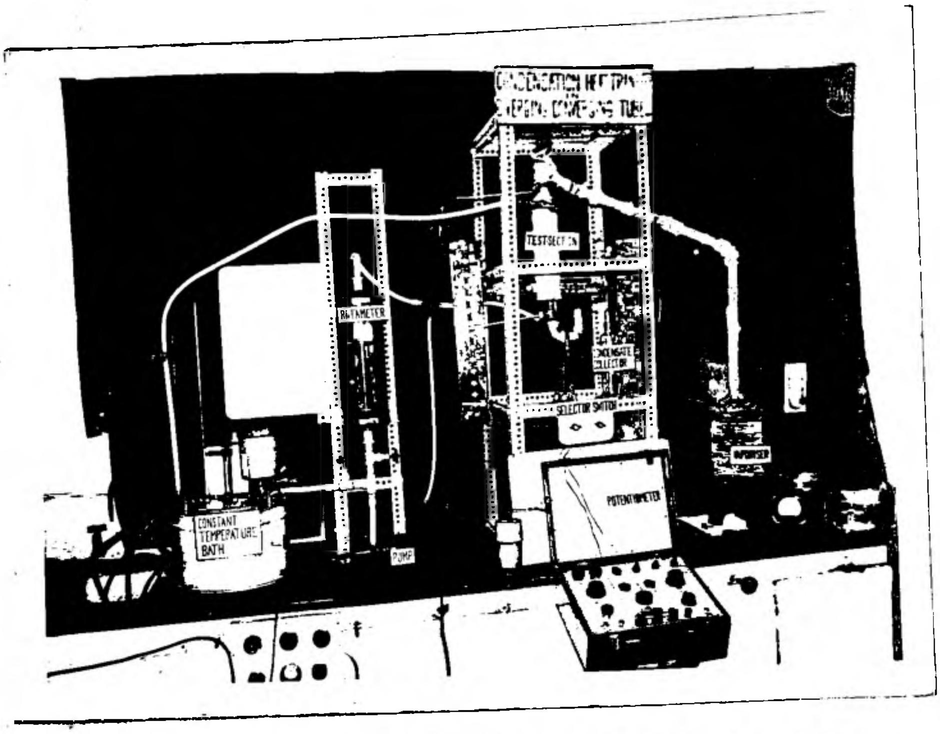
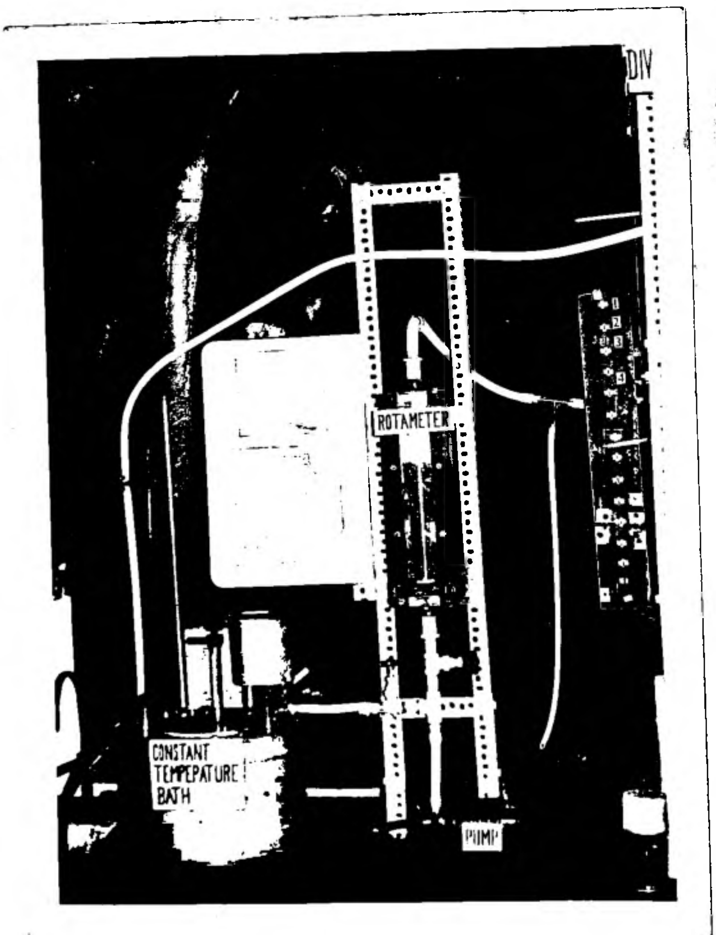


PLATE - 1 Overall View Of The Set-up



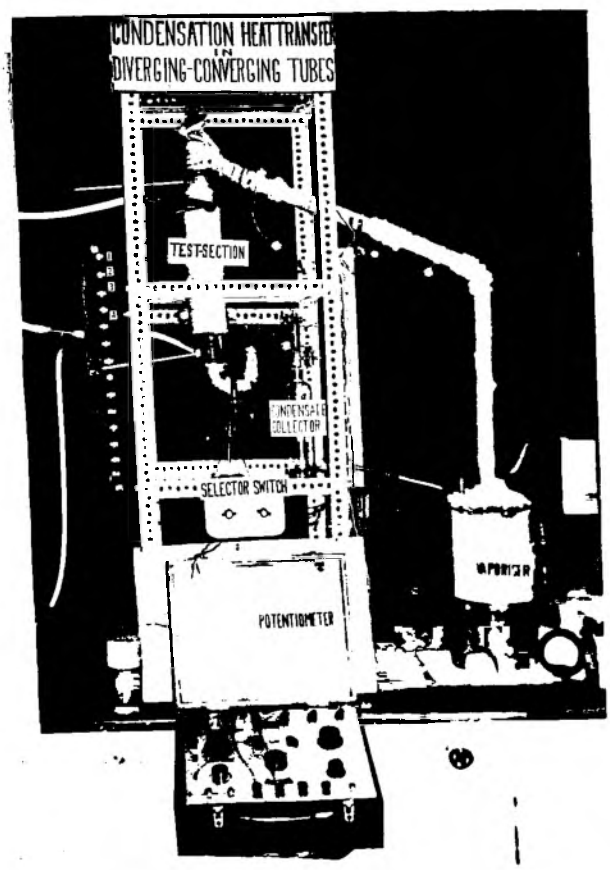


PLATE - 2 Cooling Water Recirculation Arrangement

PLATE - 3 Test Section And Vaporiser

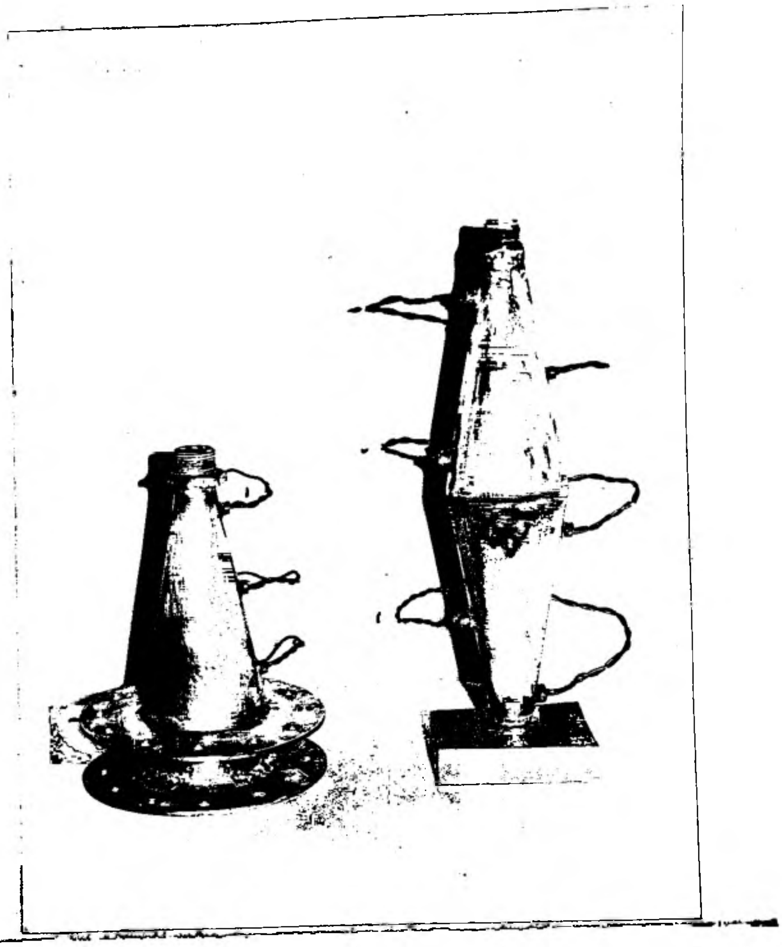


PLATE - 4 Condenser Test Section, Showing, Diverging/Converging and Diverging-Converging (combined) Test Sections.

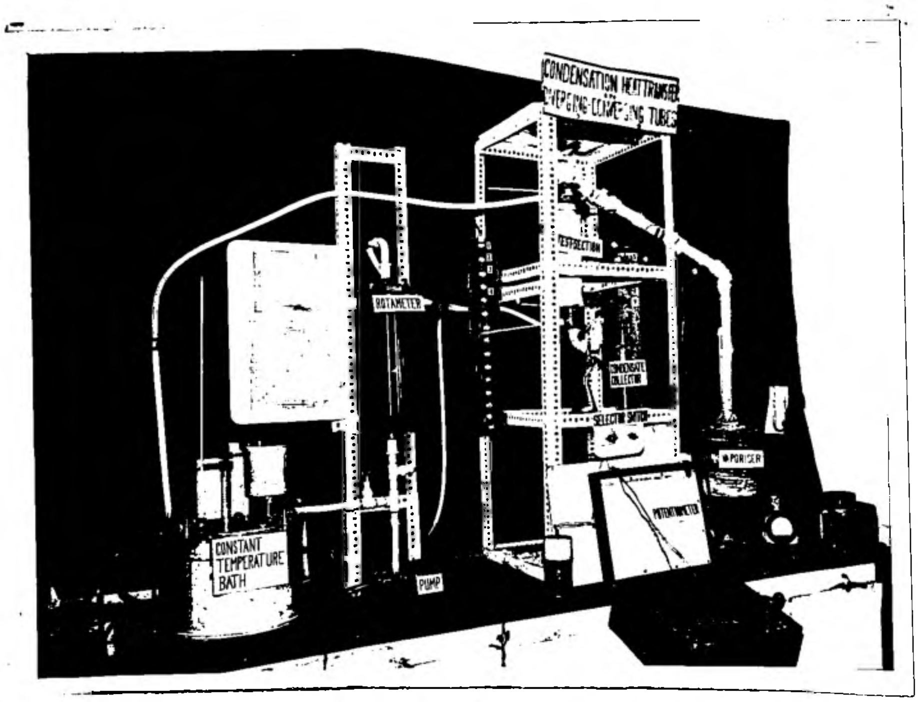


PLATE - 5 Another View Of The Set-up.

CHAPTER - 5

RESULTS AND DISCUSSION

CHAPTER - 5

RESULTS AND DISCUSSION

The objectives of the present work (as discussed in Chapter - 1) are briefly stated here. These are:

1) to develope mathematical models to study heat transfer characteristics in diverging, converging and diverging-converging combined cone sections;
2) to verify the above models through experimental investigations, and
3) to highlight the specific advantages of this constricted geometry over straight geometry, especially in enhancing the heat transfer efficiency.

As already discussed in Chapter-4, extensive experiments were carried out with specific diverging, converging and diverging-converging cone sections [detailed dimensions have been given in Appendix-II (Table AII-1 and Table AII-2)] to find out the effect of various parameters like,

1) coolant flow rate;
2) coolant inlet temperature;
3) cone angle;
4) physical properties of liquids; and
5) temperature difference (ΔT_f) between wall and condensate film;

on the condensation heat transfer characteristics of the system.

Correlations depicting these parameters have been obtained both theoretically and experimentally.

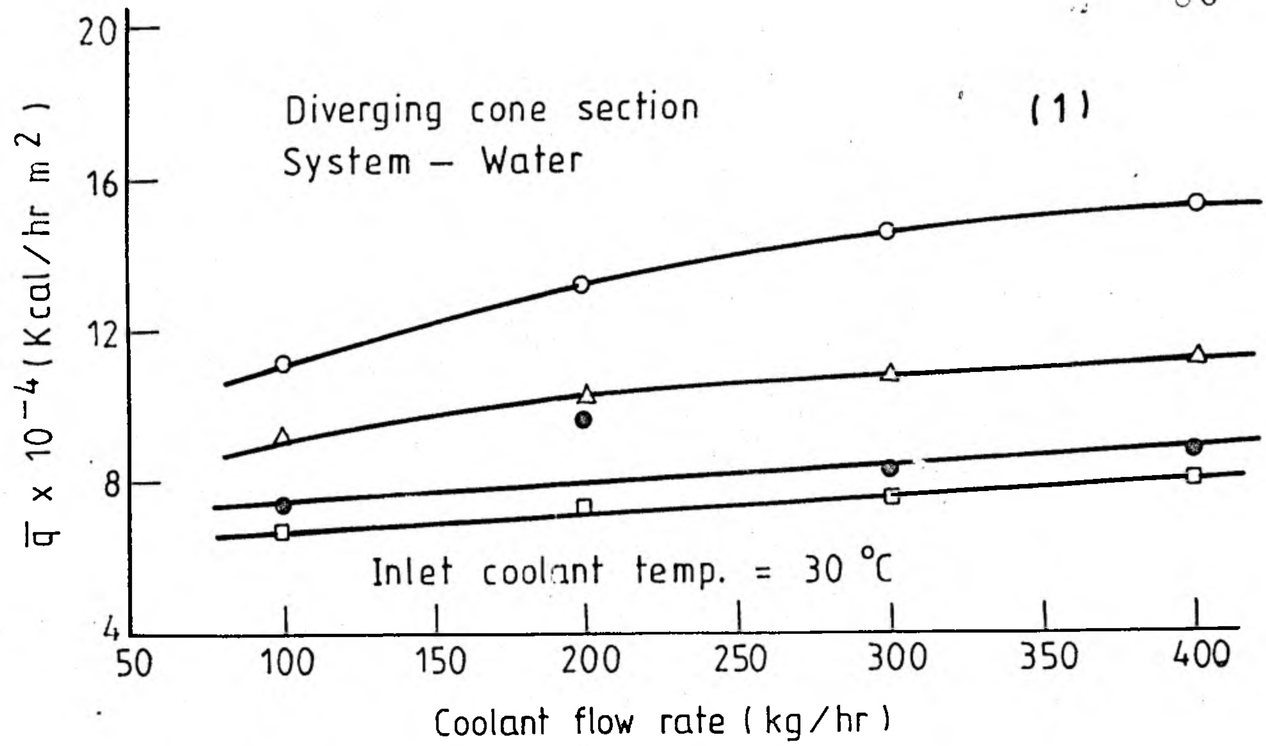
Experiments were also carried out with straight uniform tube in order to compare the performance of the constricted tube with that of straight tube. Detailed dimensions of these equivalent stright tubes are given in Appendix-II (Table AII-3 and Table AII-4).

Theoretical and experimental results are reported in Appendix-I (Table A1-1 to Table A1-63) and their graphical manifestations are furnished in figures 5.1 to 5.9.

The physical, transport and thermodynamic properties of the test liquids are given in Appendix-III (Table AIII-1). Calculations were carried out by taking the liquid properties at the saturation temperature, since it has been experimentally observed that the average condensate temperatures were very close to the saturation temperatures of the liquids. Sample calculations are given in the Appendix-VI.

5.1 Effect of Coolant Flow Rate:

The rate of heat transfer, Q, is divided by the respective heat transfer area of the test condenser sections to obtain heat flux, $\bar{q}$, which is plotted against coolant flow rate. Figure 5.1-1 and figure 5.1-2 show the effect of coolant flow rate on heat flux, $\bar{q}$, for the condensation of water and ethyl acetate vapours in diverging cone sections, for cone angles $\theta = 5°$, $10°$, $15°$, and $19°$. During these experiments the coolant



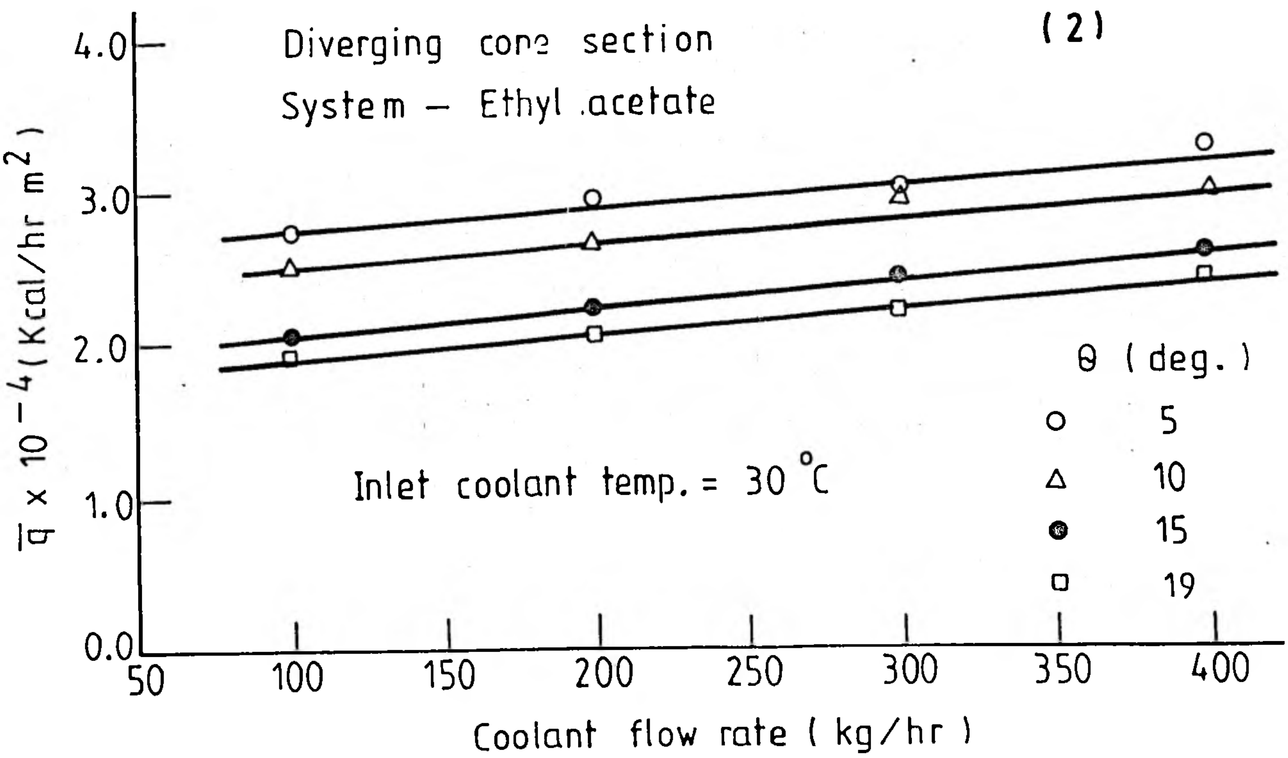
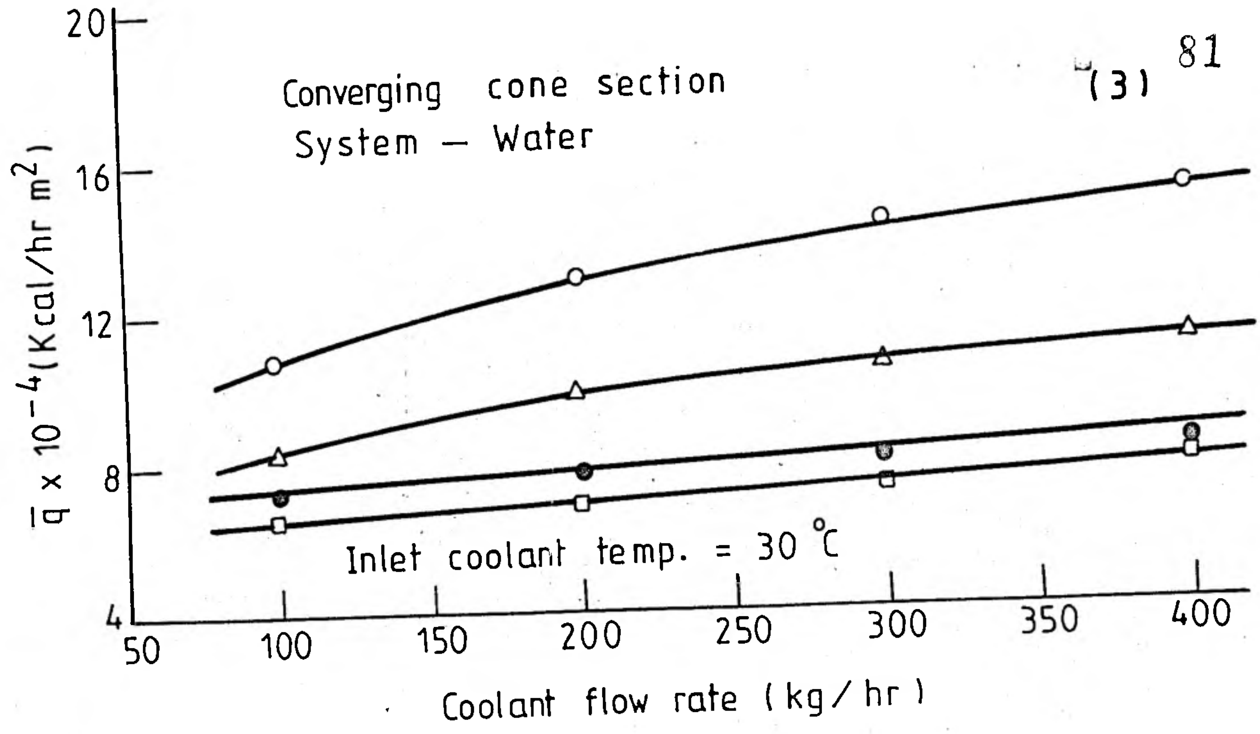


FIG.5.1. (1&2) EFFECT OF COOLANT FLOW RATE ON HEAT FLUX FOR CONDENSATION OF (1) WATER VAPOUR, (2) ETHYL ACETATE VAPOUR IN DIVERGING CONE SECTIONS



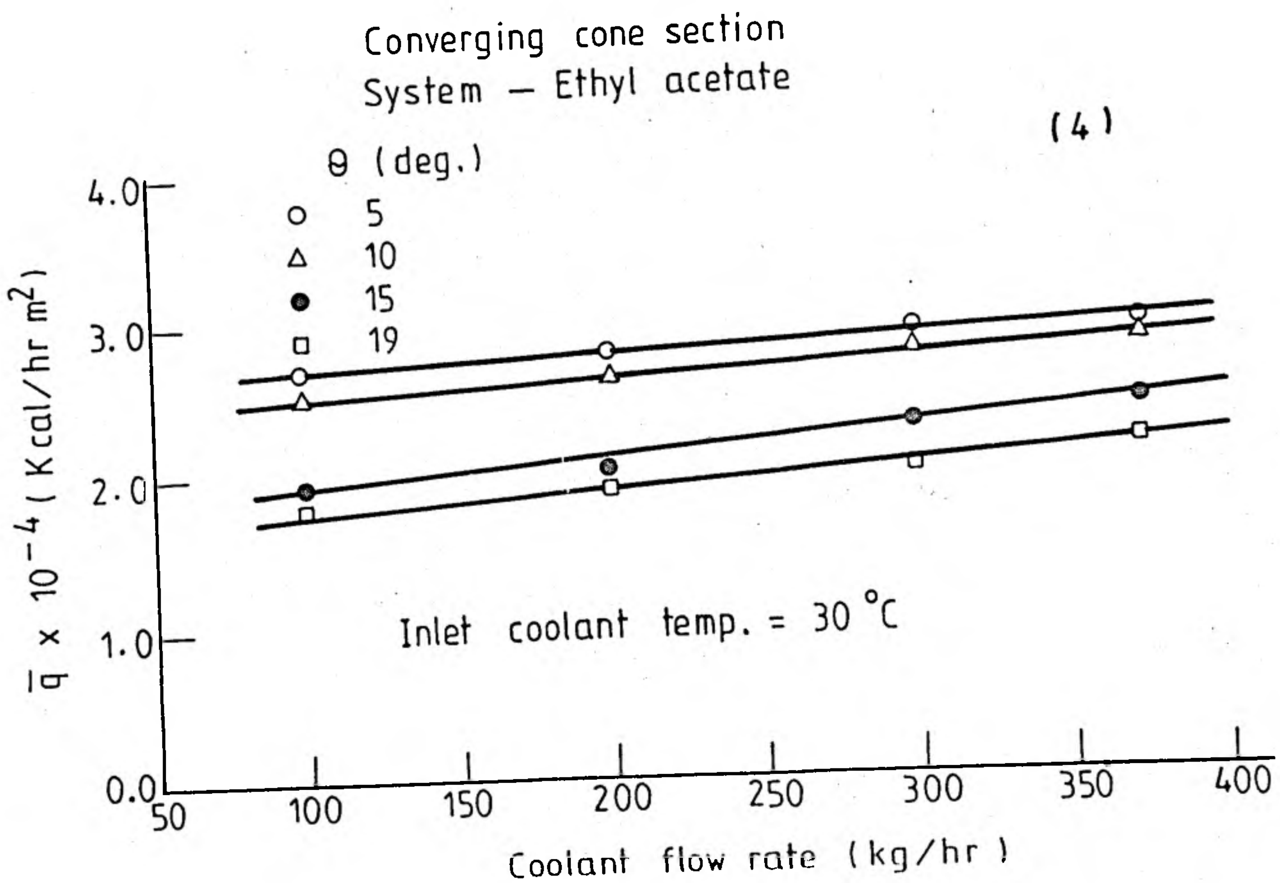
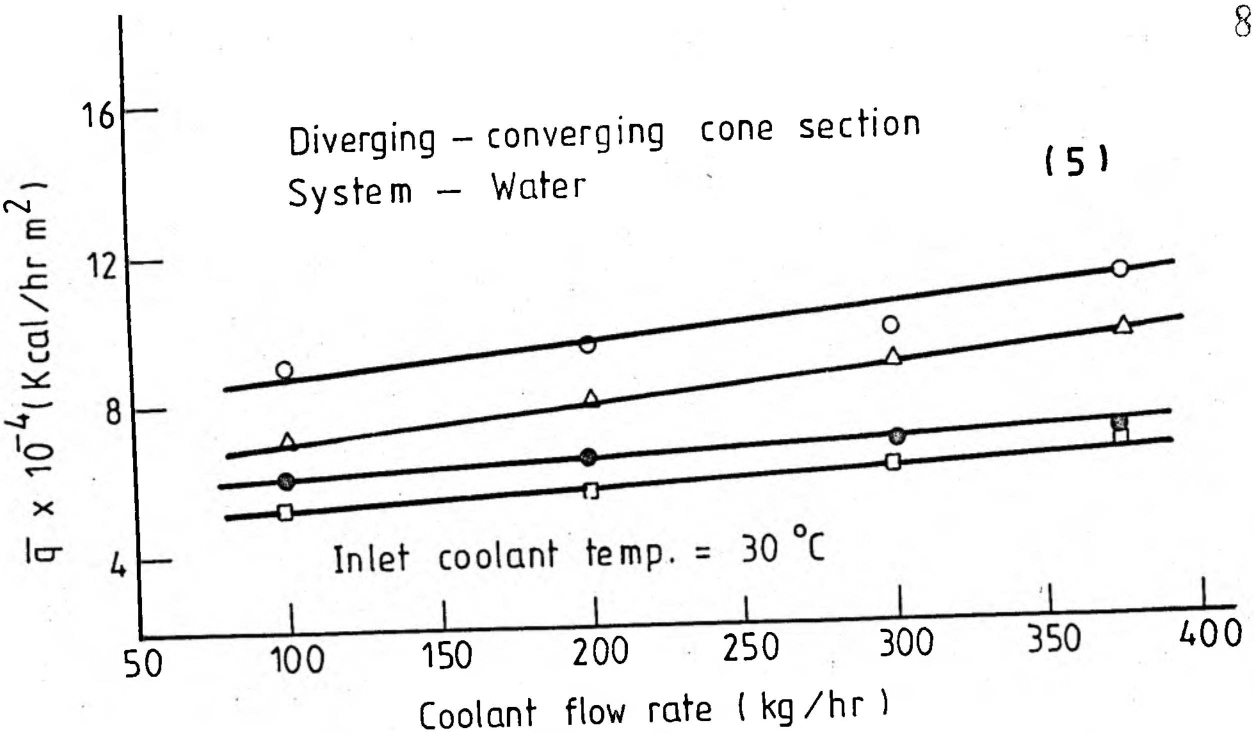


FIG.5.1. (3 & 4) EFFECT OF COOLANT FLOW RATE ON HEAT FLUX FOR CONDENSATION OF (3) WATER VAPOUR, (4) ETHYL ACETATE VAPOUR IN CONVERGING CONE SECTIONS



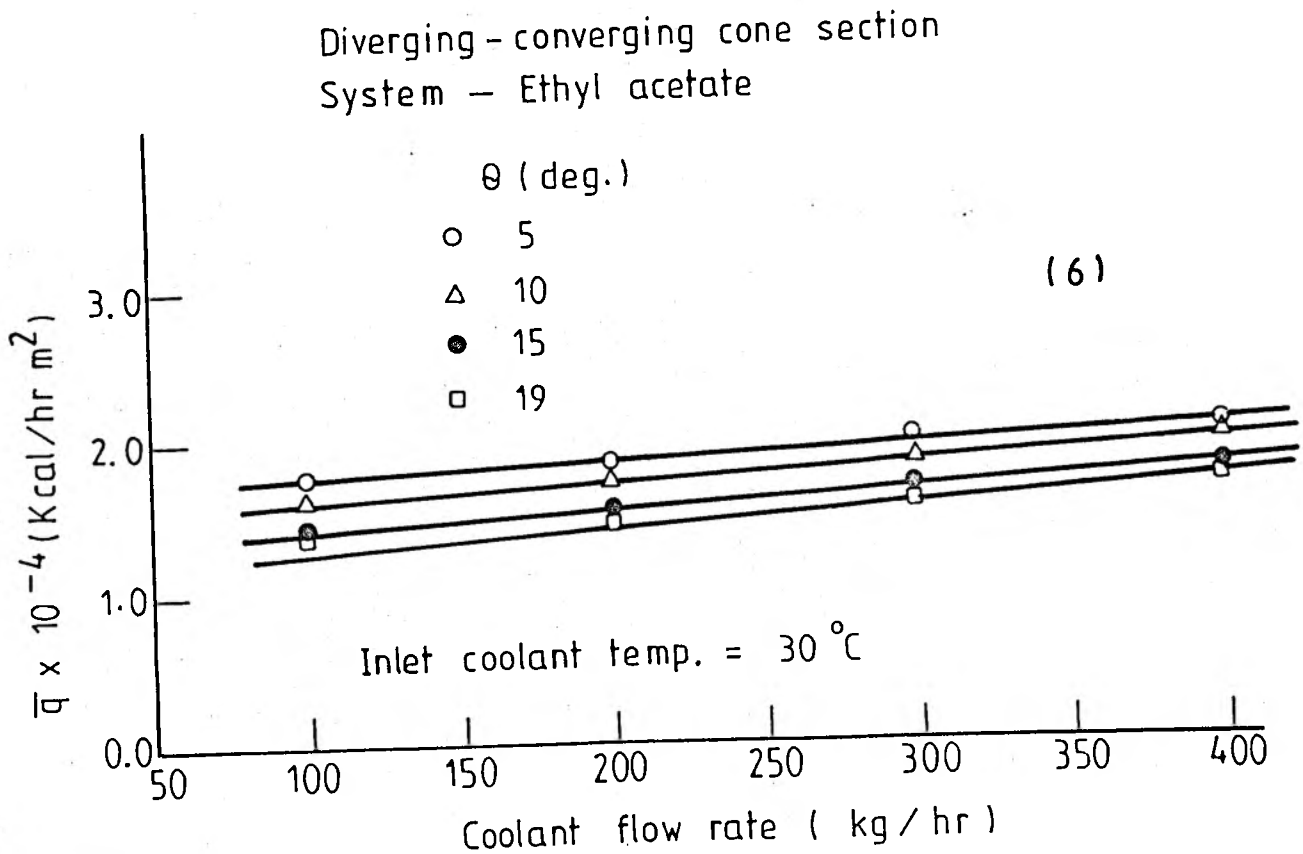
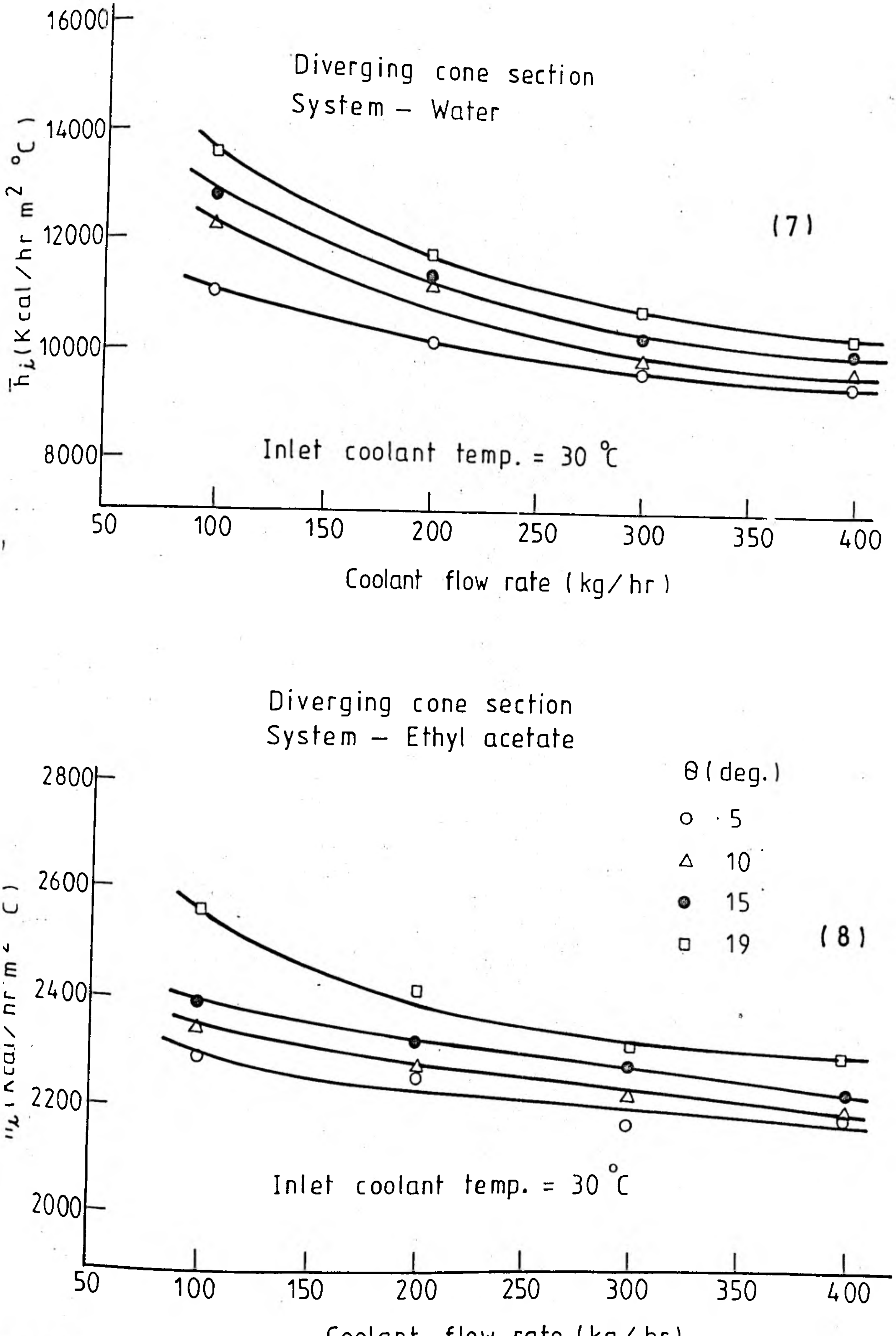


FIG.5.1. (5&6) EFFECT OF COOLANT FLOW RATE ON HEAT FLUX FOR CONDENSATION OF (5) WATER VAPOUR, (6) ETHYL ACETATE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

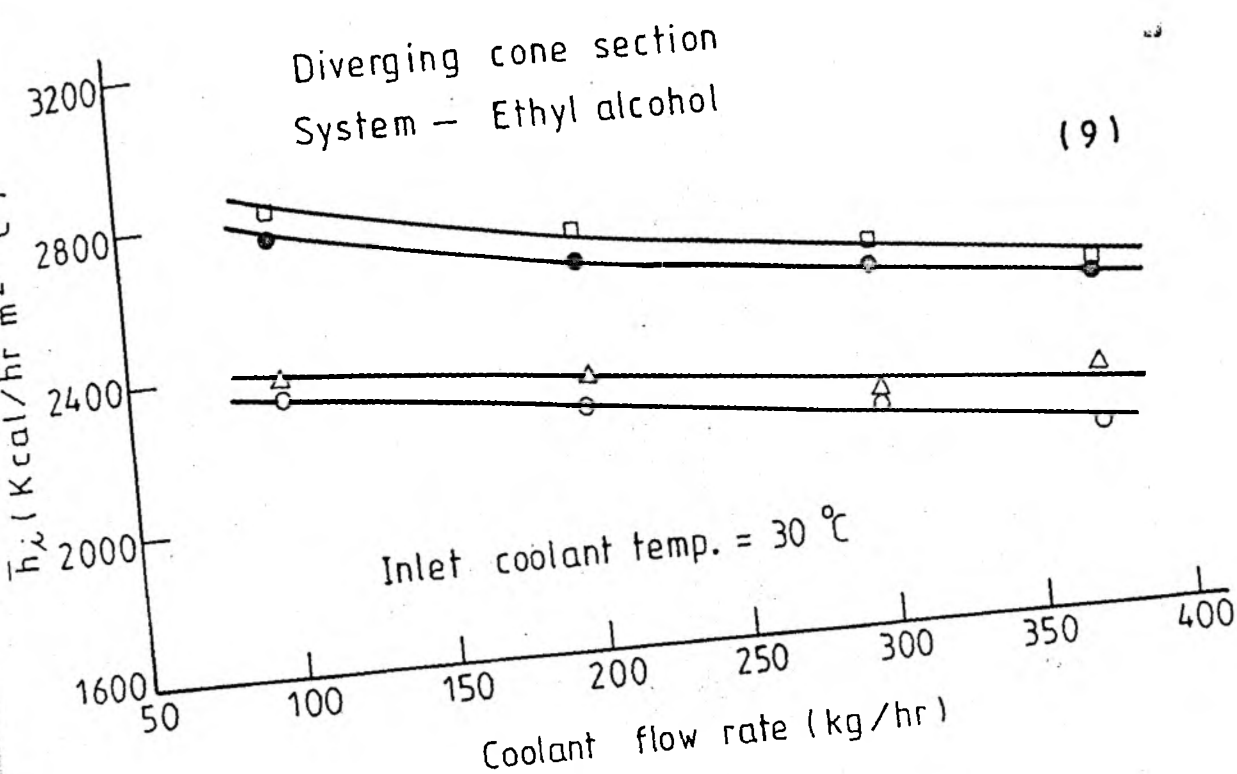
temperature was kept at 30°C.

It is evident from the plot that for the same cone angle and coolant inlet temperature, $\bar{q}$ increases with the increase in coolant flow rates. This may be attributed to the fact that the increase in coolant flow rate increases the turbulence in the annulus. As a natural consequence, the cooling water film thickness gets distributed and is reduced resulting in increase in $\bar{q}$. It is also evident from these figures that for a particular flow rate, $\bar{q}$ increases with increase in cone angle, θ. Carbon-tetra-chloride and ethyl alcohol vapours also show similar trend. Converging and diverging-converging cone sections behave in a similar manner as can be seen from figure 5.1-3 to figure 5.1-6. Here also coolant inlet temperature is 30°C. For higher coolant temperatures there is decrease in corresponding $\bar{q}$ values but the overall effect is similar, as can be seen from experimental data given in Appendix-I.

The effect of coolant flow rate on average condensing film heat transfer coefficient is shown in figure 5.1-7 to figure 5.1-10 for condensation of water, ethyl acetate, ethyl alcohol and carbon-tetra-chloride vapours in diverging cone sections with θ = 5°, 10°, 15° and 19°, the coolant inlet temperature being 30°C. It is seen that for the same cone angle, θ, the average heat transfer coefficient decreases with increase in coolant flow rates and for a particular flow rate average heat transfer coefficient increases with increase in θ. A change in the character of this dependence is possibly caused by a change



G.5.1. EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT
&8) FOR CONDENSATION OF (7) WATER VAPOUR, (8) ETHYL ACETATE VAPOUR IN DIVERGING CONE SECTION



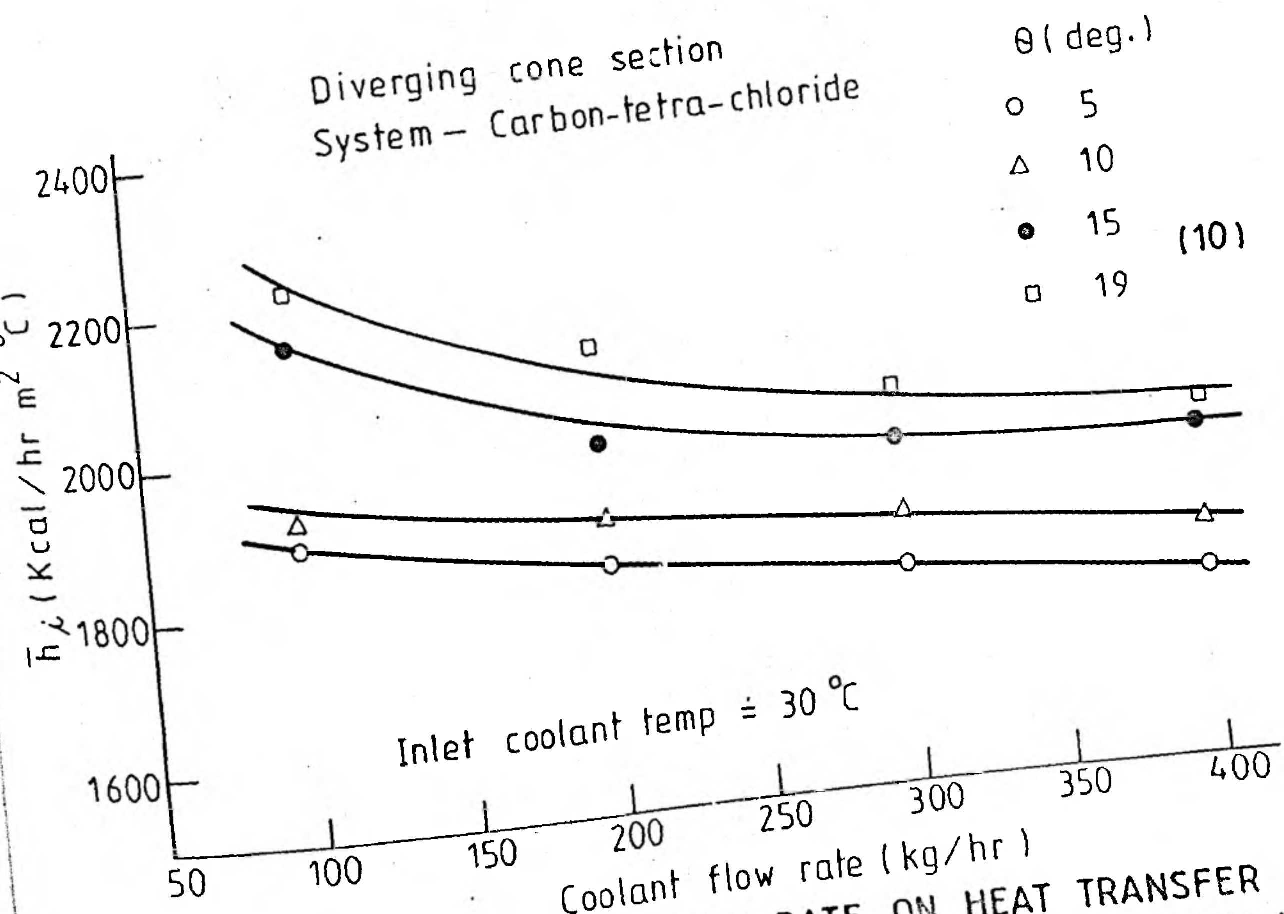
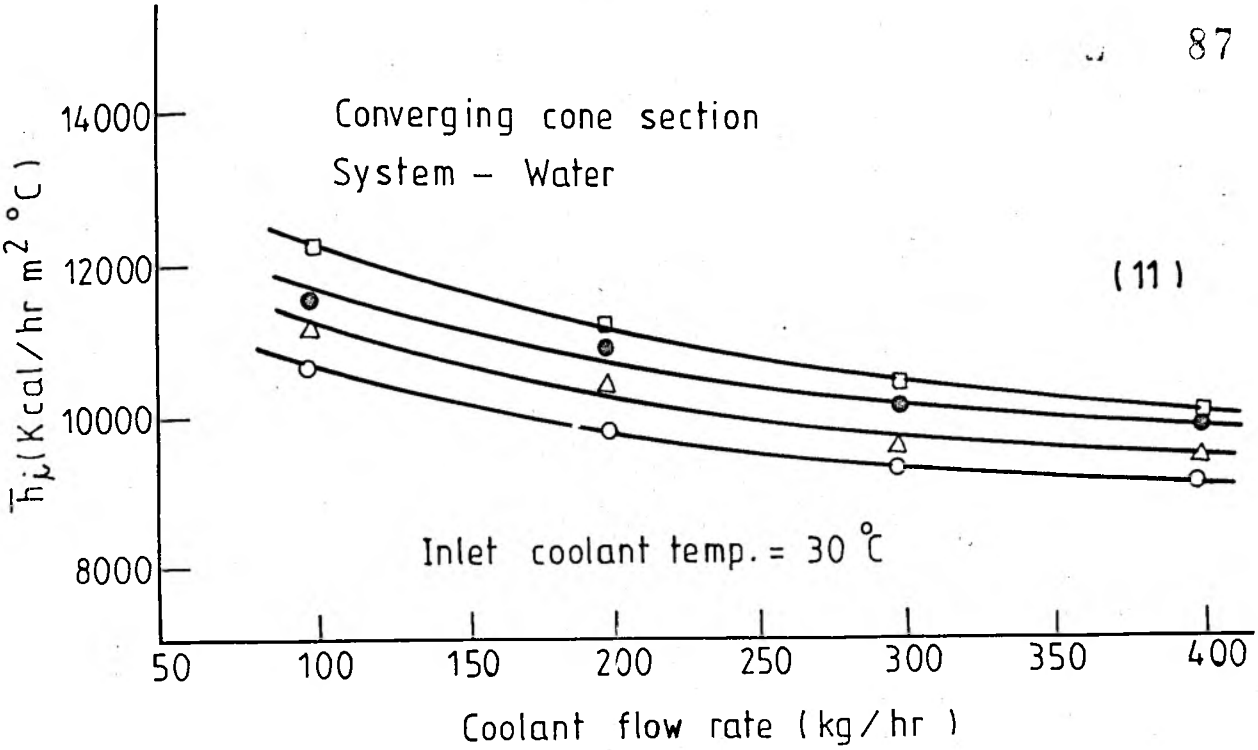


FIG.5.1. (9 & 10) EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIE FOR CONDENSATION OF (9) ETHYL ALCOHOL VAPOUR, (10) CARBO TETRA-CHLORIDE VAPOUR IN DIVERGING CONE SECTIONS

in the nature of the condensate film. With the increase in the coolant flow rate the rate of condensation increases, as a result of which the film thickness increases causing increase in resistance to heat flow across the film, vis-a-vis gradual decrease in heat transfer-coefficient. At lower range of coolant flow rates, however, the amount of condensation is not appreciable as can be seen from the figures 5.1-7 to 5.1-18, particularly, for higher cone angles. This has reflected in higher film heat transfer coefficient because of lower film thickness.

The effect of coolant flow rate on average heat transfer coefficient is found to be similar in the converging and diverging-converging cone section as well. This can be seen from figure 5.1-11 to figure 5.1-18 for the same experimental conditions. The difference is only in the magnitude of the corresponding heat transfer coefficients which are the highest in case of diverging cone sections and the lowest for diverging-converging cone sections.

The reason for higher heat transfer coefficient in case of diverging cone sections and lower value in case of converging cone sections can be explained as follows. In case of diverging cone section due to its configuration, flowarea for condensate gradually increases causing spreading of the condensate film. This reduces the film thickness, vis-a-vis resistance to heat flow. As a result of which heat transfer coefficient in diverging cone section increases. Whereas in case of converging cone section, flow area gradually decreases resulting in



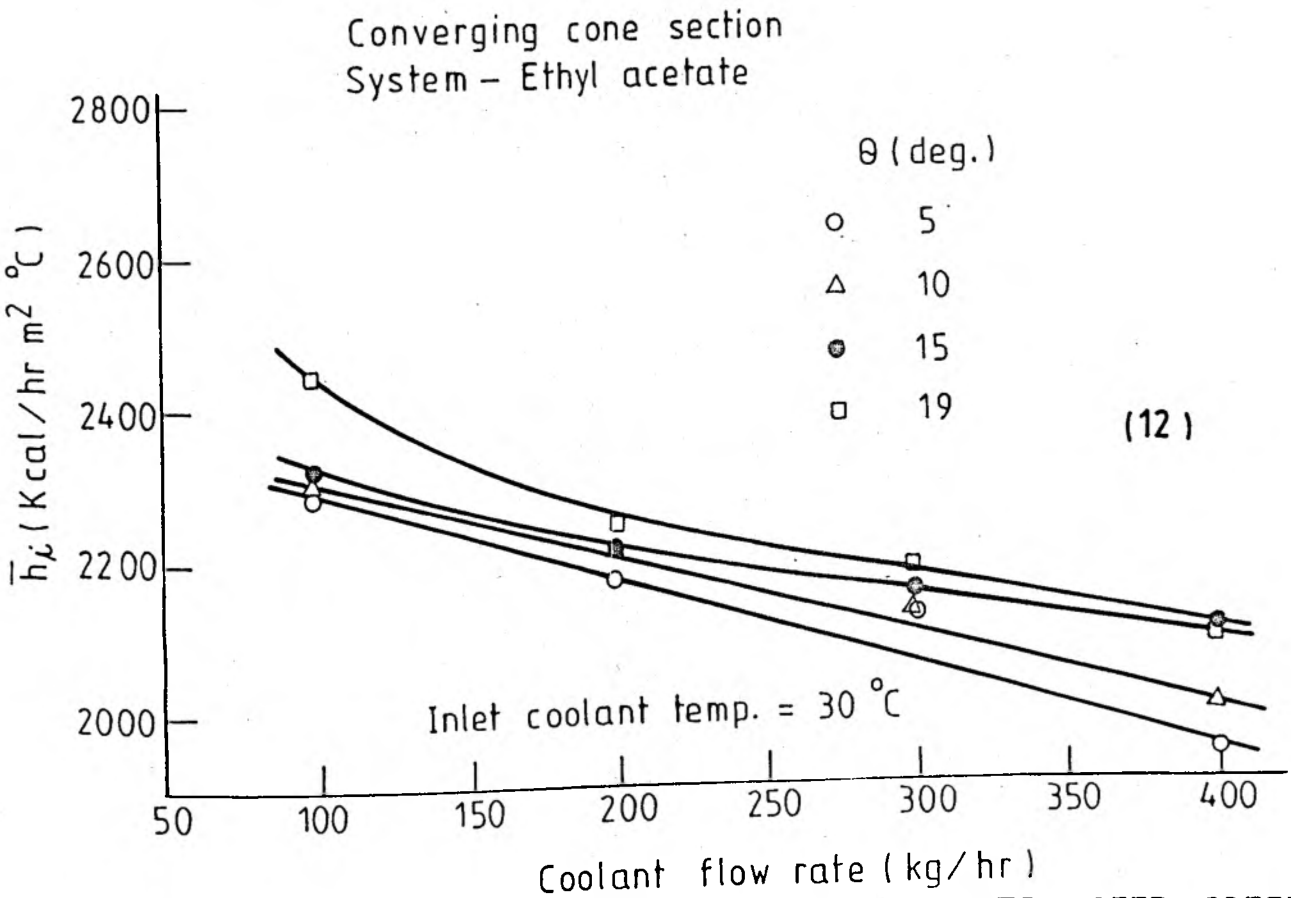
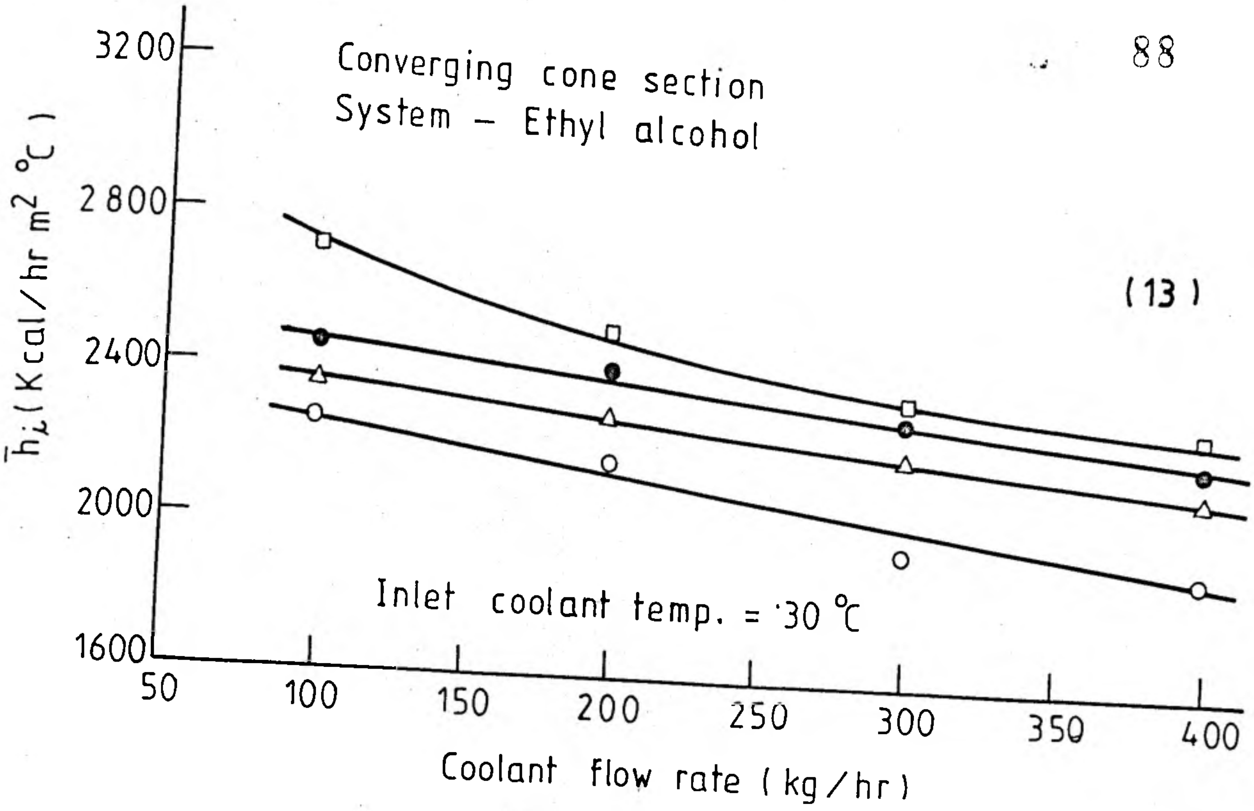


FIG. 5.1. (11 & 12) EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (11) WATER VAPOUR, (12) ETHYL ACETATE VAPOUR IN CONVERGING CONE SECTIONS



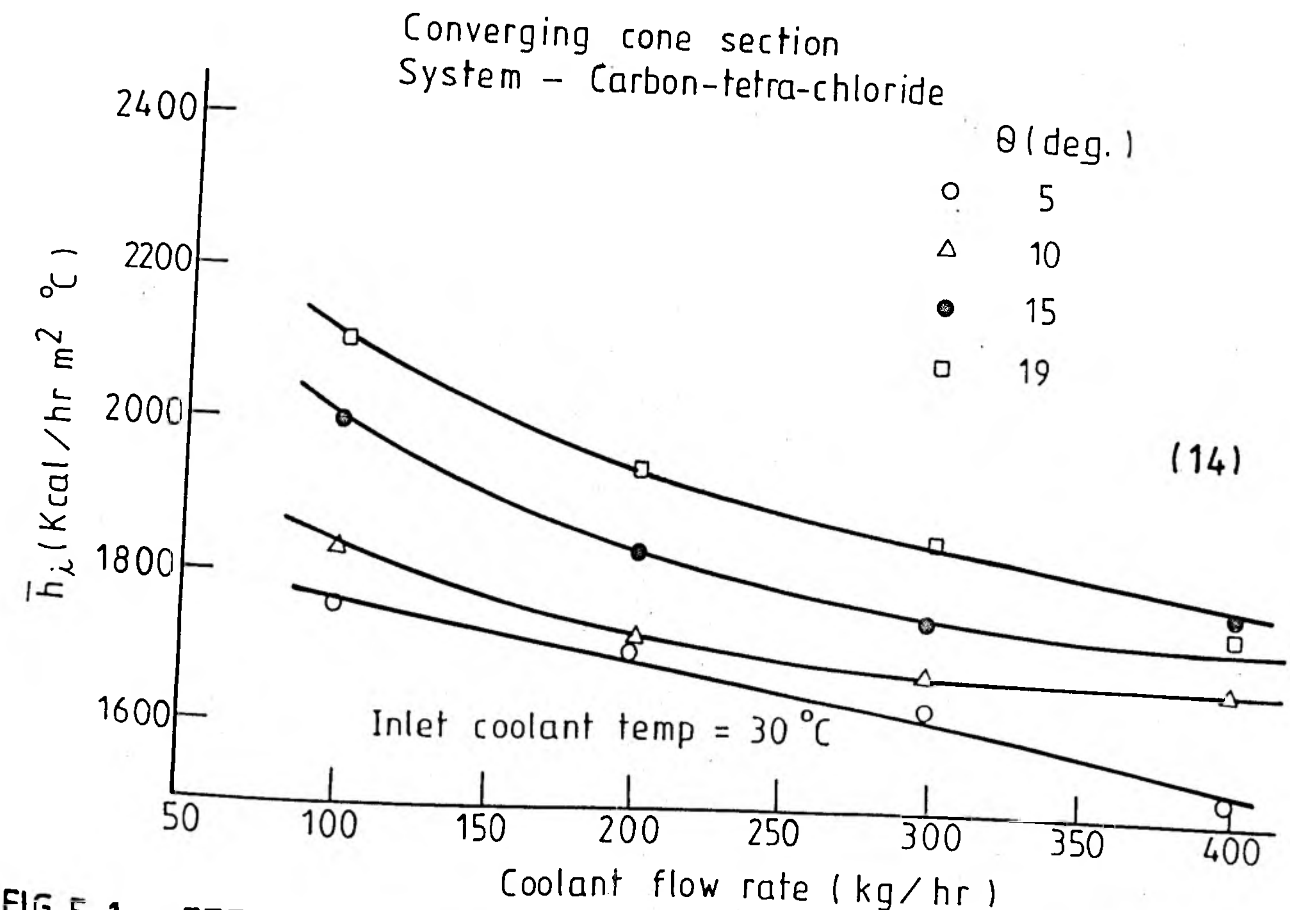
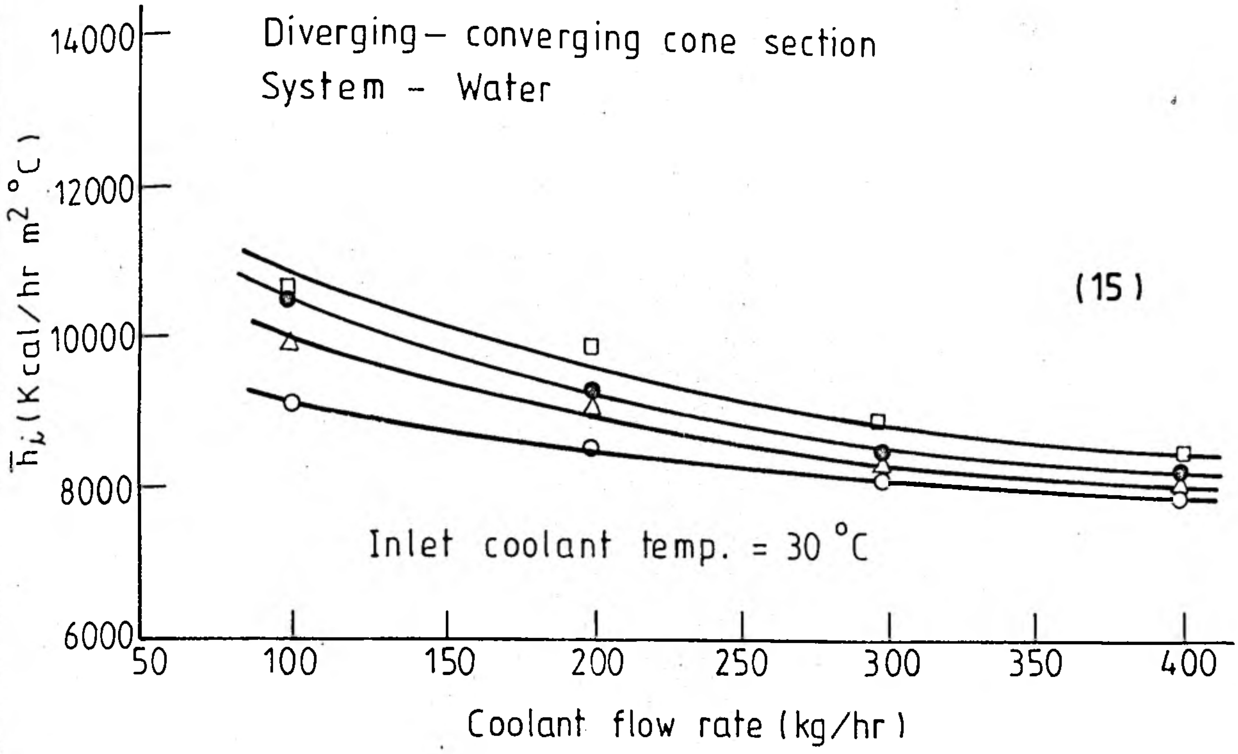


FIG.5.1. (13 & 14) EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (13) ETHYL ALCOHOL VAPOUR, (14) CARBON-TETRA-CHLORIDE VAPOUR IN CONVERGING CONE SECTIONS



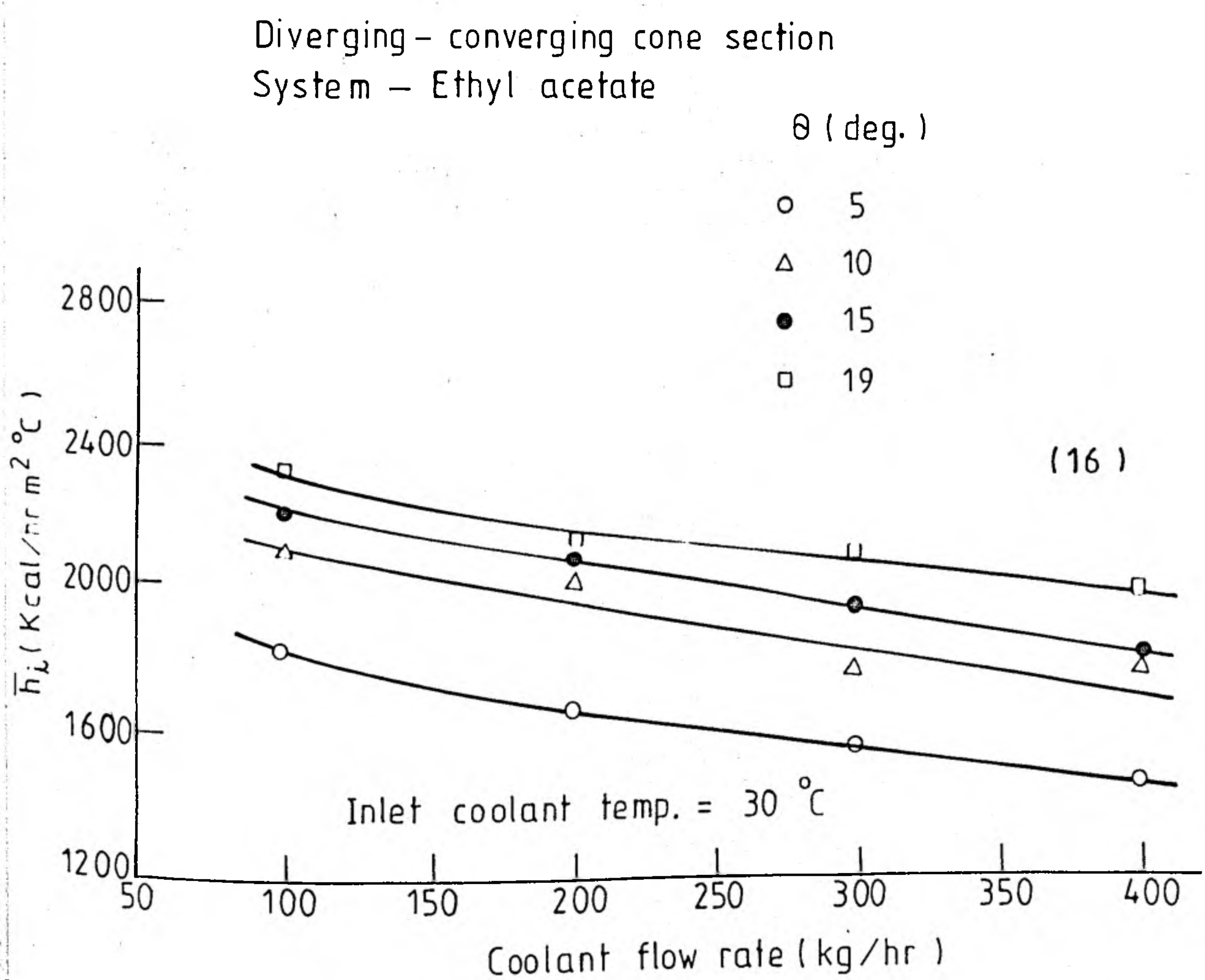
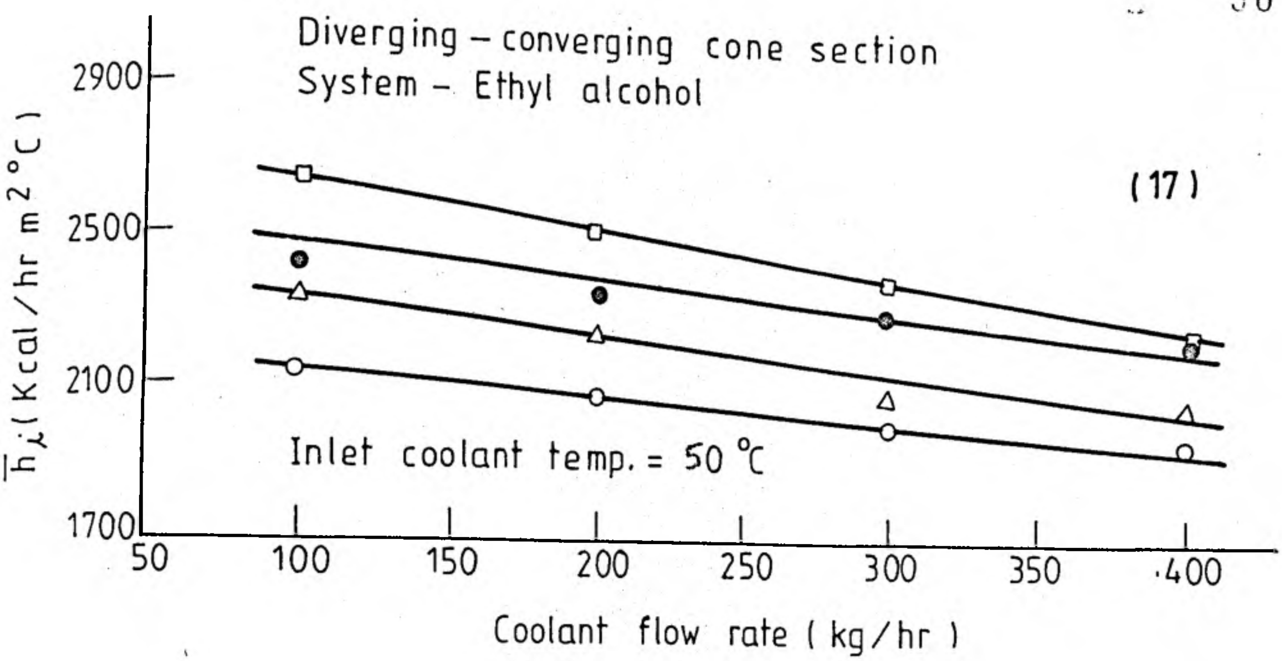


FIG.5.1. EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT
5&16) FOR CONDENSATION OF (15) WATER VAPOUR, (16) ETHYL ACETATE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS



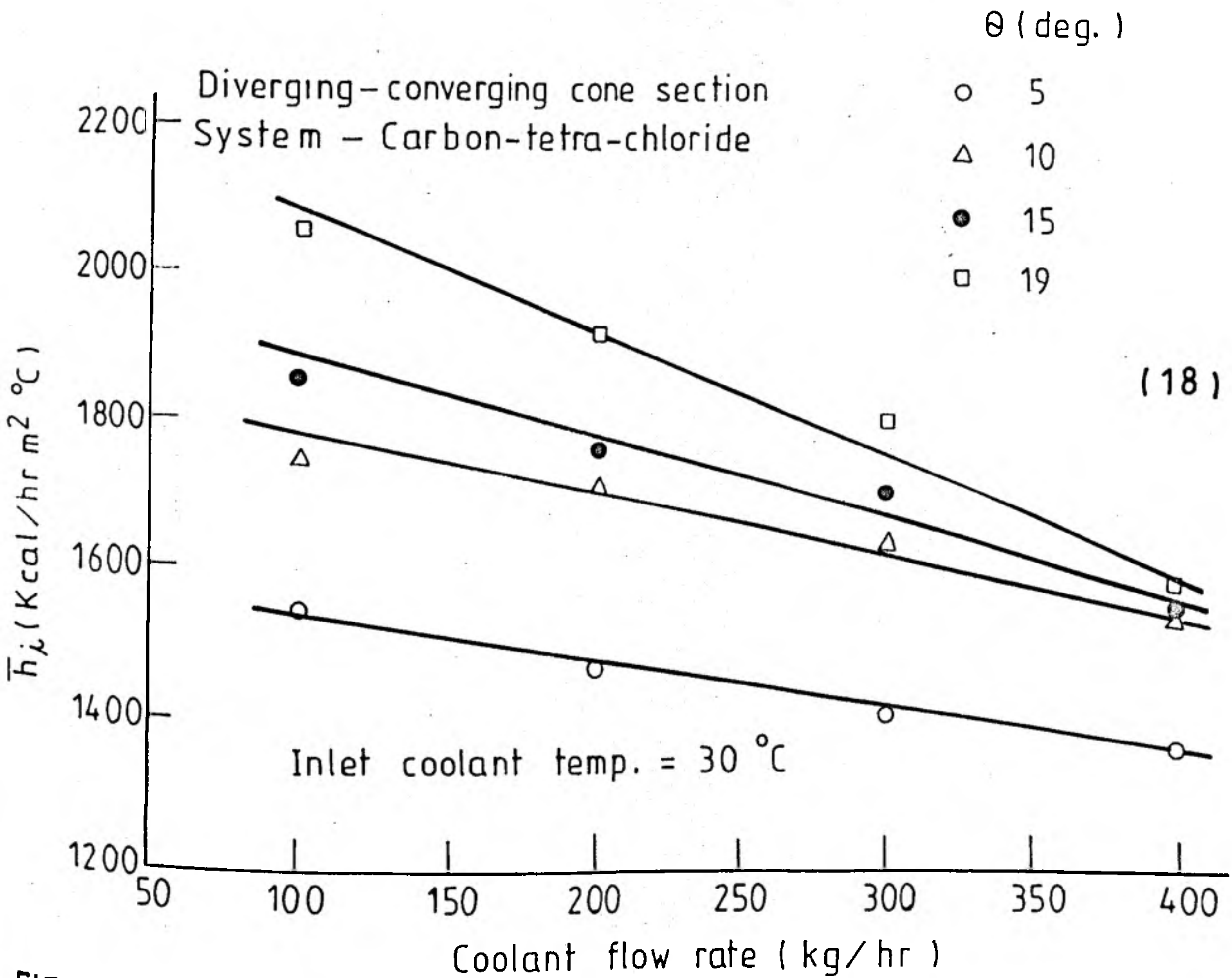
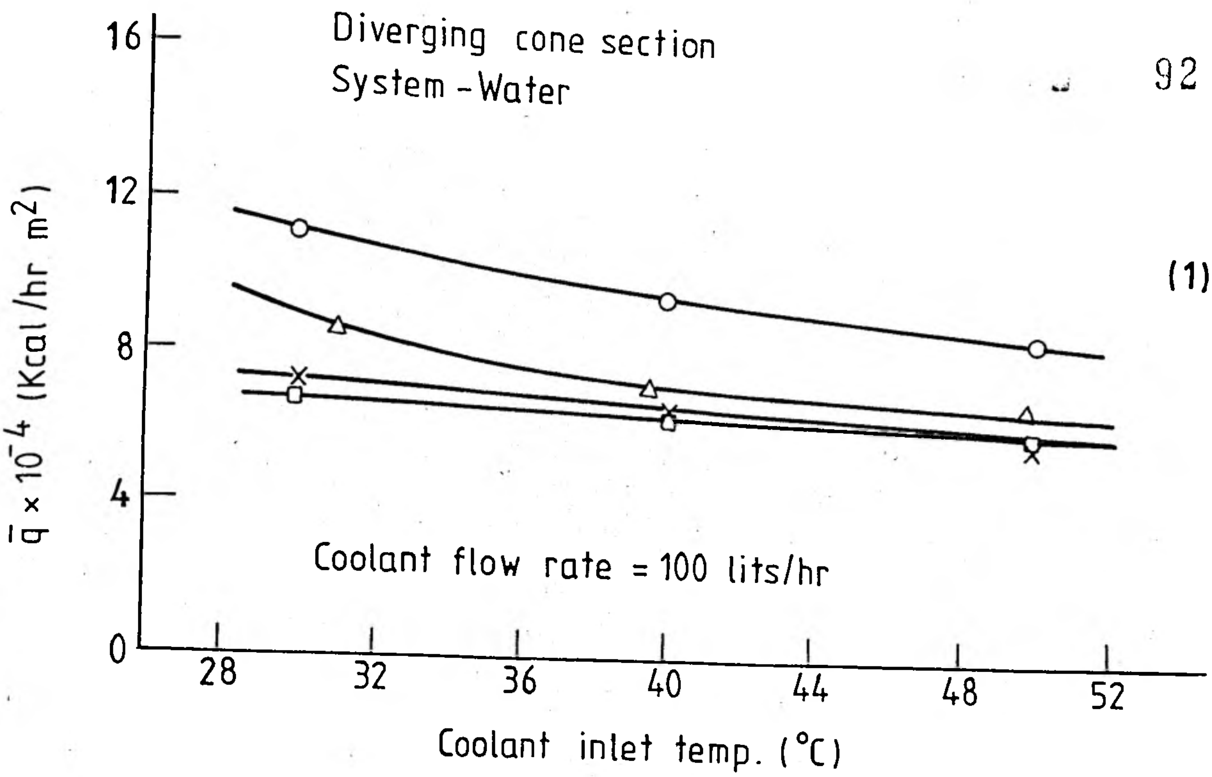


FIG.5.1. (17 & 18) EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (17) ETHYL ALCOHOL VAPOUR, (18) CARBON-TETRA-CHLORIDE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

accumulation of condensate, which causes increase in film thickness and reduction in heat transfer coefficient. In case of diverging-converging cone section, the total length is double the length of either diverging or converging section and as a natural consequence the film thickness will be higher. This is responsible for giving lower heat transfer coefficient in diverging-converging combined cone section compared to that obtained in either diverging or converging cone. The fact may be considered that the additional condensate formed in the diverging portion is also falling along the wall of the converging cone section thereby increasing the film thickness further. That is why average heat transfer coefficient value in case of diverging-converging cone section is the lowest.

5.2 Effect Of Coolant Inlet Temperature:

As already mentioned, experiments were carried out with three different coolant inlet temperatures viz. 30°, 40° and 50°C. Figure 5.2-1 to figure 5.2-6 show the effect of coolant inlet temperature on heat flux for condensation of water and ethyl-acetate vapours in diverging, converging and diverging-converging cone sections respectively. As expected, the heat flux gradually decreases with increase in coolant inlet temperatures. This is because as the inlet cooling water temperature increases, temperature difference across the wall reduces. As a result the amount of heat transferred to the coolant for a particular flow rate decreases. Consequently the rate of condensation also



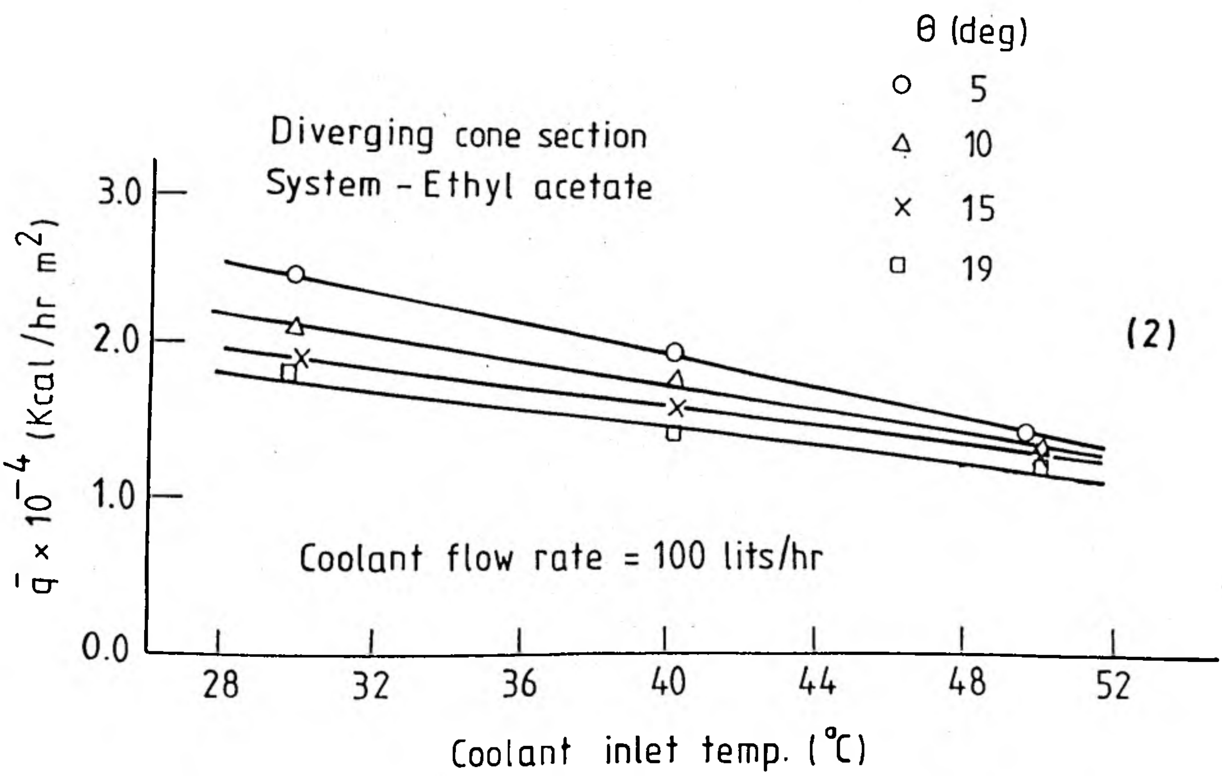
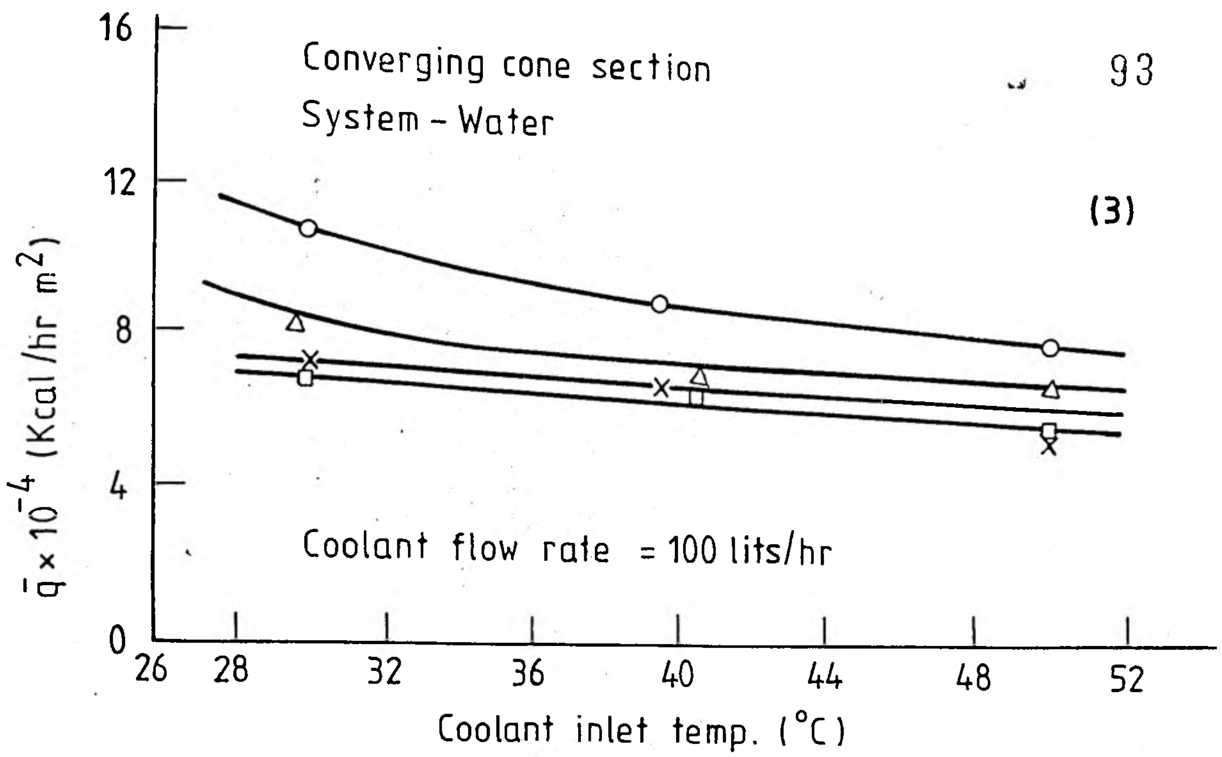


FIG.5.2. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT FLUX FOR CONDENSATION OF (1) WATER VAPOUR & (2) ETHYL-ACETATE VAPOUR IN DIVERGING CONE SECTIONS

(1 & 2)



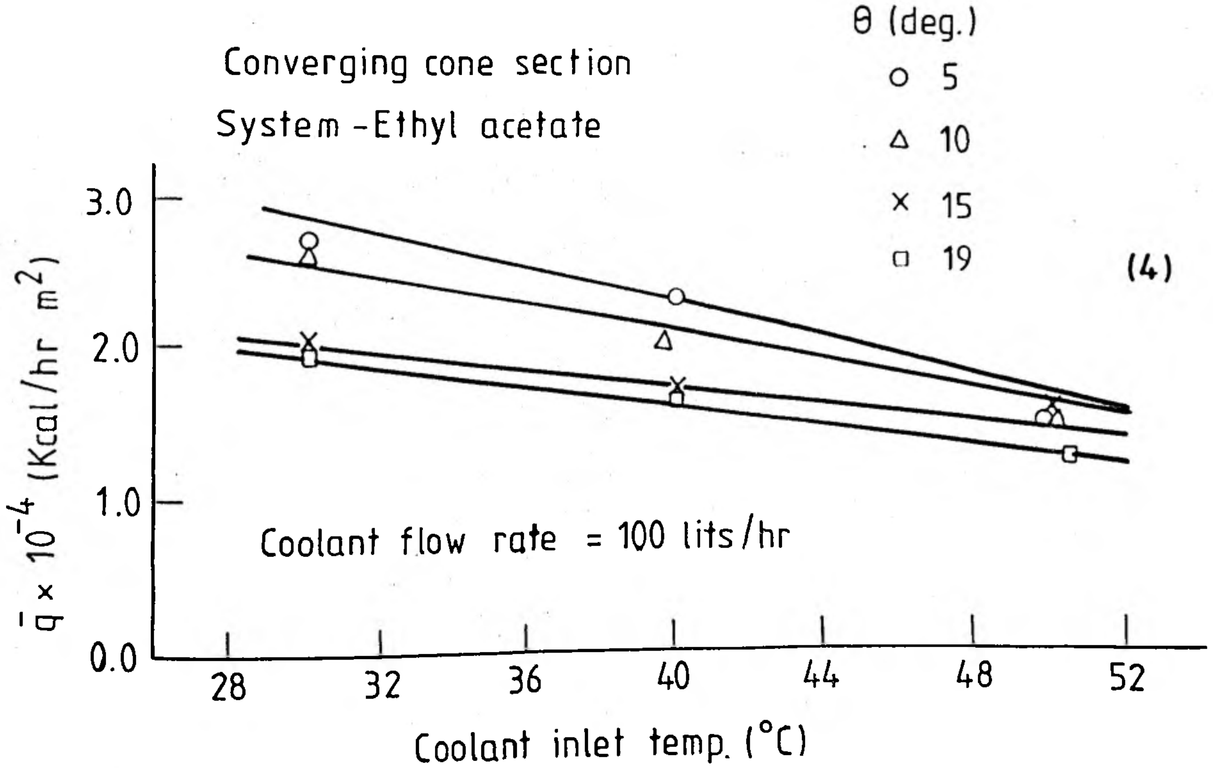
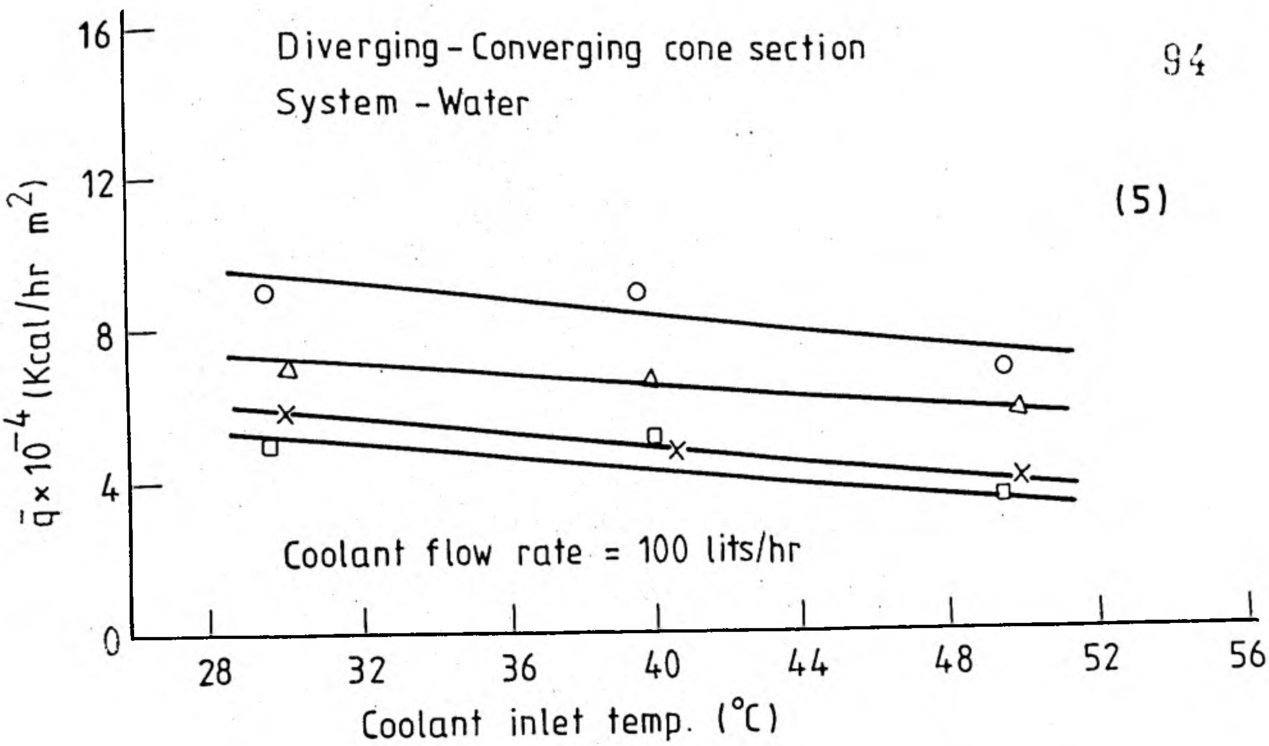


FIG.5.2. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT FLUX FOR CONDENSATION OF (3) WATER VAPOUR & (4) ETHYL-ACETATE VAPOUR IN CONVERGING CONE SECTIONS
(3&4)



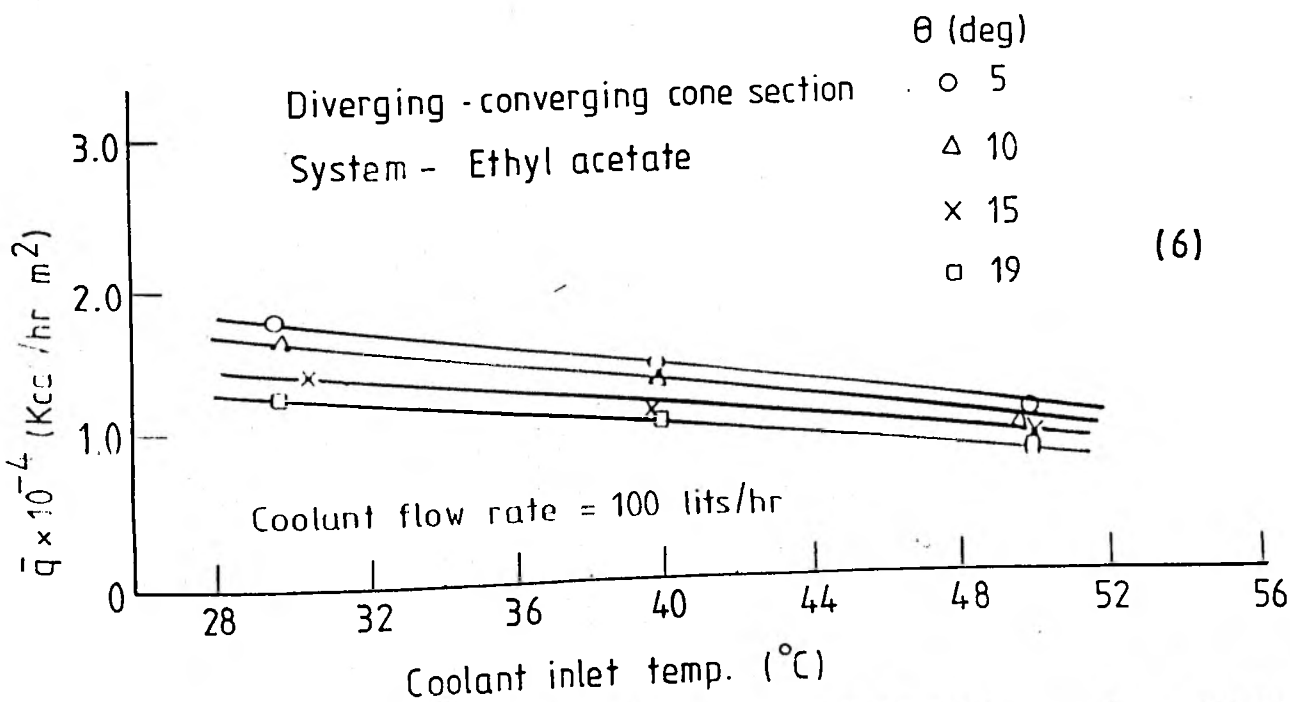
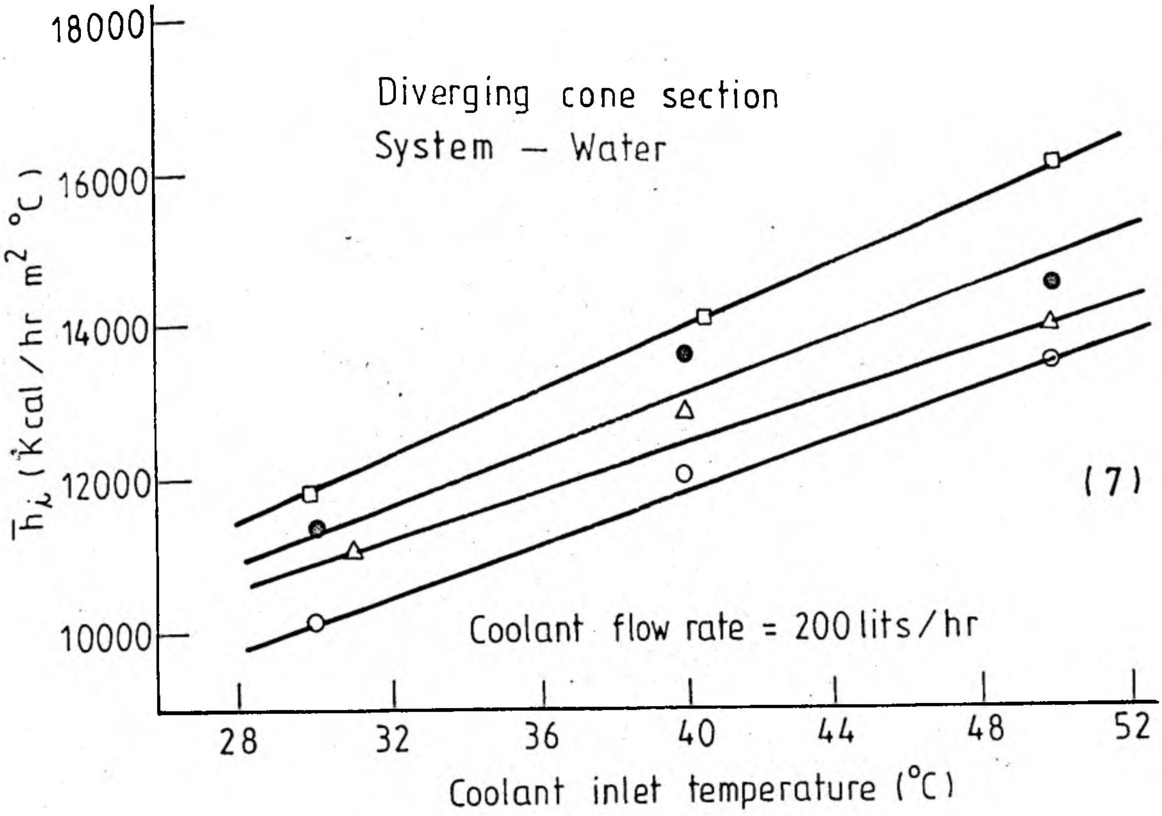


FIG. 5.2. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT FLUX FOR CONDENSATION OF (5) WATER VAPOUR & (6) ETHYL ACETATE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

(5&6)



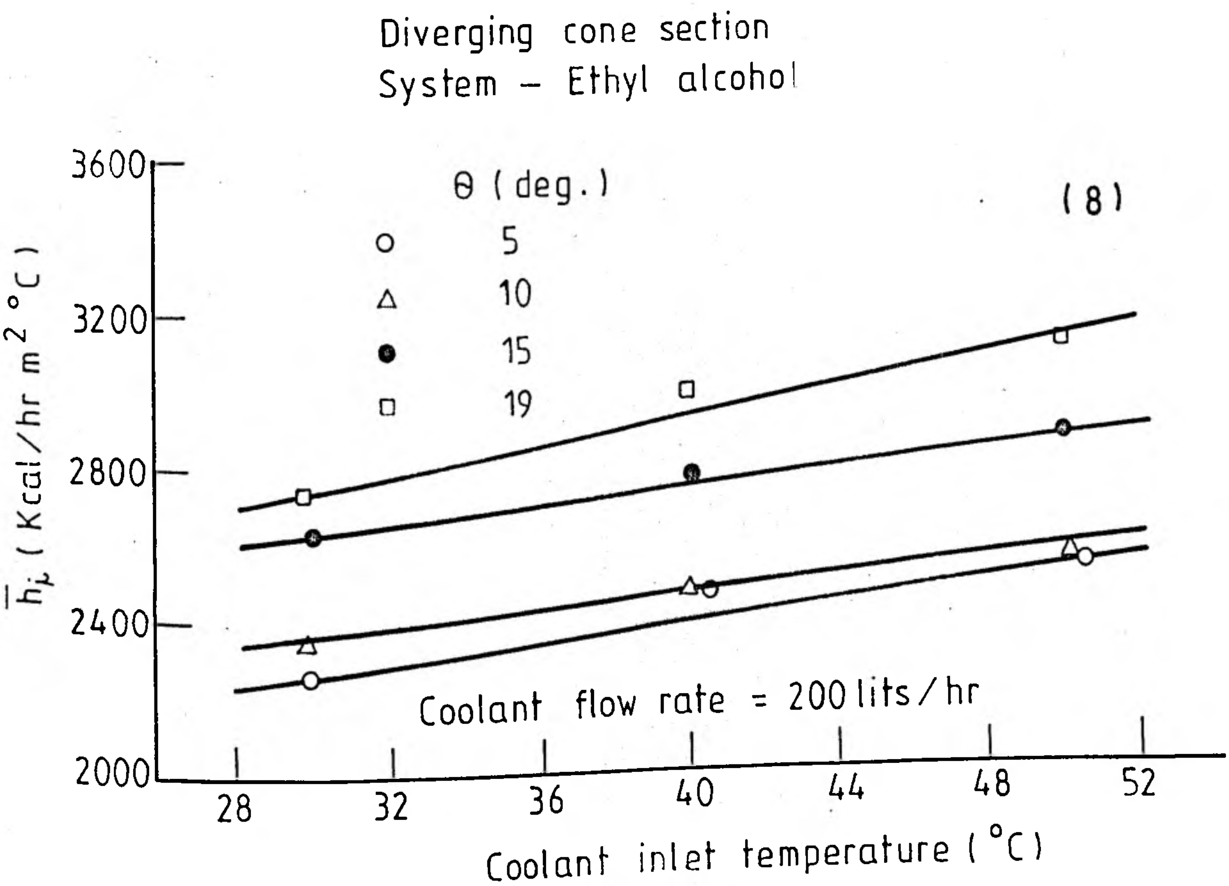
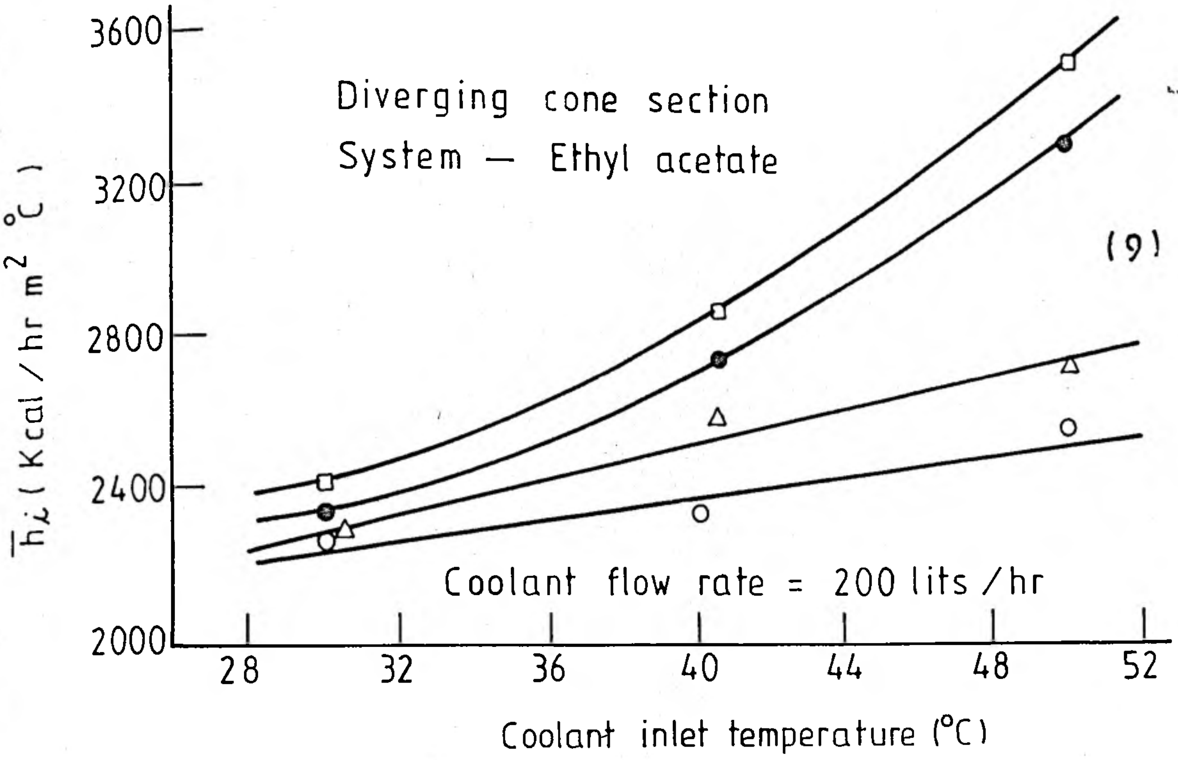


FIG.5.2. (7&8) EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (7) WATER VAPOUR, (8) ETHYL ALCOHOL VAPOUR IN DIVERGING CONE SECTIONS



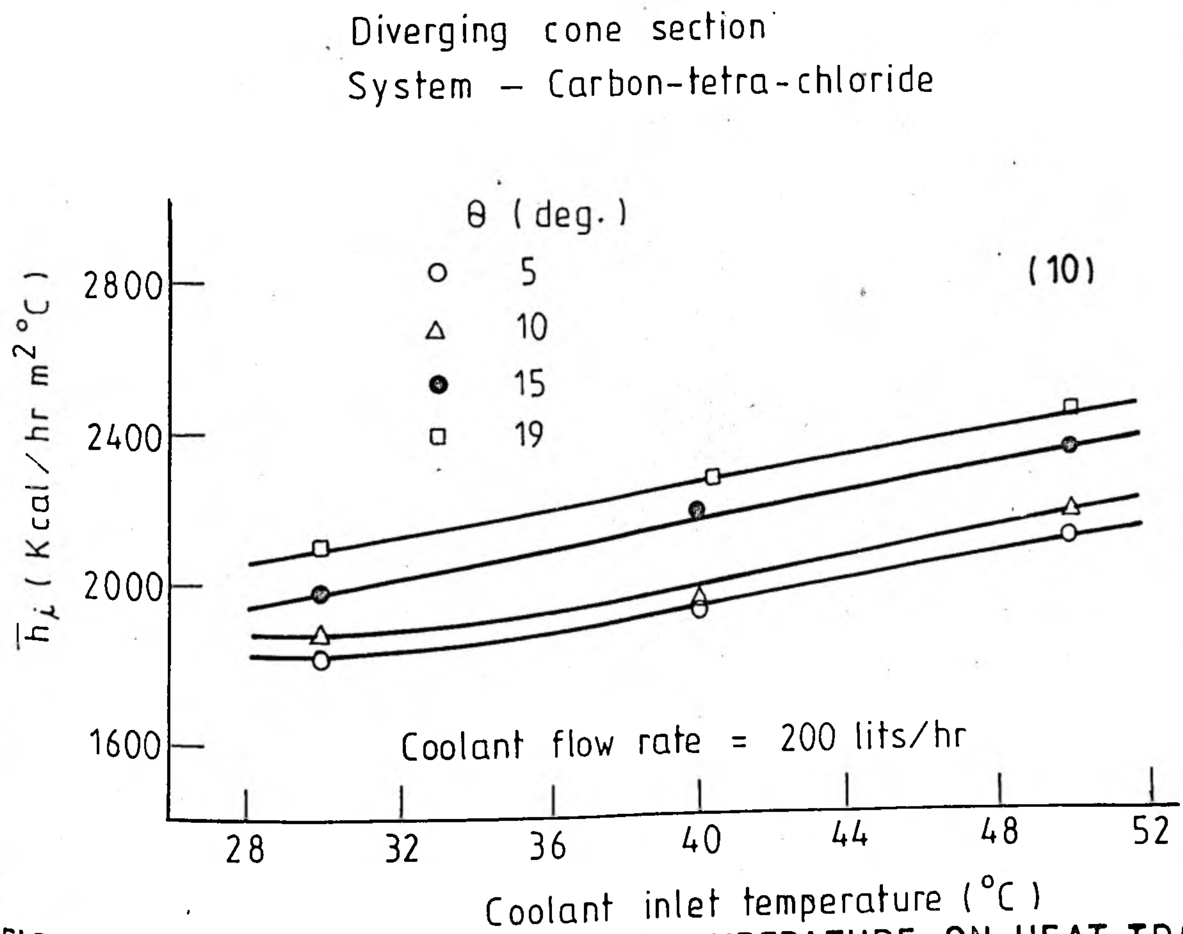
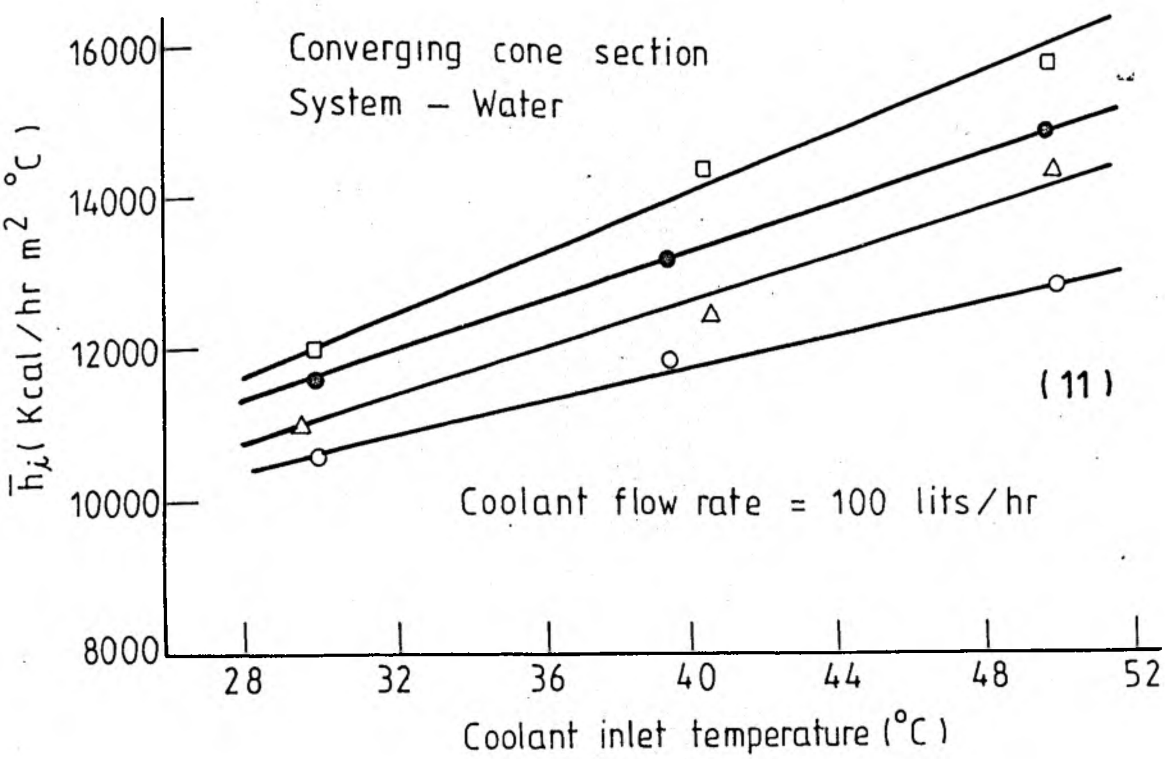


FIG.5.2. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (9) ETHYL ACETATE VAPOUR, (10) CARBON-TETRA-CHLORIDE VAPOUR IN DIVERGING CONE SECTIONS

(9 & 10)



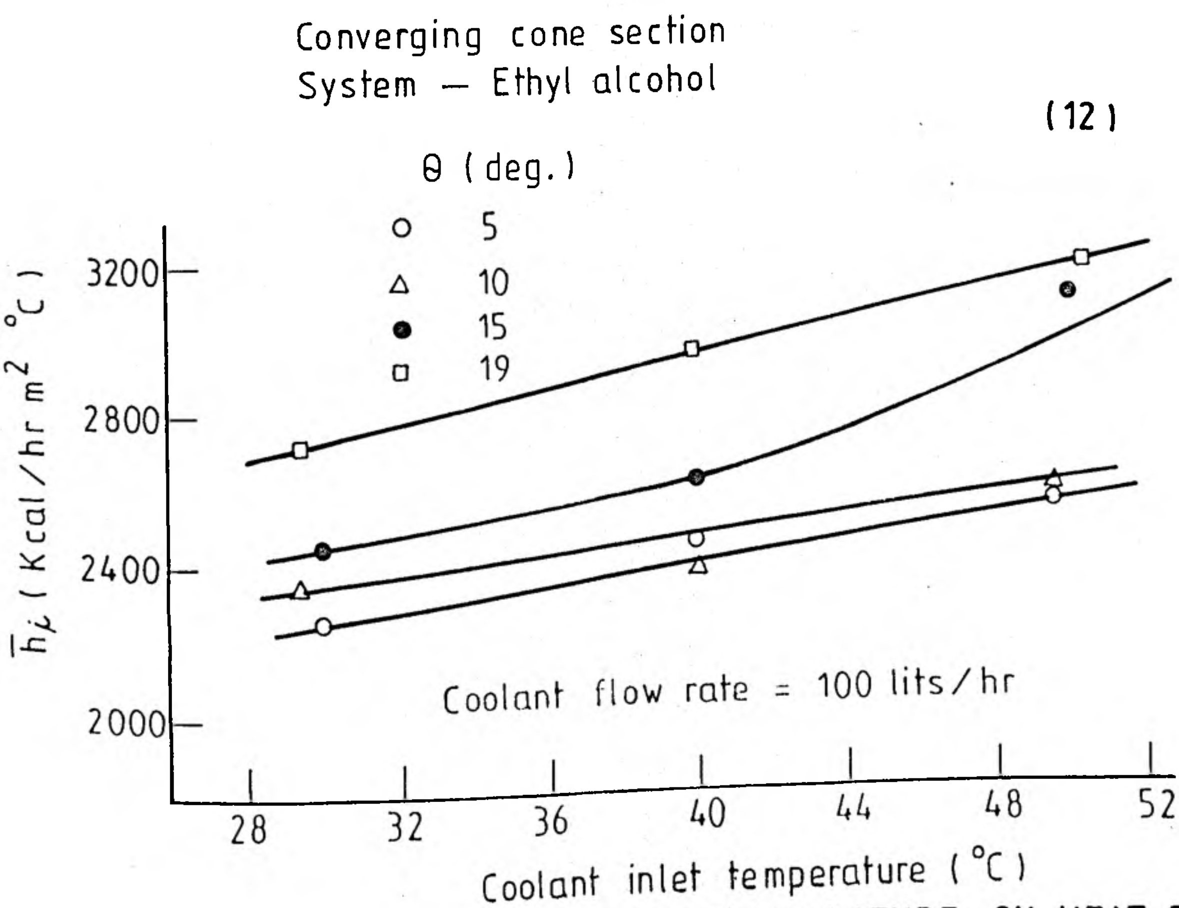
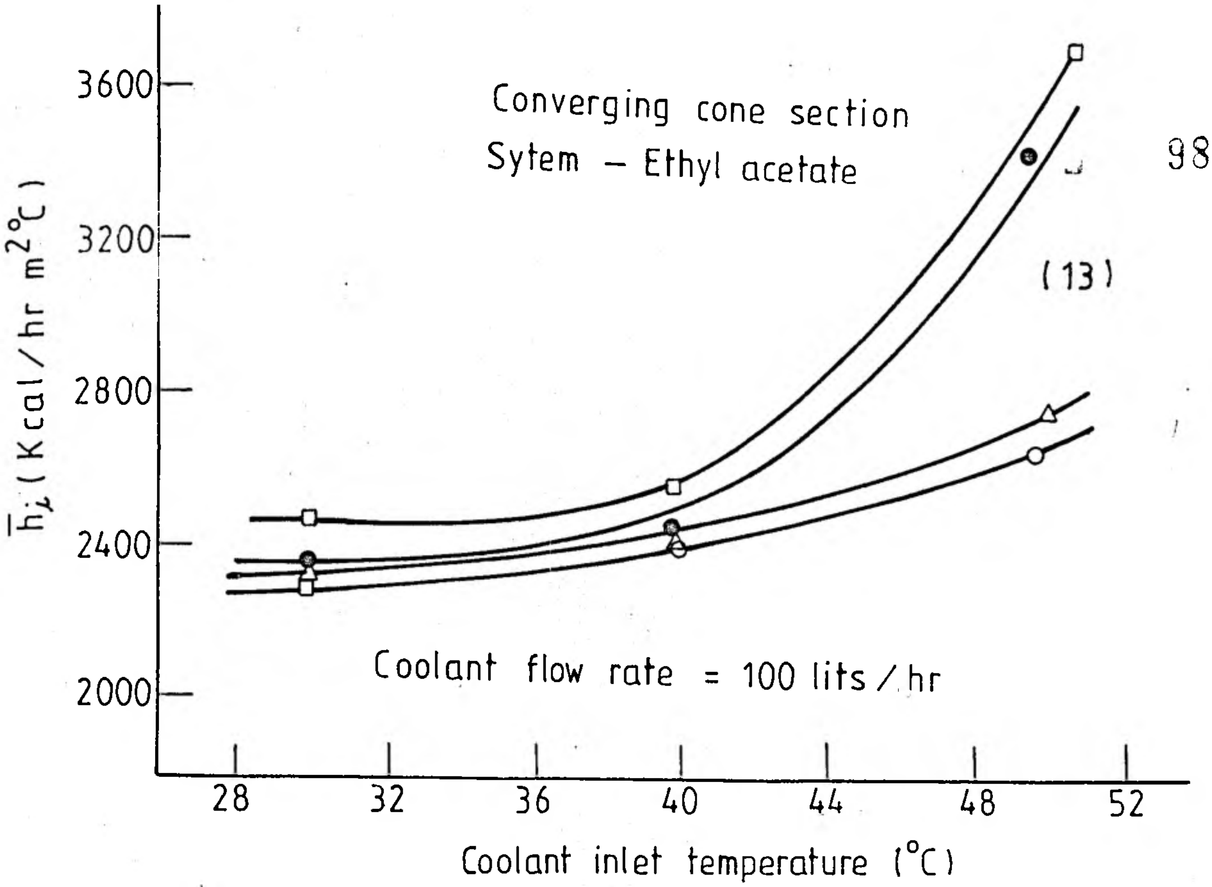


FIG.5.2. (11&12) EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (11) WATER VAPOUR, (12) ETHYL ALCOHOL VAPOUR IN CONVERGING CONE SECTIONS



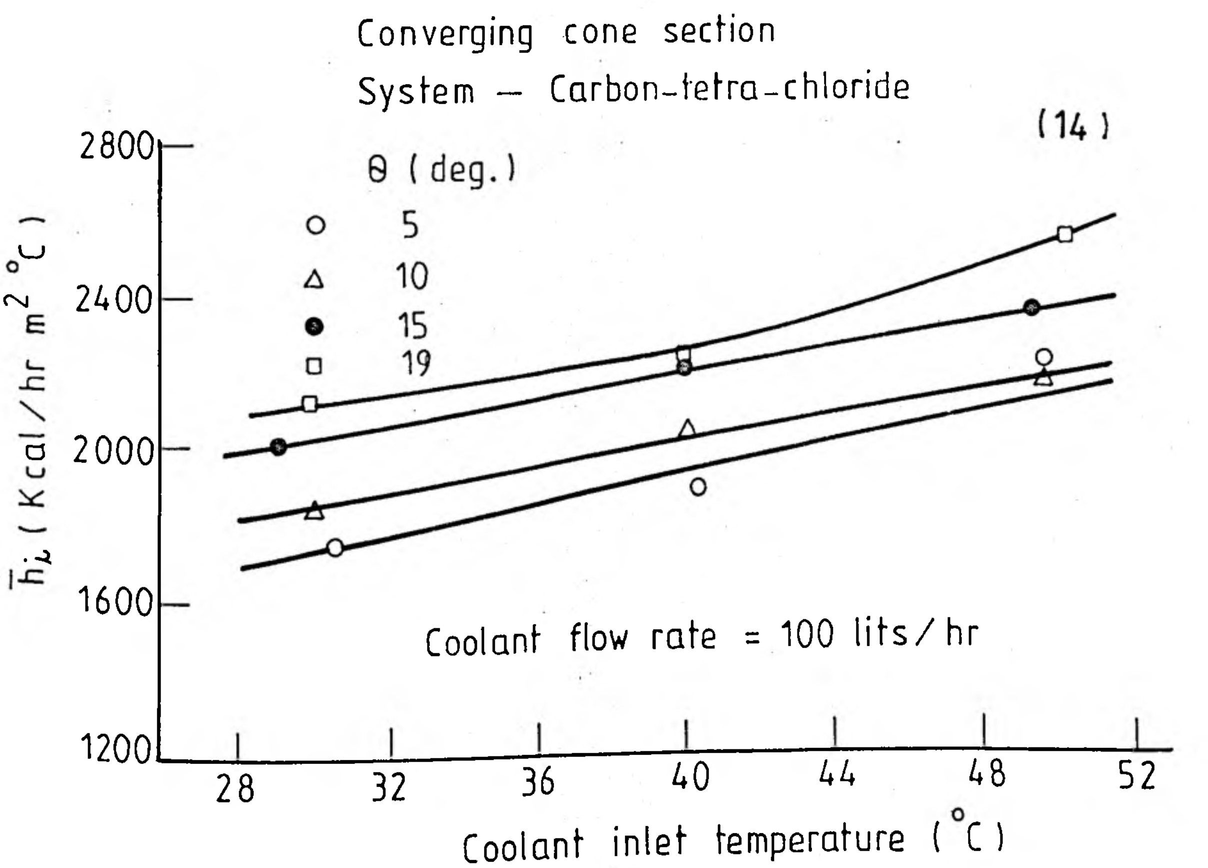


FIG.5.2. (13 & 14) EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (13) ETHYL ACETATE VAPOUR (14) CARBON-TETRA-CHLORIDE VAPOUR IN CONVERGING CONE SECTIONS

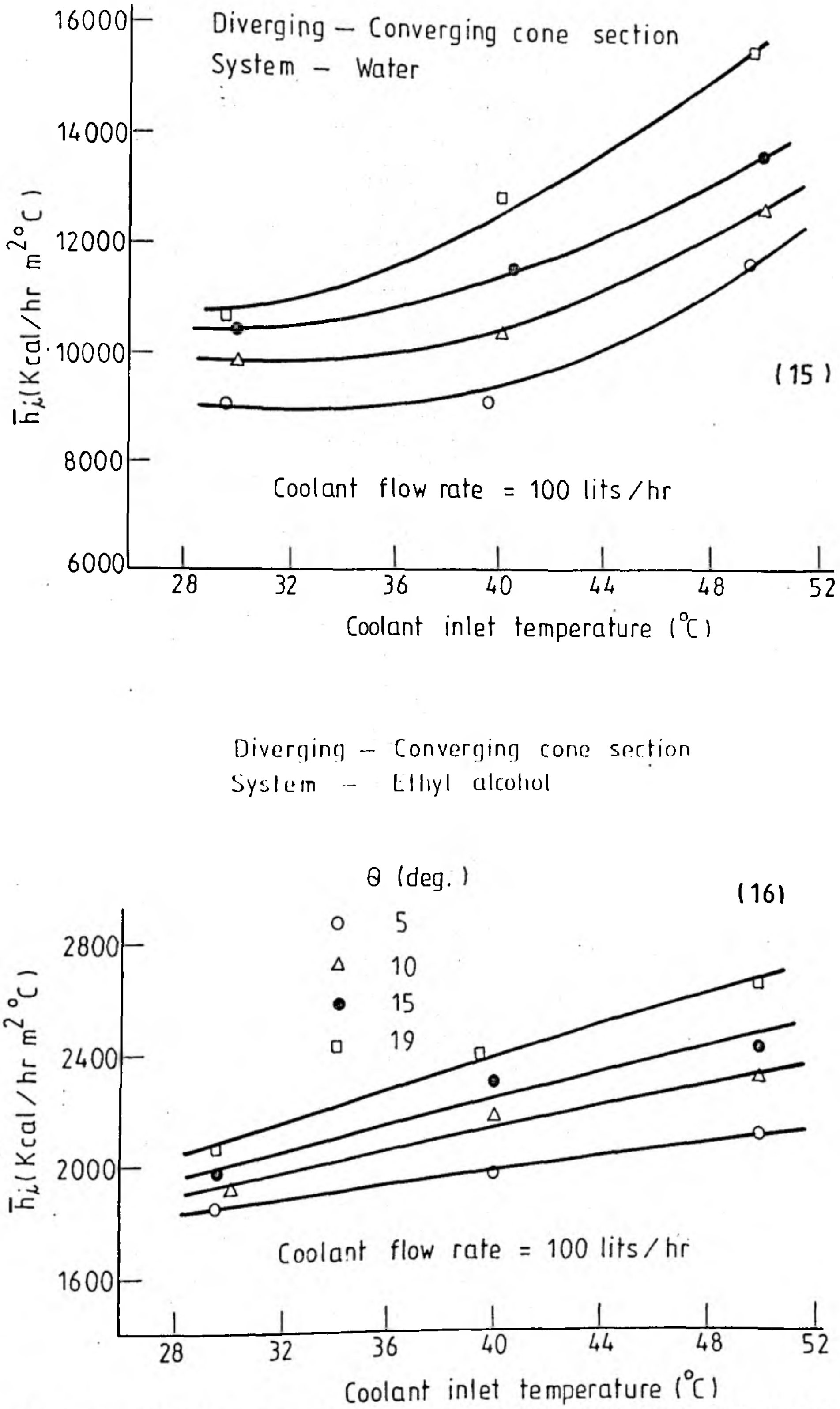
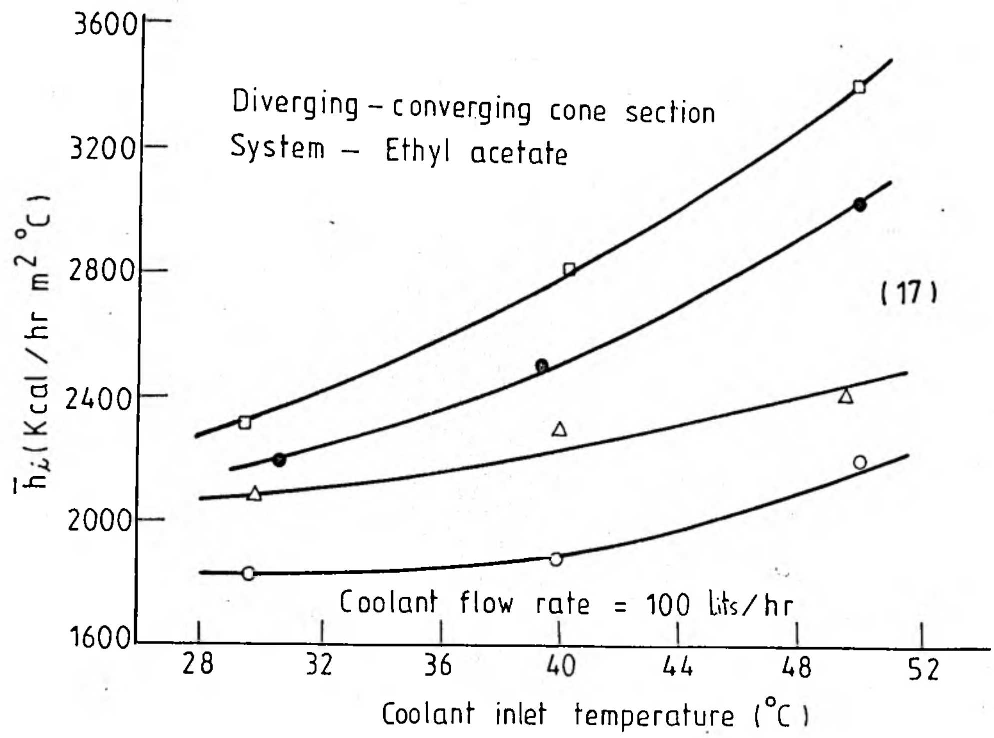


FIG.5.2. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (15) WATER VAPOUR (16) ETHYL ALCOHOL VAPOUR IN DIVERGING - CONVERGING CONE SECTIONS

(15 & 16)



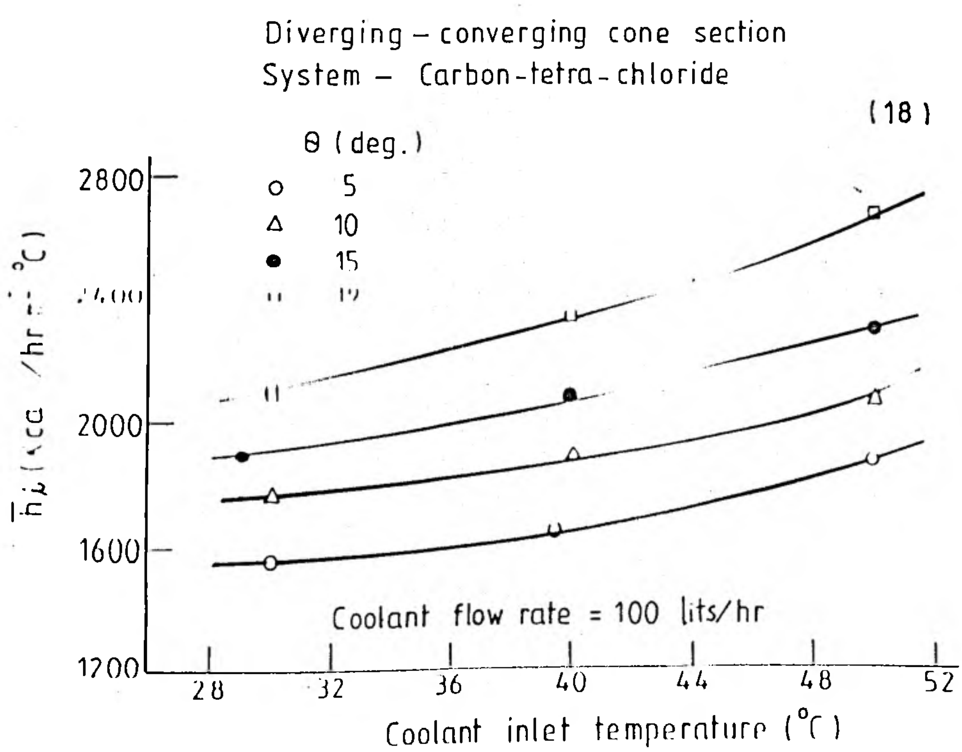


FIG.5.2. (17&18) EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (17) ETHYL ACETATE VAPOUR, (18) CARBON-TETRA-CHLORIDE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

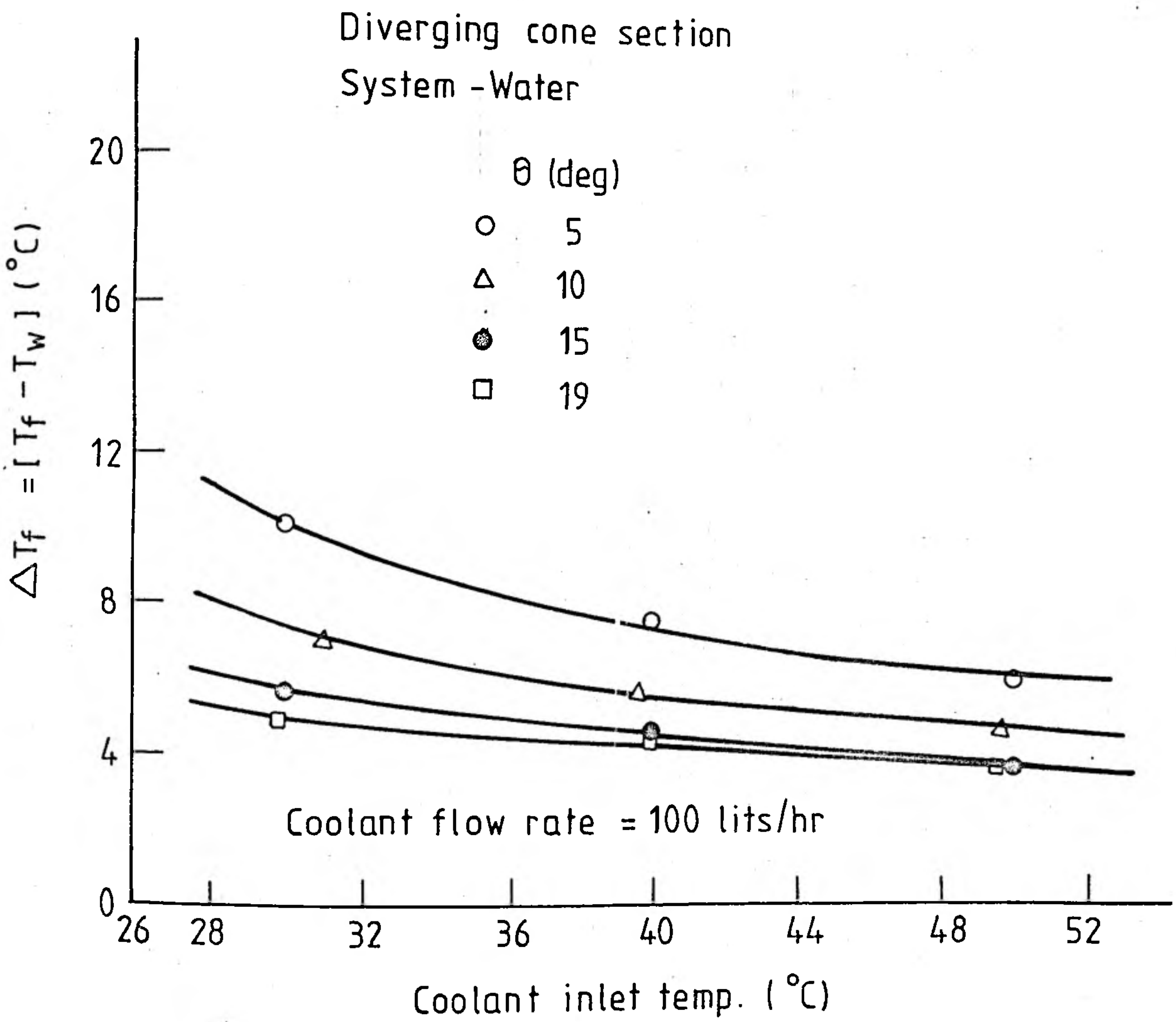


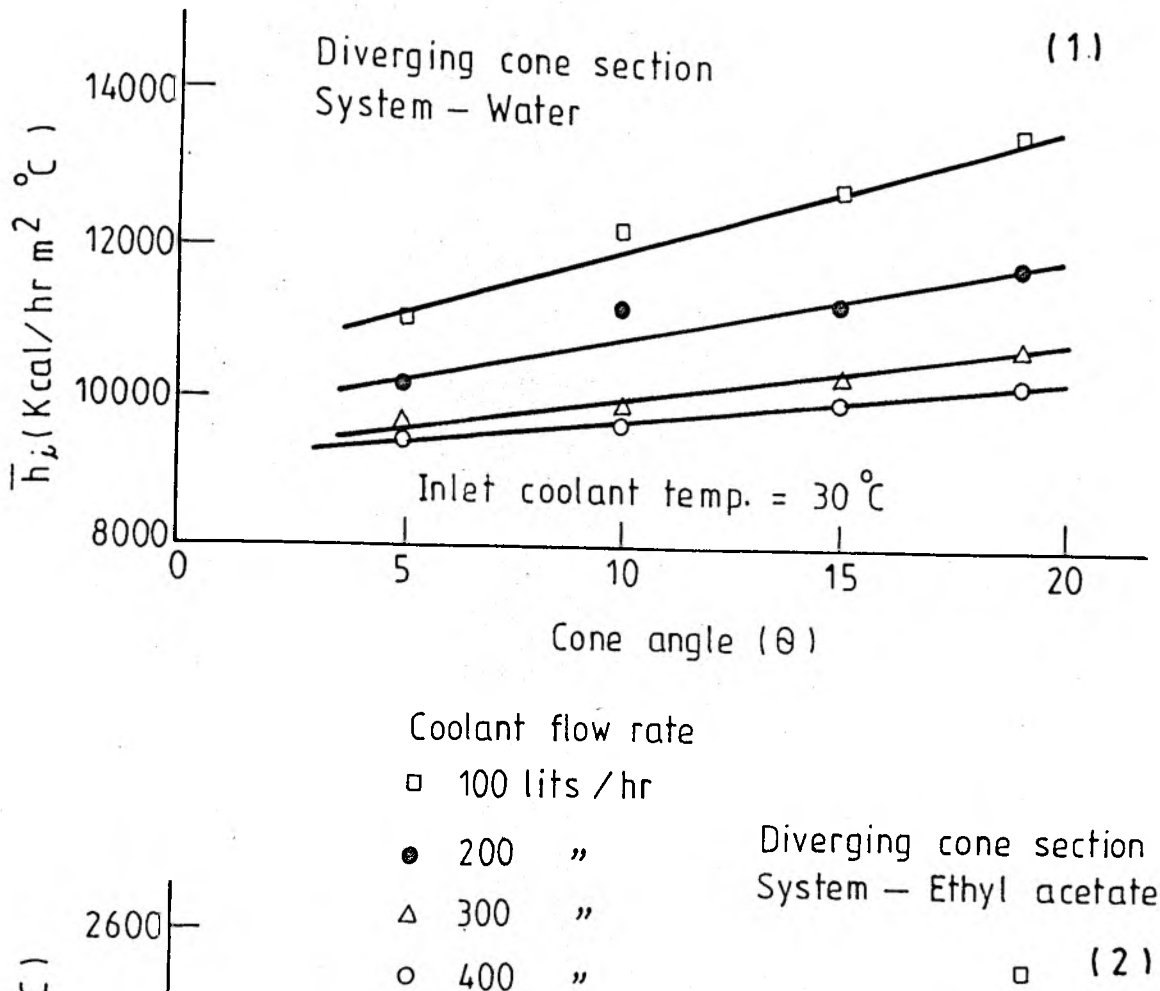
FIG. 5.2-19. EFFECT OF COOLANT INLET TEMPERATURE ON ΔT_f FOR CONDENSATION OF WATER VAPOUR IN DIVERGING CONE SECTIONS

reduces. In these plots the coolant flow rate is kept constant at 100 lits/hr. The effect of coolant flow rate on heat flux has already been discussed in the previous section. The effect of coolant inlet temperature on the condensation of ethyl alcohol and carbon-tetra-chloride vapours is found to be similar.

The effect of coolant inlet temperature on the average heat transfer coefficient has been shown in figure 5.2-7 to figure 5.2-18 for condensation of the water, ethyl alcohol, ethyl acetate and carbon-tetra-chloride vapours in diverging, converging and diverging-converging cone sections, for a constant coolant flow rate. With the increase in coolant inlet temperature, the average heat transfer coefficient also increases. The probable reason for this can be attributed to the fact that as the coolant inlet temperature increases, ΔT_f decreases (figure 5.2-19), which ultimately increases the average heat transfer coefficient. Further, with the increase of coolant inlet temperature at a particular flow rate, amount of condensate formation decreases. This is responsible for reduction in condensate film thickness vis-a-vis resistance to heat transfer. Therefore, at higher inlet temperature, increase in heat transfer coefficient is logical.

5.3 Effect Of Cone Angle:

To ascertain the influence of cone angle, θ, on the heat transfer characteristics, experiments were carried out with four different cone angles viz. θ = 5°, 10°, 15° and 19°.



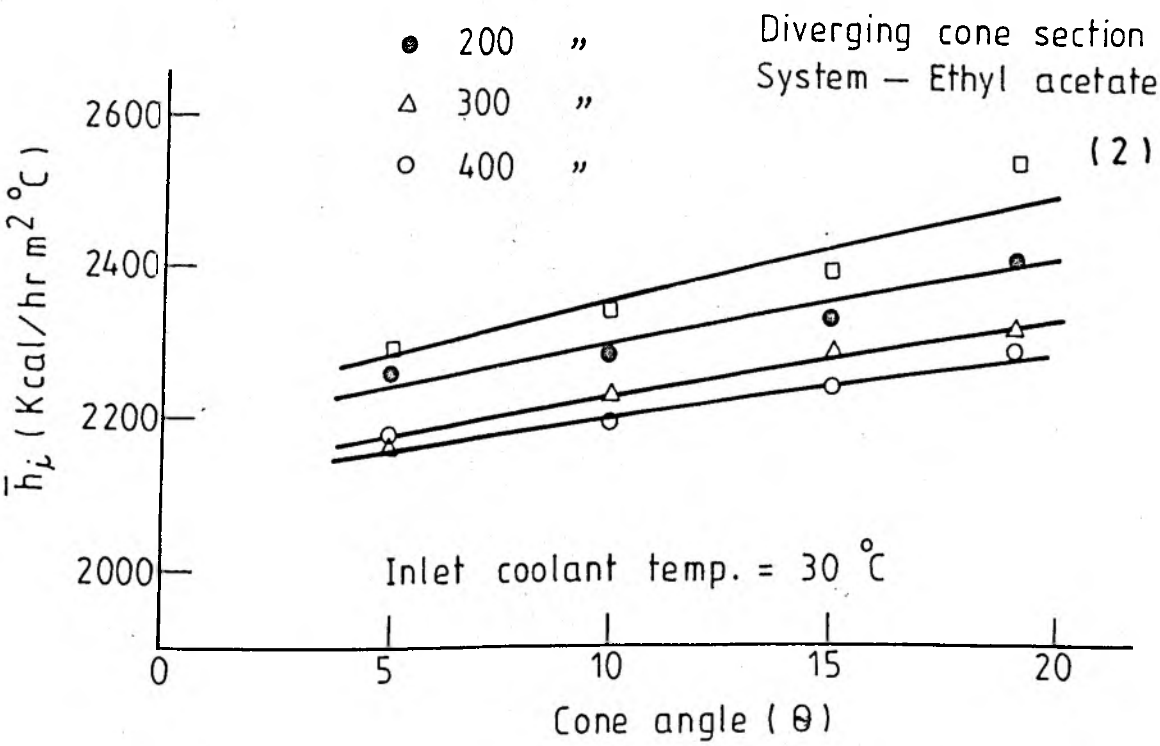


FIG.5.3. (1&2) EFFECT OF CONE ANGLE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (1) WATER VAPOUR, (2) ETHYL ACETATE VAPOUR IN DIVERGING CONE SECTIONS

The effect of cone angle on average heat transfer coefficient is shown in figure 5.3-1 and figure 5.3-2, for condensation of water and ethyl acetate vapours in diverging cone sections, for different coolant flow rates e.g. 100,200,300 and 400 Lits/hr. These plots are for coolant inlet temperature = 30°C. It will be seen that average heat transfer coefficient increases with increase in cone angle, θ. For a particular cone angle, lower coolant flow rate gives higher average heat transfer coefficient.

Converging and diverging-converging cone sections behave in a similar fashion and this has been shown in figures 5.3-3 to 5.3-6 respectively. Again magnitudewise diverging cone section gives maximum heat transfer coefficient and it becomes minimum in case of diverging-converging combined cone sections under the same experimental conditions.

Thus it is seen that as cone angle increases the average condensing film heat transfer coefficient also increases, but it should be noted that as the cone angle increases, pressure drop in the annulus becomes larger than that in the equivalent straight tube [150] . This, therefore, brings forth the conclusion that though the average heat transfer coefficient increases with increase in cone angle, an optimum value of cone angle could be chosen at which the heat transfer coefficient is appreciably large, but at the sametime pressure drop penalty is kept within reasonable limits.

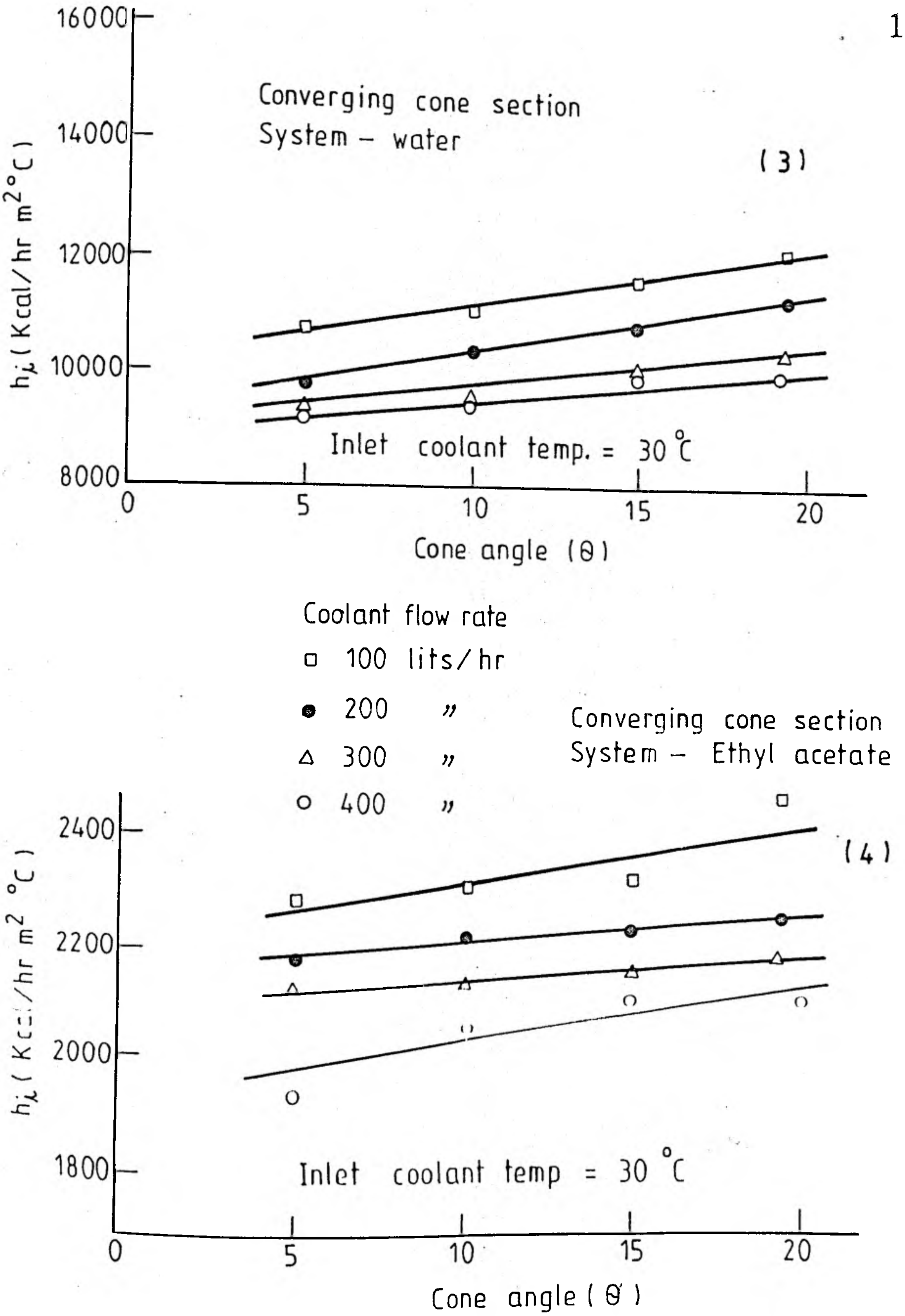
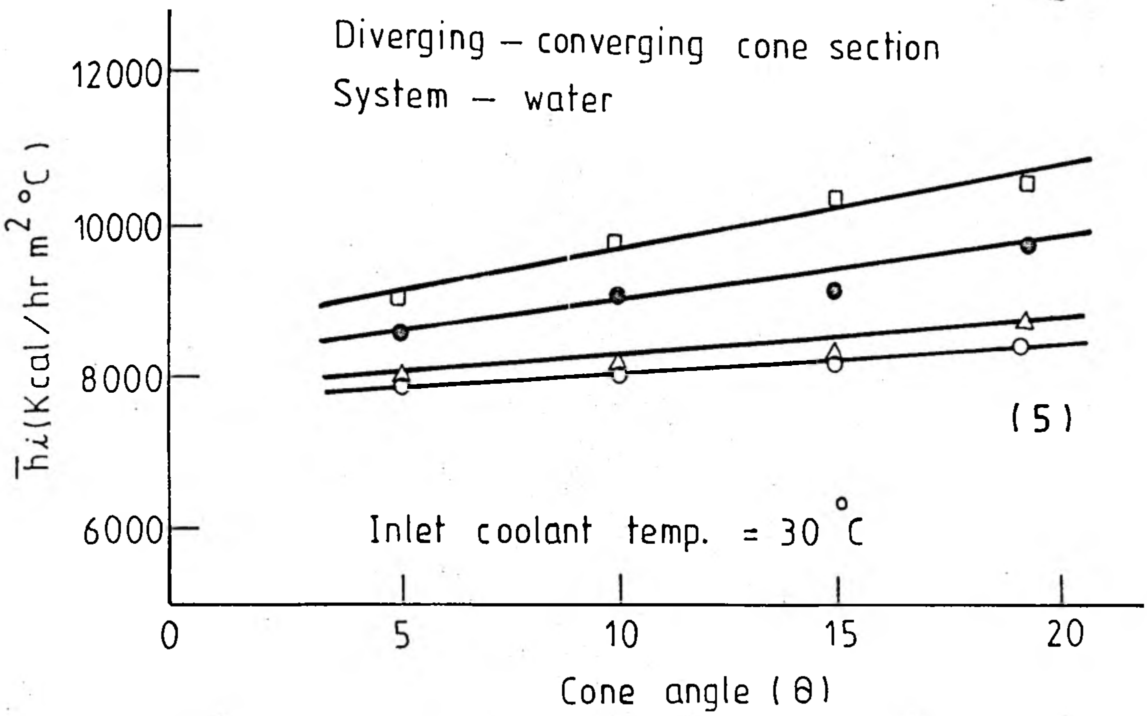


FIG. 5. 3. (3&4) EFFECT OF CONE ANGLE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (3) WATER VAPOUR, (4) ETHYL ACETATE VAPOUR IN CONVERGING CONE SECTIONS



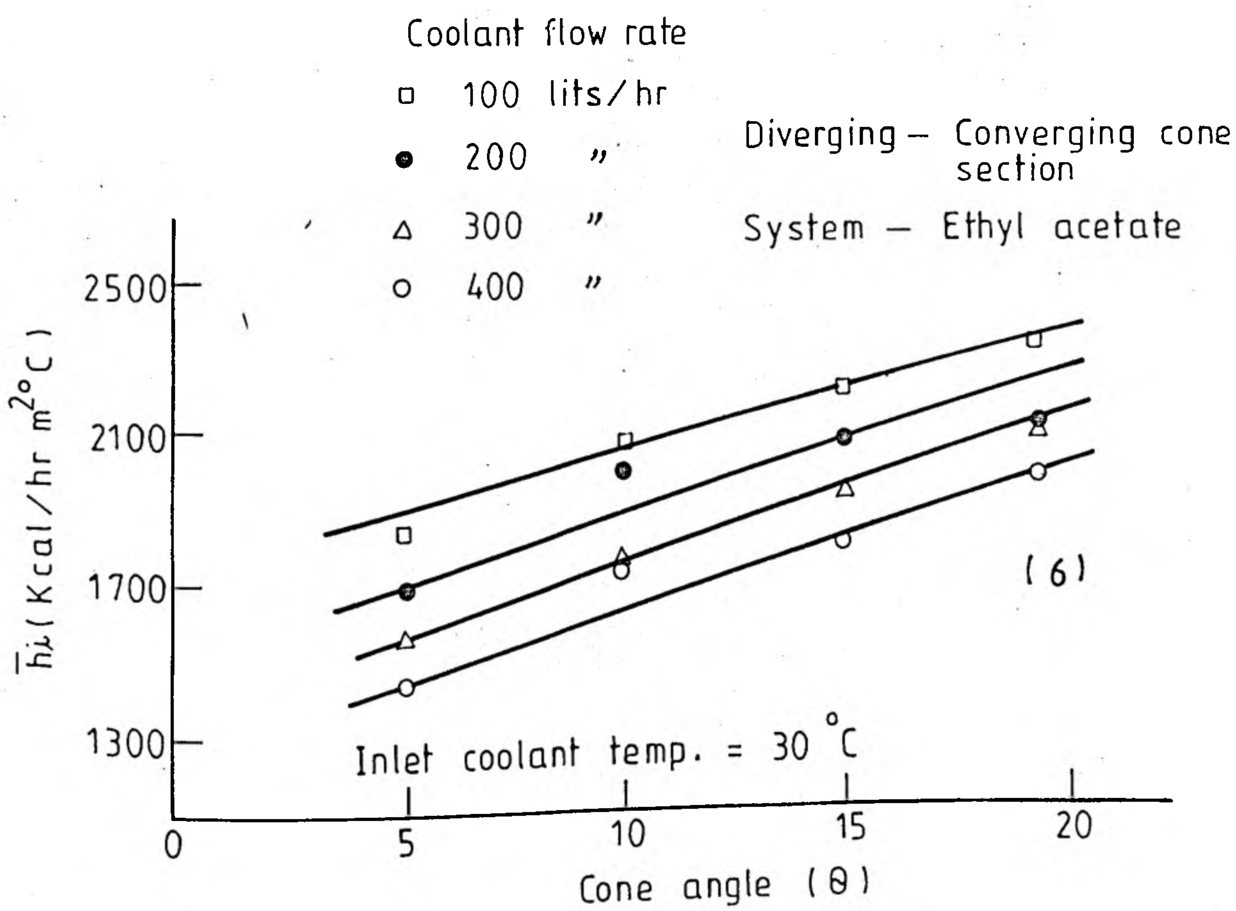


FIG.5.3. (5&6) EFFECT OF CONE ANGLE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (5) WATER VAPOUR, (6) ETHYL ACETATE VAPOUR IN DIVERGING - CONVERGING CONE SECTION

In case of heat flux, $\bar{q}$, it gradually decreases with increase in cone angle, as it is evident from figure 5.3-7 and figure 5.3-8, which are for condensation of water and carbon-tetra-chloride vapours in diverging cone sections. The coolant inlet temperature is 30°C. Variation of coolant flow rates has also been shown in these figures.

Although the heat flux decreases with increase in cone angle, the rate of heat transfer, Q, as can be seen from the heat transfer data [Appendix-I. (Table A1-1 to A1-36)], increases with increase in cone angle for all coolant flow rates and coolant inlet temperatures. The average heat transfer coefficient increases with increase in cone angle, 0, as already shown in figure 5.3-1 to figure 5.3-6.

The temperature difference (ΔT_f) between film and condenser wall plays an important role in increasing the heat transfer coefficient values. ΔT_f decreases with increase in cone angles for a particular coolant flow rate and increases with coolant flow rate for a particular cone angle, θ, (Figure 5.8-3). This is true both for converging and diverging-converging cone sections also. The effect of ΔT_f on the heat transfer characteristics will be discussed in the next section.

The behaviour of converging and diverging-converging cone sections, when cone angles are changed, is found to be similar to that of diverging cone section. This can be seen from figure 5.3-9 to figure 5.3-12, for condensation of water and ethyl alcohol vapours and for coolant inlet temperature 30°C.

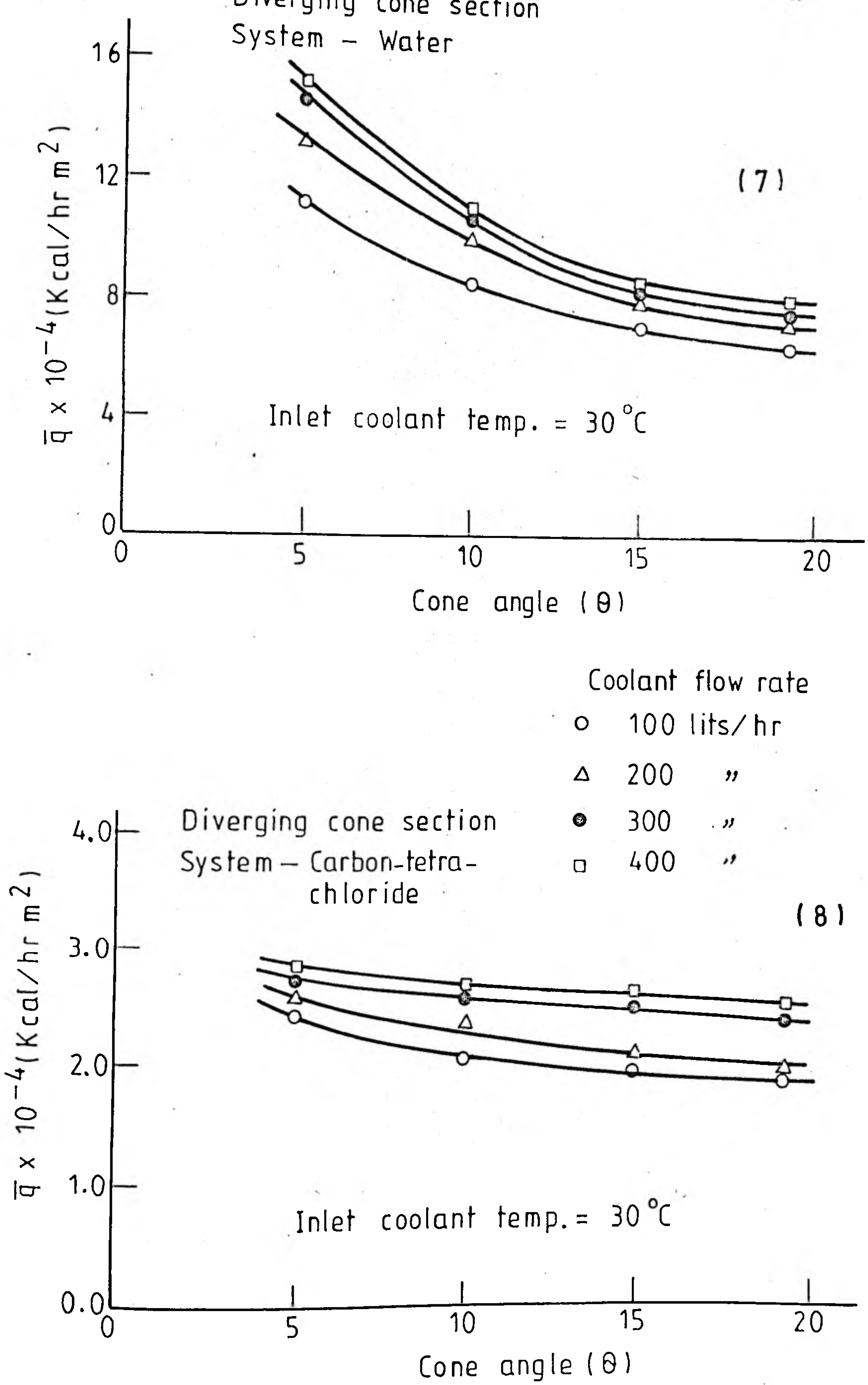
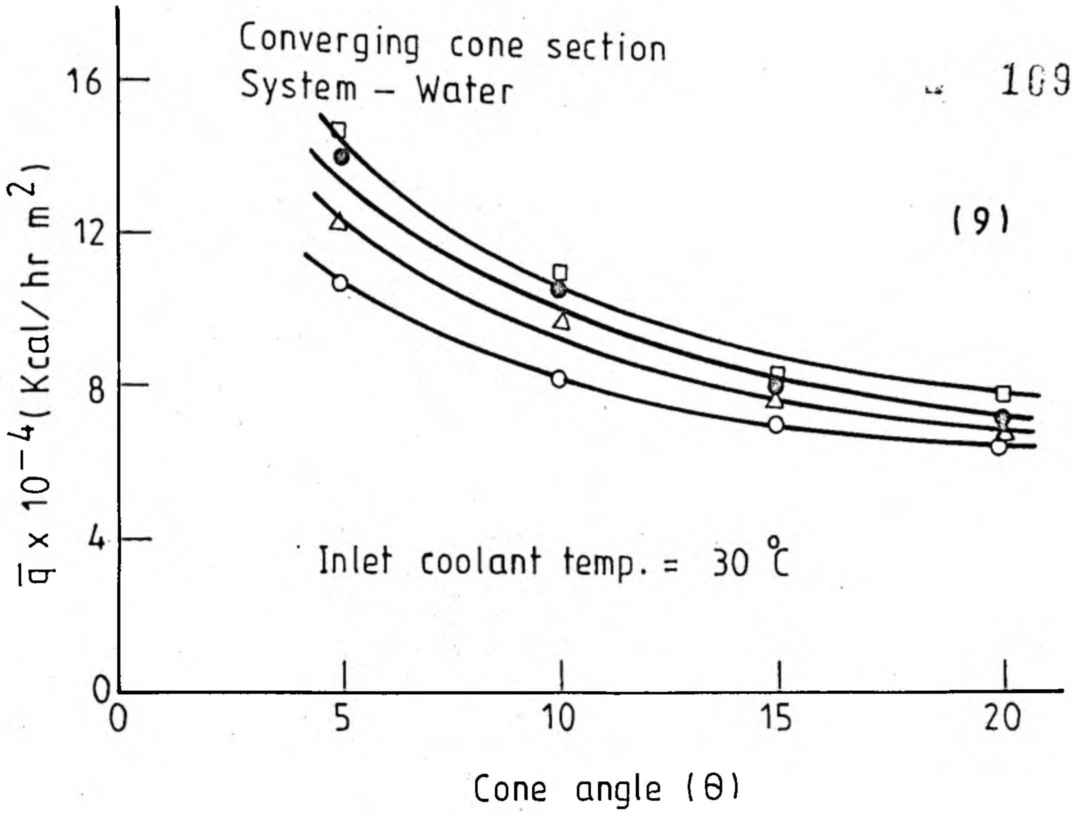


FIG. 5.3. (7&8) EFFECT OF CONE ANGLE ON HEAT FLUX FOR CONDENSATION OF (7) WATER VAPOUR, (8) CARBON-TETRA-CHLORIDE VAPOUR IN DIVERGING CONE SECTIONS



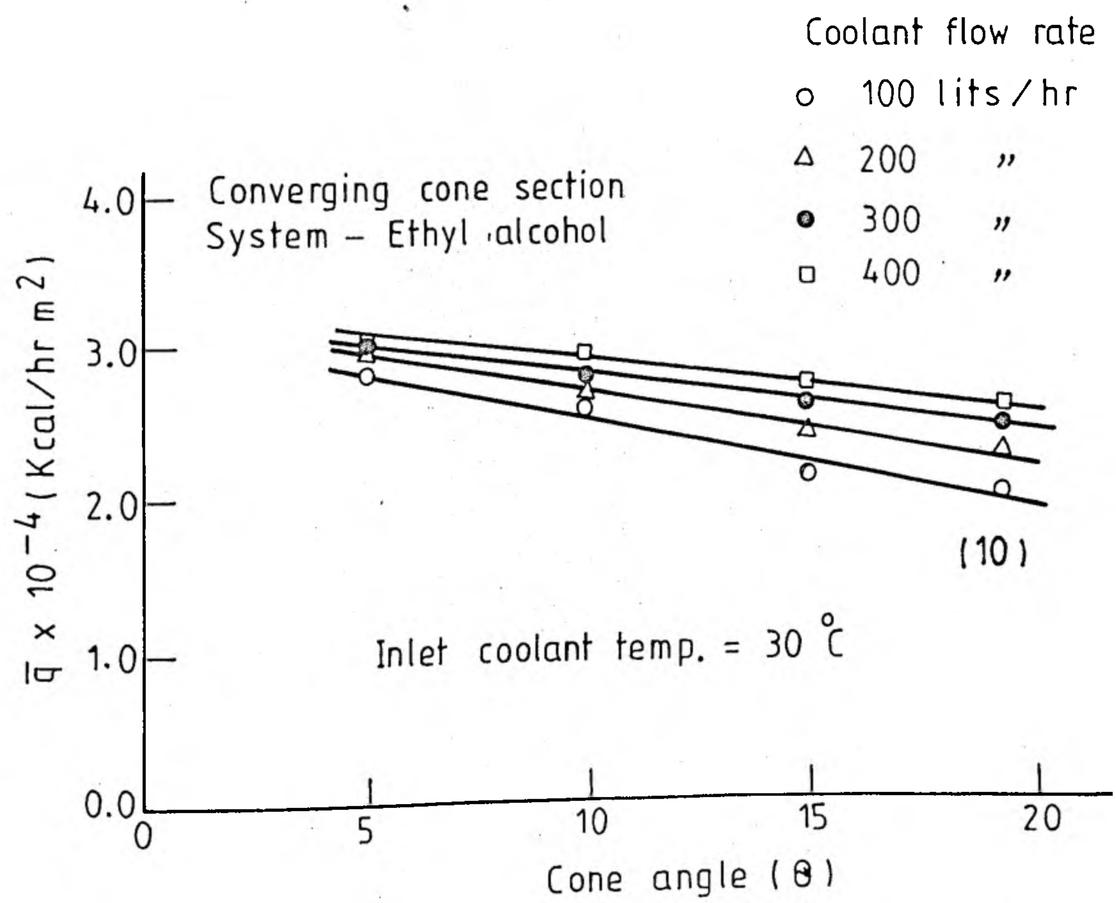
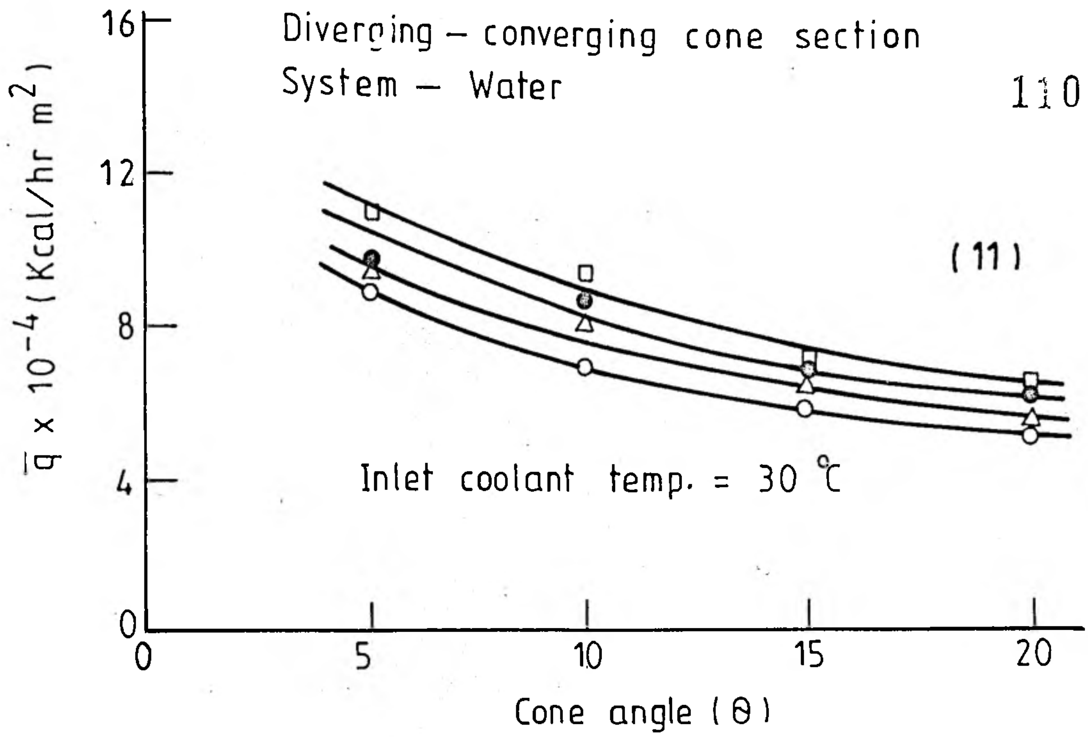


FIG.5.3. (9 & 10) EFFECT OF CONE ANGLE ON HEAT FLUX FOR CONDENSATION OF (9) WATER VAPOUR, (10) ETHYL ALCOHOL VAPOUR IN CONVERGING CONE SECTIONS



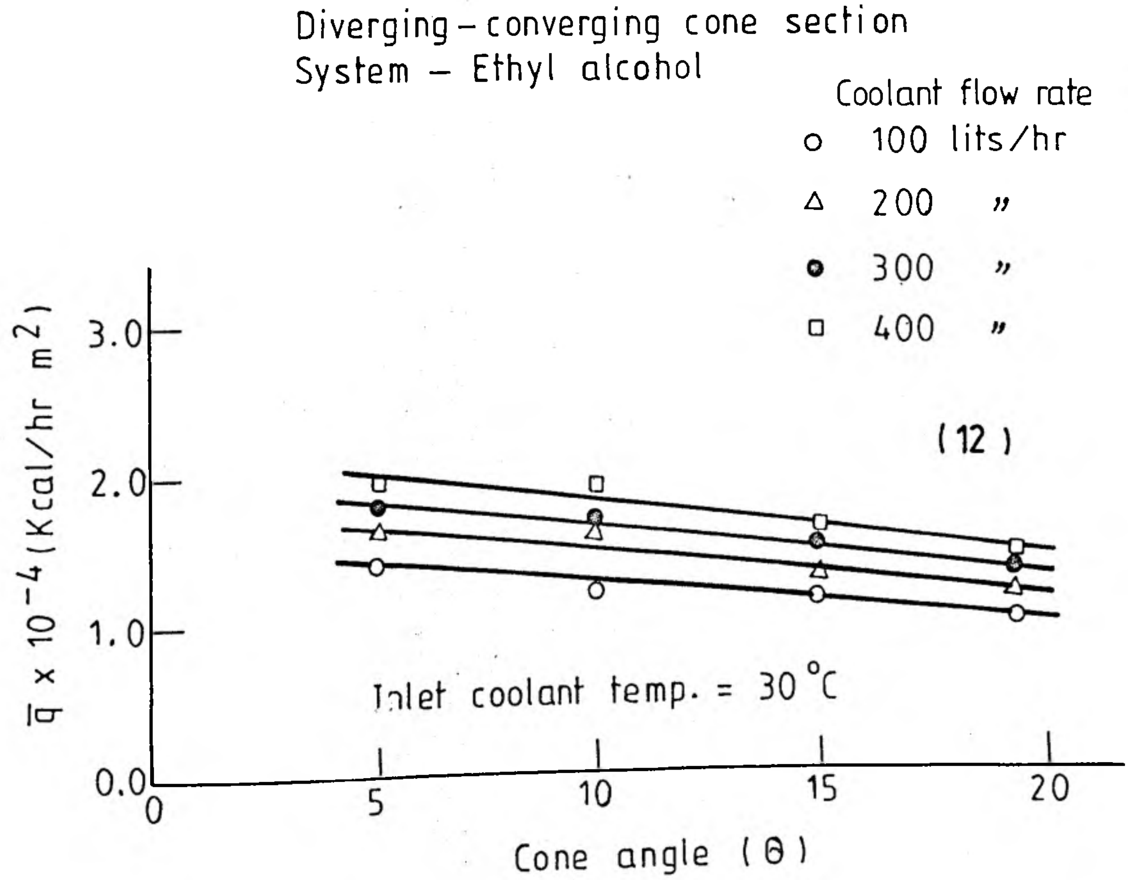


FIG.5.3. (11 & 12) EFFECT OF CONE ANGLE ON HEAT FLUX FOR CONDENSATION OF (11) WATER VAPOUR, (12) ETHYL ALCOHOL VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

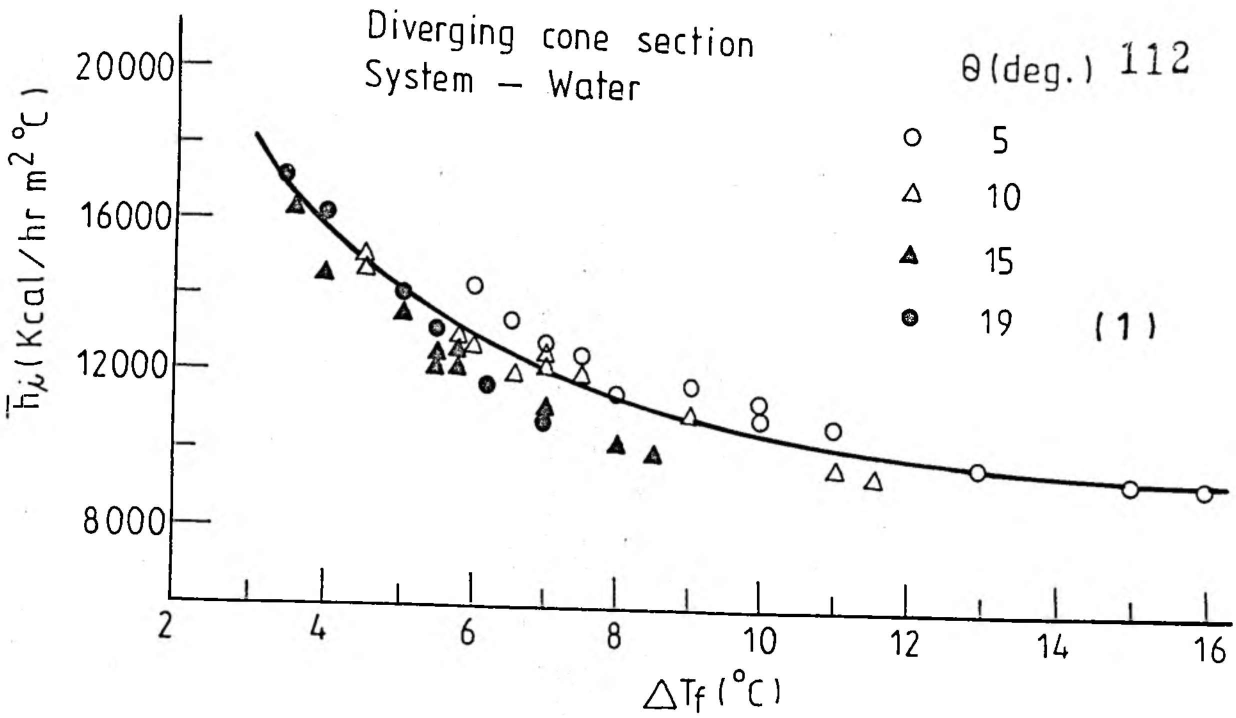
Here also the only difference is in the magnitude of heat flux values, which is maximum in case of diverging cones and minimum in case of diverging-converging cones.

5.4 Effect Of ΔT_f :

As already discussed in Chapter-4, wall temperatures of the condensing surface were measured by means of copper-constantan thermocouple wires and ΔT_f is the temperature difference between wall and film. Temperatures measured at different positions along the height of the condenser test section were found to be constant for a particular set of experiments. Wall temperatures of all the test sections and experiments have been given in Tables A1-1 to A1-36 of Appendix-I.

Figures 5.4-1 to 5.4-12 show the effect of ΔT_f on average heat transfer coefficient for condensation of water, ethyl alcohol, ethyl acetate and carbon-tetra-chloride vapours in diverging, converging and diverging-converging cone sections respectively. As expected, the average heat transfer coefficient values decrease with increase in ΔT_f. It also demonstrates the sensitivity of ΔT_f in determining average heat transfer coefficient. From plots it is evident that the effect is quite prominent in the lower range of ΔT_f values.

Figures 5.4-13 to 5.4-15 show the effect of ΔT_f on heat flux for condensation of water and organic vapours in diverging, converging and diverging-converging cone sections. Two separate lines have been obtained for water and organic vapours. It is



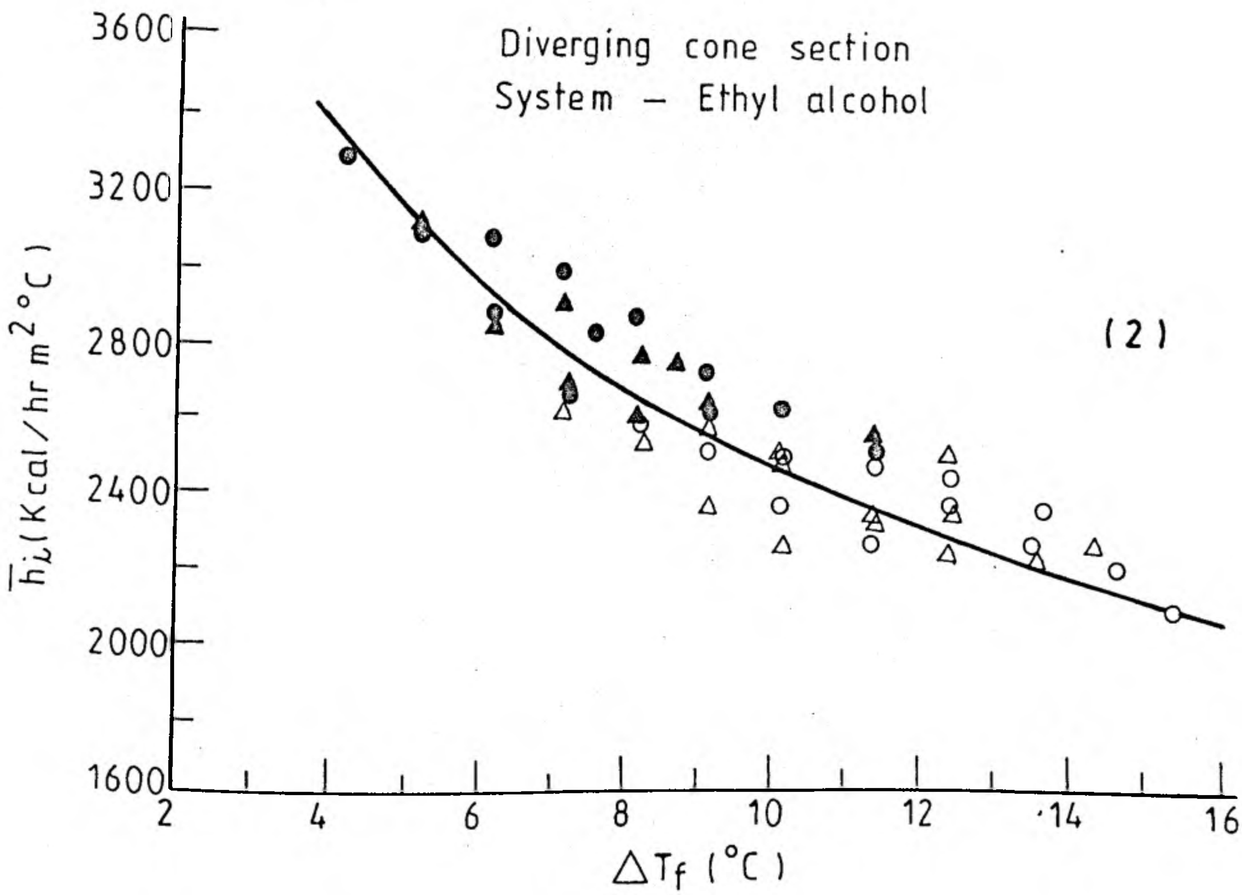
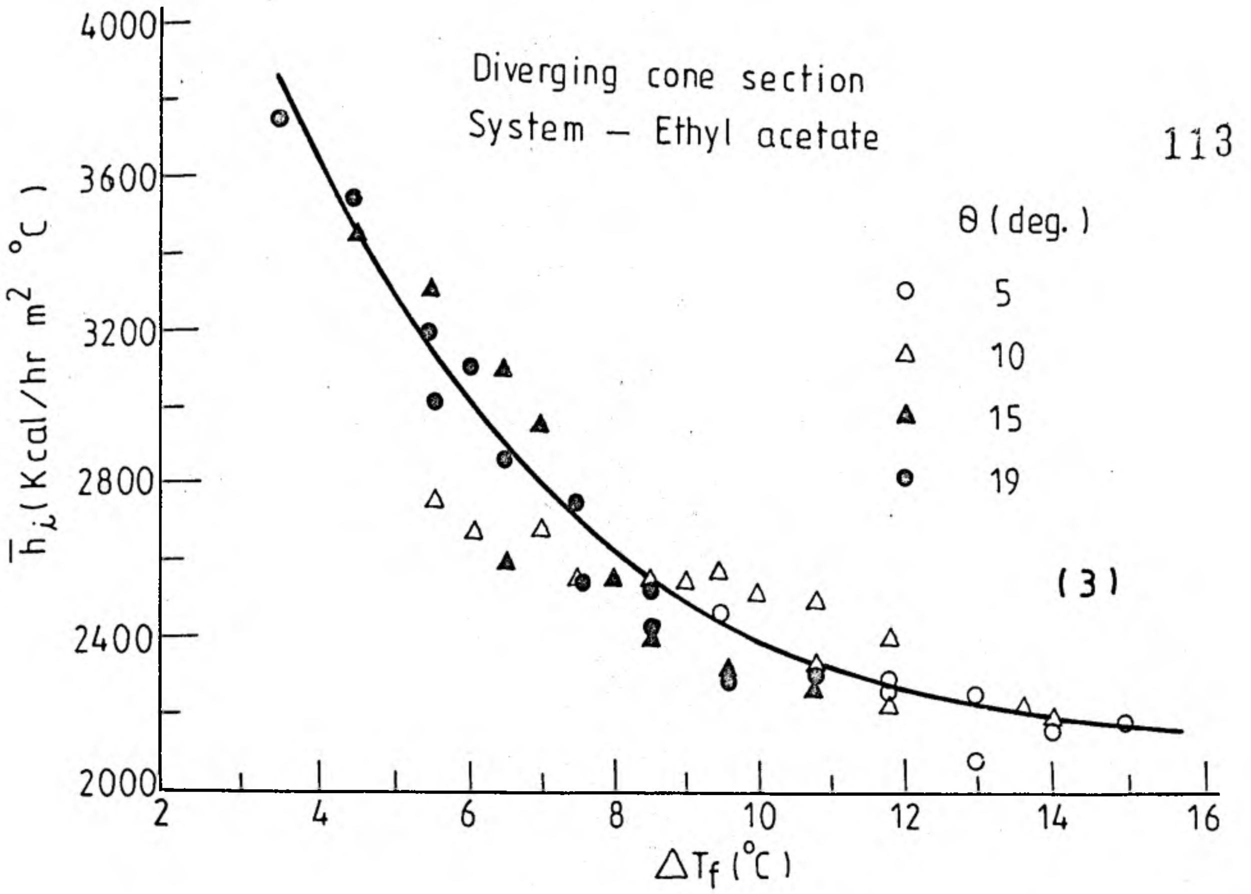


FIG.5.4. (1&2) EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (1) WATER VAPOUR, (2) ETHYL ALCOHOL VAPOUR IN DIVERGING CONE SECTIONS



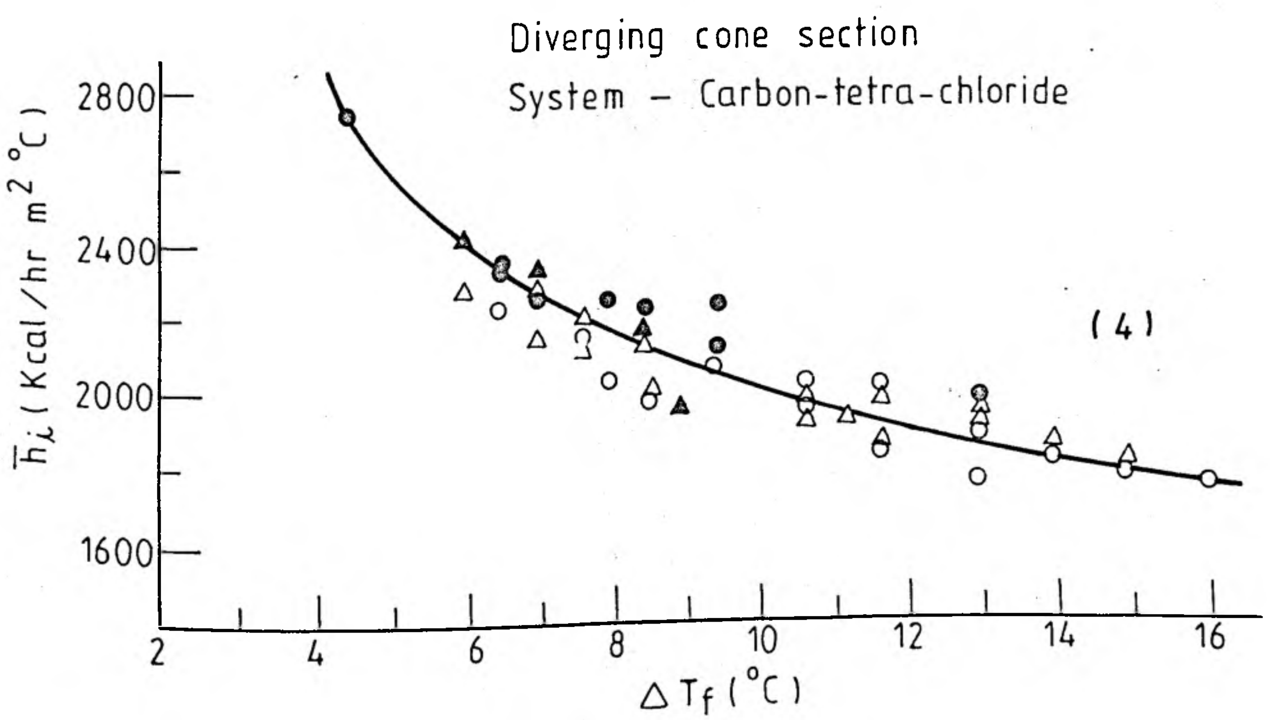
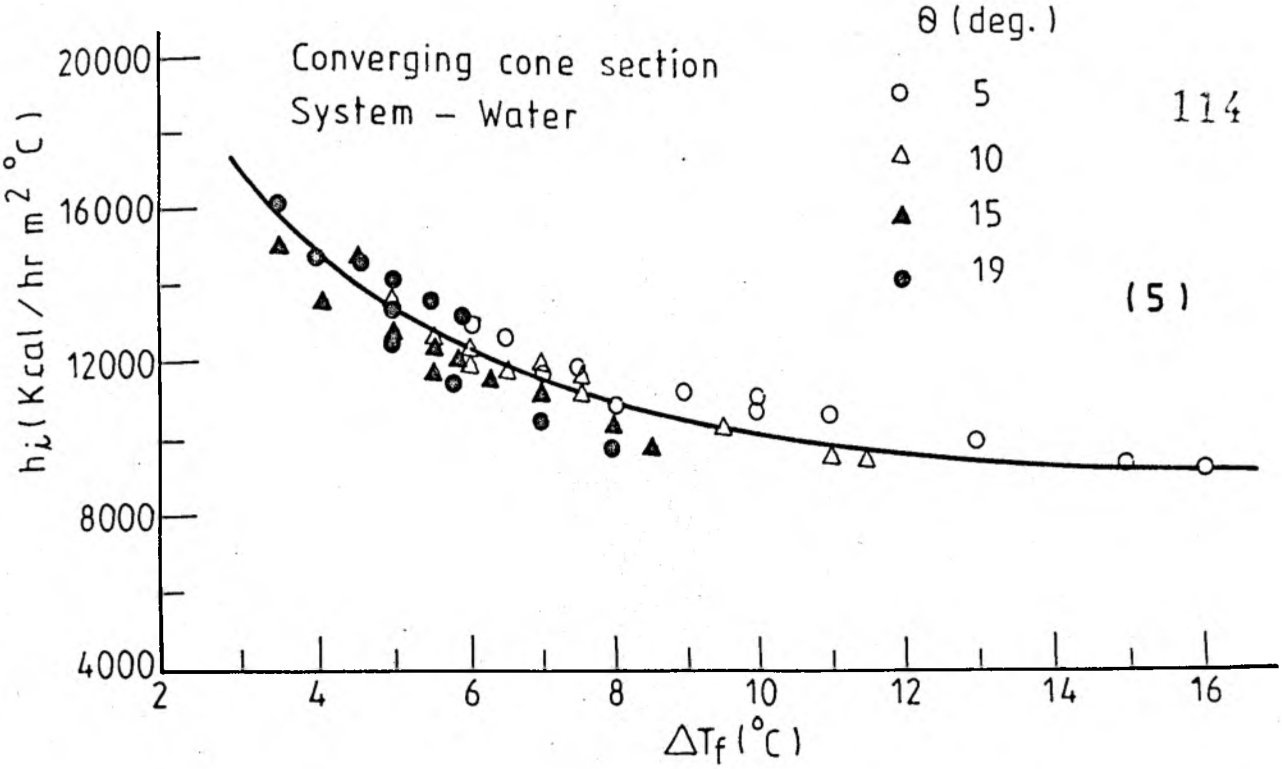


FIG.5.4. EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR
(3&4) CONDENSATION OF (3) ETHYL ACETATE VAPOUR, (4) CARBON-TETRA-CHLORIDE IN DIVERGING CONE SECTIONS



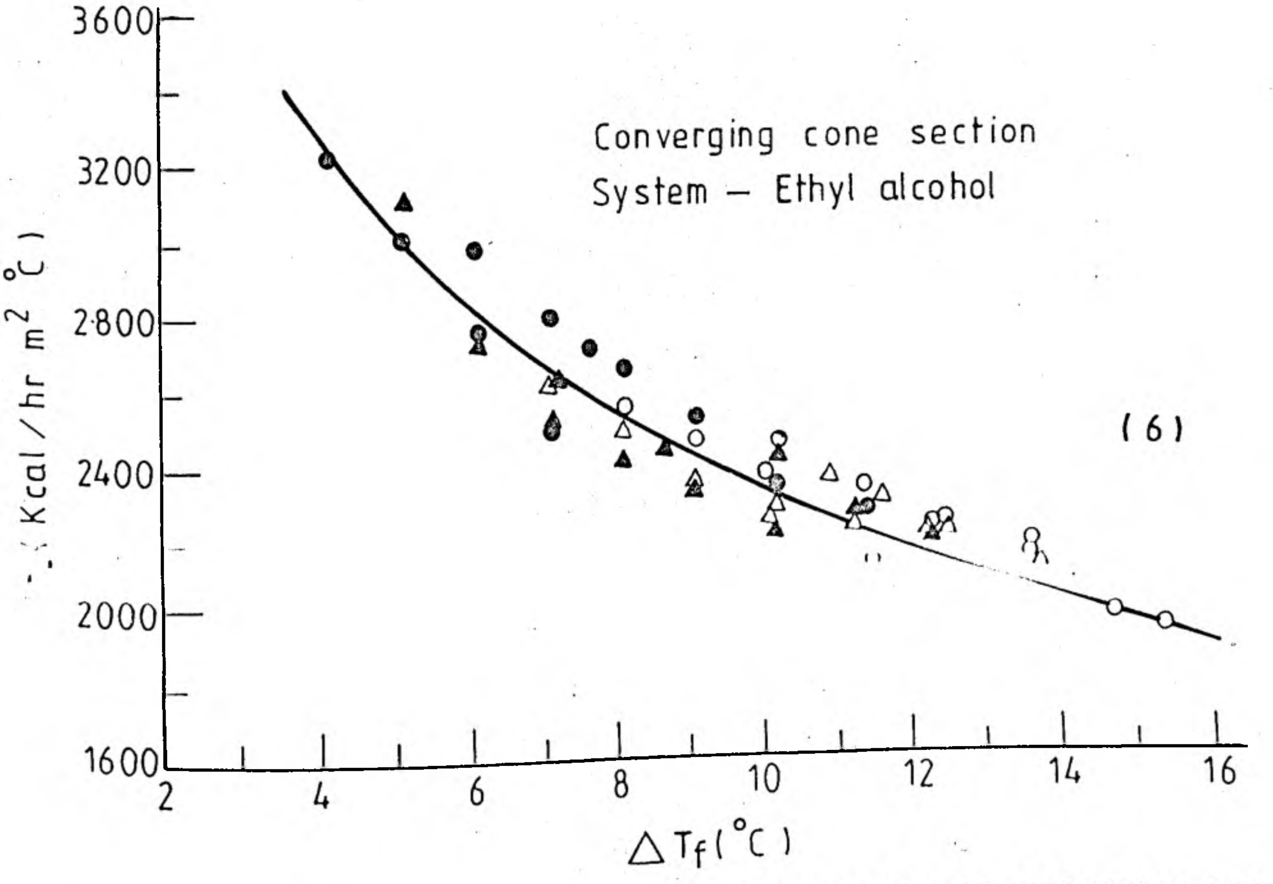
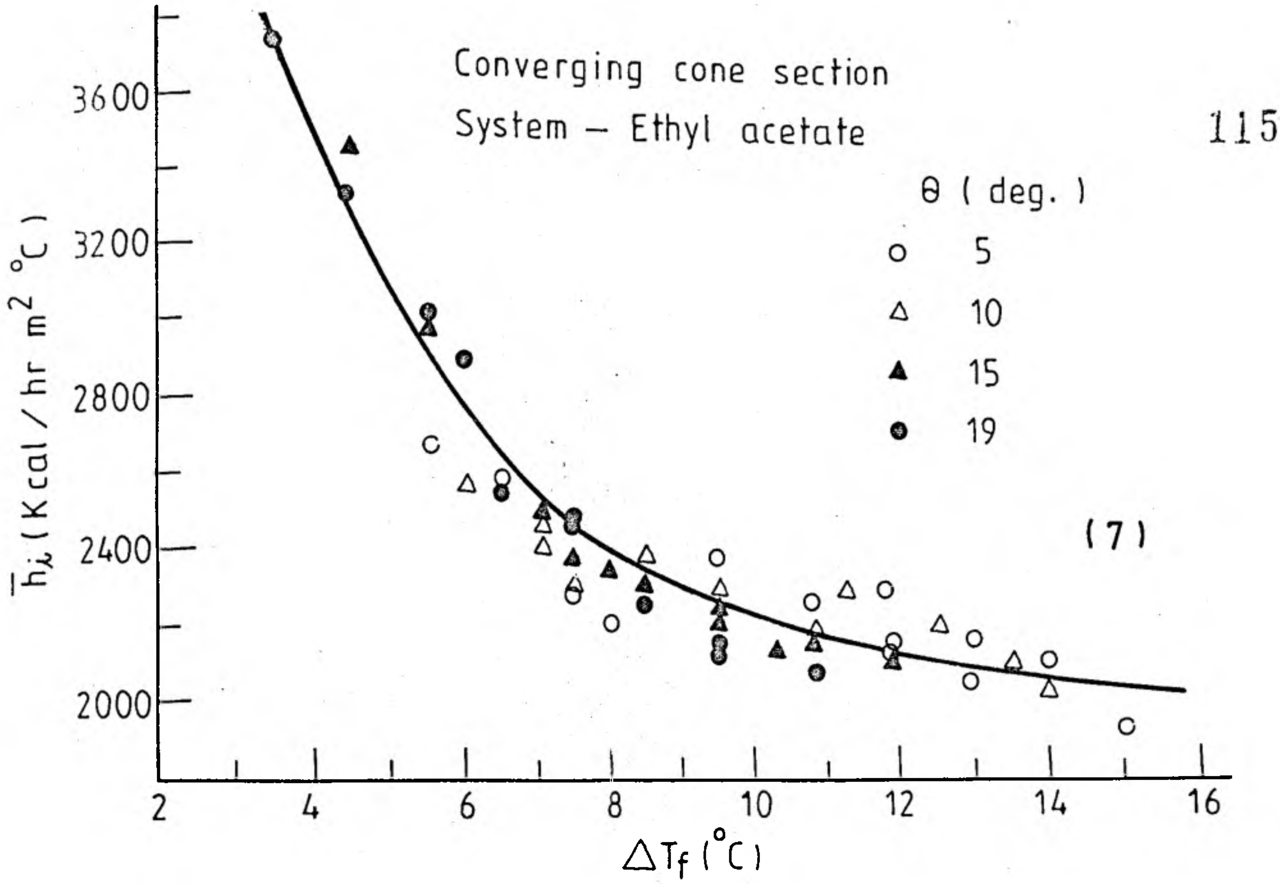


FIG.5.4. (5&6) EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (5) WATER VAPOUR, (6) ETHYL ALCOHOL VAPOUR IN CONVERGING CONE SECTIONS



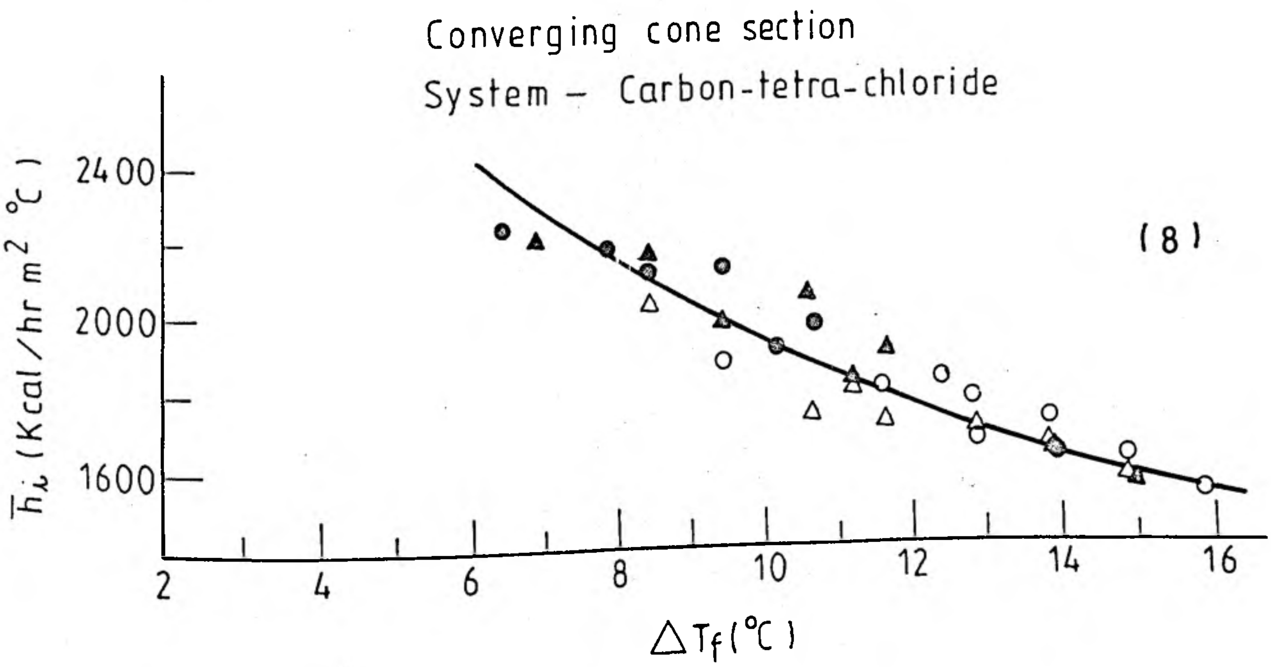
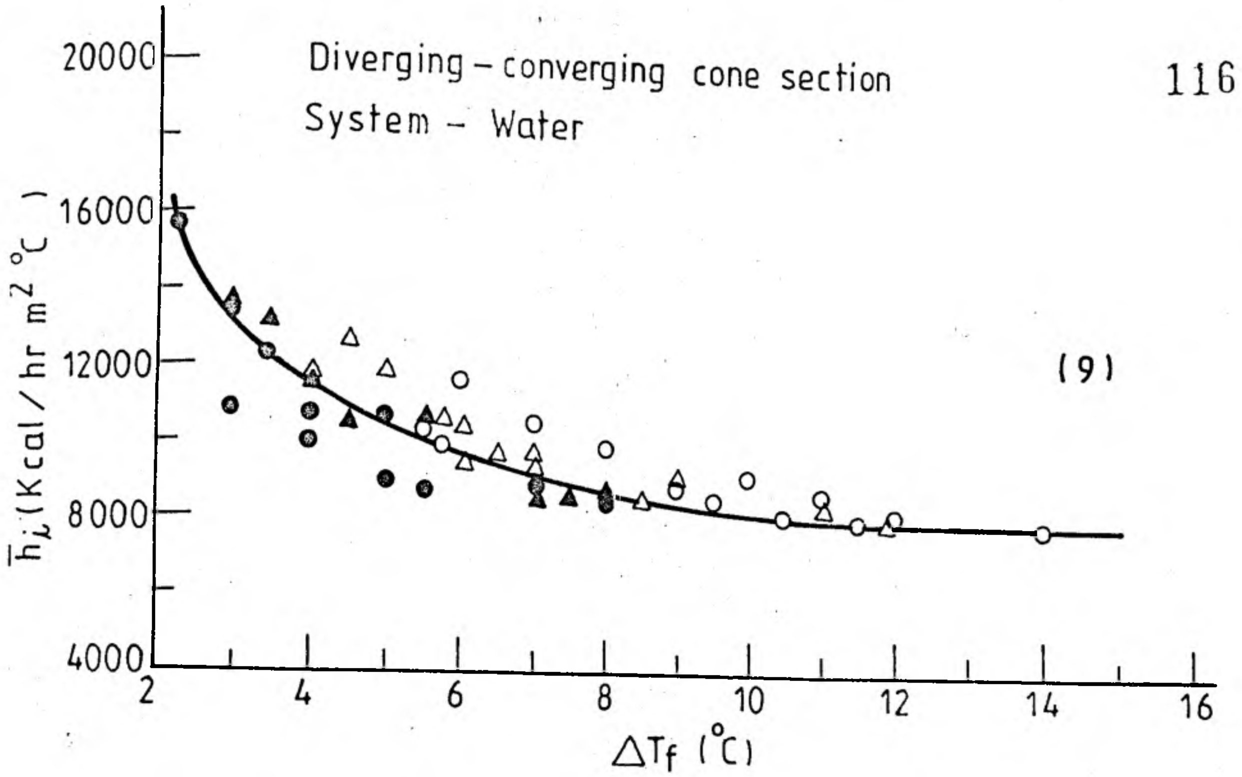


FIG.5.4. (7&8) EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (7) ETHYL ACETATE VAPOUR,(8) CARBON-TETRA-CHLORIDE VAPOUR IN CONVERGING CONE SECTIONS



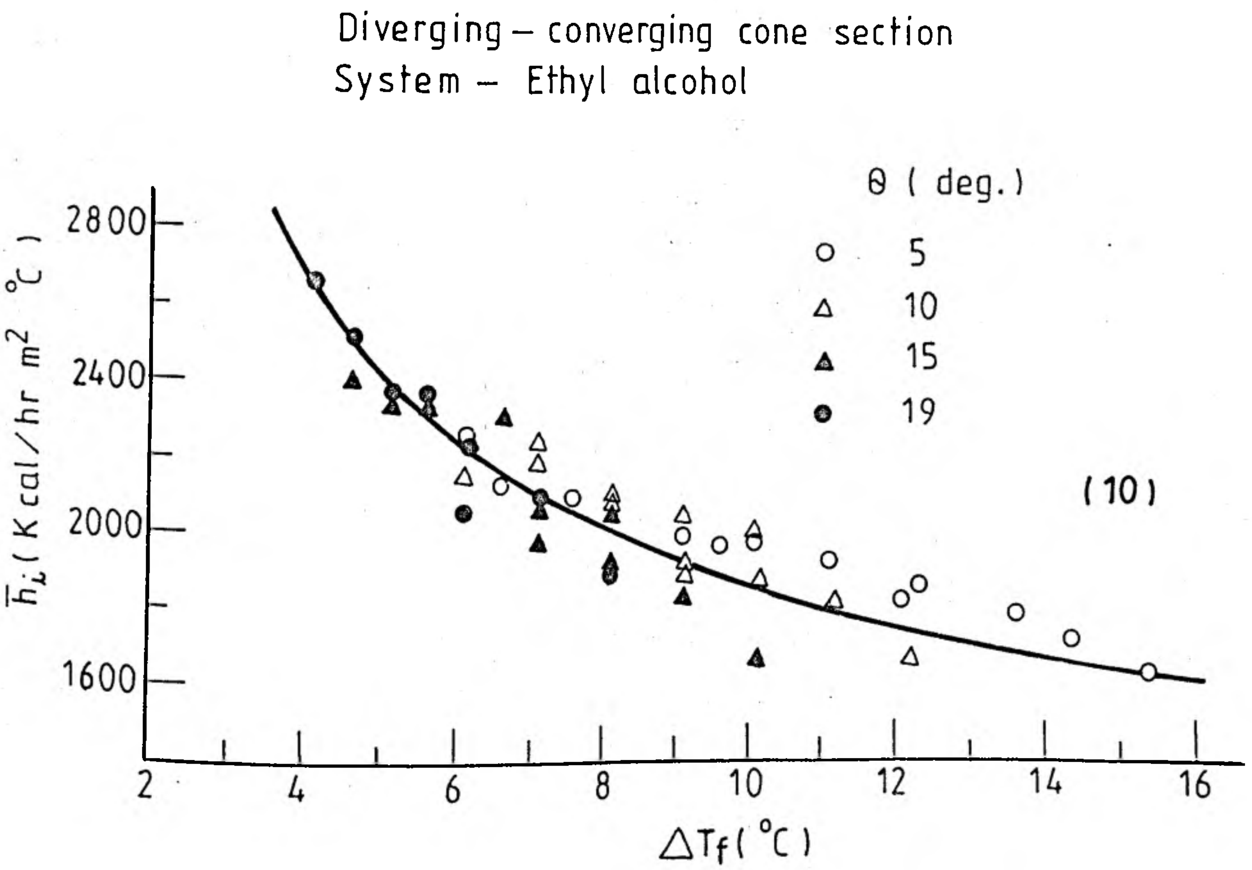
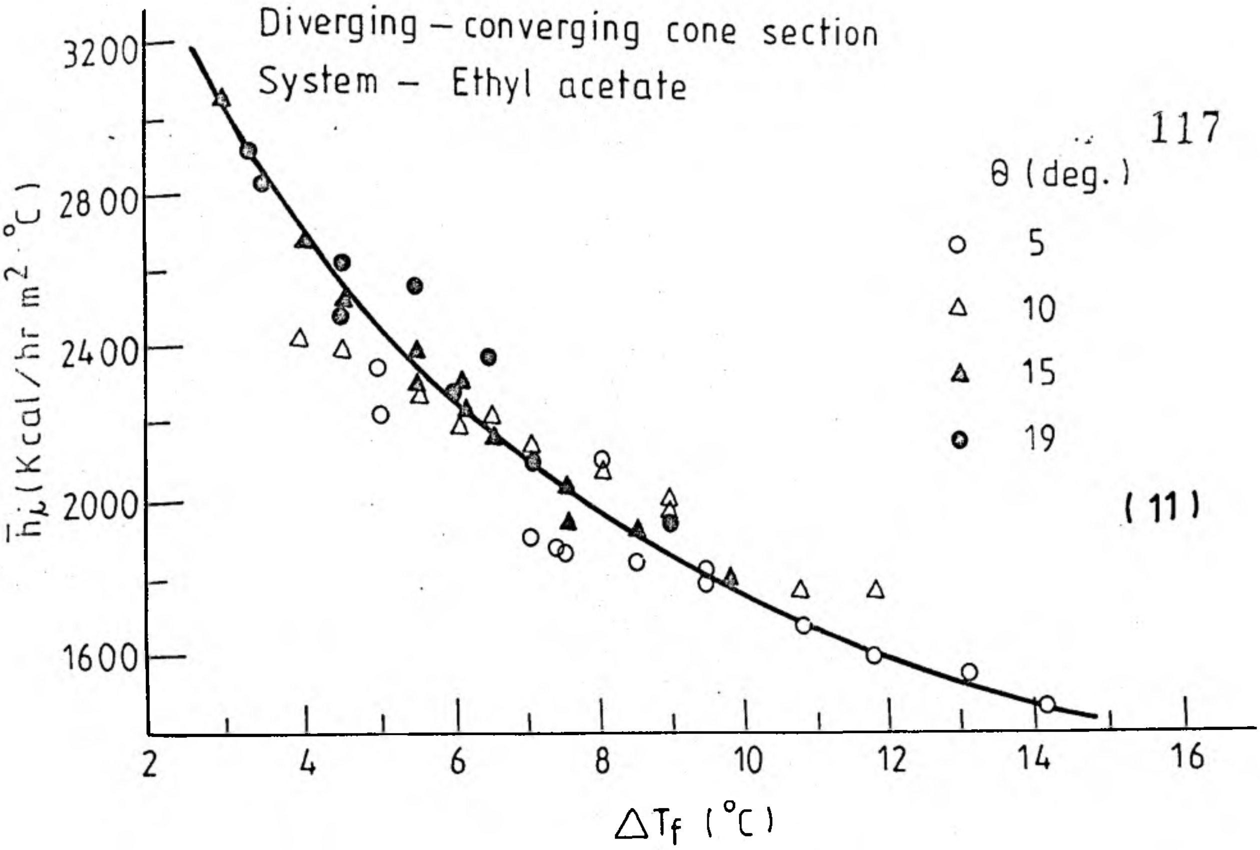


FIG. 5.4 (9 & 10) EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (9) WATER VAPOUR, (10) ETHYL ALCOHOL VAPOUR IN DIVERGING - CONVERGING CONE SECTIONS



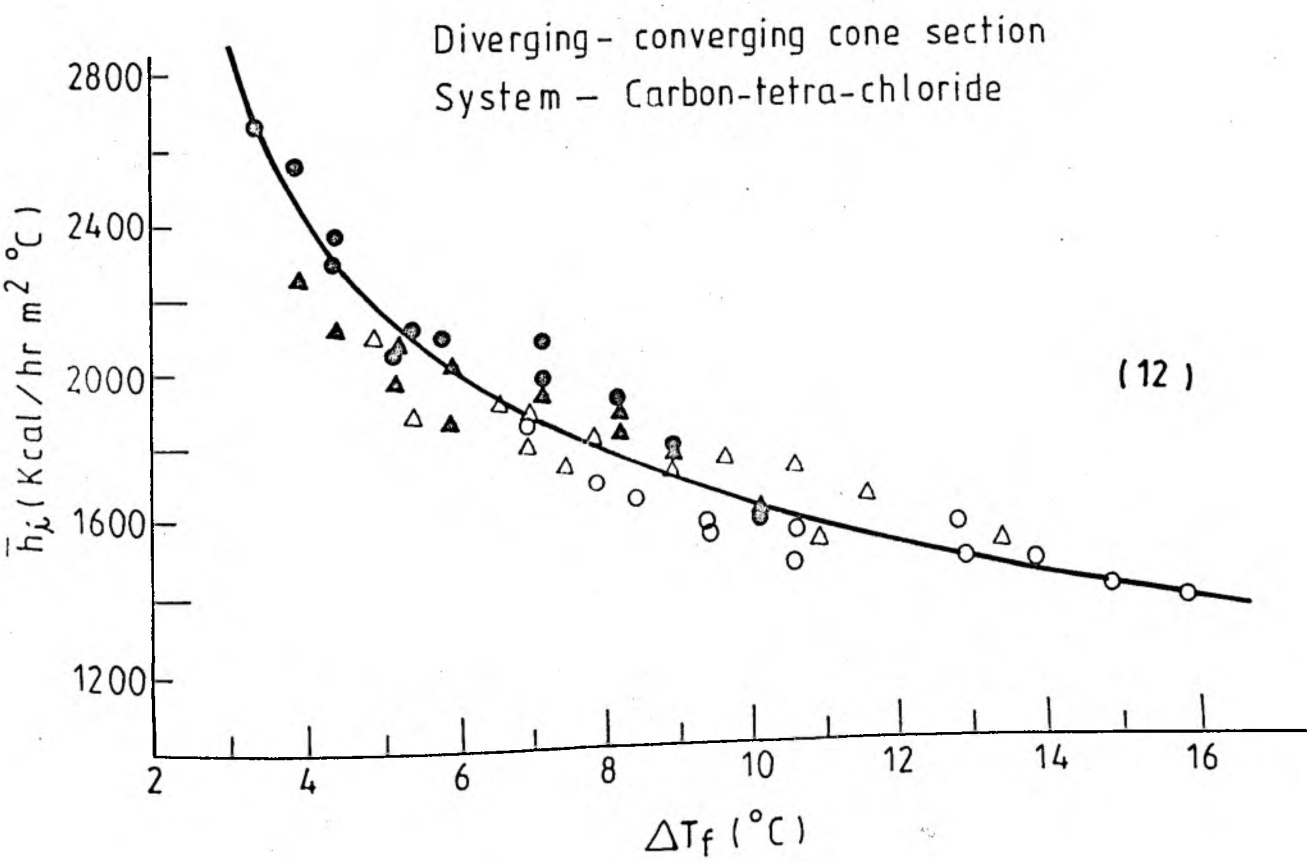


FIG.5.4. (11&12) EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF (11) ETHYL ACETATE VAPOUR, (12) CARBON-TETRA-CHLORIDE VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

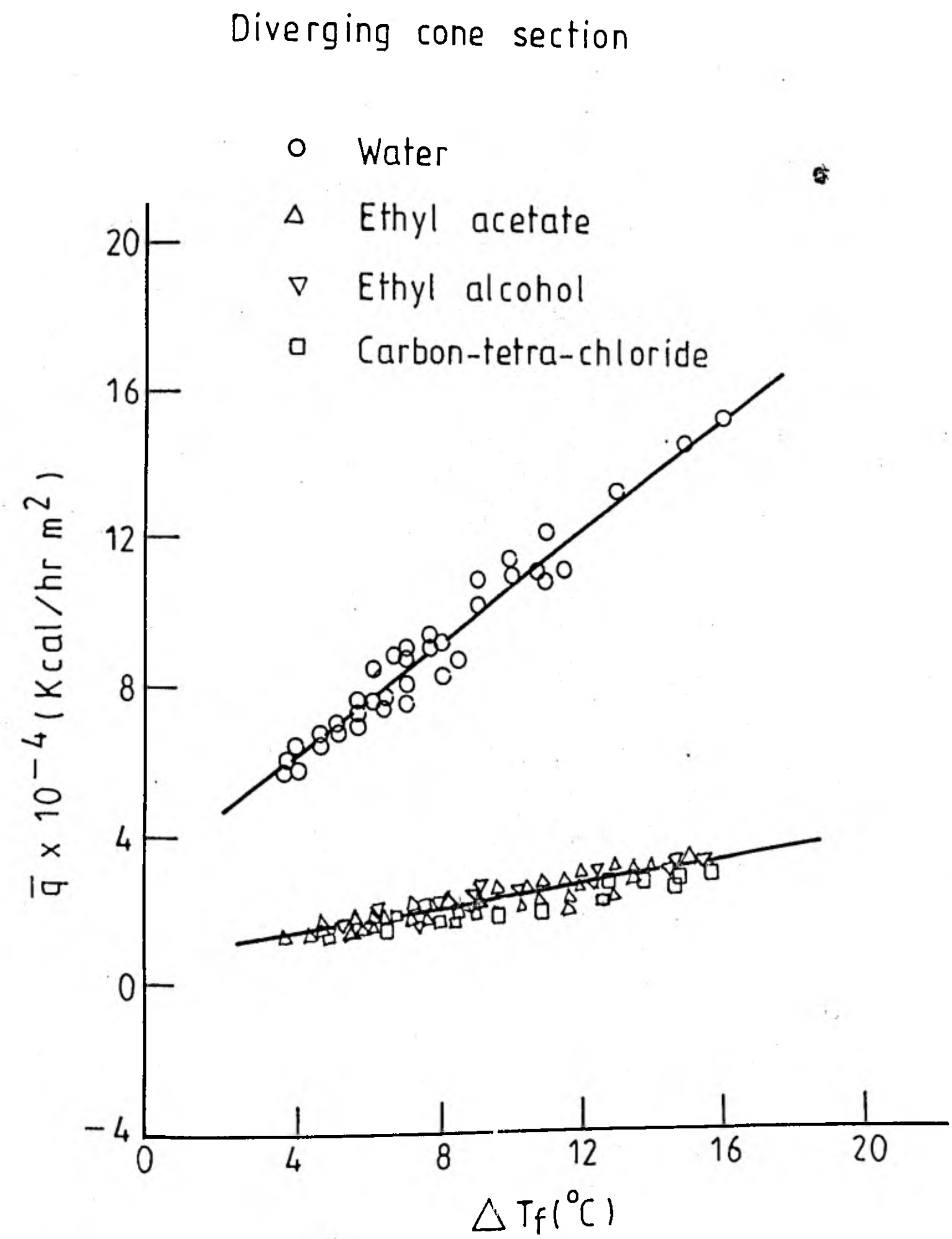


FIG.5.4-13 EFFECT OF ΔT_f ON HEAT FLUX FOR CONDENSATION OF WATER, ETHYL ACETATE, ETHYL ALCOHOL AND CARBON-TETRA-CHLORIDE VAPOURS IN DIVERGING CONE SECTIONS

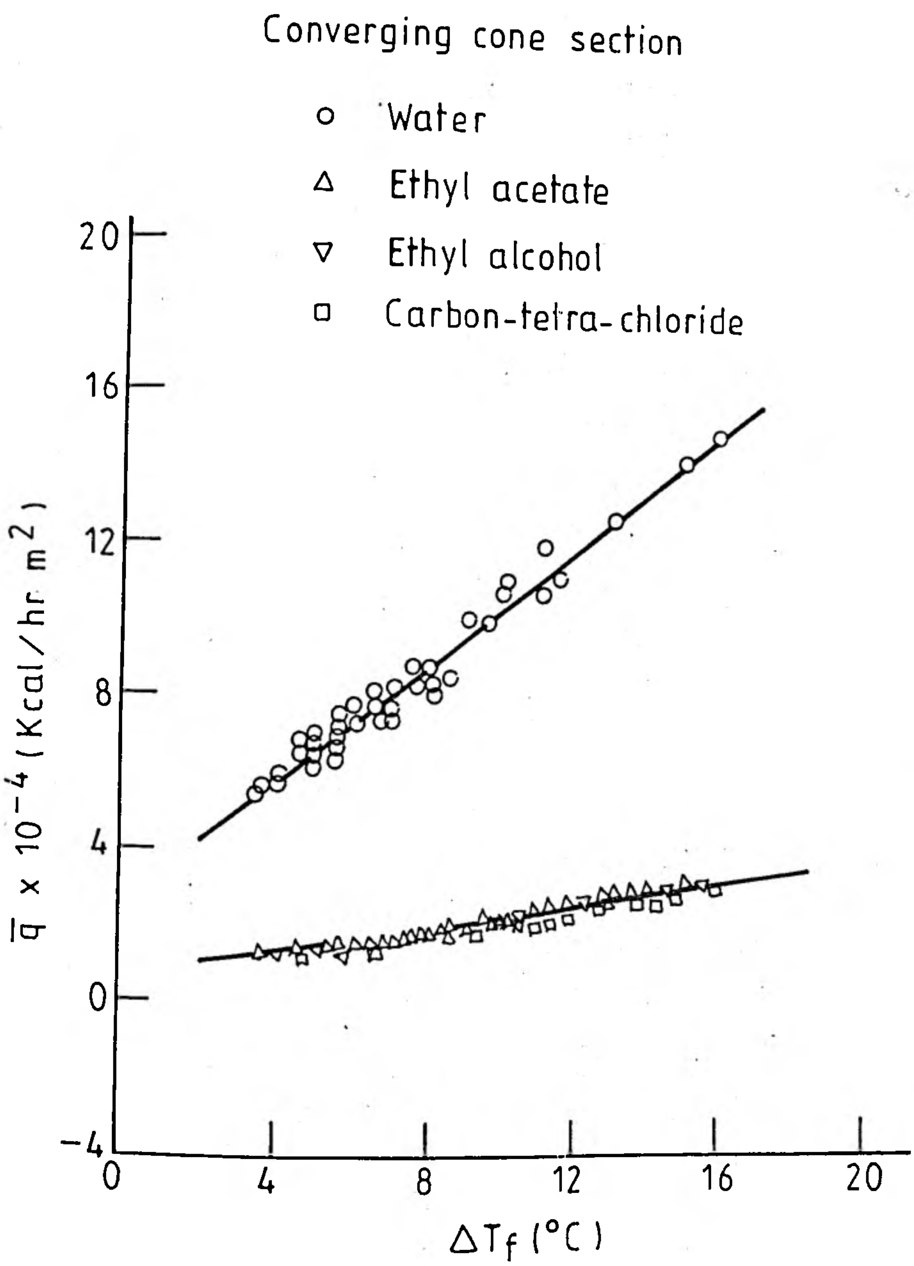


FIG.5.4-14. EFFECT OF ΔT_f ON HEAT FLUX FOR CONDENSATION OF WATER, ETHYL ACETATE, ETHYL ALCOHOL AND CARBON-TETRA-CHLORIDE VAPOURS IN CONVERGING CONE SECTIONS

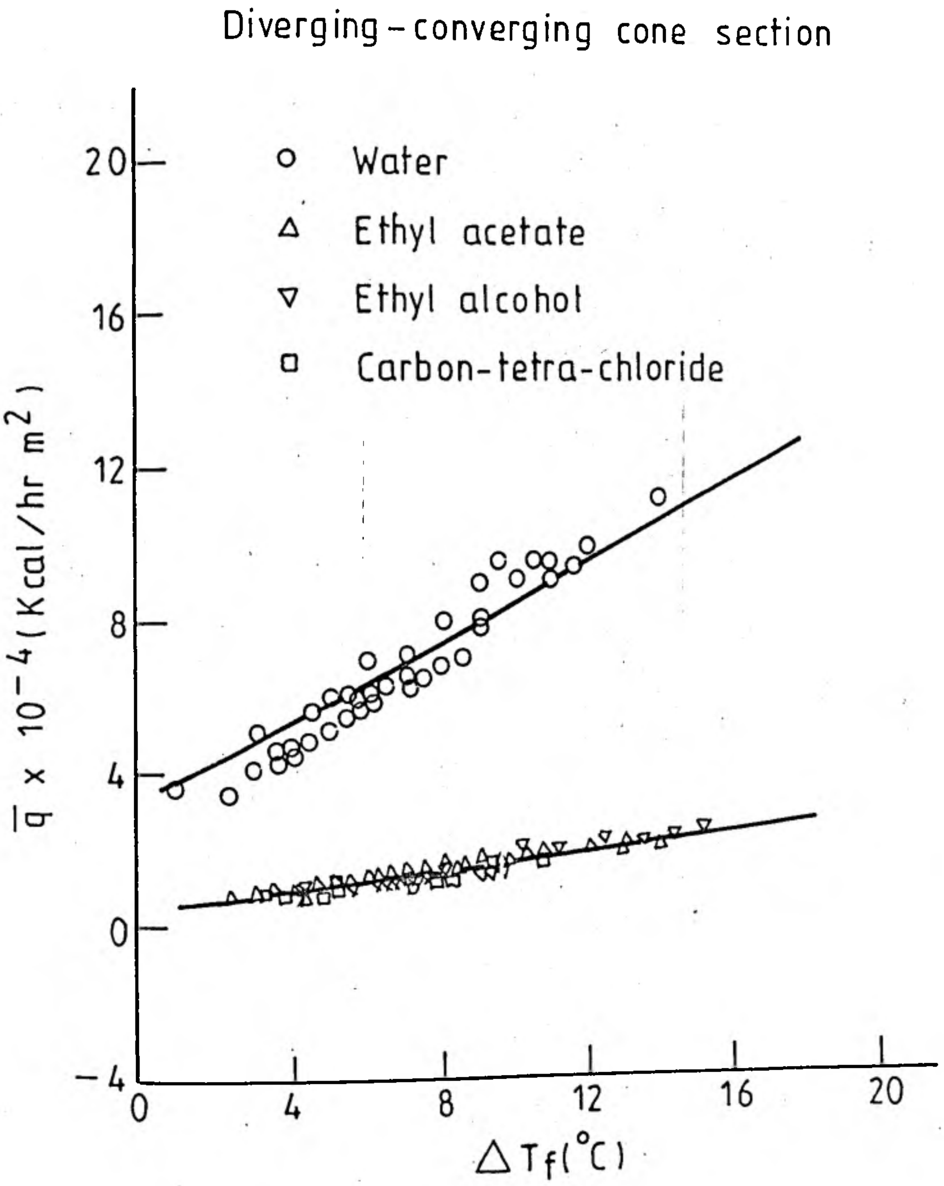


FIG.5.4-15. EFFECT OF ΔT_f ON HEAT FLUX FOR CONDENSATION OF WATER, ETHYL ACETATE, ETHYL ALCOHOL AND CARBON-TETRA-CHLORIDE VAPOURS IN DIVERGING-CONVERGING CONE SECTIONS

evident from the graphs that the heat flux increases with increase in ΔT_f, and this commensurates with the basic principle of heat transfer.

5.5 Comparison Of Theoretical And Experimental Results:

Figures 5.5-1 and 5.5-2 compare the experimental values of average heat transfer coefficient with that obtained theoretically by using equation (3.2.22) for condensation of water and organic vapours respectively in diverging cone sections. Data points for all the cones viz., $\theta = 5°$, 10°, 15° and 19° have been plotted. Calculations were carried out on the basis of the same ΔT_f values, obtained experimentally.

Figures 5.5-3 and 5.5-4 compare the experimental values of average heat transfer coefficient and that obtained theoretically by using equation (3.3.17), for condensation of water and organic vapours respectively in converging cone sections. Figures 5.5-5 and 5.5-6 represent similar comparison for diverging-converging cone sections. For diverging-converging cone sections theoretical values have been calculated from equation (3.4.11).

In all the three cases, the experimental values of heat transfer coefficient are found to be higher than the theoretical ones for most of the data points. This may be attributed to the fact that the various assumptions, as already discussed in connection with the mathematical analysis, might have resulted in some sort of simplification of the actual experimental

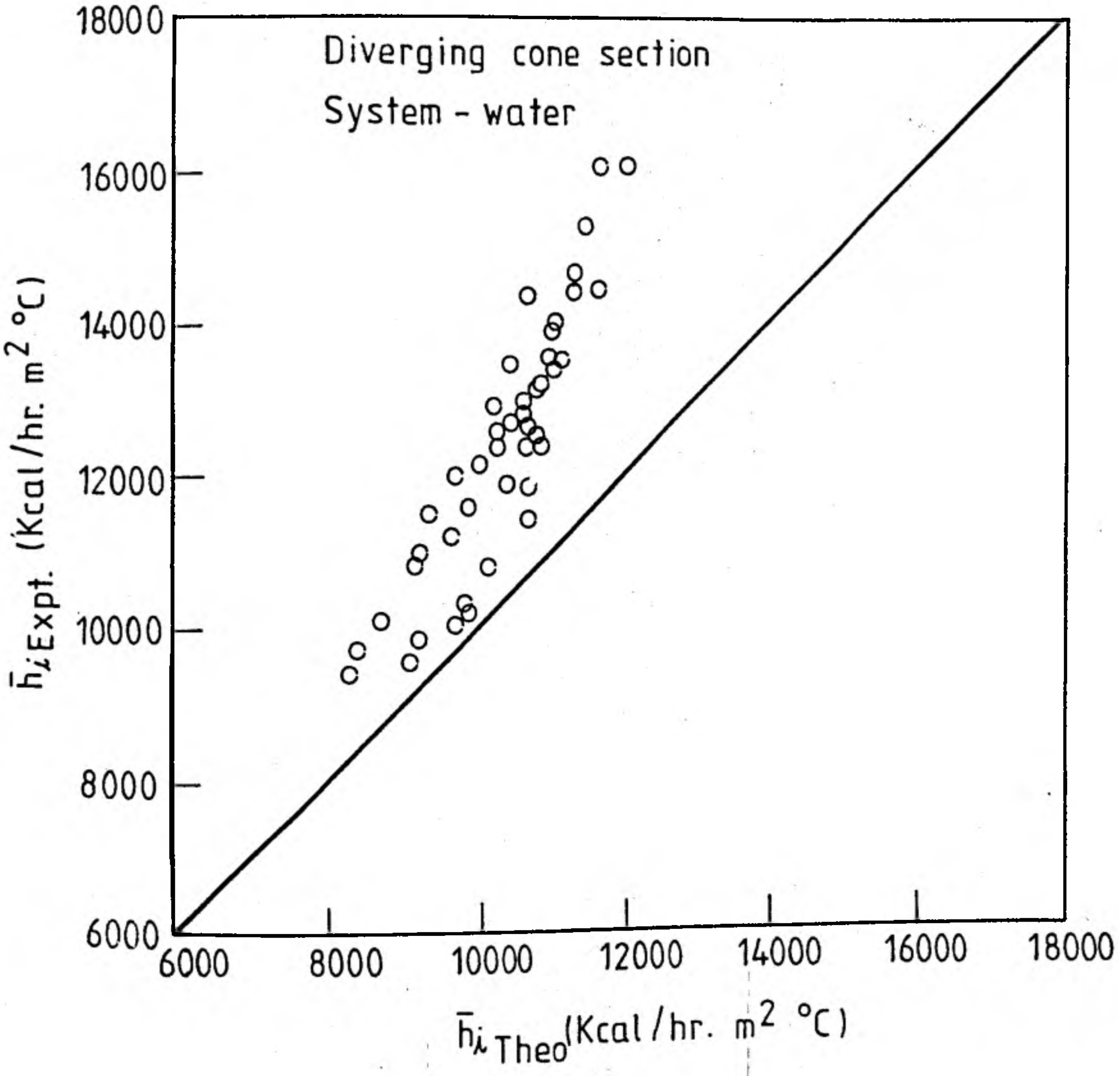


FIG. 5.5-1. COMPARISON OF EXPERIMENTAL AND THEORETICAL VALUES OF HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOUR IN DIVERGING CONE SECTIONS

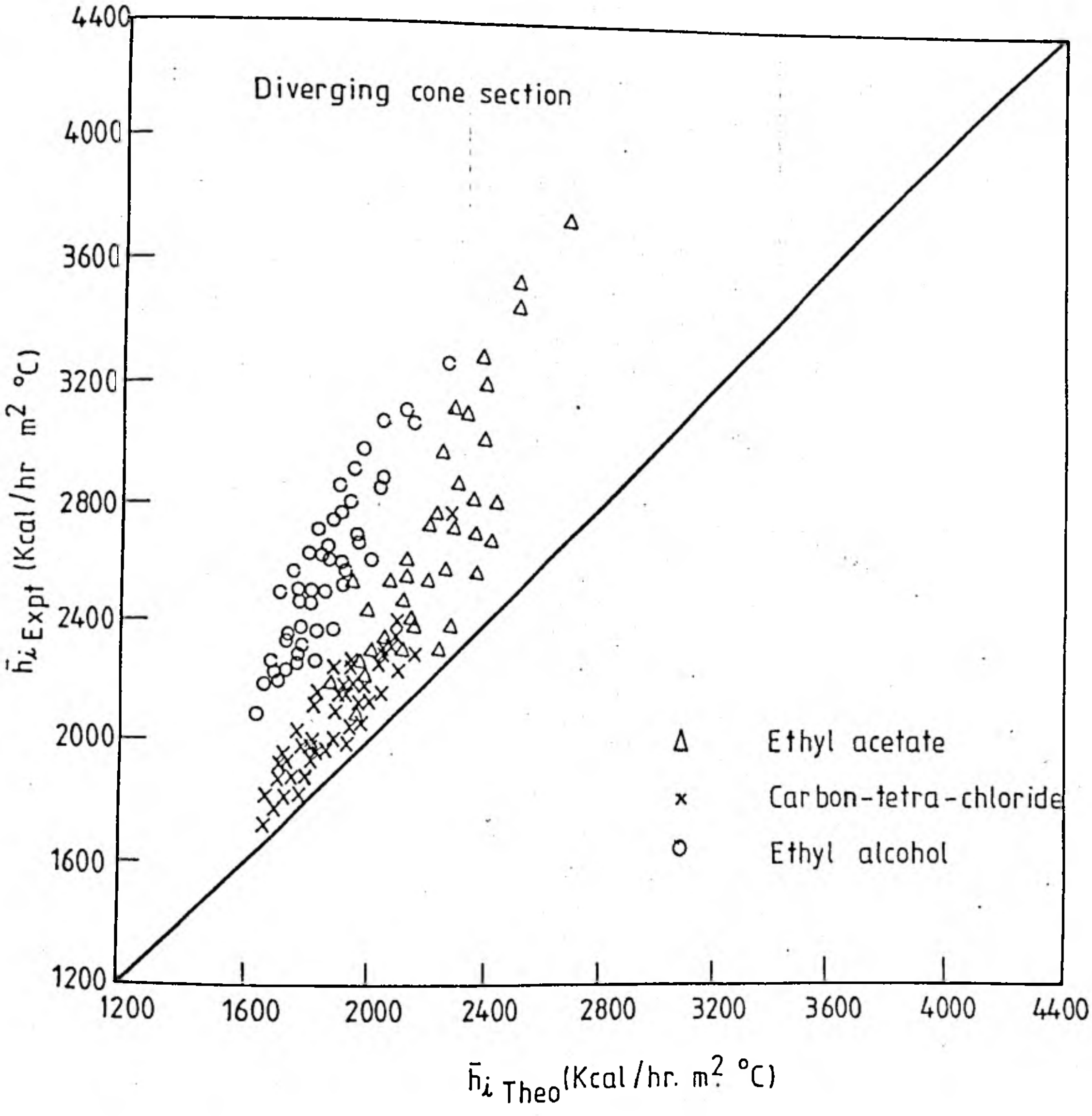


FIG.5.5-2. COMPARISON OF EXPERIMENTAL AND THEORETICAL VALUES OF HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF ETHYL-ACETATE, CTC AND ETHYL ALCOHOL VAPOURS IN DIVERGING CONE SECTIONS

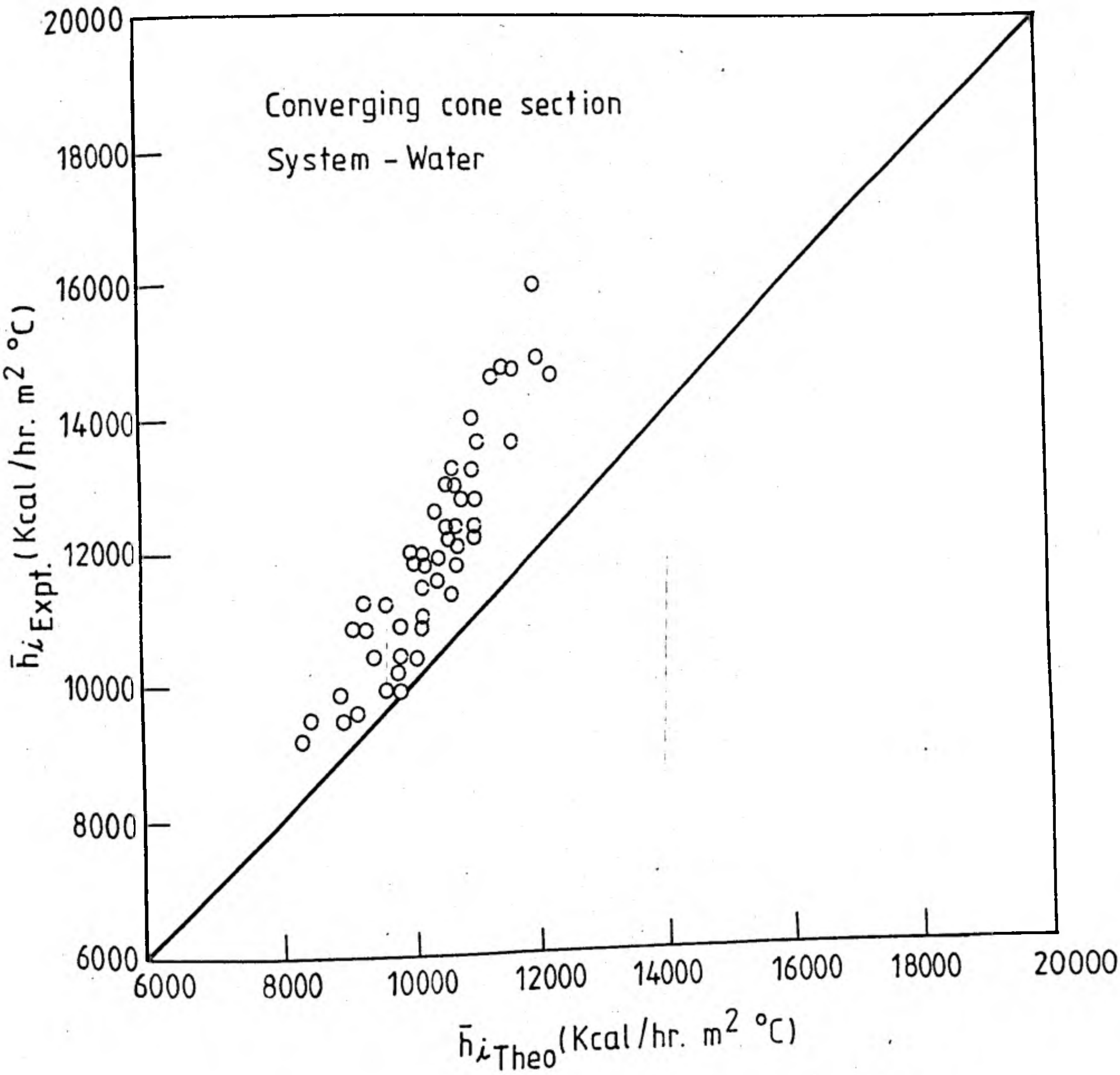


FIG. 5.5-3. COMPARISON OF EXPERIMENTAL AND THEORETICAL VALUES OF HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOUR IN CONVERGING CONE SECTIONS

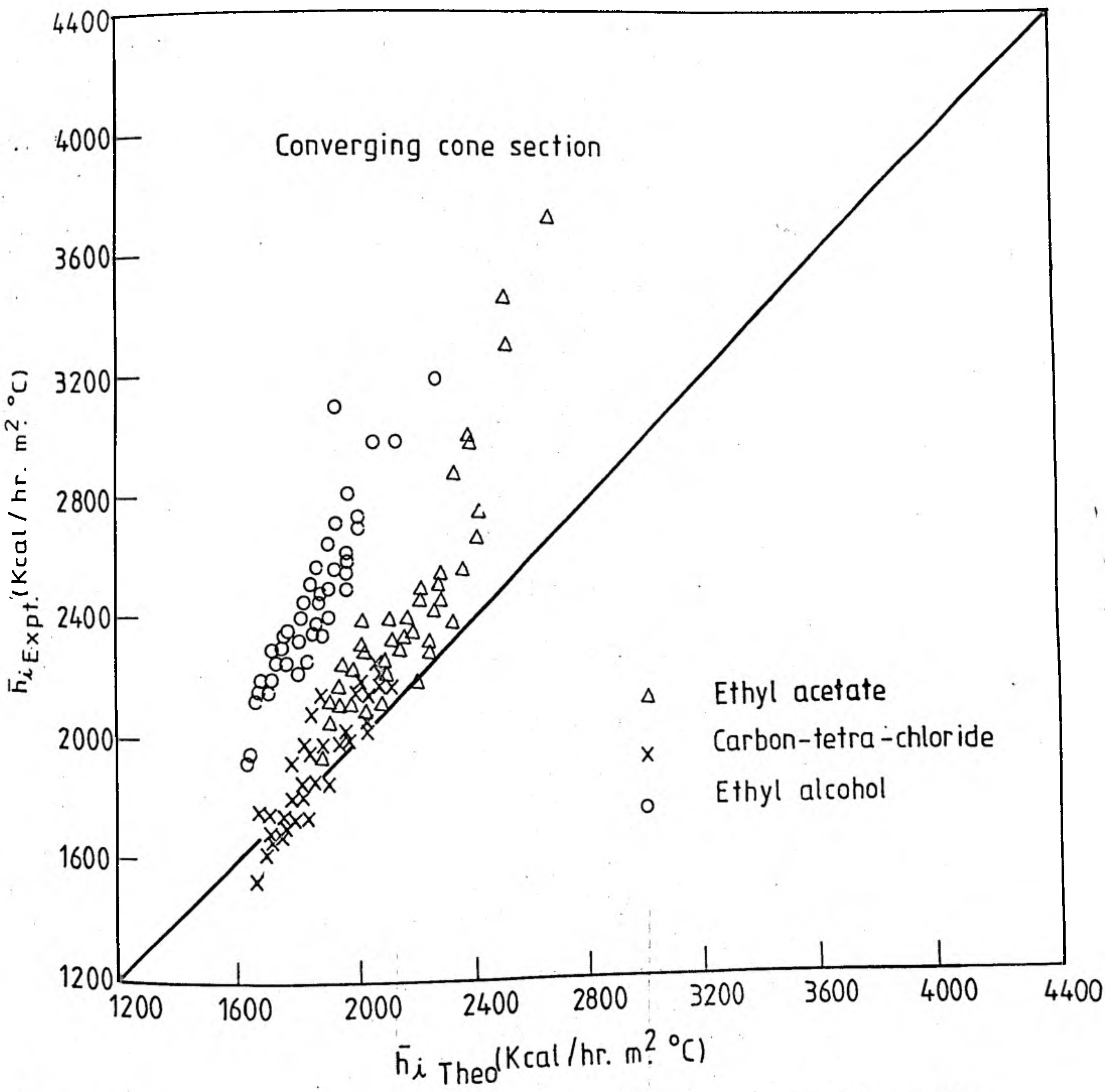


FIG. 5.5-4. COMPARISON OF EXPERIMENTAL AND THEORETICAL HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF ETHYL ACETATE, CTC AND ETHYL ALCOHOL VAPOURS IN CONVERGING CONE SECTIONS

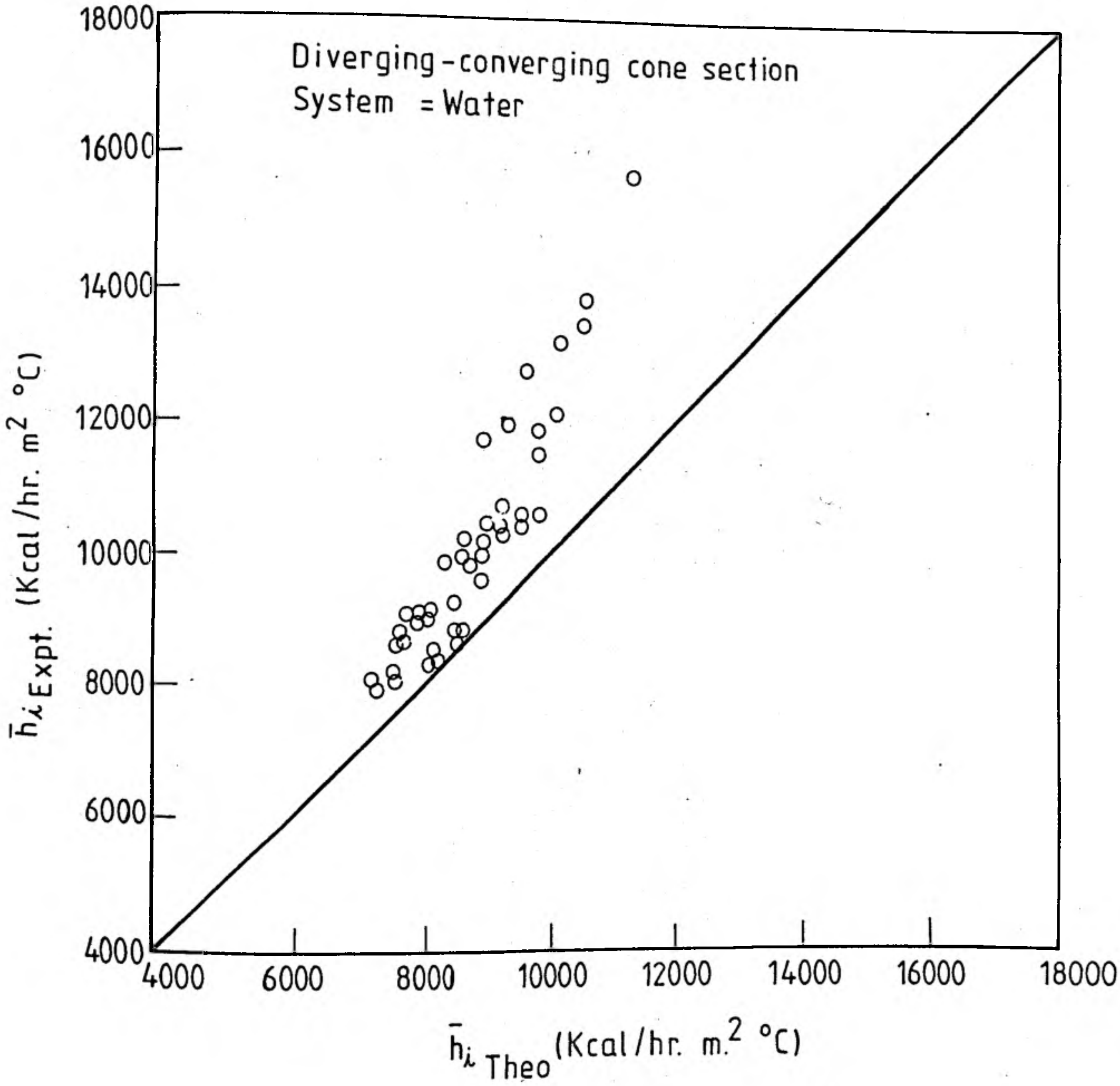


FIG. 5.5-5. COMPARISON OF EXPERIMENTAL AND THEORETICAL HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOURS IN DIVERGING-CONVERGING CONE SECTION

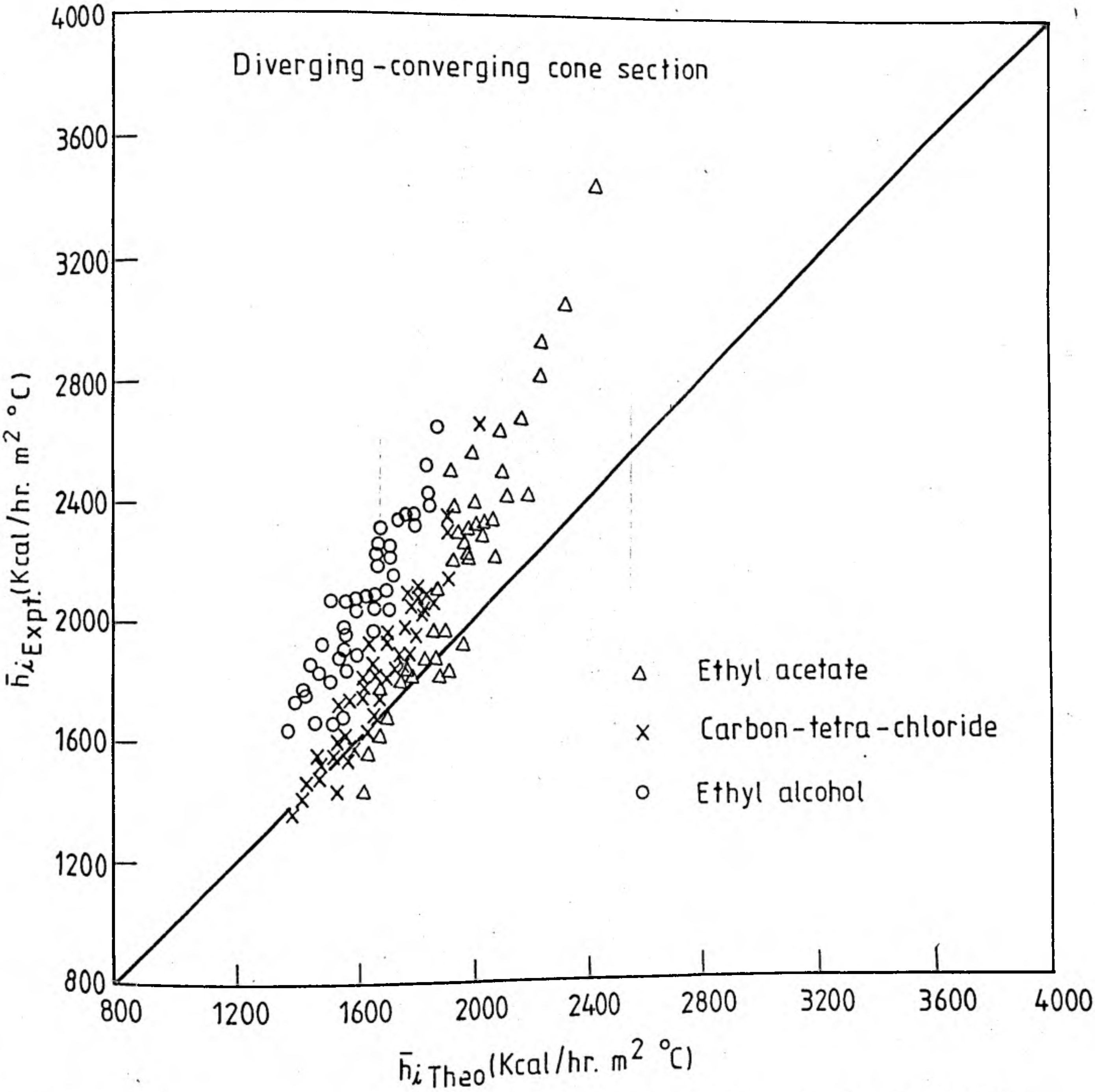


FIG.5.5-6. COMPARISON OF EXPERIMENTAL AND THEORETICAL HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF ETHYL ACETATE, CTC AND ETHYL ALCOHOL VAPOURS IN DIVERGING-CONVERGING CONE SECTIONS

conditions, thereby reducing the theoretical values of heat transfer coefficient. However, the overall standard deviation of theoretical and experimental heat transfer coefficients for the diverging, converging and diverging-converging sections are found to be 19.5, 15.6 and 16.5 percent respectively.

5.6 Variation Of Mean Nusselt Number and Condensate Reynolds Number:

Figure 5.6-1 depicts the variation of mean Nusselt number with condensate Reynolds number for condensation of water vapour in diverging cone section with θ as the third parameter. From the plot it is evident that the mean Nusselt number gradually decreases with the increase in Reynolds number. With the increase in the coolant flow rate at a particular coolant inlet temperature, the rate of condensation increases and as a result of that film thickness increases. This causes decrease in heat transfer coefficient, which commensurates with the fact that the condensation is of film type, atleast up to the range of condensate Reynolds number that has been obtained in the experiments. Condensation of carbon-tetra-chloride vapour (Figure 5.6-2) and other vapours follow the same basic trend. From the plot it is evident that for a particular condensate Reynolds number, mean Nusselt number increases with increase in cone angle, θ, which indicates that film thickness decreases with increase in cone angle, which has already been discussed in section 5.3.

For converging and diverging-converging cone sections

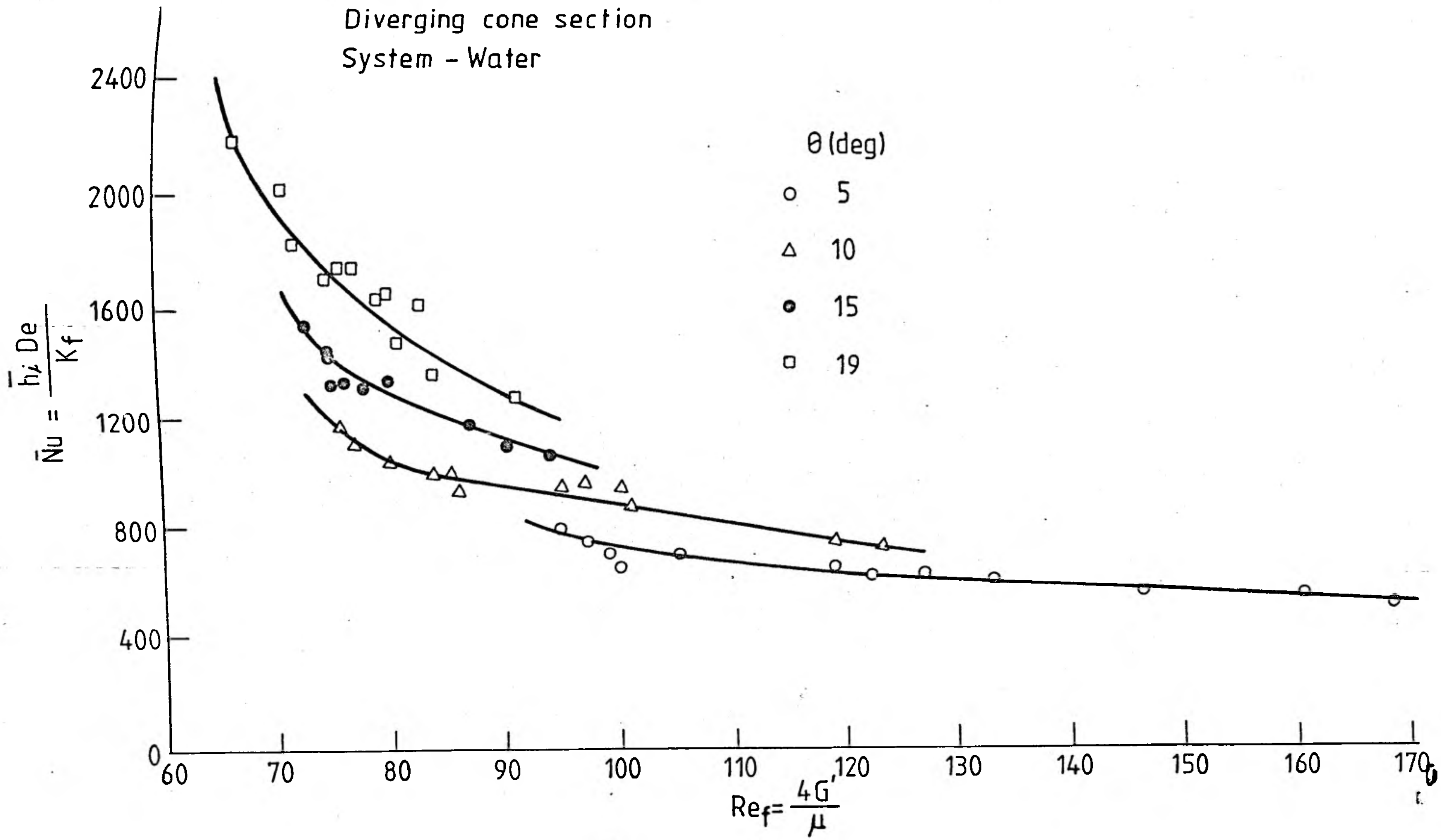


FIG.5.6 -1. VARIATION OF Re_f WITH $\bar{N}u$ FOR CONDENSATION OF WATER VAPOUR IN DIVERGING CONE SECTIONS

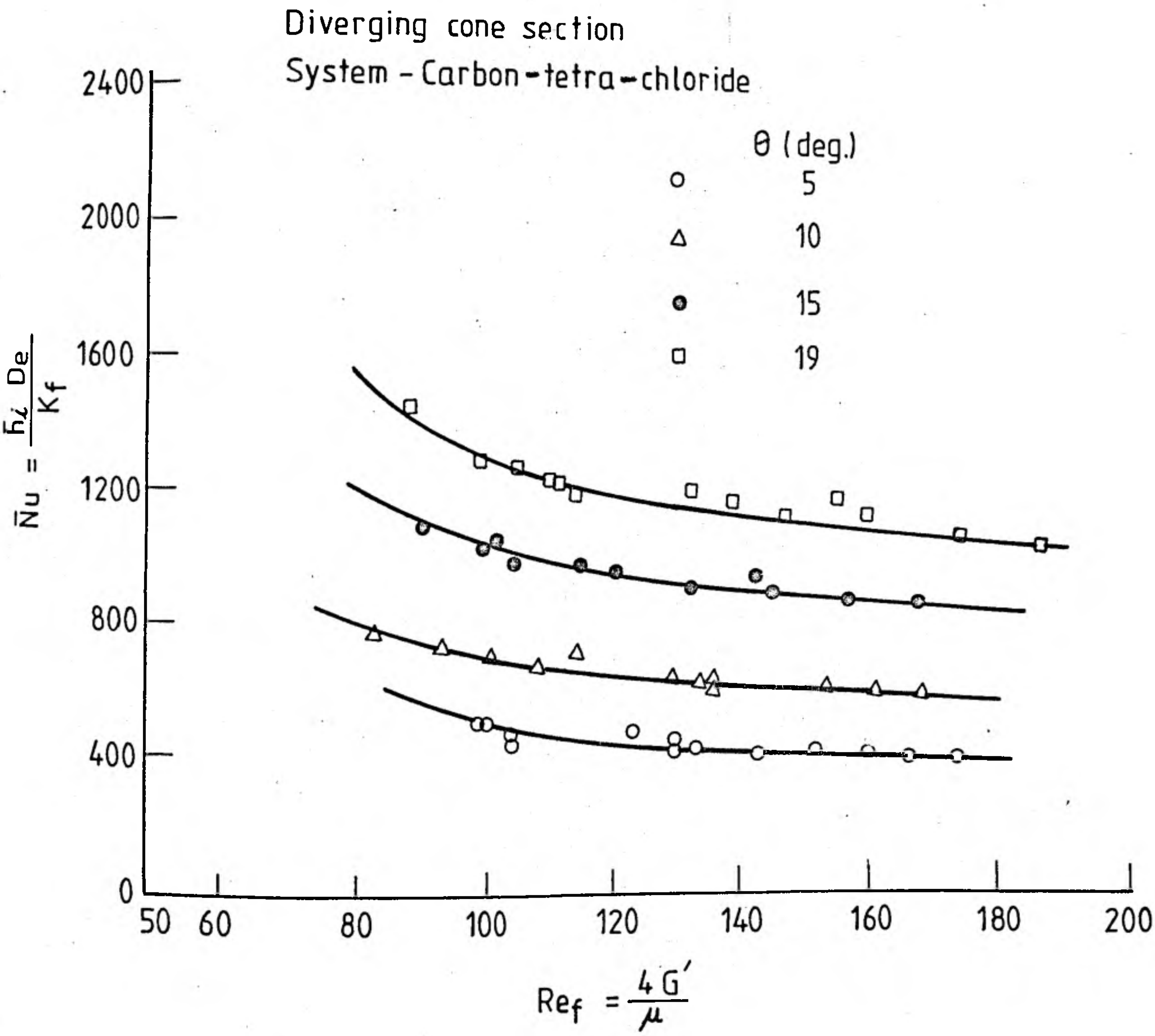


FIG. 5.6-2. VARIATION OF Re_f WITH $\bar{Nu}$ FOR CONDENSATION OF CTC VAPOUR IN DIVERGING CONE SECTIONS

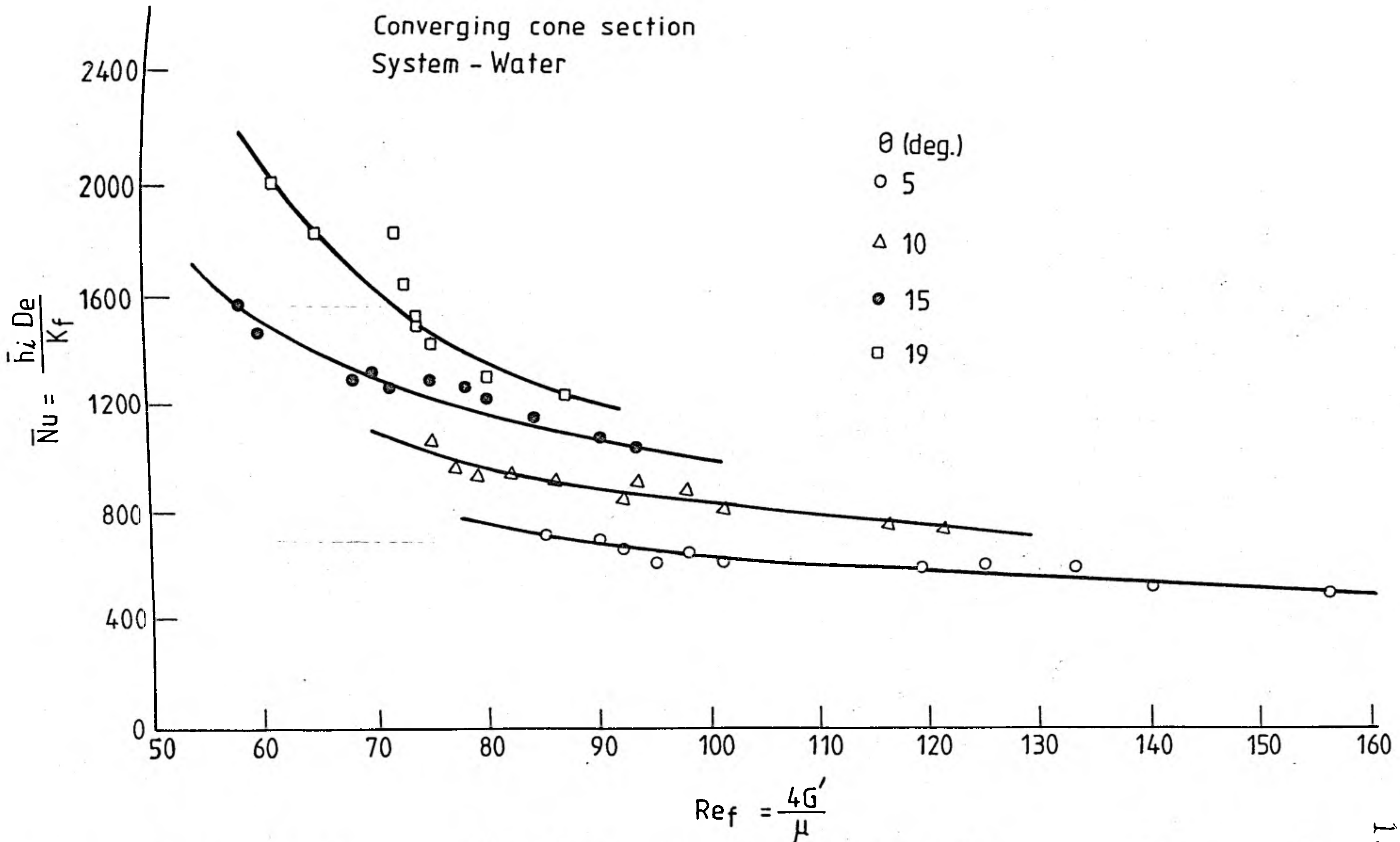


FIG. 5.6 -3. VARIATION OF Re_f WITH $\bar{N}u$ FOR CONDENSATION OF WATER VAPOUR IN CONVERGING CONE SECTIONS

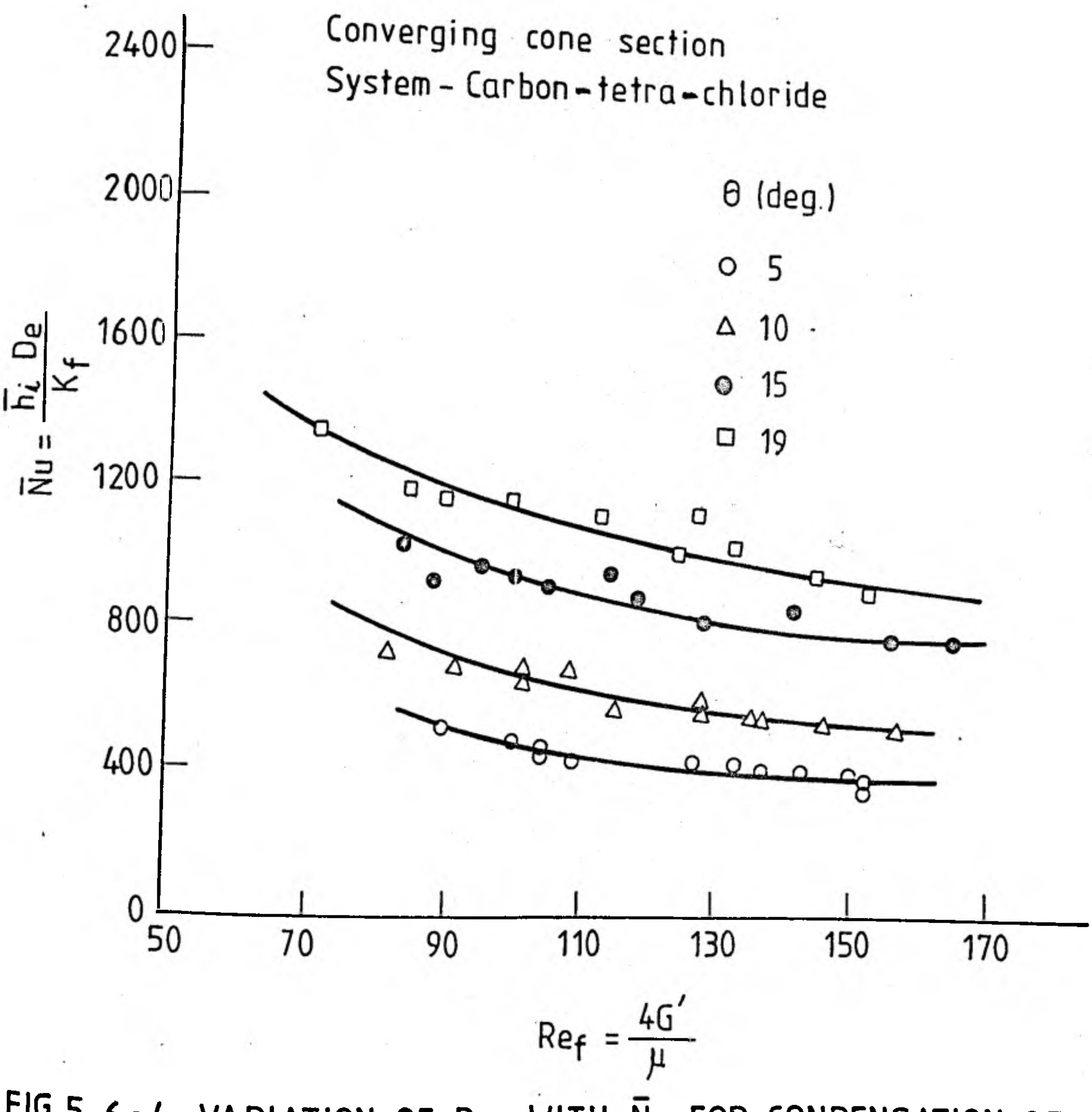


FIG.5.6-4. VARIATION OF Re_f WITH $\bar{N}u$ FOR CONDENSATION OF CTC VAPOUR IN CONVERGING CONE SECTIONS

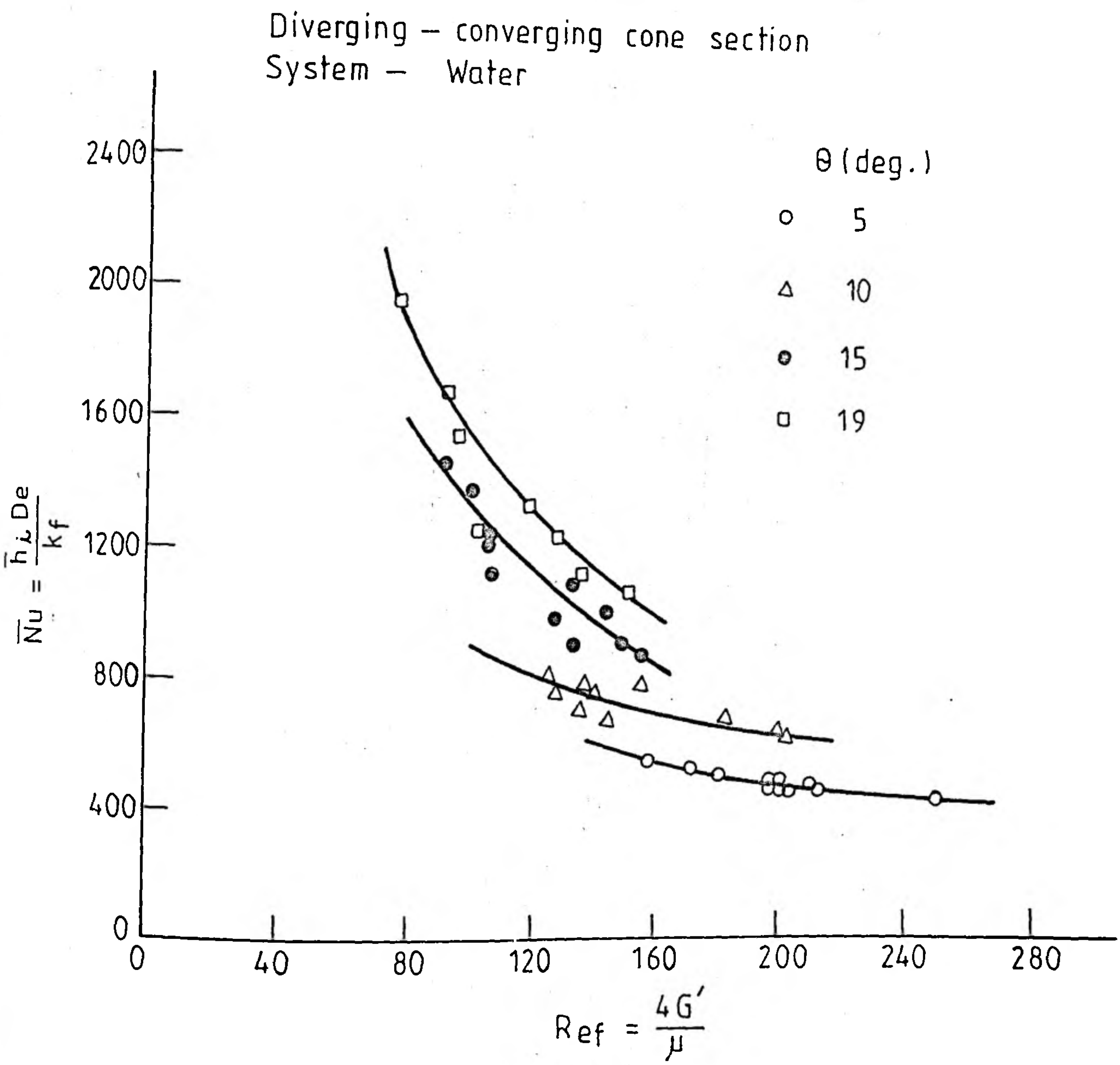


FIG.5.6-5. VARIATION OF Re_f WITH $\overline{Nu}$ FOR CONDENSATION OF WATER VAPOUR IN DIVERGING - CONVERGING CONE SECTIONS

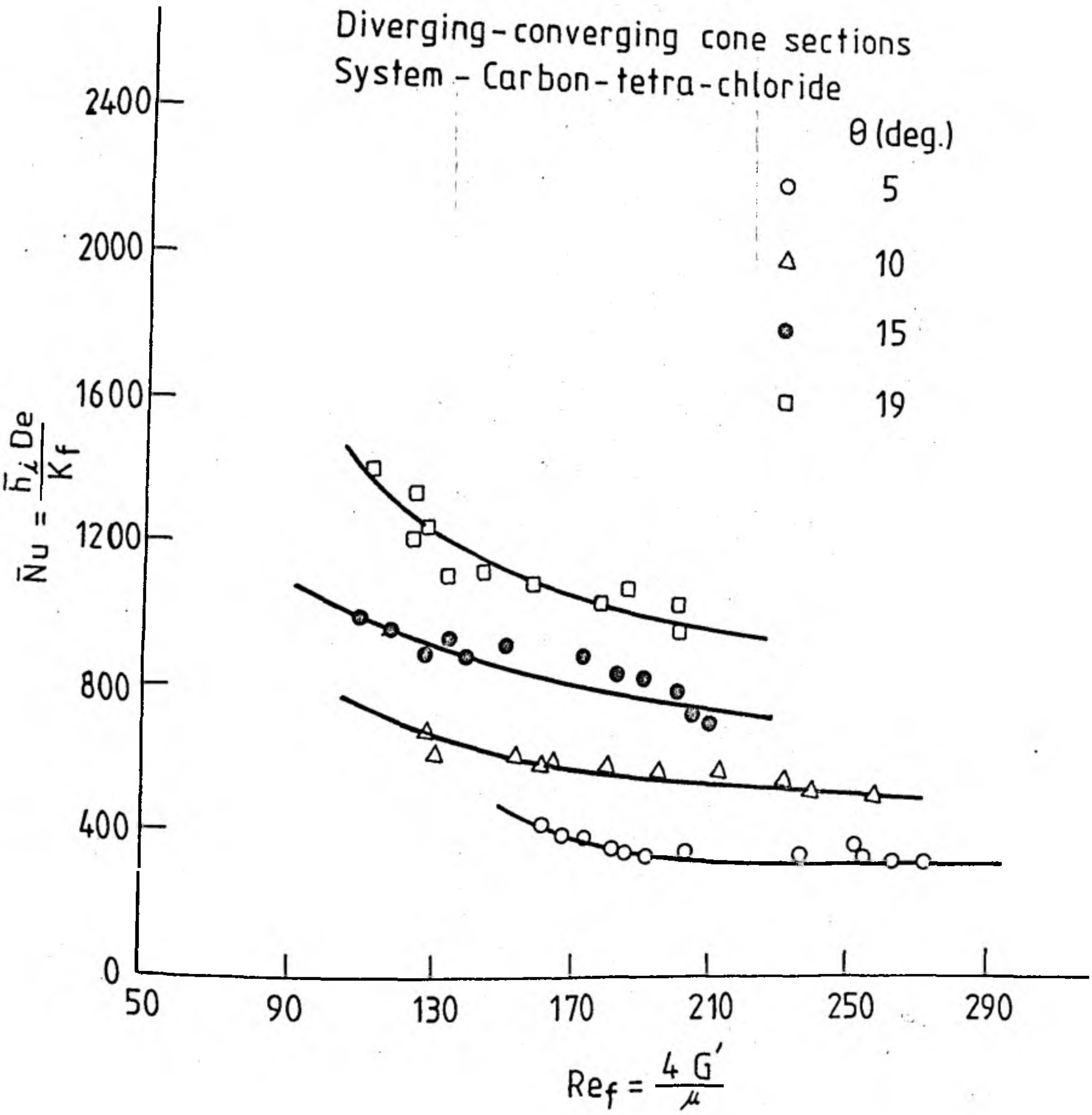


FIG.5.6-6. VARIATION OF Re_f WITH $\bar{N}u$ FOR CONDENSATION OF CTC VAPOUR IN DIVERGING-CONVERGING CONE SECTIONS

the basic trend is similar as it is evident from figure 5.6-3 to figure 5.6-6.

5.7 Correlation For Average Heat Transfer Data:

5.7.1 Relation Between $\bar{h}_i \left[\frac{\mu_f^2}{\rho_f^2 g\, k_f^3}\right]^{1/3}$ and Re_f.

From theoretical analysis of filmwise condensation in diverging, converging and diverging-converging cone sections we can arrive at a general relationship between $\bar{h}_i \left[\mu_f^2 / \rho_f^2 g\, k_f^3\right]^{1/3}$ (dimensionless) and condensate Reynolds number.

From equations (3.5.7), (3.5.15) and (3.5.23), following relationship is obtained.

$$\bar{h}_i \left[\frac{\mu_f^2}{\rho_f^2 g\, k_f^3}\right]^{1/3} = 1.585\ fc.\ Re_f^{-1/3} \qquad \ldots (5.7.1)$$

$$\text{where,}\quad fc = \left[\frac{(1-a)^{1/3}(1-a^{7/3})}{(1-a^2)^{4/3}} \cdot \cos\theta\right]^{2/3}$$

and, $a = r_1/r_2$

The factor $\bar{h}_i \left[\mu_f^2 / \rho_f^2 g\, k_f^3\right]^{1/3}$ and the Re_f have also been calculated from the experimental results and when plotted give a relationship of the form,

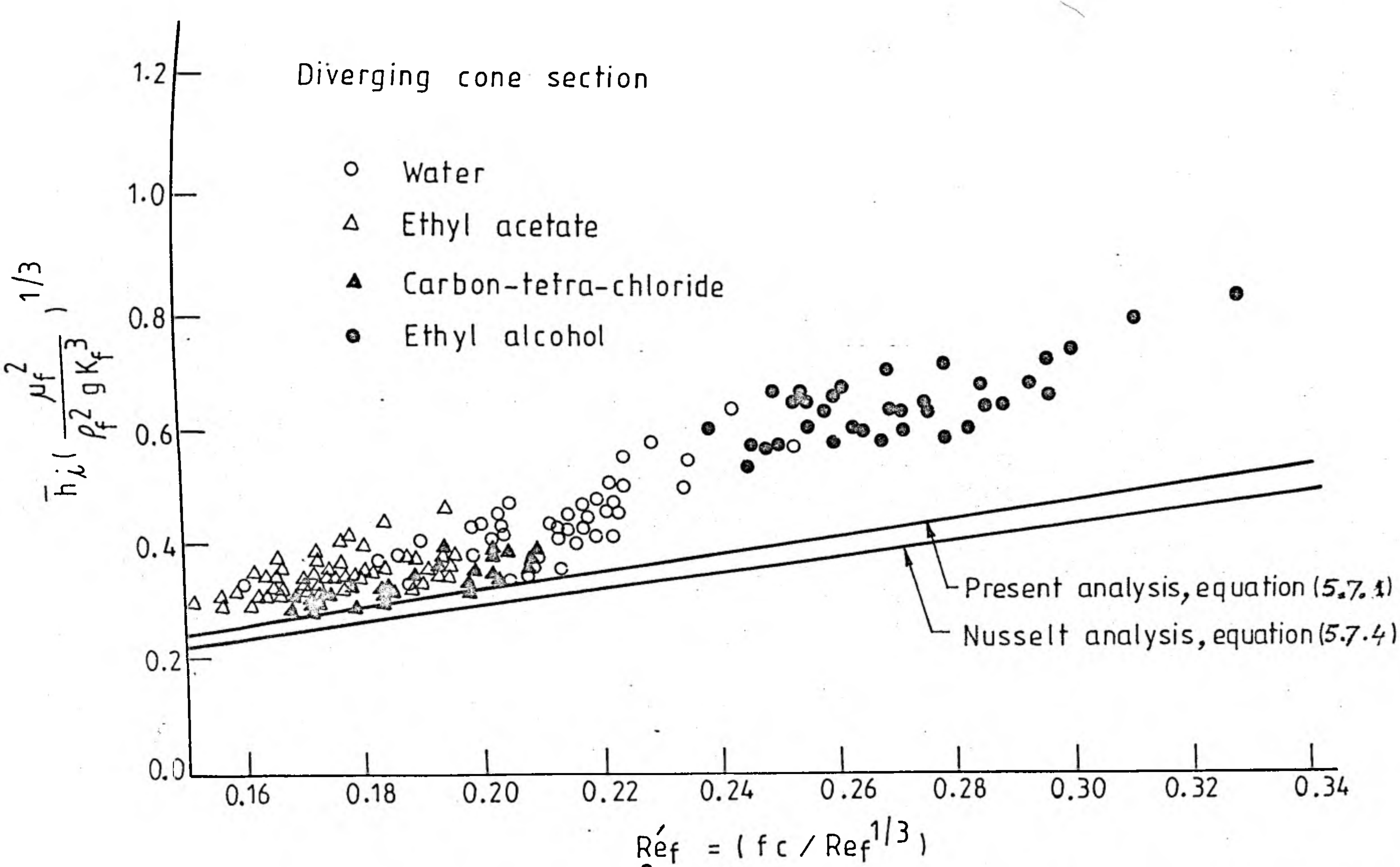


FIG.5.7-1.PLOT OF Re'_f VS $\bar{h}_l \left(\frac{\mu_f^2}{\rho_f^2 g K_f^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING CONE SECTIONS

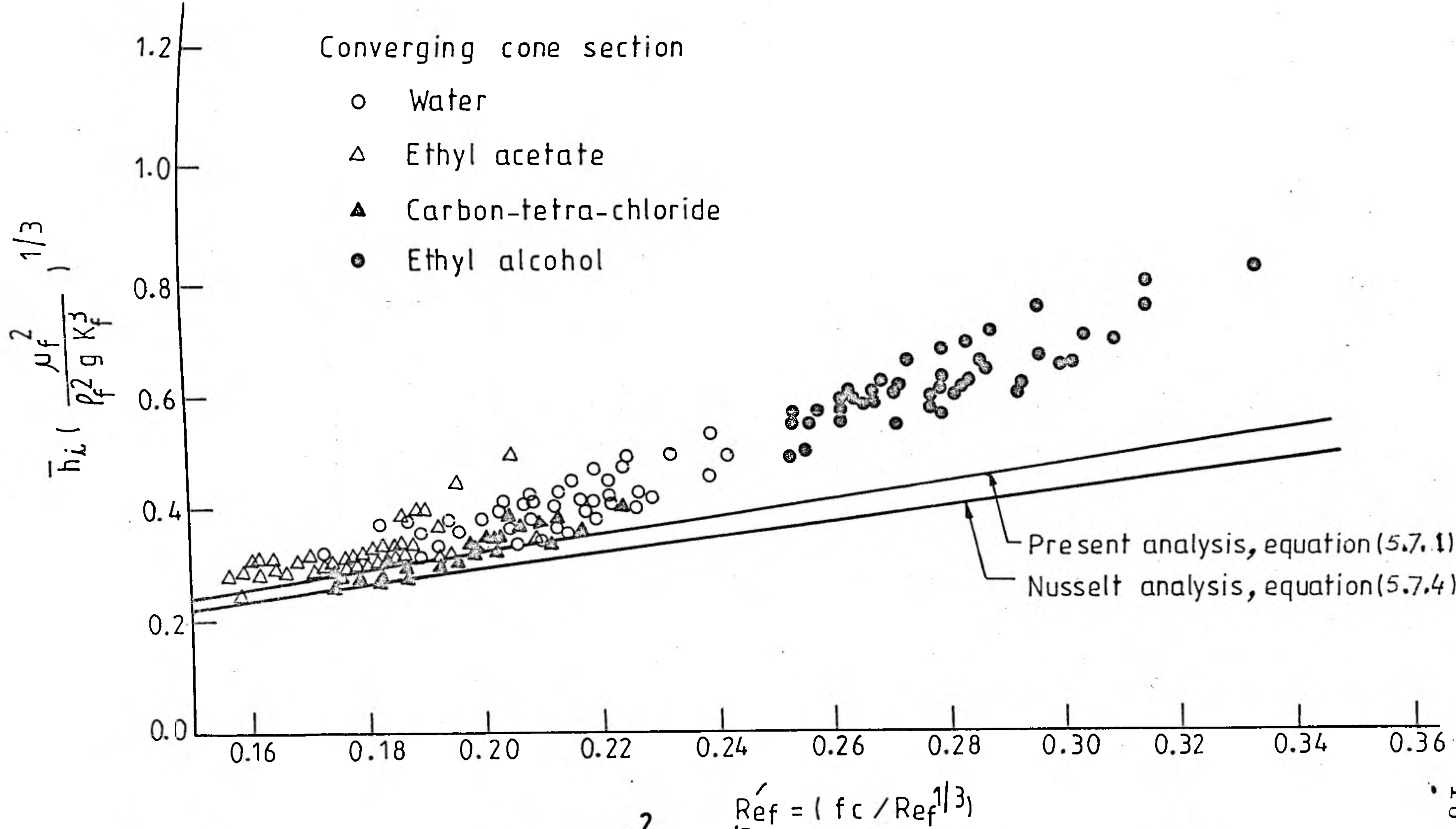


FIG.5.7-2. PLOT OF Re'_f VS $\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g K_f^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLDRIDE AND ETHYL ALCOHOL VAPOURS IN CONVERGING CONE SECTIONS

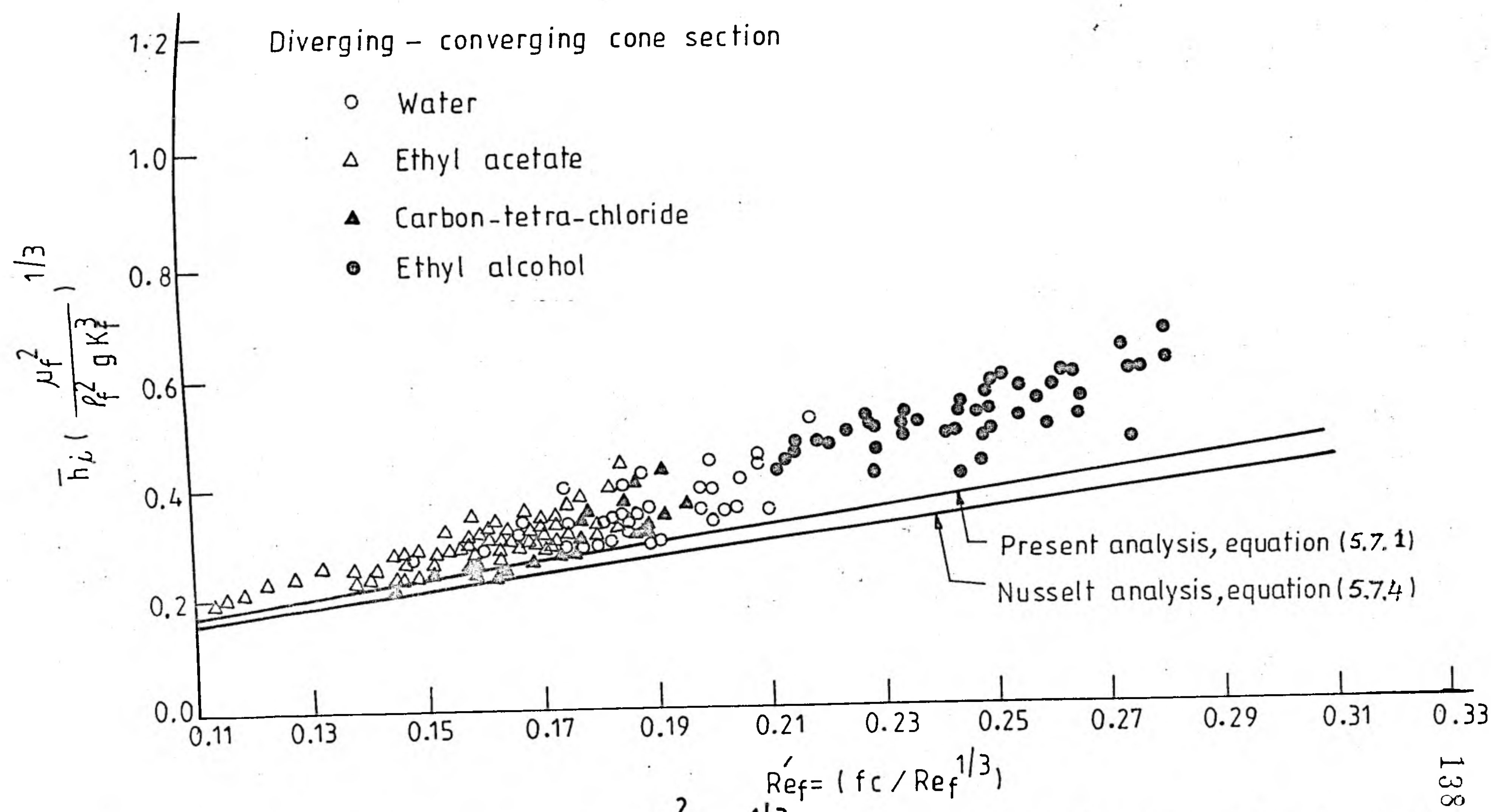


FIG. 5.7–3, PLOT OF Re_f' VS $\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g K_T^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING-CONVERGING CONE SECTIONS

$$\bar{h}_i \left[\frac{\mu_f^2}{\rho_f^2 g\, k_f^3}\right]^{1/3} = C'_{.}\, f_c .\, Re_f^{-1/3} \qquad \ldots (5.7.2)$$

where, C' is the slope.

The magnitude of the slope C' has been determined for the three cases viz. diverging, converging and diverging-converging cone sections, by the method of least squares using HCL 2000 Computer.

Replacing C' by numerical figure, equation (5.7.2) gives the following expression which is common for all the three systems.

$$\bar{h}_i \left[\frac{\mu_f^2}{\rho_f^2 g\, k_f^3}\right]^{1/3} = 2.00\ fc.\ Re_f^{-1/3} \qquad \ldots (5.7.3)$$

The correlation coefficient for the above expression is 0.70.

Figure 5.7-1 to figure 5.7-3 have been plotted with experimental values of $\bar{h}_i\, [\mu_f^2/\rho_f^2\, g\, k_f^3]^{1/3}$ and $[fc/Re_f]^{1/3}$ for condensation of water, ethyl alcohol, ethyl acetate and carbon-tetra-chloride vapours in diverging, converging and diverging-converging cone sections respectively.

The lowest curve is the Nusselt's theoretical solution of filmwise condensation on a smooth vertical surface [157], which

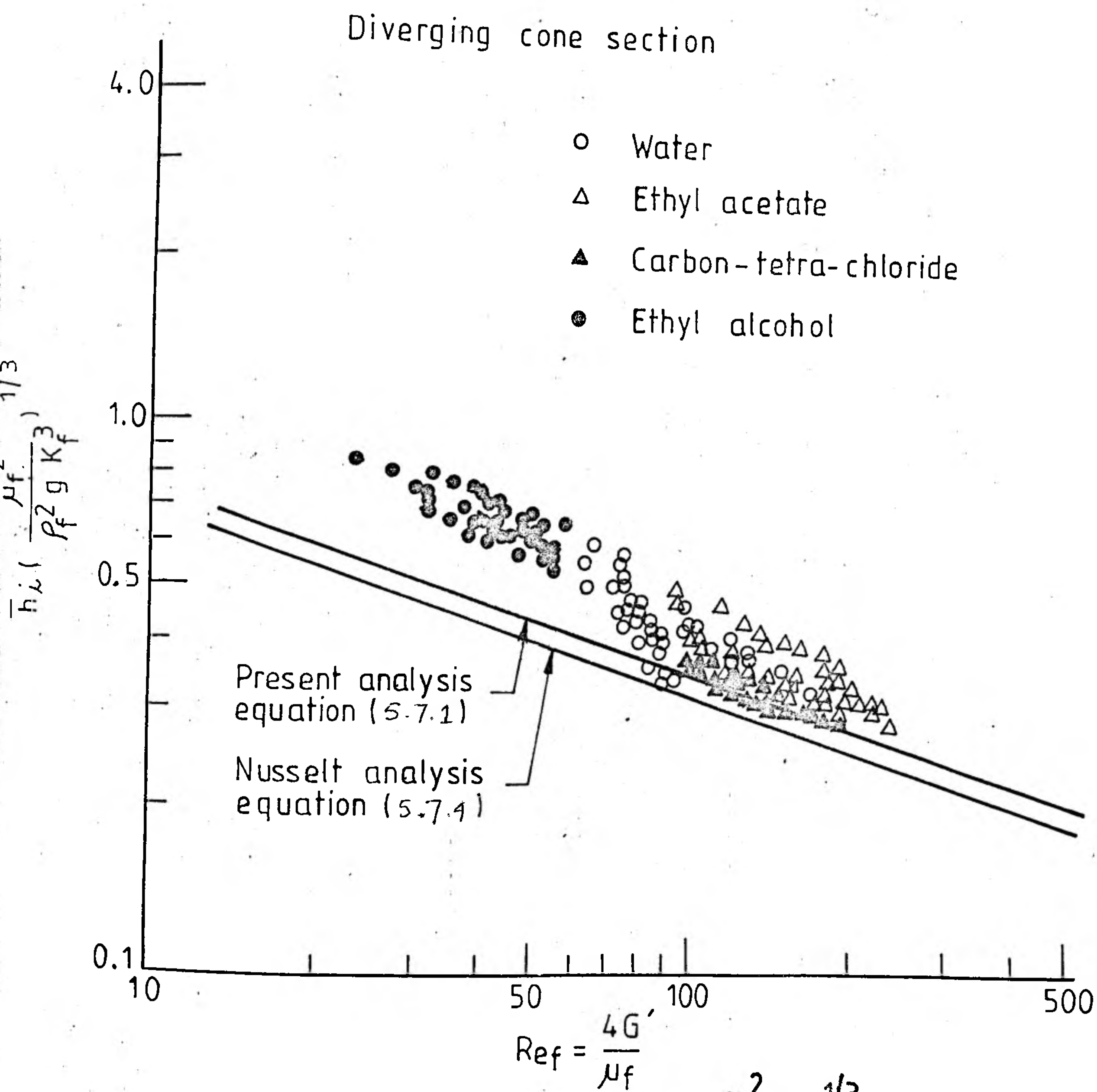


FIG. 5.7 - 4. VARIATION OF Re_f WITH $\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g K_f^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING CONE SECTIONS

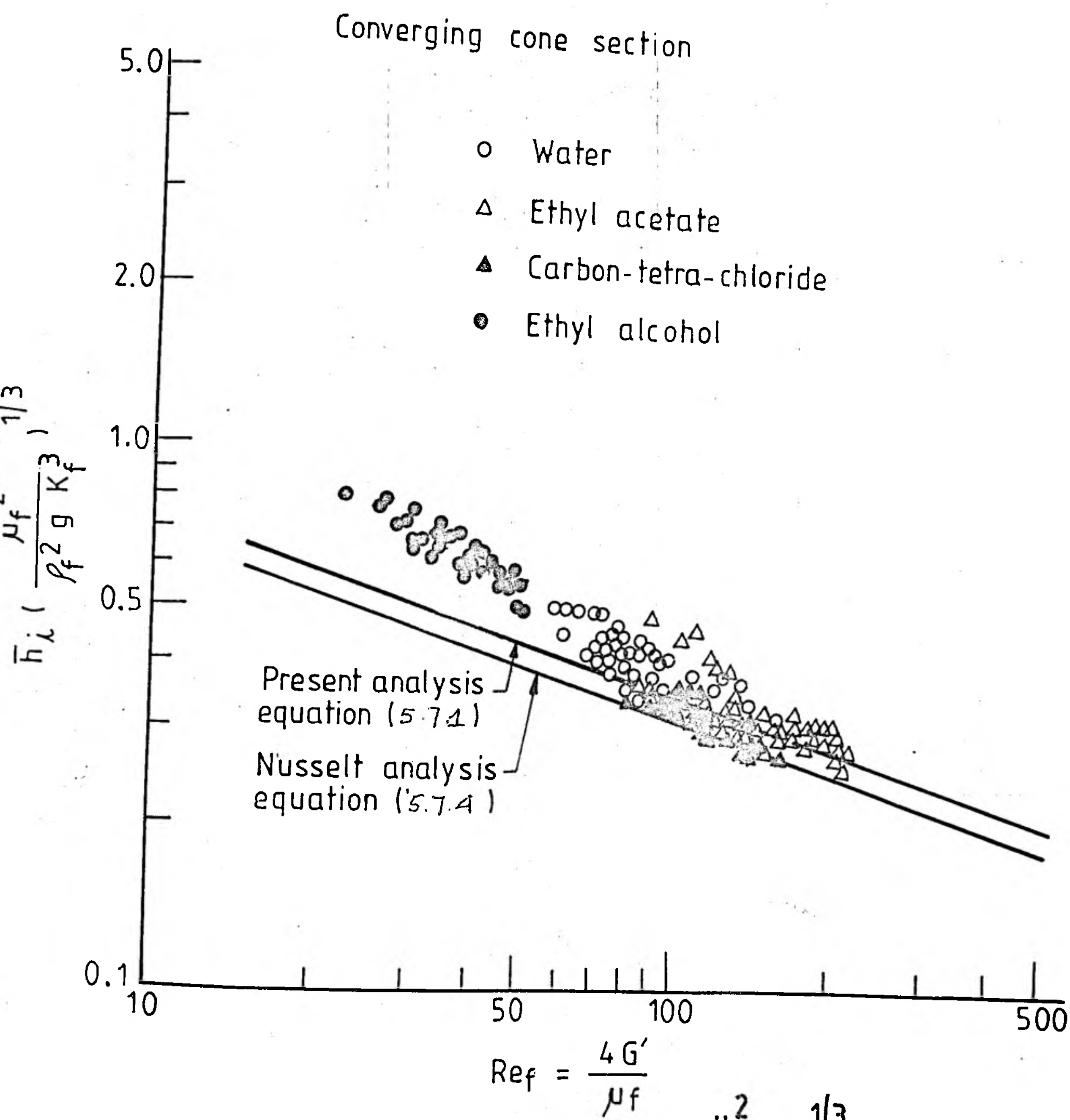


FIG.5.7-5.VARIATION OF Ref VS $\bar{h}_l \left(\frac{\mu_f^2}{\rho_f^2 g K_f^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN CONVERGING CONE SECTIONS

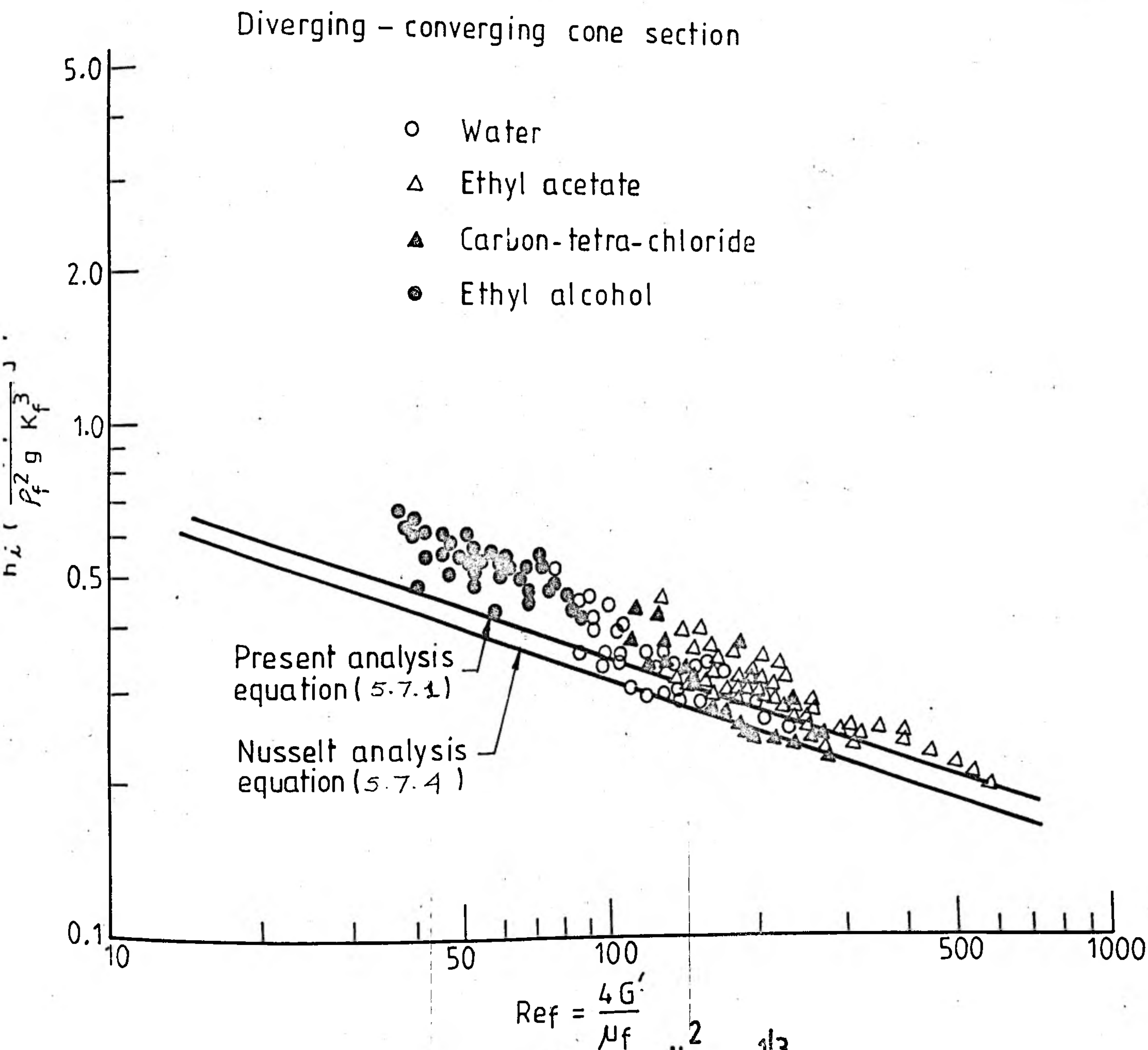


FIG.5.7-6.VARIATION OF Ref WITH $\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g K_f^3} \right)^{1/3}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING-CONVERGING CONE SECTIONS

gives relationship of the form,

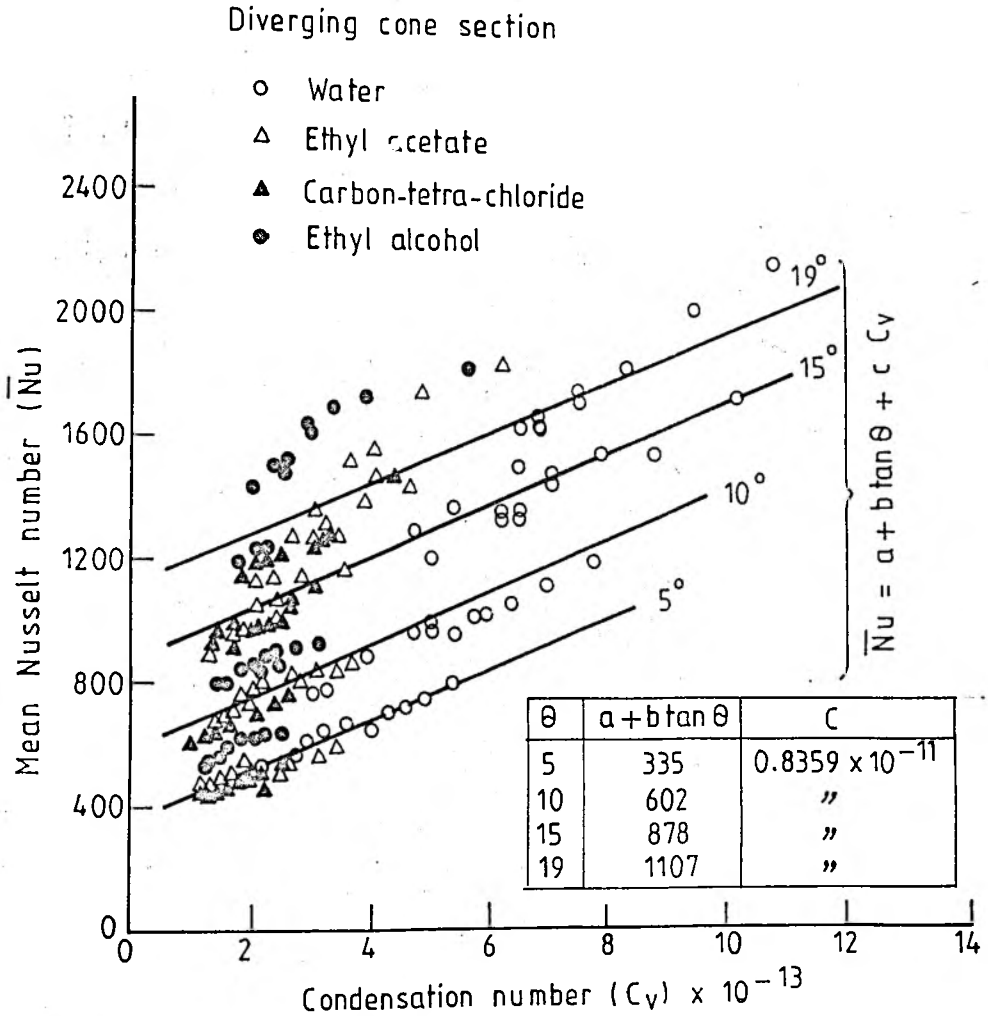
$$\bar{h}_i \left[\frac{\mu_f^2}{\rho_f^2 g \; k_f^3} \right]^{1/3} = 1.47 \; Re^{-1/3} \qquad \ldots (5.7.4)$$

The curve above the Nusselt's solution represents theoretical curve for constricted tubes, given by equation (5.7.1).

Figures 5.7-4 to 5.7-6 give the plot of experimental values of $\bar{h}_i \left[\mu_f^2 / \rho_f^2 g \; k_f^3 \right]^{1/3}$ and Re_f for condensation of water, ethyl alcohol, ethyl acetate and carbon-tetra-chloride vapours in diverging, converging and diverging-converging cone sections respectively. The lowest line in each figure is the Nusselt's theoretical solution of filmwise condensation on smooth vertical surface. The corresponding upper line represents the present theoretical solution of filmwise condensation in vertical constricted tubes.

5.7.2 Relation Between Mean Nusselt Number ($\overline{Nu}$) and Condensation Number (C_v).

The variation of various parameters and their effects have been shown in figure 5.1-1 to figure 5.7-6 for different situations to get a clear understanding of heat transfer behaviour of the diverging-converging tube system. Dimensional analysis has been carried out (Appendix-V) to get a relationship incorporating all the influencing parameters. From dimensional analysis it has been observed that a relationship between $\overline{Nu}$ and



θ	a+b tan θ	C
5	335	0.8359×10^{-11}
10	602	"
15	878	"
19	1107	"

FIG.5.7-7. PLOT OF C_v VS $\overline{Nu}$ FOR CONDENSATION OF WATER, ETHYL ACETATE CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING CONE SECTIONS

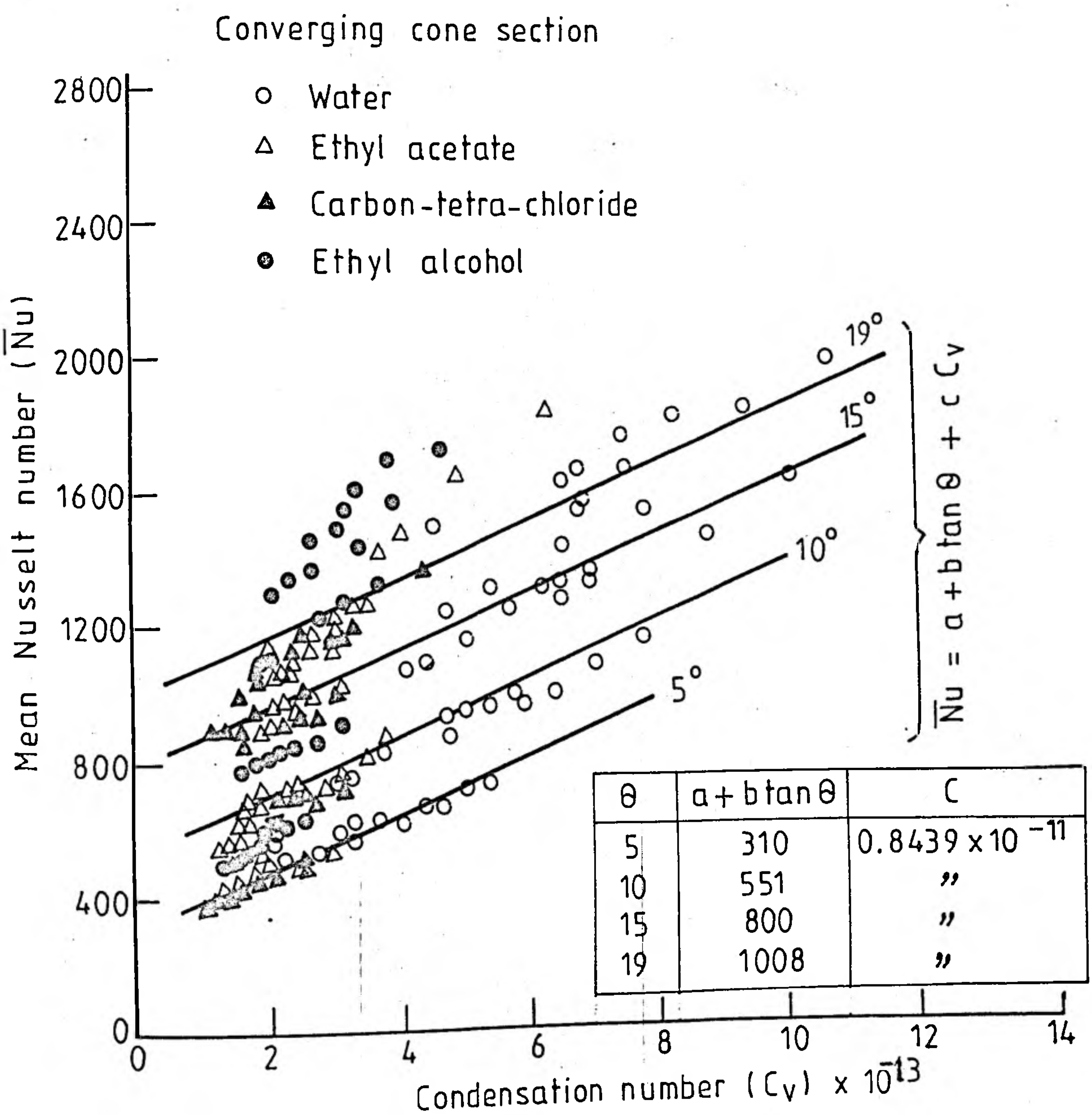


FIG.5.7-8. PLOT OF C_V VS $\overline{Nu}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN CONVERGING CONE SECTIONS

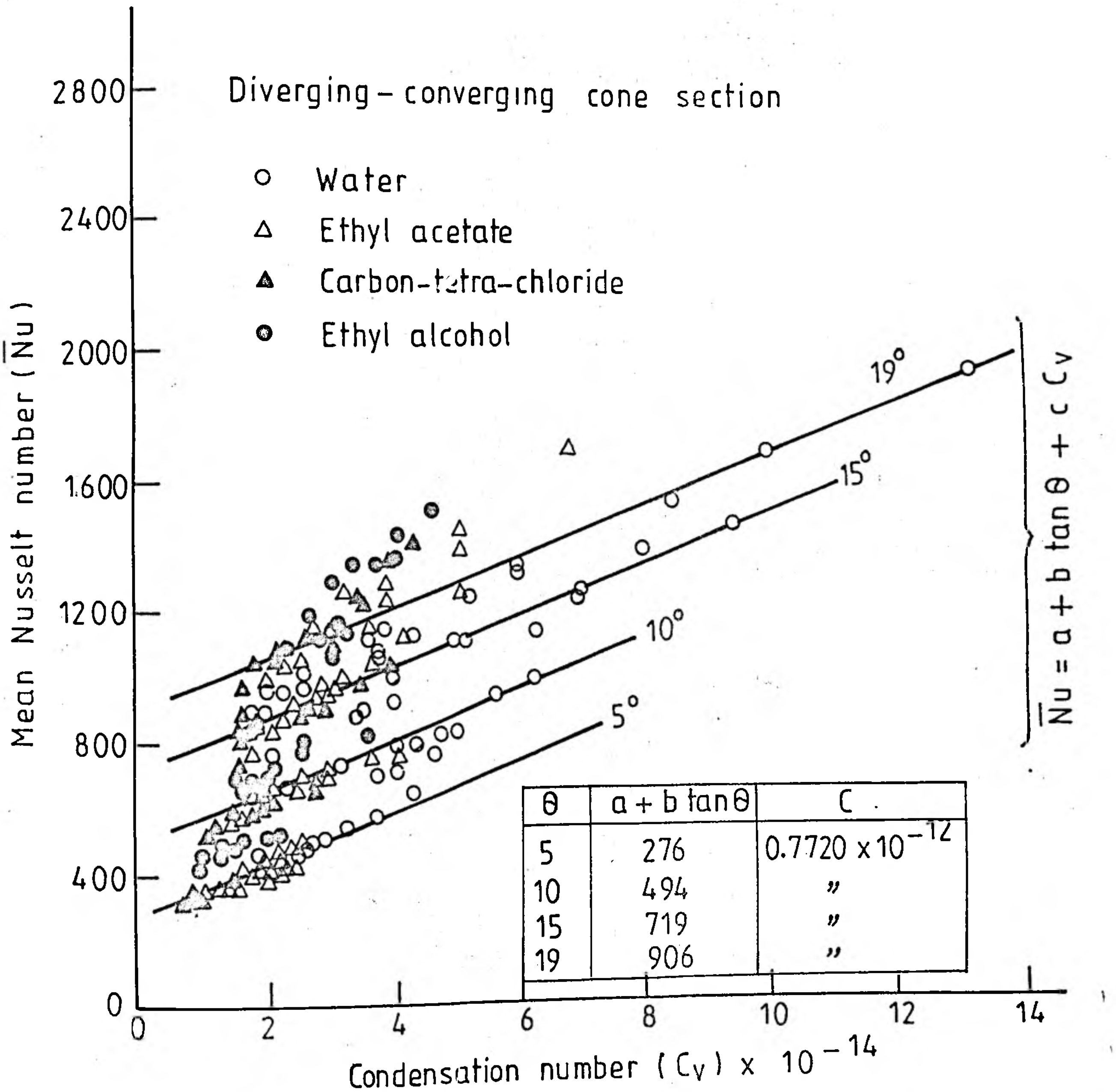


FIG. 5.7-9. PLOT OF C_V VS $\overline{Nu}$ FOR CONDENSATION OF WATER, ETHYL ACETATE, CARBON-TETRA-CHLORIDE AND ETHYL ALCOHOL VAPOURS IN DIVERGING - CONVERGING CONE SECTIONS

C_v can define the system well.

Figure 5.7-7 to figure 5.7-9 show plots of $\overline{Nu}$ v/s C_v for condensation of water, ethyl acetate, ethyl alcohol and carbon-tetra-chloride vapours in diverging, converging and diverging-converging cone sections respectively. Observing the trend of the graphs a generalized correlation of the following nature has been suggested,

$$\overline{Nu} = a + b\tan\theta + cC_v \qquad \ldots (5.7.5).$$

where, (a + btan θ) represents the intercept and 'c' is the slope ; a, b and c are correlation constants. The values of these correlation constants have been determined by the method of least squares using a digital computer HCL BC2/82.

Equation (5.7.5) givesthe following expressions for different systems, where a, b, and c are substituted with numericals,

Diverging : $\overline{Nu} = 72 + 3007\tan\theta + 0.8359 \times 10^{-11}\cdot C_v$ ...(5.7.6)

Converging: $\overline{Nu} = 72 + 2717\tan\theta + 0.8439 \times 10^{-11}\cdot C_v$ ...(5.7.7)

Diverging-Converging: $\overline{Nu} = 61 + 2454\tan\theta + 0.7720 \times 10^{-12}\cdot C_v$ ...(5.7.8)

The correlation coefficientsfor the above expressions (5.7.6), (5.7.7) and (5.7.8) are 0.92, 0.92 and 0.96 respectively.

5.8 Comparison Of Condensation Performance:

Figure 5.8-1 and figure 5.8-2 show the comparison of

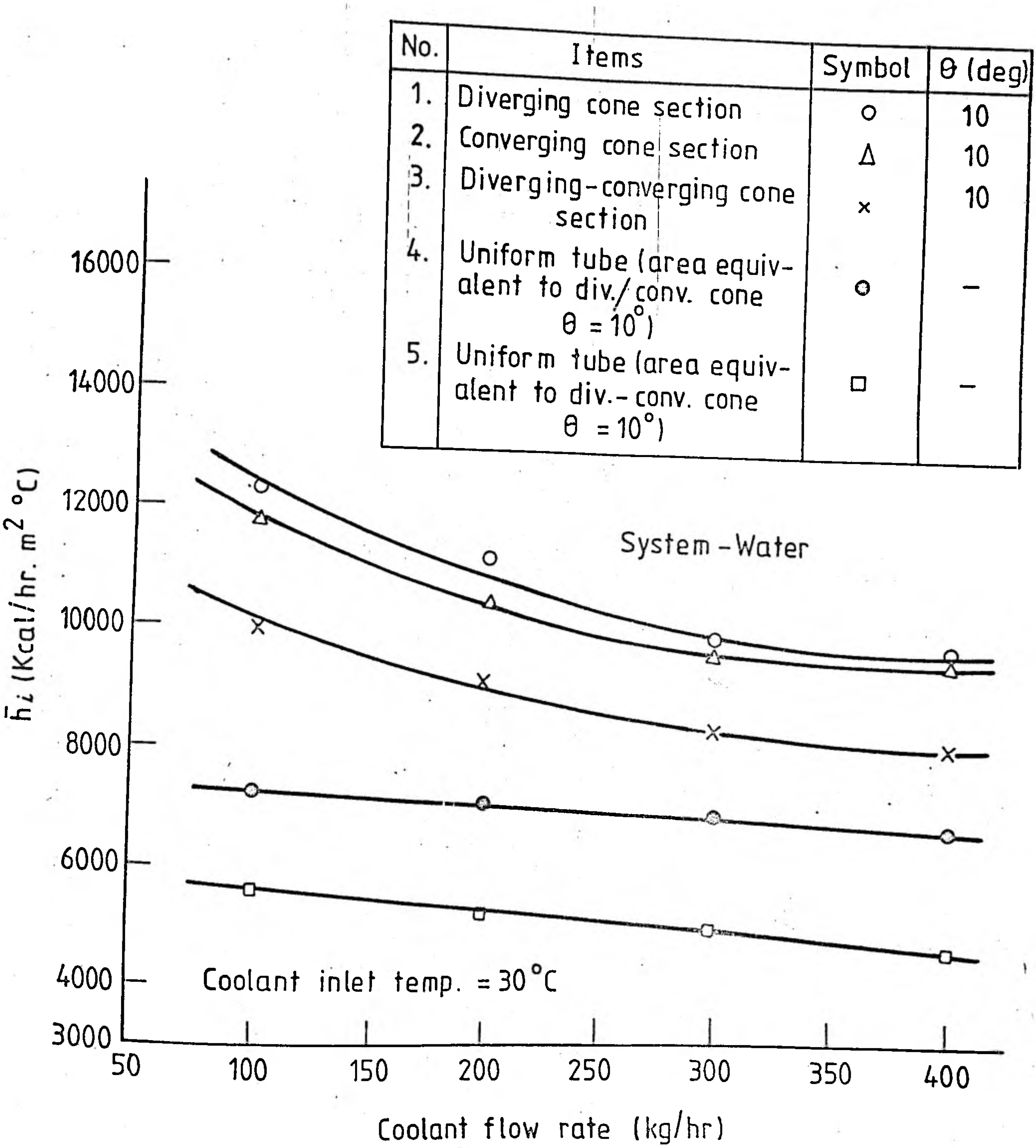


FIG. 5.8 -1. COMPARISON OF PERFORMANCE OF DIVERGING-CONVERGING CONE SYSTEMS WITH THAT OF STRAIGHT UNIFORM TUBE HAVING SAME HEAT TRANSFER AREA AND LENGTH (FOR COOLANT INLET TEMP. = 30°C, CONE ANGLE $\theta = 10°$ AND SYSTEM = WATER)

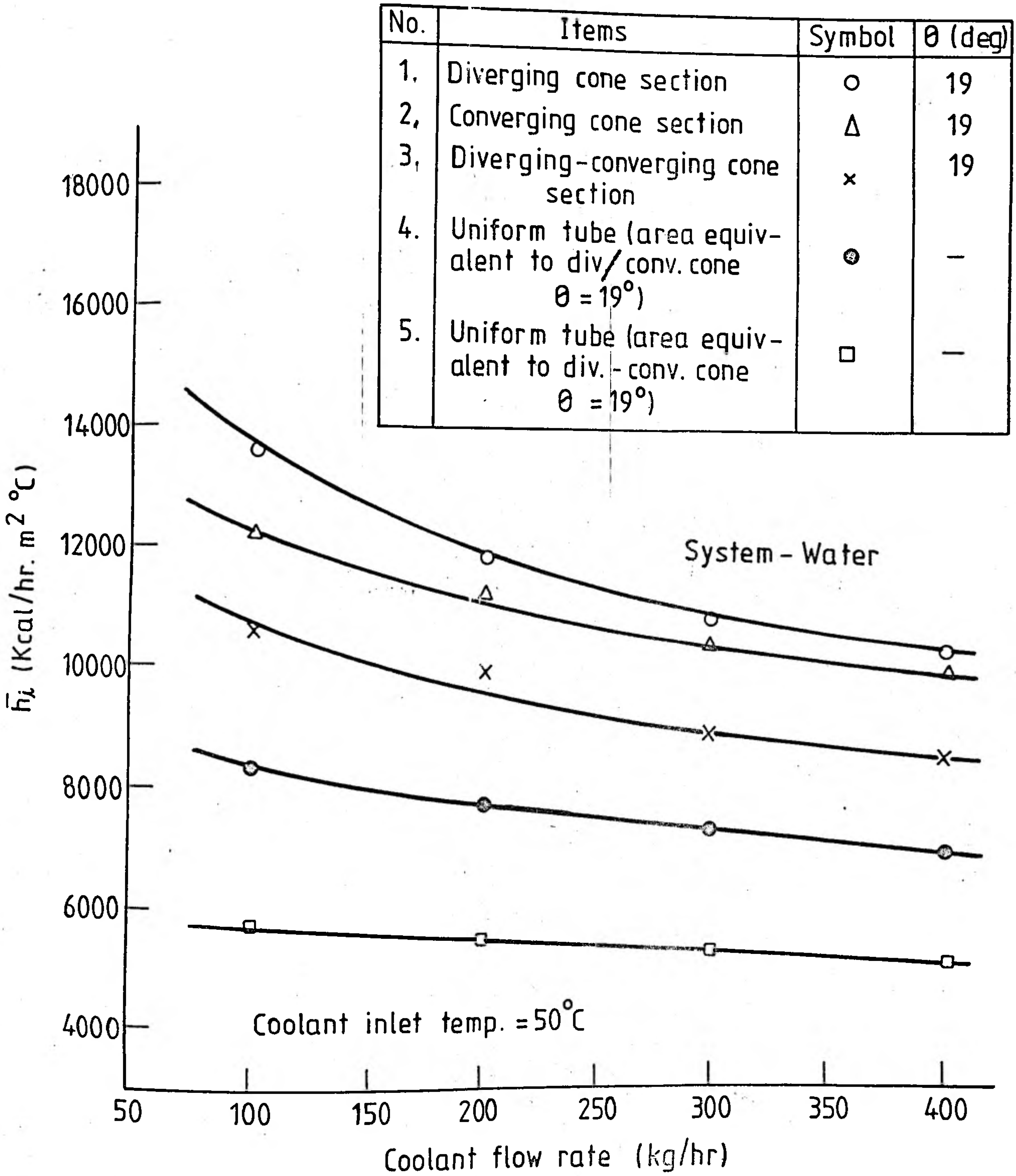


FIG.5.8-2. COMPARISON OF PERFORMANCE OF DIVERGING-CONVERGING CONE SYSTEMS WITH THAT OF STRAIGHT UNIFORM TUBE HAVING SAME HEAT TRANSFER AREA AND LENGTH. (FOR COOLANT INLET TEMP. = 50°C, CONE ANGLE θ = 19° AND SYSTEM = WATER)

condensation performance of diverging, converging and diverging-converging cone sections, for condensation of water vapour at coolant inlet temperature 30°C (for $\theta = 10°$) and 50°C (for $\theta = 19°$) respectively. To compare the performance of the constricted tube system with that of straight uniform tube, experiments were conducted with straight uniform tube having same heat transfer area and length (or height). This was discussed in details in Chapter-4. To keep the heat transfer area and length same, the diameters of the straight tubes were changed according to the equivalent diameters of the different cone sections. For testing the performance, three straight tubes with area equivalent to cones having $\theta = 10°$, 15° and 19° and three liquid systems viz. water, ethyl acetate and carbon-tetra-chloride were used.

The dimensions of the corresponding shells were kept same for both cone and straight tube test condenser sections during the experiment in order to nullify the effect that may be caused due to change in shell dimensions.

From figures 5.8-1 and 5.8-2 it is evident that the average heat transfer coefficient of the diverging cone sections is higher than that of the converging cone sections. The variation has been estimated to be 2 to 10 percent on the basis of the same heat transfer area and length (height) for most of the data points.

It will be seen that the average heat transfer coefficient for converging cone sections is 5 to 15 percent higher, for most of the data points, over that obtained for diverging-converging cone sections.

It is further observed from the plots that change of configuration from uniform cylindrical tube to diverging-converging tube system increases the average condensation heat transfer coefficient. The degree of augmentation is about 40% to 80% for most of the data points.

Similar trend was observed in case of condensation of other vapour systems and at other coolant inlet temperatures. The difference was only in the magnitude of $\bar{h}_i$ values. The temperature difference (ΔT_f) between film and wall plays an important role in enhancing the condensation heat transfer coefficient in case of diverging-converging systems. Figure 5.8-3 shows that ΔT_f values of equivalent straight tubes are higher than that of the diverging cone sections, with other conditions remaining same. This is true for converging and diverging-converging cone sections also.

In case of vertical uniform cylindrical tubes the flow of film condensate is more or less unidimensional whereas in diverging/converging system the flow is expected to be two-dimensional in nature, due to its configuration. As a result of which, in case of diverging cone, the film thickness gets spreaded affecting the boundary layer formation. This reduces the film thickness and thus the heat transfer coefficient is increased. Surface tension is also expected to play an important role here in reducing the film thickness.

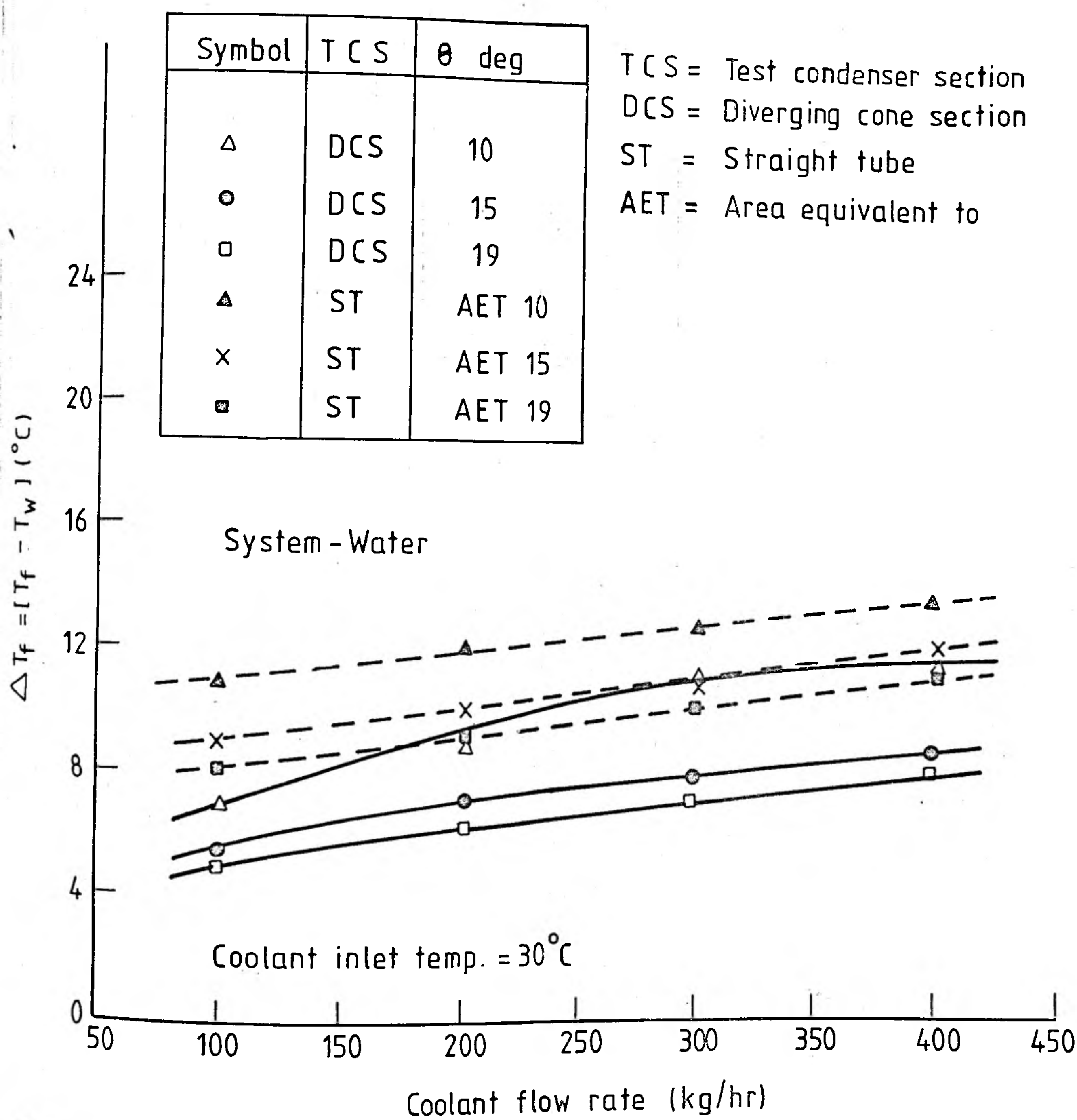


FIG.5.8-3. VARIATION OF ΔT_f WITH COOLANT FLOW RATE FOR CONDENSATION OF WATER VAPOUR IN DIVERGING CONE SECTIONS AND STRAIGHT TUBES WITH EQUIVALENT AREAS

Converging cone section on the other hand, due to its configuration, is expected to behave in a manner opposite to diverging cone. As the condensate film flows downward along the converging surface, flow area decreases and film thickness increases. For this reason converging cone section gives less heat transfer coefficient and is less efficient compared to the diverging cone section. But due to the contraction of the converging surface, film flow disturbs boundary layer formation unlike uniform straight tube. This phenomenon is probably responsible for getting high heat transfer coefficient in converging cone section compared to that achieved through straight tube under similar operating conditions.

5.9 Effects of Multiple Diverging-Converging Units In Series:

As mentioned earlier in Chapter-4, to establish the practical suitability of the proposed system, experiments were carried out with tubes containing one, two, three and four diverging-converging units. The detailed informations have been given in Table AII-4. These experiments were carried out to find out the behaviour of these diverging-converging units when joined together in series. The construction was such that the condensate from the upper diverging-converging unit did not fall on the surface of the lower units. This was done by projecting the lower portion of each diverging-converging unit inside the next lower unit (Figure 4.5).

With this modification it has been observed experimentally

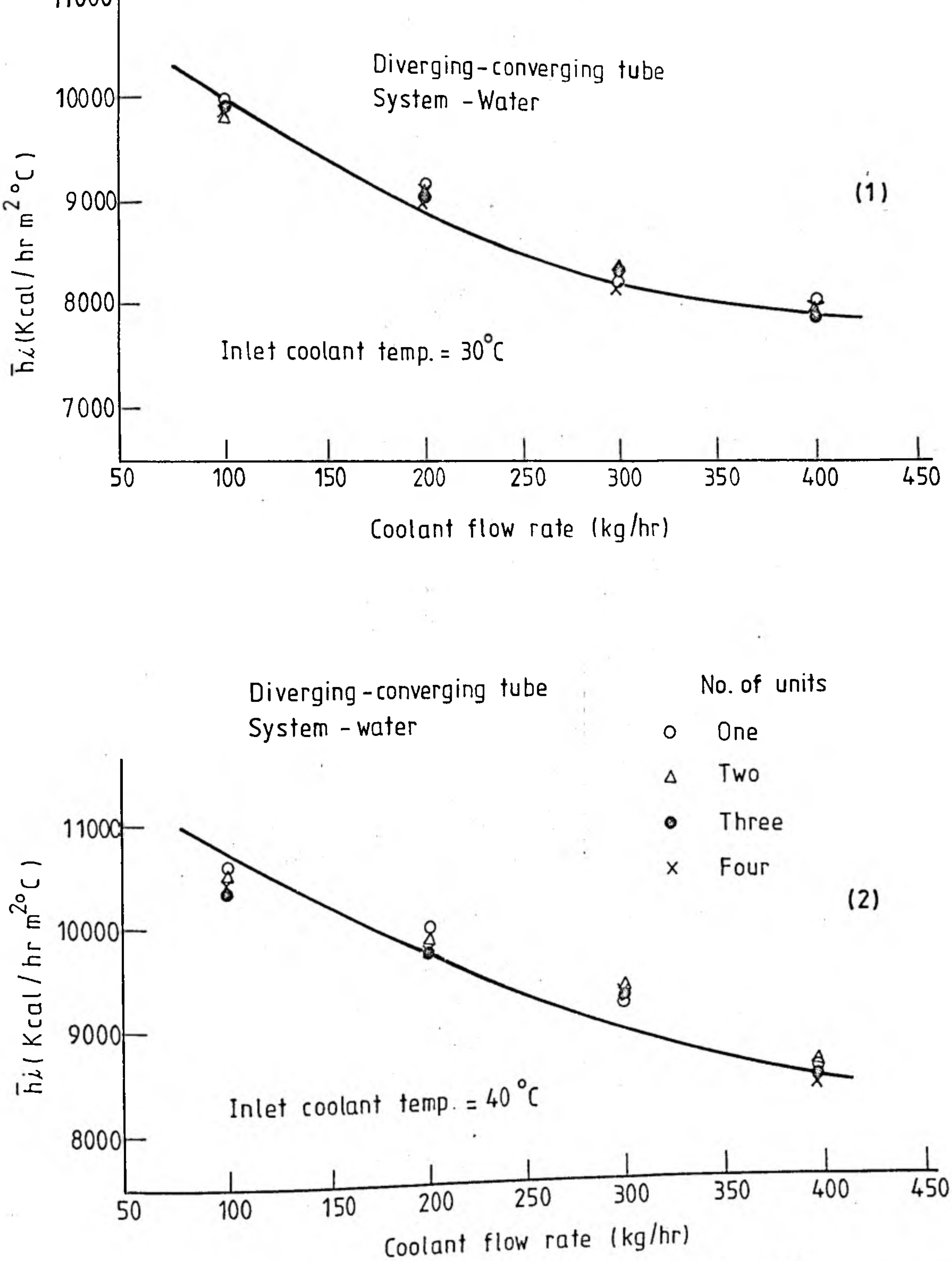


FIG. 5.9. EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOUR IN DIVERGING-CONVERGING TUBES, COOLANT INLET TEMPERATURE (1) = 30°C & (2) = 40°C
(1&2)

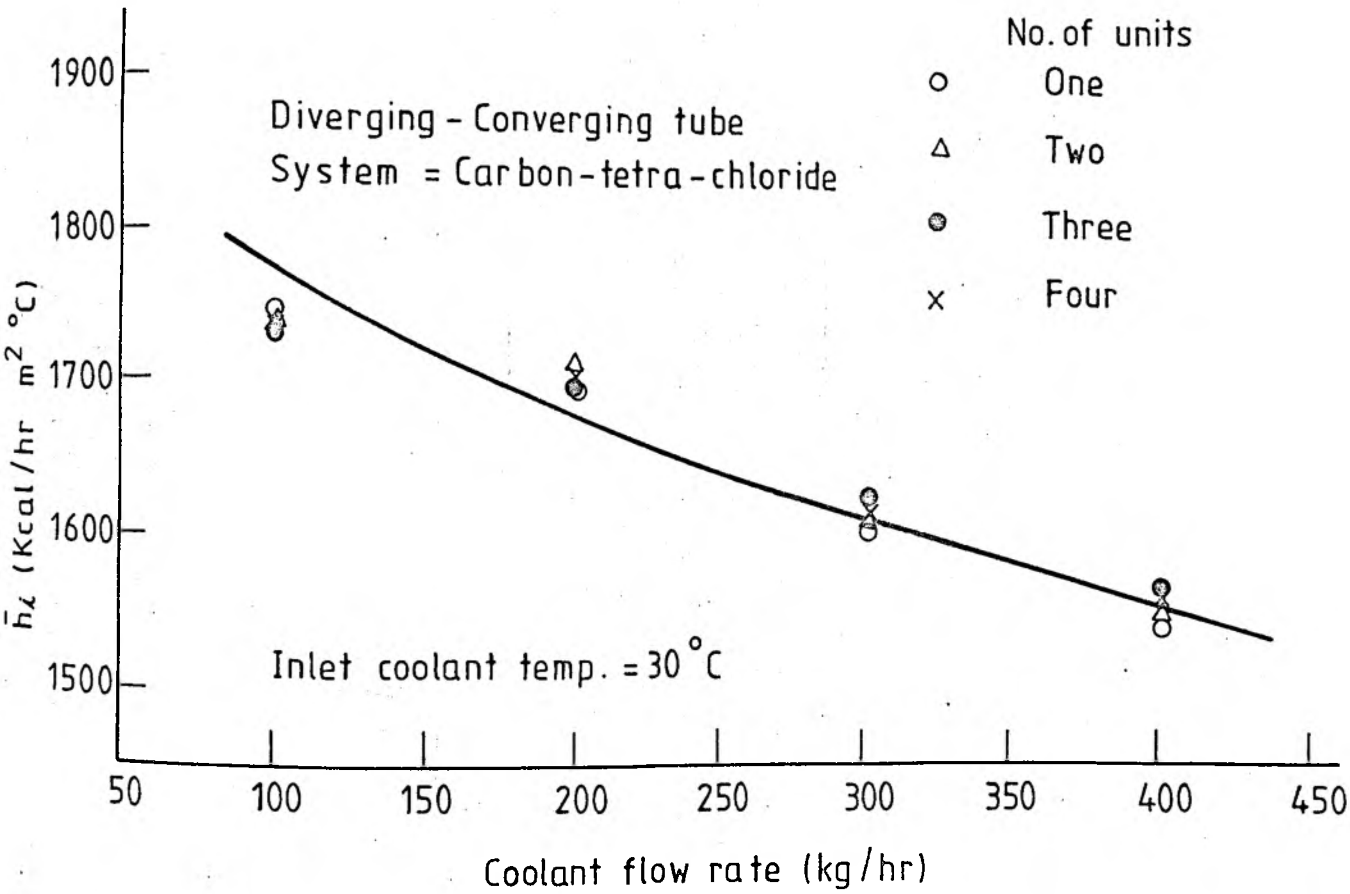


FIG.5.9-3. EFFECT OF COOLANT FLOW RATE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF CTC VAPOUR IN DIVERGING-CONVERGING TUBES, COOLANT TEMPERATURE = 30 °C

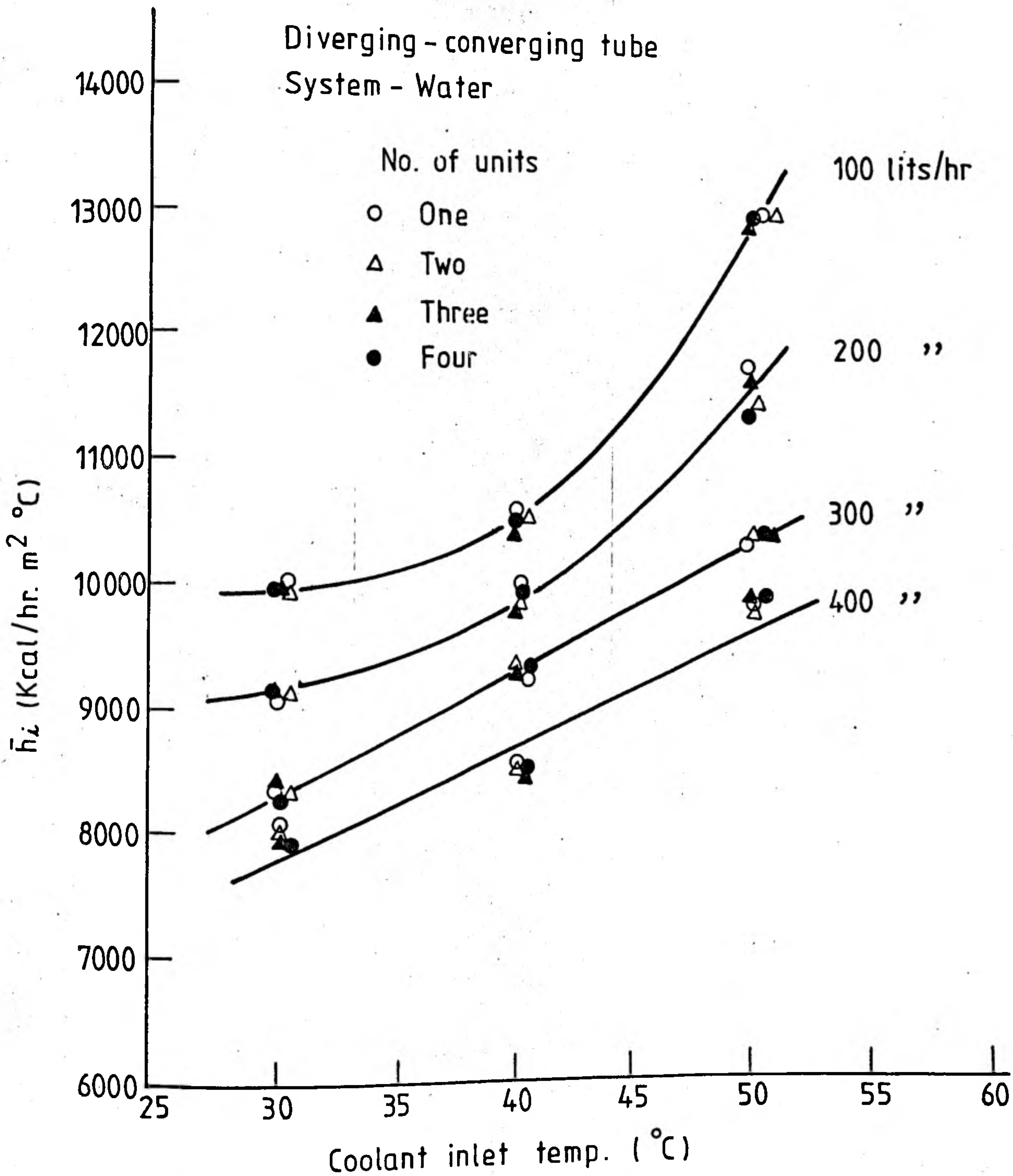


FIG.5.9-4. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOUR IN DIVERGING-CONVERGING TUBES

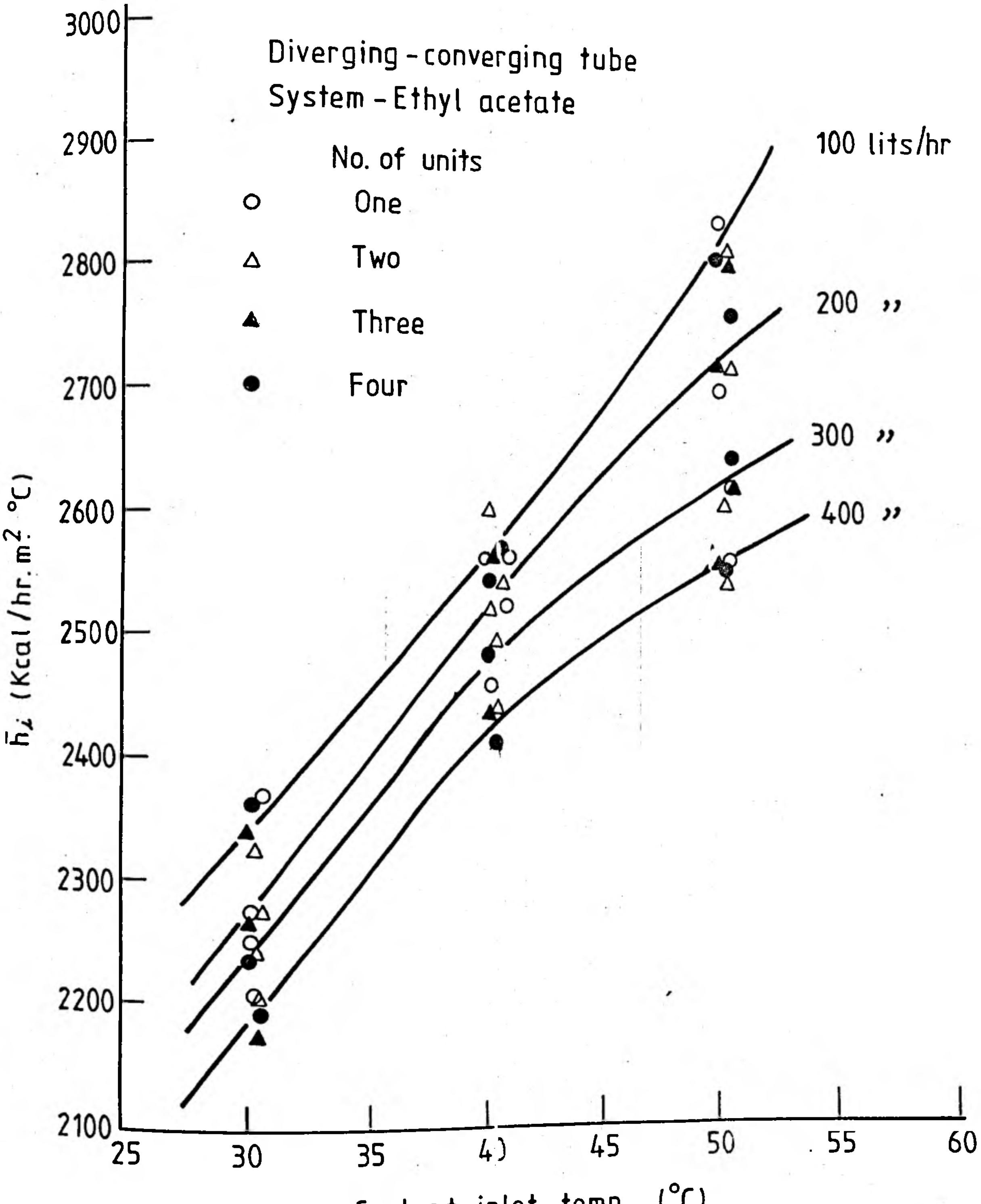


FIG.5.9-5. EFFECT OF COOLANT INLET TEMPERATURE ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF ETHYL ACETATE VAPOUR IN DIVERGING-CONVERGING TUBES

that the number of units in series does not have any significant influence on the average condensation heat transfer coefficient, $\bar{h}_i$. This can be seen readily from figure 5.9-1 to figure 5.9-5.

To find out the effect of coolant inlet temperatures and ΔT_f on average heat transfer coefficient, experiments were carried out with three liquid systems viz. water, ethyl acetate and carbon-tetra-chloride. The effect of coolant inlet temperature on $\bar{h}_i$ for tubes containing one, two, three and four units of diverging-converging sections has been shown in figure 5.9-4 and figure 5.9-5 for condensation of water and ethyl acetate vapour at different flow rates as parameters.

Figures 5.9-6 and 5.9-7 show the effect of ΔT_f on average heat transfer coefficient for condensation of water and ethyl acetate vapours in constricted tubes having multiple units. As expected, $\bar{h}_i$ decreases with the increase in ΔT_f values.

So a tube with augmented heat transfer and having practically no influence of length (or height) on the heat transfer characteristics, is of immense practical importance and poised for future development of heat transfer equipment.

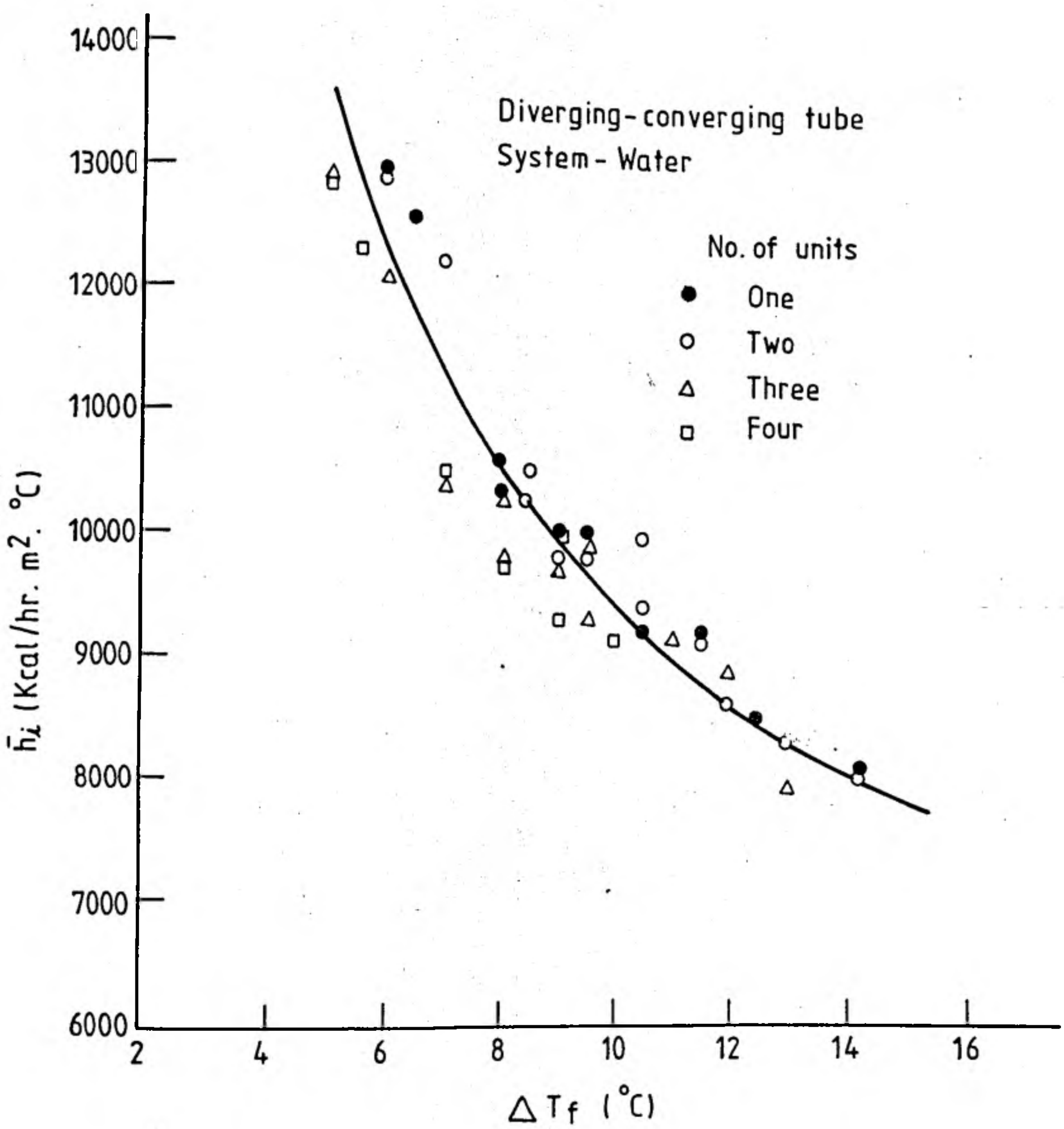


FIG.5.9-6. EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF WATER VAPOUR IN DIVERGING-CONVERGING TUBES

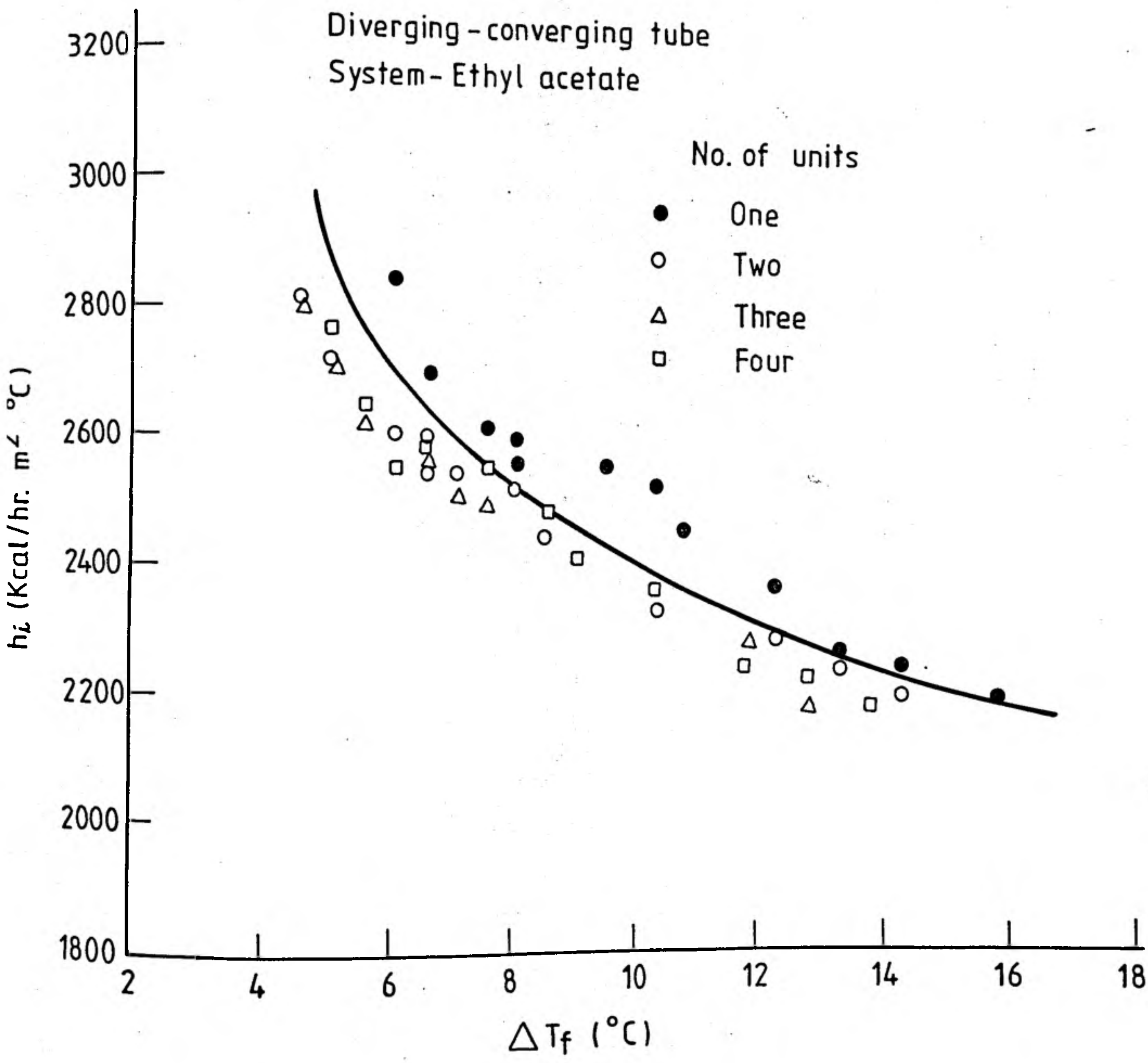


FIG.5.9-7. EFFECT OF ΔT_f ON HEAT TRANSFER COEFFICIENT FOR CONDENSATION OF ETHYL-ACETATE VAPOUR IN DIVERGING-CONVERGING TUBES

CHAPTER - 6

CONCLUSION

CHAPTER - 6

CONCLUSION

In the present investigation attempts have been made to shed sufficient light on the general heat transfer characteristics for steady, laminar condensation of pure vapours with negligible surface tension effects in constricted geometry, namely, diverging, converging and diverging-converging cone sections.

To sum up:

1) Mathematical models have been developed for condensation of pure vapours in diverging, converging and diverging-converging cone systems to determine average velocity and film thickness of the condensate, total heat flow across the condensate layer, local and average heat transfer coefficients.

2) Experiments were carried out with diverging, converging and diverging-converging cone sections to check the reliability of these mathematical models. The experimental results are found in good agreement with the theoretical predictions.

3) The heat transfer characteristics of the system are found to be a strong function of coolant flow rate, coolant inlet temperature, ΔT_f, fluid properties and system configuration.

4) Experiments were carried out with four cone angles, θ = 5, 10, 15 and 19 degrees. It has been found that for the same heat transfer length (or height) the average heat transfer coefficient increases with increase in cone angles. This is true for all systems, i.e., diverging, converging and diverging-converging systems.

5) The average condensation heat transfer coefficient for diverging cone system is higher than that for the converging cone system. The variation is about 2 percent to 10 percent, on the basis of same heat transfer area and length (or height) for most of the data points.

6) The average condensation heat transfer coefficients for converging cone system are higher than that for the diverging-converging cone system. The variation is about 5 percent to 15 percent for most of the data points.

7) From the experimental investigations it has been observed that the temperature difference (ΔT_f) between wall and film plays an important role in the process of condensation and the accuracy of the experiment depends upon how accurately the condensing surface temperature are measured.

8) Experiments were carried out with uniform cylindrical tubes to compare the performance of straight tubes with that of diverging-converging tube systems. It has been

observed that change of configuration from cylindrical tubes to diverging-converging system having the same heat transfer area and length, increases the average condensation heat transfer coefficient. The degree of augmentation is about 40 percent to 80 percent for most of the data points.

9) The experimental data could be correlated by equations (5.7.3) & (5.7.6); (5.7.3) & (5.7.7); and (5.7.3) & (5.7.8) for condensation of pure vapour in diverging, converging and diverging-converging tube systems respectively.

10) It is observed experimentally that the condensation heat transfer coefficient can be made more or less independent of total heat transfer length (or height) by minor modification in the construction.

11) In the construction of heat exchangers, if uniform tube is replaced by constricted tube, the higher efficiency of the system can be achieved.

On the whole it may be stated that the field is quite fertile for future investigations, as the constricted geometry exhibits encouraging advantages for the construction of the heat transfer equipment. The increase in fabrication cost for such geometry is hoped to be compensated by its advantages. More elaborate studies could be conducted to device a well-planned procedure for optimizing the design parameters such as θ, r and H.

Studies could be extended to higher Reynolds number range and also to other test fluids having wide different physical, transport and thermodynamic properties. Work may also be carried out using non-Neutonian fluids as well as fluids with variable physical and transport properties.

APPENDICES

TABLE - A1-1

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
1	100 (99.568)	5°	30.0	46.4	16.4	80	93	10
2	200 (199.136)		30.1	39.6	9.5	74	90	13
3	300 (298.70)		30.1	37.3	7.2	70	89	15
4	400 (398.27)		30.3	35.9	5.6	68	87	16
5	100 (99.568)	10°	31.0	49.0	18.0	86	94	7
6	200 (199.136)		30.5	41.0	10.5	82	93	9
7	300 (298.70)		30.5	37.9	7.4	78	92	11
8	400 (398.27)		30.0	35.8	5.8	77	90	11.5
9	100 (99.568)	15°	30.0	50.0	20.0	88.5	95	5.75
10	200 (199.136)		30.5	41.2	10.7	86	94	7.00
11	300 (298.70)		30.1	37.8	7.7	84	92	8.00
12	400 (398.27)		30.0	36.0	6.0	83	91	8.5
13	100 (99.568)	19°	29.8	52.6	22.8	90	96	5.0
14	200 (199.136)		30.0	42.5	12.5	87.5	95.5	6.25
15	300 (298.70)		30.0	38.5	8.5	86	94	7.0
16	400 (398.27)		30.2	37.1	6.9	84	92	8.0

TABLE - A1-1 (Contd.)

Serial No.	Average Condensate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat Flux($\bar{q}$) 10^{-4} Kcal/hrm^2	$\bar{h}_{id}$ Expt. $\frac{\text{Kcal}}{\text{hr.m}^2.\ °\text{C}}$	$\bar{h}_{id}$ Theo. $\frac{\text{Kcal}}{\text{hr.m}^2.\ °\text{C}}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
1	3.045	1641	1633	11.04	11043	9336
2	3.645	1964	1892	13.21	10167	8744
3	4.007	2160	2150	14.53	9690	8436
4	4.195	2260	2230	15.21	9505	8302
5	3.339	1800	1792	8.62	12313	10190
6	3.898	2100	2090	10.05	11172	9570
7	4.177	2250	2210	10.77	9794	9102
8	4.305	2320	2309	11.11	9660	9001
9	3.863	2082	1991	7.32	12723	10634
10	4.167	2246	2130	7.89	11273	10123
11	4.330	2334	2300	8.20	10251	9791
12	4.51	2430	2390	8.54	10045	9644
13	4.246	2287	2270	6.78	13560	10996
14	4.631	2492	2489	7.40	11821	10618
15	4.758	2558	2539	7.58	10834	10109
16	5.149	2777	2748	8.23	10269	9777

TABLE - A1-1 (contd.)

Serial No.	$h_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
1	0.375	122	0.188	609	3.22
2	0.345	146	0.177	561	2.68
3	0.329	160	0.172	534	2.15
4	0.322	168	0.169	524	2.01
5	0.418	95	0.204	967	5.02
6	0.379	111	0.194	877	3.90
7	0.332	119	0.160	769	3.19
8	0.318	123	0.188	759	3.05
9	0.432	80	0.217	1352	6,17
10	0.383	87	0.210	1198	5.07
11	0.348	90	0.208	1090	4.44
12	0.341	94	0.205	1068	4.17
13	0.460	75	0.222	1697	7.50
14	0.401	81	0.216	1480	6.51
15	0.368	84	0.213	1356	5.35
16	0.348	91	0.208	1285	4.70

TABLE - A1-2

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Average Coolant Temperature Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
17	100(99.225)		40.0	54.0	14.0	85	94	7.5
18	200(198.45)	5°	40.1	48.1	8.0	82	93	9
19	300((297.7)		40.1	45.7	5.6	80	93	10
20	400(396.9)		40.2	44.6	4.4	78	91	11
21	100(99.225))		39.5	54.6	15.1	89	95	5.5
22	200(198.45)	10°	40.0	48.0	8.0	88	94.5	6.0
23	300(297.7)		40.0	46.0	6.0	86	93	7.0
24	400(396.9)		40.2	44.9	4.7	85	92.5	7.5
25	100(99.225)		40.0	56.8	16.8	91	96	4.5
26	200(198.45)	15°	40.0	49.2	9.2	90	95	5.0
27	300(297.7)		40.5	47.0	6.5	89	94.5	5.5
28	400(396.9)		40.0	45	5.0	88.5	94	5.75
29	100(99.225)		40.0	60.0	20.0	91	96	4.5
30	200(198.45)	19°	40.5	51.1	10.6	90	95	5.0
31	300(297.7)		40.0	47.6	7.6	89	94	5.5
32	400(396.9)		40.0	45.8	5.8	88.5	94	5.75

TABLE - A1-2 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - received by Coolant Kcal/hr	Heat - Flux($\bar{q}$) X10^{-4} Kcal/hr	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.\ °C}$	$\bar{h}_{id}$ Theo. $\frac{Kcal}{hr.m^2.\ °C}$
(0)	( )	(3)	(9)	(10)	(11)	(12)
17	2.623	1414	1389	9.51	12687	10032
18	2.981	1607	1588	10.81	12015	9586
19	3.164	1705	1667	11.47	11473	9337
20	3.323	1791	1746	12.05	10957	9117
21	2.818	1519	1498	7.27	13225	10924
22	2.969	1600	1587	7.66	12769	10591
23	3.419	1842	1788	8.82	12600	10190
24	3.524	1900	1865	9.10	12130	10016
25	3.506	1890	1670	6.64	14751	11306
26	3.607	1944	1826	6.83	13661	11012
27	3.670	1976	1935	6.94	12624	10753
28	3.762	2028	1985	7.13	12392	10634
29	4.119	2221	1985	6.58	14633	11289
30	4.367	2356	2103	6.98	13969	10996
31	4.569	2460	2262	7.29	13260	10737
32	4.694	2524	2302	7.48	13014	10618

TABLE - A1-2 (contd.)

Serial no.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
17	0.431	105	0.198	699	4.29
18	0.408	119	0.189	662	3.57
19	0.389	127	0.186	633	3.22
20	0.372	133	0.183	604	2.92
21	0.449	80	0.218	1039	6.39
22	0.433	85	0.213	1003	5.85
23	0.428	97	0.204	989	5.02
24	0.411	100	0.202	953	4.68
25	0.501	73	0.223	1568	7.88
26	0.463	75	0.222	1452	7.09
27	0.428	76	0.221	1342	6.45
28	0.421	78	0.219	1318	6.17
29	0.497	72	0.223	1831	8.33
30	0.474	77	0.218	1748	7.50
31	0.450	80	0.215	1660	6.81
32	0.441	83	0.212	1629	6.51

TABLE - A1-3

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
33	100 (98.807)	5°	50.1	62.7	12.6	88	95	6.0
34	200 (197.614)		50.0	56.4	6.4	87	94	6.5
35	300 (296.421)		50.2	54.6	4.4	86	93	7
36	400 (395.228)		50.2	53.4	3.2	84	92	8
37	100 (98.807)	10°	49.8	64.2	14.4	91	95	4.5
38	200 (197.614)		50.0	57.3	7.3	90	95	5.0
39	300 (296.421)		50.0	55.3	5.3	88	94	6.0
40	400 (395.228)		50.5	54.6	4.1	86	93.5	6.5
41	100 (98.807)	15°	50.0	66.1	16.1	93	97	3.5
42	200 (197.614)		50.0	58.2	8.2	92	96	4.0
43	300 (296.421)		50.0	56.4	6.4	90	95	5.0
44	400 (395.228)		50.5	55.3	4.8	89	95	5.5
45	100 (98.807)	19°	49.5	70.3	20.8	93	97	3.5
46	200 (197.614)		50.0	61.0	11.0	92	96	4.0
47	300 (296.421)		50.0	57.8	7.8	90	96	5.0
48	400 (395.228)		50.2	56.3	6.1	89	9.5	5.5

TABLE - A1-3 (contd.)

Serial no.	Average Condensate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hrm^2	$\bar{h}_{id}$ Expt. Kcal/hr.m^2.°C	$\bar{h}_{id}$ Theo. Kcal/hr.m^2. °C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
33	2.381	1283	1245	8.63	14390	10608
34	2.419	1304	1265	8.80	13500	10398
35	2.489	1341	1304	9.02	12892	10207
36	2.559	1379	1344	9.27	11600	9872
37	2.661	1434	1423	6.87	15258	11381
38	2.716	1463	1442	7.00	14011	11085
39	2.964	1597	1571	7.65	12745	10591
40	3.006	1620	1620	7.76	11934	10381
41	3.006	1620	1591	5.69	16263	12039
42	3.056	1647	1620	5.79	14468	11644
43	3.597	1938	1897	6.81	13619	11012
44	3.607	1944	1897	6.83	12381	10753
45	3.833	2057	2055	6.09	17449	12012
46	4.059	2184	2173	6.47	16187	11627
47	4.367	2356	2312	6.98	13969	10996
48	4.510	2430	2410	7.20	13099	10737

TABLE - A1-3 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
33	0.488	95	0.205	793	5.36
34	0.458	97	0.203	745	4.95
35	0.437	99	0.201	711	4.59
36	0.394	102	0.199	640	4.02
37	0.518	76	0.221	1198	7.81
38	0.476	77	0.220	1100	7.03
39	0.433	84	0.214	1001	5.86
40	0.405	86	0.213	937	5.45
41	0.552	63	0.235	1729	10.13
42	0.491	64	0.234	1538	8.87
43	0.462	75	0.222	1448	7.09
44	0.420	75	0.222	1316	6.45
45	0.592	67	0.228	2184	10.7
46	0.549	71	0.224	2026	9.37
47	0.474	76	0.219	1749	7.50
48	0.444	79	0.216	1639	6.81

TABLE - A1-4

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Agnel (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(3)	(3)			(4)	(5)	(6)
49	100 (99.568)	5°	30.0	34.0	4.0	53.5	68	11.8
50	200 (199.136)		30.0	32.2	2.2	51.0	68	13.05
51	300 (298.70)		30.5	32.0	1.5	49.0	66	14.05
52	400 (398.27)		30.0	31.2	1.2	47.0	65	15.05
53	100 (99.568)	10°	30.0	35.3	5.3	55.5	70	10.8
54	200 (199.136)		30.5	33.3	2.8	53.5	68	11.8
55	300 (298.70)		30.0	32.1	2.1	50.0	67	13.55
56	400 (398.27)		30.2	31.8	1.6	49.0	66	14.05
57	100 (99.568)	15°	30.2	35.8	5.6	60.0	72	8.55
58	200 (199.136)		30.0	33.0	3.0	58.0	70	9.55
59	300 (298.70)		30.0	32.3	2.3	55.5	69	10.8
60	400 (398.27)		30.0	31.8	1.8	53.5	69	11.8
61	100 (99.568)	19°	29.5	35.7	6.2	62.0	73	7.55
62	200 (199.136)		30.0	33.3	3.3	60.0	71	8.55
63	300 (298.70)		30.0	32.3	2.3	58.0	70	9.55
64	400 (398.27)		30.2	32.2	2.0	55.5	69	10.8

TABLE - A1-4 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat Flux($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. Kcal / hr.m^2.°C	$\bar{h}_{id}$ Theo. Kcal / hr.m^2. °C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
49	4.703	403	398	2.71	2298	1992
50	5.120	439	438	2.95	2263	1942
51	5.260	451	448	3.03	2160	1907
52	5.737	491	478	3.30	2195	1874
53	6.162	528	528	2.53	2340	2033
54	6.557	561	557	2.69	2276	1988
55	7.354	630	627	3.02	2226	1921
56	7.478	641	637	3.07	2187	1904
57	6.790	581	557	2.04	2387	2141
58	7.390	634	597	2.23	2325	2083
59	8.217	700	687	2.45	2277	2020
60	8.750	750	716	2.63	2233	1976
61	7.606	651	617	1.93	2556	2205
62	8.110	696	657	2.06	2413	2138
63	8.640	742	687	2.19	2303	2079
64	9.500	836	796	2.48	2295	2016

TABLE - A1-4 (contd.)

Serial No.	$h_i \left(\frac{\mu_f^2}{\rho_f^2 g\ k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) X10^{-13}
(0)	(13)	(14)	(15)	(16)	(17)
49	0.312	195	0.161	499	1.60
50	0.308	213	0.156	491	1.45
51	0.294	218	0.155	469	1.35
52	0.298	238	0.151	477	1.26
53	0.318	182	0.166	724	1.92
54	0.309	194	0.162	704	1.75
55	0.302	218	0.156	689	1.53
56	0.297	221	0.155	677	1.47
57	0.324	147	0.177	1000	2.44
58	0.316	160	0.172	973	2.19
59	0.309	177	0.166	954	1.94
60	0.304	189	0.163	935	1.77
61	0.348	164	0.171	1260	2.92
62	0.328	175	0.167	1190	2.58
63	0.313	187	0.163	1136	2.31
64	0.312	205	0.158	1132	2.04

TABLE - A1.5

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
65	100(99.225)	5°	39.5	43.0	3.5	58.0	71	9.55
66	200(198.45)		40.0	41.8	1.8	55.5	71	10.8
67	300(297.70)		40.0	41.3	1.3	53.5	70	11.8
68	400(396.90)		40.2	41.2	1.0	51.0	69	13.05
69	100(99.225)	10°	41.0	45.2	4.2	61.0	72	8.05
70	200(198.45)		40.5	43.0	2.5	58.0	71	9.55
71	300(297.70)		40.5	41.8	1.8	55.5	70	10.8
72	400(396.90)		40.0	41.5	1.5	53.5	69	11.8
73	100(99.225)	15°	40.5	44.7	4.2	65	73	6.05
74	200(198.45)		40.5	43.3	2.8	62	72	7.55
75	300(297.70)		40.0	42.2	2.2	59	71	9.05
76	400(396.90)		40.0	41.7	1.7	56.5	70	10.3
77	100(99.225)	19°	40.0	45.4	5.4	66	74	5.55
78	200(198.45)		40.5	43.5	3.0	64	73	6.55
79	300(297.70)		40.0	42.2	2.2	62	72	7.55
80	400(396.90)		40.0	41.7	1.7	60	70	8.55

TABLE - A1-5 (contd.)

Serial No.	Average Condensate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat Flux($\bar{q}$) X10^{-4} Kcal/hrm^2	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.\ °C}$	$\bar{h}_{id}$ Theo. $\frac{Kcal}{hr.m^2.\ °C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
65	4.08	350	347	2.36	2466	2100
66	4.24	363	357	2.44	2261	2036
67	4.59	393	387	2.64	2241	1992
68	4.70	403	397	2.71	2078	1942
69	5.367	460	416	2.20	2736	2188
70	6.051	518	496	2.48	2597	2096
71	6.525	559	536	2.68	2527	2033
72	7.006	601	596	2.88	2434	1989
73	5.670	486	416	1.71	2823	2335
74	6.880	590	556	2.07	2746	2209
75	5.120	660	655	2.34	2562	2111
76	8.320	744	674	2.61	2538	2044
77	6.570	563	536	1.67	3007	2382
78	7.395	634	595	1.88	2869	2285
79	8.217	704	655	2.09	2764	2205
80	8.530	731	675	2.17	2534	2138

TABLE - A1-5 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number $(\overline{Nu})$	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
65	0.335	169	0.169	536	1.98
66	0.307	176	0.167	491	1.75
67	0.305	190	0.162	487	1.61
68	0.283	195	0.161	452	1.45
69	0.372	159	0.172	847	2.57
70	0.353	179	0.166	804	2.17
71	0.344	193	0.161	782	1.92
72	0.331	207	0.158	753	1.76
73	0.384	122	0.188	1183	3.45
74	0.374	149	0.176	1150	2.77
75	0.348	111	0.194	1073	2.31
76	0.345	180	0.165	1063	2.03
77	0.409	142	0.179	1483	3.97
78	0.390	160	0.172	1267	3.37
79	0.376	177	0.166	1363	2.92
80	0.345	184	0.164	1250	2.58

TABLE - A1-6

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Average Coolant Temperature Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
81	100 (98.807)	5°	50.0	52.1	2.1	66	72	5.55
82	200 (197.614)		50.0	51.1	1.1	65	72	6.05
83	300 (296.421)		50.0	50.8	0.8	63	71	7.05
84	400 (395.228)		50.2	50.8	0.6	62	70	7.55
85	100 (98.807)	10°	50.5	53.6	3.1	66	73	5.55
86	200 (197.614)		50.0	51.7	1.7	65	73	6.05
87	300 (296.421)		50.0	51.3	1.3	63	72	7.05
88	400 (395.228)		50.0	51.0	1.0	62	70	7.55
89	100 (98.807)	15°	50.2	54.2	4.0	68	74	4.55
90	200 (197.614)		50.0	52.4	2.4	66	73	5.55
91	300 (296.421)		50.0	51.8	1.8	64	72	6.55
92	400 (395.228)		50.0	51.4	1.4	63	71	7.05
93	100 (98.807)	19°	50.0	54.5	4.5	70	74	3.55
94	200 (197.614)		50.2	52.7	2.5	68	74	4.55
95	300 (296.421)		50.0	52.0	2.0	66	73	5.55
96	400 (395.228)		50.0	51.6	1.6	65	72	6.05

TABLE - A1-6 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. Kcal/hr.m^2. °C	$\bar{h}_{id}$ Theo. Kcal/hr.m^2. °C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
81	2.569	220	198	1.48	2667	2405
82	2.690	231	217	1.55	2569	2354
83	2.906	249	237	1.67	2376	2265
84	3.000	257	237	1.73	2290	2227
85	3.740	320	307	1.53	2761	2401
86	3.962	340	336	1.63	2690	2350
87	4.497	385	385	1.84	2721	2262
88	4.720	405	395	1.94	2568	2223
89	5.254	450	395	1.58	3475	2507
90	6.125	525	474	1.84	3324	2385
91	6.790	582	533	2.05	3122	2289
92	6.980	598	554	2.10	2980	2247
93	5.241	449	445	1.33	3750	2663
94	6.340	543	494	1.61	3538	2503
95	7.006	600	593	1.78	3205	2382
96	7.395	634	632	1.88	3107	2331

TABLE - A1-6 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{\;1/3}}$	Mean Nusselt Number ($\bar{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
81	0.362	107	0.195	579	3.41
82	0.349	112	0.194	558	3.13
83	0.323	121	0.189	516	2.69
84	0.311	125	0.187	497	2.51
85	0.375	111	0.195	854	3.74
86	0.366	118	0.191	832	3.42
87	0.370	133	0.184	842	2.94
88	0.349	140	0.181	794	2.74
89	0.472	113	0.193	1455	4.59
90	0.452	132	0.184	1392	3.77
91	0.424	147	0.177	1308	3.19
92	0.405	151	0.176	1248	2.97
93	0.509	95	0.203	1850	6.22
94	0.481	115	0.190	1745	4.85
95	0.436	128	0.184	1581	3.98
96	0.422	135	0.181	1532	3.65

TABLE - A1-7

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Average Coolant Temperature Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
97	100(99.568)	5°	29.8	34.1	4.3	53.5	69	12.4
98	200(199.136)		30.0	32.4	2.4	51.0	68	13.65
99	300(298.70)		30.0	31.6	1.6	49.0	66	14.65
100	400(398.27)		30.2	31.4	1.2	47.5	65	15.40
101	100(99.568)	10°	30.0	35.4	5.4	55.5	69	11.4
102	200(199.136)		29.8	32.8	3.0	53.5	68	12.4
103	300(298.70)		30.0	32.1	2.1	51.0	68	13.65
104	400(398.27)		30.0	31.7	1.7	49.5	66	14.4
105	100(99.568)	15°	31.0	37.8	6.8	61.0	73	8.65
106	200(199.136)		30.0	33.8	3.8	58.0	71	10.15
107	300(298.70)		30.0	32.8	2.8	55.5	70	11.40
108	400(398.27)		30.2	32.4	2.2	53.5	69	12.40
109	100(99.568)	19°	30.0	37.3	7.3	63.0	74	7.65
110	200(199.136)		29.8	34.0	4.2	60.0	72	9.15
111	300(298.70)		30.0	33.0	3.0	58.0	71	10.15
112	400(398.27)		30.2	32.6	2.4	55.5	70	11.40

TABLE - A1-7 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{id}$ Theo $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
97	2.057	432	428	2.91	2344	1725
98	2.181	458	458	3.08	2257	1685
99	2.278	478	477	3.22	2195	1655
100	2.278	478	477	3.21	2089	1635
101	2.681	563	538	2.69	2365	1759
102	2.884	605	597	2.89	2340	1723
103	3.007	631	627	3.02	2213	1682
104	3.244	681	677	3.26	2264	1660
105	3.469	680	677	2.39	2762	1873
106	4.006	758	756	2.66	2624	1800
107	4.281	838	836	2.94	2583	1748
108	4.596	881	876	3.09	2496	1711
109	3.469	728	726	2.16	2821	1928
110	4.006	841	836	2.49	2724	1844
111	4.280	898	896	2.66	2622	1797
112	4.595	965	955	2.86	2509	1745

TABLE - A1-7 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
97	0.609	49	0.255	591	1.64
98	0.587	52	0.250	569	1.49
99	0.570	55	0.246	554	1.39
100	0.543	55	0.246	527	1.32
101	0.615	46	0.262	849	1.95
102	0.608	49	0.239	840	1.79
103	0.575	51	0.252	795	1.63
104	0.588	55	0.247	813	1.54
105	0.718	43	0.267	1343	2.60
106	0.682	50	0.254	1373	2.21
107	0.671	53	0.249	1256	1.97
108	0.649	57	0.243	1213	1.81
109	0.733	37	0.278	1614	3.09
110	0.708	42	0.267	1559	2.59
111	0.681	45	0.261	1501	2.33
112	0.652	48	0.255	1436	2.08

TABLE - A1-8

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. °C	Avg. Condensate Temp (T_c) °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
113	100(99.225)	5°	40.0	43.8	3.8	58.0	71	10.15
114	200(198.45)		40.5	42.6	2.1	55.5	70	11.40
115	300(297.70)		40.0	41.5	1.5	51.0	68	13.65
116	400(396.90)		40.0	41.2	1.2	49.0	66	14.65
117	100(99.225)	10°	39.5	44.3	4.8	60.0	72	9.15
118	200(198.45)		40.0	42.6	2.6	58.0	70	10.15
119	300(297.70)		40.0	41.8	1.8	55.5	70	11.40
120	400(396.90)		40.0	41.4	1.4	53.5	69	12.40
121	100(99.225)	15°	40.0	46.0	6.0	64.0	73	7.15
122	200(198.45)		40.0	43.2	3.2	62.0	72	8.15
123	300(297.70)		40.2	42.5	2.3	60.0	71	9.15
124	400(396.90)		40.2	42.0	1.8	58.0	70	10.15
125	100(99.225)	19°	40.2	46.6	6.4	66.0	74	6.15
126	200(198.45)		40.0	43.6	3.6	64.0	73	7.15
127	300(297.70)		40.0	42.6	2.6	62.0	72	8.15
128	400(396.90)		40.0	42.0	2.0	60.0	72	9.15

TABLE - A1-8 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat-Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{id}$ Expt. Kcal / hr.m^2. °C	$\bar{h}_{id}$ Theo. Kcal / hr.m^2. °C
(0	(7)	(8)	(9)	(10)	(11)	(12)
113	1.799	378	377	2.54	2506	1814
114	1.981	416	416	2.80	2456	1762
115	2.140	450	447	3.03	2218	1685
116	2.288	480	476	3.23	2205	1655
117	2.328	488	476	2.34	2559	1859
118	2.480	521	516	2.49	2458	1811
119	2.628	552	536	2.64	2318	1759
120	2.765	581	555	2.78	2244	1723
121	3.050	596	595	2.09	2928	1964
122	3.433	641	635	2.25	2763	1901
123	3.738	688	684	2.42	2642	1846
124	3.819	723	714	2.54	2502	1800
125	3.050	641	635	1.90	3090	2036
126	3.433	721	714	2.14	2989	1961
127	3.738	785	774	2.33	2856	1898
128	3.818	802	793	2.38	2599	1844

TABLE - A1-8 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number $(\overline{Nu})$	Condensation Number (C_v) $x10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
113	0.651	43	0.267	632	2.00
114	0.638	48	0.257	619	1.78
115	0.576	51	0.252	559	1.49
116	0.573	55	0.246	556	1.39
117	0.665	40	0.274	919	2.43
118	0.639	42	0.270	882	2.19
119	0.602	45	0.264	833	1.95
120	0.583	47	0.260	806	1.80
121	0.761	38	0.278	1423	3.13
122	0.718	43	0.267	1343	2.75
123	0.687	47	0.259	1285	2.45
124	0.650	48	0.257	1216	2.21
125	0.803	32	0.292	1768	3.85
126	0.777	36	0.281	1711	3.31
127	0.742	40	0.271	1635	2.90
128	0.675	40	0.271	1488	2.59

TABLE - A1-9

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_v) = 78.3° C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. °C	Avg. Condensate Temp. (T_c) °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
129	100(98.807)		50.0	53.0	3.0	62.0	72	8.15
130	200(197.614)	5°	50.5	52.2	1.7	60.0	71	9.15
131	300(296.421)		50.2	51.4	1.2	58.0	70	10.15
132	400(395.228)		50.0	50.9	0.9	55.5	69	11.40
133	100(98.807)		50.0	53.8	3.8	64.0	73	7.15
134	200(197.614)	10°	50.2	52.2	2.0	62.0	73	8.15
135	300(296.421)		50.0	51.4	1.4	60.0	72	9.15
136	400(395.228)		50.0	51.2	1.2	58.0	70	10.15
137	100(98.807)		49.8	54.3	4.5	68.0	75	5.15
138	200(197.614)	15°	50.0	52.5	2.5	66.0	74	6.15
139	300(296.421)		50.0	51.8	1.8	64.0	74	7.15
140	400(395.228)		50.0	51.5	1.5	62.0	72	8.15
141	100(98.807)		50.5	55.0	4.5	70.0	76	4.15
142	200(197.614)	19°	50.2	52.9	2.7	68.0	75	5.15
143	300(296.421)		50.0	52.0	2.0	66.0	74	6.15
144	400(395.228)		50.0	51.6	1.6	64.0	72	7.15

TABLE - A1-9 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by Vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{id_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
129	1.474	310	296	2.08	2559	1916
130	1.623	341	336	2.29	2508	1862
131	1.701	357	356	2.40	2367	1814
132	1.811	381	356	2.56	2249	1762
133	1.849	388	375	1.86	2598	1977
134	2.049	430	395	2.06	2526	1913
135	2.149	451	414	2.16	2360	1859
136	2.288	481	474	2.30	2269	1811
137	2.171	456	444	1.60	3111	2132
138	2.546	499	494	1.75	2850	2039
139	2.846	543	533	1.91	2668	1964
140	3.050	601	595	2.11	2591	1901
141	2.171	456	444	1.35	3257	2247
142	2.545	535	533	1.59	3079	2129
143	2.846	597	592	1.77	2877	2036
144	3.050	641	632	1.90	2657	1961

TABLE - A1-9 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
129	0.665	35	0.285	645	2.50
130	0.652	39	0.275	632	2.20
131	0.615	41	0.271	597	2.00
132	0.584	43	0.267	567	1.78
133	0.675	32	0.295	933	3.11
134	0.656	35	0.287	907	2.73
135	0.613	37	0.282	848	2.43
136	0.589	39	0.277	815	2.19
137	0.808	27	0.312	1512	4.35
138	0.741	32	0.295	1385	3.65
139	0.693	36	0.284	1297	3.14
140	0.673	38	0.278	1260	2.75
141	0.846	23	0.326	1865	5.71
142	0.800	27	0.309	1762	4.60
143	0.748	30	0.298	1647	3.85
144	0.690	32	0.292	1521	3.31

TABLE - A1-10

LIQUID SYSTEM = CARBON-TETRA-CHLORIDE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP(T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Average Coolant Temperature Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
145	100 (99.568)	5°	29.5	33.1	3.6	51.0	67	12.875
146	200 (199.136)		30.0	31.8	1.8	49.0	66	13.875
147	300 (298.70)		30.0	31.3	1.3	47.0	64	14.875
148	400 (398.27)		30.2	31.2	1.0	45.0	63	15.875
149	100 (99.568)	10°	30.0	34.3	4.3	54.5	68	11.125
150	200 (199.136)		30.2	32.4	2.2	51.0	67	12.875
151	300 (298.70)		29.8	31.3	1.5	49.0	65	13.875
152	400 (398.27)		30.0	31.4	1.4	47.0	65	14.875
153	100 (99.568)	15°	29.5	34.6	5.1	59.0	70	8.875
154	200 (199.136)		30.0	33.0	3.0	55.5	68	10.625
155	300 (298.70)		30.0	32.3	2.3	51.0	66	12.875
156	400 (398.27)		30.2	32.0	1.8	49.0	66	13.875
157	100 (99.568)	19°	29.5	35.5	6.0	60.0	70	8.375
158	200 (199.136)		30.0	33.2	3.2	58.0	70	9.375
159	300 (298.70)		30.0	32.5	2.5	53.5	68	11.625
160	400 (398.27)		30.0	32.0	2.0	51.0	67	12.875

TABLE - A1-10 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{id_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
145	7.610	360	358	2.42	1881	1741
146	7.980	377	358	2.54	1828	1709
147	8.360	395	388	2.66	1786	1680
148	8.720	412	398	2.77	1746	1653
149	9.480	446	428	2.14	1919	1803
150	10.800	509	438	2.44	1893	1738
151	11.340	535	477	2.56	1846	1706
152	11.870	560	558	2.68	1802	1676
153	11.510	544	508	1.91	2153	1895
154	12.740	601	597	2.11	1987	1812
155	15.120	713	687	2.51	1945	1727
156	16.200	766	718	2.69	1939	1695
157	13.340	630	597	1.87	2230	1920
158	14.175	669	637	1.98	2115	1867
159	16.800	793	747	2.35	2022	1769
160	18.000	850	797	2.52	1957	1725

TABLE - A1-10 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3} \right)^{1/3}$	Re_f	$Re_f' \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\bar{N}u$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
145	0.308	152	0.175	439	1.25
146	0.299	160	0.172	427	1.16
147	0.292	167	0.169	417	1.08
148	0.286	174	0.167	408	1.01
149	0.314	134	0.183	638	1.58
150	0.310	154	0.175	630	1.37
151	0.302	161	0.173	614	1.27
152	0.295	169	0.170	599	1.18
153	0.353	120	0.189	969	2.00
154	0.326	132	0.184	894	1.67
155	0.318	157	0.173	875	1.38
156	0.318	168	0.169	873	1.28
157	0.365	139	0.179	1182	2.24
158	0.346	147	0.176	1121	2.00
159	0.331	175	0.166	1072	1.62
160	0.320	187	0.162	1037	1.46

TABLE - A1-11

LIQUID SYSTEM = CARBON-TETRA-CHLORIDE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_V) = 76.75°C

Serial No.	Coolant Rate lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
161	100(99.225)	5°	40.0	42.8	2.8	58.0	69	9.375
162	200(198.45)		40.2	41.7	1.5	55.5	69	10.625
163	300(297.70)		39.8	40.8	1.0	53.5	68	11.625
164	400(396.90)		40.0	40.8	0.8	51.0	66	12.875
165	100(99.225)	10°	41.0	44.7	3.7	60.0	70	8.375
166	200(198.45)		40.0	42.0	2.0	55.5	69	10.625
167	300(297.70)		40.0	41.5	1.5	54.5	69	11.125
168	400(396.90)		40.0	41.1	1.1	53.5	68	11.625
169	100(99.225)	15°	40.0	44.4	4.4	63.0	72	6.875
170	200(198.45)		40.0	42.6	2.6	60.0	70	8.375
171	300(297.70)		40.2	42.3	2.1	55.5	69	10.625
172	400(396.90)		40.0	41.6	1.6	53.5	67	11.625
173	100(99.225)	19°	40.0	45.0	5.0	64.0	73	6.375
174	200(198.45)		40.5	43.3	2.8	61.0	71	7.875
175	300(297.70)		40.2	42.5	2.3	58.0	70	9.375
176	400(396.90)		40.0	41.8	1.8	56.5	68	10.125

TABLE - A1-11

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{id}$ Theo. $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
161	6.13	289	278	1.95	2074	1885
162	6.49	307	298	2.06	1944	1827
163	6.62	313	298	2.10	1811	1786
164	7.15	338	317	2.27	1766	1741
165	7.98	377	367	1.80	2155	1935
166	9.11	430	397	2.06	1937	1824
167	9.53	450	447	2.15	1936	1803
168	9.53	450	437	2.15	1853	1783
169	9.52	450	436	1.58	2299	2020
170	11.01	520	516	1.83	2186	1923
171	13.79	652	625	2.29	2156	1812
172	13.98	661	635	2.32	1998	1772
173	10.54	498	496	1.47	2316	2056
174	12.67	598	556	1.77	2251	1950
175	15.01	709	685	2.10	2242	1867
176	15.32	724	714	2.15	2120	1832

TABLE - A1-11 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\bar{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
161	0.339	123	0.188	484	1.72
162	0.318	130	0.184	454	1.51
163	0.297	133	0.183	423	1.38
164	0.289	143	0.179	412	1.25
165	0.353	114	0.194	717	2.10
166	0.317	130	0.185	644	1.66
167	0.317	136	0.183	644	1.58
168	0.304	136	0.183	616	1.52
169	0.376	99	0.202	1035	2.58
170	0.358	114	0.193	984	2.12
171	0.353	143	0.179	970	1.67
172	0.327	145	0.178	899	1.53
173	0.379	110	0.195	1228	2.90
174	0.369	132	0.184	1193	2.40
175	0.367	156	0.174	1188	2.00
176	0.347	159	0.173	1124	1.84

TABLE - A1-12

LIQUID SYSTEM = CARBON - TETRA- CHLORIDE

TYPE OF CONDENSER = DIVERGING CONE SECTION

VAPOUR TEMP (T_V) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Average Coolant Temperature Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_v - T_w}{2}$
				°C		°C	°C	°C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
177	100 (98.807)		49.0	51.5	2.0	64.0	72	6.375
178	200 (197.614)	5°	50.0	51.1	1.1	62.0	72	7.550
179	300 (296.421)		50.0	50.8	0.8	61.0	70	7.875
180	400 (395.228)		50.2	50.8	0.6	60.0	70	8.375
181	100 (98.807)		50.2	52.6	2.4	65.0	72	5.875
182	200 (197.614)	10°	50.0	51.5	1.5	63.0	71	6.875
183	300 (296.421)		50.0	51.0	1.0	62.0	71	7.550
184	400 (395.228)		50.2	51.1	0.9	60.0	70	8.375
185	100 (98.807)		49.8	53.6	3.8	65.0	74	5.875
186	200 (197.614)	15°	50.0	52.2	2.2	63.0	73	6.875
187	300 (296.421)		50.0	51.5	1.5	62.0	71	7.550
188	400 (395.228)		50.0	51.3	1.3	60.0	71	8.375
189	100 (98.807)		49.8	53.6	3.8	68.0	74	4.375
190	200 (197.614)	19°	50.0	52.2	2.2	65.0	73	5.875
191	300 (296.421)		50.0	51.7	1.7	64.0	72	6.375
192	400 (395.228)		50.2	51.5	1.3	63.0	72	6.875

1193

TABLE - A1-12

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ $Kcal/hr.m^2$	$\bar{h}_{id}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{id}$ Theo. $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(1)	(11)	(12)
177	4.45	210	198	1.41	2216	2076
178	5.01	237	217	1.60	2112	1990
179	5.21	246	237	1.66	2025	1969
180	5.22	247	237	1.66	1985	1939
181	5.93	280	237	1.34	2282	2115
182	6.55	310	297	1.48	2159	2033
183	7.08	335	297	1.64	2124	1986
184	7.58	358	355	1.71	2046	1936
185	8.61	406	375	1.43	2428	2101
186	9.72	459	424	1.61	2343	2021
187	10.01	473	445	1.66	2201	1974
188	11.05	520	514	1.83	2181	1923
189	8.590	406	375	1.20	2751	2259
190	10.125	478	435	1.41	2412	2098
191	10.698	505	503	1.50	2349	2056
192	11.010	520	514	1.54	2242	2017

TABLE - A1-12

Serial No.	$\bar{h}_i \cdot \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
177	0.363	89	0.209	518	2.53
178	0.346	100	0.201	493	2.13
179	0.332	104	0.198	473	2.04
180	0.325	104	0.198	463	1.92
181	0.374	84	0.214	759	2.99
182	0.354	93	0.207	718	2.56
183	0.348	101	0.202	706	2.33
184	0.335	108	0.197	680	2.10
185	0.398	89	0.209	1093	3.03
186	0.383	101	0.201	1054	2.59
187	0.361	104	0.199	991	2.36
188	0.357	115	0.192	982	2.12
189	0.451	89	0.207	1456	4.30
190	0.395	105	0.196	1278	3.20
191	0.385	111	0.193	1245	2.95
192	0.367	114	0.191	1188	2.73

TABLE - A1-13

LQUID SYSTEM = WATER

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
193	100(99.568)	5°	30.0	45.8	15.8	80.0	93	10.0
194	200(199.136)		30.5	39.7	9.2	74.0	91	13.0
195	300(298.70)		30.5	37.4	6.9	70.0	89	15.0
196	400(398.27)		31.0	36.4	5.4	68.0	87	16.0
197	100(99.568)	10°	29.5	46.5	17.0	85.0	94	7.5
198	200(199.136)		30.0	40.0	10.0	81.0	92	9.5
199	300(298.70)		30.2	37.2	7.0	78.0	91	11.0
200	400(398.27)		30.0	35.5	5.5	77.0	90	11.5
201	100(99.568)	15°	30.0	49.5	19.5	87.5	94	6.25
202	200(199.136)		30.5	40.9	10.4	86.0	94	7.0
203	300(298.70)		30.0	37.6	7.6	84.0	92	8.0
204	400(398.27)		30.0	35.9	5.9	83.0	92	8.5
205	100(99.568)	19°	29.8	51.0	21.2	90.0	95	5.0
206	200(199.136)		30.0	41.4	11.4	88.5	95	5.75
207	300(298.70)		30.0	37.8	7.8	86.0	94	7.0
208	400(398.27)		30.2	36.2	6.0	84.0	92	8.0

TABLE - A1-13 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat Released by Vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{id}$ Expt. $\frac{\text{Kcal}}{\text{hr.m}^2.\,°\text{C}}$	$\bar{h}_{id}$ Theo. $\frac{\text{Kcal}}{\text{hr.m}^2.\,°\text{C}}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
193	2.967	1600	1573	10.77	10767	9336
194	3.503	1887	1832	12.69	9768	8920
195	3.898	2101	2061	14.14	9425	8436
196	4.078	2197	2151	14.78	9240	8302
197	3.216	1734	1693	8.30	11070	10016
198	3.855	2078	1991	9.95	10473	9441
199	4.087	2202	2091	10.54	9585	9102
200	4.241	2285	2190	10.94	9514	9001
201	3.833	2006	1942	7.26	11615	10415
202	4.028	2171	2071	7.63	10897	10123
203	4.318	2326	2270	8.17	10216	9791
204	4.451	2398	2349	8.42	9913	9644
205	4.207	2267	2110	6.72	12220	10996
206	4.243	2287	2270	6.78	11300	10618
207	4.569	2460	2329	7.29	10419	10109
208	4.953	2670	2389	7.92	9894	9777

TABLE - A1-13 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) X10^{-13}
(0)	(13)	(14)	(15)	(16)	(17)
193	0.365	119	0.189	593	8.22
194	0.331	140	0.179	542	2.68
195	0.319	156	0.173	520	2.15
196	0.314	163	0.171	510	2.01
197	0.375	92	0.208	869	4.68
198	0.355	110	0.196	822	3.69
199	0.325	116	0.192	753	3.19
200	0.323	121	0.189	747	3.05
201	0.394	80	0.217	1235	5.68
202	0.369	84	0.213	1158	5.07
203	0.347	90	0.209	1086	4.44
204	0.336	93	0.206	1054	4.17
205	0.415	74	0.221	1530	6.81
206	0.383	75	0.219	1414	6.52
207	0.354	80	0.215	1304	5.35
208	0.336	87	0.209	1238	4.69

TABLE - A1-14

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
209	100 (99.225)	5°	39.5	52.7	13.2	85.0	94	7.5
210	200 (198.45)		40.0	47.5	7.5	82.0	93	9.0
211	300 (297.70)		40.0	45.5	5.5	80.0	93	10.0
212	400 (396.9)		40.2	44.5	4.3	78.0	91	11.0
213	100 (99.225)	10°	40.5	54.6	14.1	89.0	95	5.5
214	200 (198.45)		40.0	47.5	7.5	88.0	94	6.0
215	300 (297.70)		40.2	45.8	5.6	86.0	93	7.0
216	400 (396.9)		40.0	44.5	4.5	85.0	92	7.5
217	100 (99.225)	15°	39.5	56.7	17.2	90.0	95	5.0
218	200 (198.45)		40.0	49.0	9.0	90.0	94	5.0
219	300 (297.70)		40.2	46.7	6.5	89.0	94	5.5
220	400 (396.9)		40.0	45.0	5.0	88.5	93	5.75
221	100 (99.225)	19°	40.5	59.6	19.1	91.0	96.0	4.5
222	200 (198.45)		40.5	51.0	10.5	90.0	95.5	5.0
223	300 (297.70)		40.0	47.6	7.6	89.0	95.0	5.5
224	400 (396.9)		40.0	45.8	5.8	88.5	94.5	5.75

TABLE - A1-14 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{ic}$ Theo. $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
209	2.453	1322	1310	8.89	11862	10032
210	2.773	1501	1488	10.10	11223	9586
211	3.122	1683	1639	11.33	11325	9337
212	3.324	1791	1707	12.05	10957	9117
213	2.702	1456	1399	6.97	12676	10824
214	2.887	1556	1488	7.45	12418	10591
215	3.247	1750	1668	8.40	11970	10190
216	3.446	1857	1786	8.89	11856	10016
217	3.495	1884	1706	6.62	13240	11012
218	3.369	1815	1786	6.38	12754	11012
219	3.607	1944	1935	6.83	12419	10763
220	3.729	2008	1985	7.06	12270	10634
221	4.122	2221	1895	6.58	14633	11289
222	4.373	2356	2083	6.98	13969	10996
223	4.563	2460	2262	7.29	13260	10737
224	4.684	2524	2302	7.48	13014	10618

TABLE - A1-14 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
209	0.402	98	0.203	654	4.29
210	0.381	111	0.194	619	3.57
211	0.384	125	0.187	625	3.22
212	0.372	133	0.183	604	2.92
213	0.430	77	0.221	995	6.39
214	0.421	82	0.216	975	5.85
215	0.406	93	0.207	940	5.02
216	0.402	98	0.204	931	4.68
217	0.449	73	0.224	1407	7.88
218	0.433	70	0.227	1356	7.09
219	0.422	75	0.222	1320	6.45
220	0.416	78	0.219	1305	6.17
221	0.496	72	0.223	1831	8.32
222	0.474	77	0.218	1748	7.50
223	0.450	80	0.215	1660	6.81
224	0.442	82	0.213	1629	6.52

TABLE - A1-15

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Conden-sate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
225	100(98.807)		50.0	61.6	11.6	88.0	94.0	6.0
226	200(197.614)		50.5	56.4	5.9	87.0	93.5	6.5
227	300(296.421)	5°	50.0	54.2	4.2	86.0	93.0	7.0
228	400(395.228)		50.4	53.6	3.2	84.0	93.0	8.0
229	100(98.807)		50.0	63.7	13.7	91.0	95.0	4.5
230	200(197.614)		50.5	57.7	7.2	90.0	95.0	5.0
231	300(296.421)	10°	50.5	55.5	5.0	88.0	94.0	6.0
232	400(395.228)		50.0	54.0	4.0	87.0	93.5	6.5
233	100(98.807)		50.0	63.0	13.0	93.0	97.0	3.5
234	200(197.614)		50.0	57.7	7.7	92.0	96.5	4.0
235	300(296.421)	15°	50.5	56.1	5.6	90.0	96.0	5.0
236	400(395.228)		50.2	54.8	4.6	89.0	95.0	5.5
237	100(98.807)		50.0	68.0	18.0	93.0	97.0	3.5
238	200(197.614)		50.0	60.0	10.0	92.0	96.5	4.0
239	300(296.421)	19°	50.2	57.6	7.4	90.0	95.0	5.0
240	400(395.228)		50.0	55.6	5.6	89.0	95.0	5.5

TABLE - A1-15 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. Kcal / hr.m^2.°C	$\bar{h}_{ic}$ Theo. Kcal / hr.m^2 °C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
225	2.133	1150	1146	7.74	12898	10608
226	2.260	1218	1165	8.19	12610	10398
227	2.291	1234	1185	8.30	11863	10207
228	2.373	1279	1268	8.61	10758	9872
229	2.569	1385	1353	6.63	14737	11381
230	2.640	1423	1423	6.81	13627	11085
231	2.780	1498	1480	7.17	11955	10591
232	3.007	1620	1581	7.76	11934	10381
233	2.765	1490	1304	5.23	14958	12039
234	2.887	1555	1522	5.46	13659	11644
235	3.247	1750	1660	6.15	12298	11012
236	3.430	1847	1818	6.49	11799	10753
237	3.485	1878	1778	5.56	15907	12012
238	3.682	1984	1976	5.88	14705	11627
239	4.144	2234	2193	6.62	13246	10996
240	4.220	2273	2213	6.74	12253	10737

TABLE - A1-15 (contd.)

Serial No.	$\bar{h}_1 \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
225	0.438	85	0.212	711	5.36
226	0.428	90	0.208	695	4.95
227	0.403	92	0.207	654	4.59
228	0.365	95	0.205	593	4.02
229	0.500	73	0.224	1157	7.81
230	0.462	75	0.222	1070	7.03
231	0.406	79	0.218	939	5.86
232	0.405	86	0.213	937	5.40
233	0.507	58	0.242	1590	10.13
234	0.463	60	0.239	1452	8.87
235	0.417	68	0.229	1307	7.09
236	0.400	71	0.226	1253	6.45
237	0.539	61	0.238	1991	10.7
238	0.500	65	0.232	1841	9.37
239	0.449	73	0.224	1658	7.50
240	0.416	74	0.223	1534	6.80

TABLE - A1-16

LIQUID = ETHYL ACETATE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
241	100 (99.568)		30.0	34.0	4.0	53.5	68	11.80
242	200 (199.136)		30.0	32.1	2.1	51.0	68	13.05
243	300 (298.70)	5°	30.2	31.6	1.4	49.0	66	14.05
244	300 (398.27)		30.0	31.0	1.0	47.0	65	15.05
245	100 (99.568)		30.0	34.9	4.9	54.5	68	11.30
246	200 (199.136)		30.5	33.1	2.6	52.0	67	12.55
247	300 (298.70)	10°	30.2	32.2	2.0	50.0	67	13.55
248	400 (398.27)		30.0	31.5	1.5	49.0	66	14.05
249	100 (99.568)		29.8	35.2	5.4	60.0	72	8.55
250	200 (199.136)		30.0	32.8	2.8	58.0	70	9.55
251	300 (298.70)	15°	30.0	32.1	2.1	55.5	69	10.8
252	400 (398.27)		30.2	31.9	1.7	53.5	69	11.85
253	100 (99.568)		30.0	36.0	6.0	62.0	73	7.55
254	200 (199.136)		30.2	33.3	3.1	60.0	71	8.55
255	300 (298.70)	19°	30.2	32.4	2.2	58.0	70	9.55
256	400 (398.27)		30.0	31.8	1.8	55.5	69	10.80

TABLE - A1-16 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. Kcal / hr.m^2.°C	$\bar{h}_{ic}$ Theo. Kcal / hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
241	4.704	403	398	2.71	2298	1992
242	4.912	421	418	2.83	2170	1942
243	5.200	445	418	2.99	2131	1907
244	5.042	432	398	2.91	1931	1874
245	6.369	545	488	2.61	2309	2010
246	6.792	582	518	2.78	2220	1958
247	7.043	603	597	2.89	2130	1921
248	7.043	603	597	2.89	2055	1904
249	6.590	563	537	1.98	2316	2141
250	7.006	601	558	2.11	2207	2083
251	7.739	663	627	2.33	2157	2020
252	8.270	709	677	2.49	2102	1976
253	7.120	627	597	1.86	2462	2205
254	7.390	651	617	1.93	2257	2138
255	8.00	704	657	2.09	2186	2079
256	8.67	763	717	2.26	2094	2016

TABLE - A1-16 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) X10^{-13}
(0)	(13)	(14)	(15)	(16)	(17)
241	0.312	195	0.161	499	1.60
242	0.295	204	0.159	472	1.45
243	0.289	216	0.156	463	1.35
244	0.262	209	0.157	420	1.26
245	0.314	189	0.164	715	1.83
246	0.302	201	0.160	687	1.65
247	0.289	208	0.158	659	1.53
248	0.279	208	0.158	636	1.47
249	0.315	142	0.179	971	2.44
250	0.300	151	0.176	924	2.19
251	0.293	167	0.170	903	1.94
252	0.285	179	0.166	880	1.76
253	0.334	130	0.183	1214	2.92
254	0.307	135	0.181	1113	2.58
255	0.297	146	0.176	1078	2.31
256	0.285	158	0.171	1033	2.04

TABLE - A1-17

LIQUID SYSTEM = ETHYL-ACETATE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
257	100 (99.225)	5°	40.0	43.4	3.4	58.0	71	9.55
258	200 (198.45)		40.5	42.3	1.8	55.5	71	10.8
259	300 (297.70)		40.2	41.4	1.2	53.5	70	11.8
260	400 (396.90)		40.0	41.0	1.0	51.0	69	13.05
261	100 (99.225)	10°	39.5	43.5	4.0	60.0	71	8.55
262	200 (198.45)		40.0	42.1	2.1	58.0	71	9.55
263	300 (297.70)		40.0	41.6	1.6	55.5	70	10.8
264	400 (396.90)		40.0	41.3	1.3	53.5	69	11.8
265	100 (99.225)	15°	40.0	44.2	4.2	63.0	71	7.05
266	200 (198.45)		40.0	42.6	2.6	61.0	71	8.05
267	300 (297.70)		40.0	42.0	2.0	58.0	70	9.55
268	400 (396.90)		40.2	41.7	1.5	56.5	69	10.3
269	100 (99.225))	19°	40.0	45.4	5.4	64.0	73	6.55
270	200 (198.45)		40.2	43.2	3.0	62.0	72	7.55
271	300 (297.70)		40.0	42.2	2.2	60.0	71	8.55
272	400 (396.90)		40.2	41.9	1.7	58.0	70	9.55

TABLE - A1-17 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. Kcal/hr.m^2. °C	$\bar{h}_{ic_{Theo.}}$ Kcal/hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
257	3.985	341	337	2.29	2403	2100
258	4.239	363	357	2.44	2261	2036
259	4.408	378	357	2.54	2155	1992
260	4.704	403	397	2.72	2078	1942
261	5.000	429	397	2.05	2403	2156
262	5.410	464	417	2.22	2326	2096
263	5.787	496	476	2.37	2199	2033
264	6.190	530	516	2.54	2150	1989
265	5.690	486	416	1.71	2427	2247
266	6.310	541	516	1.90	2361	2174
267	7.150	615	595	2.16	2255	2083
268	7.350	631	595	2.22	2149	2044
269	6.390	563	536	1.67	2548	2285
270	7.200	634	595	1.88	2489	2205
271	7.500	665	655	1.97	2306	2138
272	7.790	685	675	2.03	2126	2080

TABLE - A1 - 17 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\bar{N}u$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
257	0.327	166	0.170	522	1.98
258	0.307	176	0.167	491	1.75
259	0.293	183	0.164	468	1.61
260	0.283	195	0.161	452	1.45
261	0.327	148	0.177	744	2.42
262	0.316	160	0.173	720	2.17
263	0.299	171	0.169	681	1.92
264	0.292	183	0.165	665	1.76
265	0.330	123	0.188	1016	2.97
266	0.321	136	0.182	989	2.60
267	0.307	154	0.175	945	2.19
268	0.292	159	0.173	900	2.03
269	0.346	116	0.190	1257	3.37
270	0.338	131	0.182	1226	2.92
271	0.313	137	0.180	1137	2.58
272	0.289	142	0.178	1048	2.31

TABLE - A1-18

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Conden-sate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
273	100(98.807)	5°	49.8	51.8	2.0	66.0	72	5.55
274	200(197.614)		50.0	51.1	1.1	64.0	72	6.55
275	300(296.421)		50.0	50.8	0.8	62.0	71	7.55
276	400(395.228)		50.2	50.8	0.6	61.0	70	8.05
277	100(98.807)	10°	50.0	53.0	3.0	66.0	73	5.5
278	200(197.614)		50.5	52.0	1.5	65.0	73	6.0
279	300(296.421)		50.2	51.4	1.2	63.0	72	7.0
280	400(395.228)		50.0	50.9	0.9	62.0	71	7.5
281	100(98.807)	15°	49.5	53.5	4.0	68.0	74	4.55
282	200(197.614)		50.0	52.2	2.2	66.0	73	5.55
283	300(296.421)		50.0	51.6	1.6	63.0	72	7.05
284	400(395.228)		50.2	51.5	1.3	62.0	71	7.55
285	100(98.807)	19°	50.5	55.0	4.5	70.0	74	3.55
286	200(197.614)		50.0	52.4	2.4	68.0	74	4.55
287	300(296.421)		50.0	51.8	1.8	66.0	73	5.55
288	400(395.228)		50.2	51.6	1.4	65.0	72	6.05

TABLE - A1-18 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{ic_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
273	2.569	220	198	1.48	2667	2405
274	2.694	231	217	1.55	2373	2308
275	3.025	259	237	1.74	2308	2227
276	3.067	263	237	1.77	2198	2192
277	3.740	320	296	1.53	2761	2401
278	3.790	325	296	1.56	2572	2350
279	4.240	363	356	1.74	2465	2262
280	4.240	363	356	1.74	2302	2223
281	5.240	450	395	1.58	3467	2507
282	5.523	473	435	1.66	2994	2385
283	5.890	503	474	1.77	2516	2247
284	6.050	517	514	1.81	2411	2209
285	5.100	449	445	1.33	3750	2663
286	5.800	511	474	1.51	3329	2503
287	6.410	564	533	1.67	3012	2382
288	6.740	593	553	1.76	2906	2331

TABLE - A1 - 18 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusslet Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
273	0.362	107	0.197	580	3.41
274	0.322	112	0.194	516	2.89
275	0.314	126	0.186	502	2.51
276	0.299	127	0.186	478	2.35
277	0.375	111	0.195	854	3.74
278	0.349	112	0.195	796	3.42
279	0.335	126	0.187	763	2.94
280	0.313	126	0.187	712	2.74
281	0.471	113	0.193	1452	4.59
282	0.407	119	0.190	1254	3.77
283	0.342	127	0.186	1054	2.97
284	0.328	131	0.184	1010	2.77
285	0.510	93	0.204	1849	6.21
286	0.453	106	0.196	1642	4.85
287	0.409	117	0.189	1486	3.98
288	0.395	123	0.186	1433	3.65

TABLE - A1-19

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
289	100 (99.568)	5°	30.0	34.0	4.0	53.5	69	12.40
290	200 (199.136)		30.2	32.4	2.2	51.0	68	13.65
291	300 (298.70)		30.0	31.4	1.4	49.0	66	14.65
292	400 (398.27)		29.8	30.9	1.1	47.5	65	15.40
293	100 (99.568)	10°	29.5	34.5	5.0	56.5	69	10.90
294	200 (199.136)		30.0	32.8	2.8	55.0	68	11.65
295	300 (298.7)		30.0	31.9	1.9	53.5	66	12.40
296	400 (398.27)		30.2	31.7	1.5	51.0	65	13.65
297	100 (99.568)	15°	30.0	36.0	6.0	61.0	73	8.65
298	200 (199.136)		30.5	33.9	3.4	58.0	71	10.15
299	300 (298.70)		30.0	32.2	2.2	55.5	70	11.40
300	400 (398.27)		30.0	31.8	1.8	53.5	69	12.40
301	100 (99.568)	19°	29.6	36.6	7.0	63.0	74	7.65
302	200 (199.136)		30.0	33.8	3.8	60.0	72	9.15
303	300 (298.70)		30.0	32.6	2.6	58.0	71	10.15
304	400 (398.27)		30.2	32.4	2.2	55.5	70	11.40

TABLE - A1-19 (contd.)

Serial No.	Average Condensate Rate	Heat - Released by vapour	Heat - Received by Coolant	Heat - Flux ($\bar{q}$) X10^{-4}	$\bar{h}_{ic_{Expt.}}$	$\bar{h}_{ic_{Theo.}}$
	kg/hr	Kcal/hr	Kcal/hr	Kcal/hr.m^2	$\frac{Kcal}{hr.m^2.\ °C}$	$\frac{Kcal}{hr.m^2\ °C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
289	1.976	415	398	2.79	2252	1725
290	2.109	443	438	2.98	2188	1685
291	2.026	425	418	2.86	1952	1655
292	2.086	438	438	2.95	1913	1635
293	2.576	541	497	2.59	2376	1780
294	2.674	561	557	2.69	2305	1723
295	2.722	572	567	2.74	2208	1682
296	2.900	609	597	2.92	2136	1660
297	2.869	602	597	2.11	2442	1873
298	3.326	698	677	2.45	2416	1800
299	3.550	746	657	2.62	2296	1748
300	3.712	779	717	2.74	2207	1711
301	3.349	703	697	2.08	2725	1928
302	3.725	782	757	2.32	2534	1844
303	3.818	801	777	2.37	2339	1797
304	4.179	878	876	2.60	2283	1745

TABLE - A1-19 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
289	0.585	48	0.257	568	1.64
290	0.568	50	0.253	551	1.49
291	0.507	49	0.255	492	1.39
292	0.497	50	0.253	482	1.32
293	0.617	44	0.266	853	2.04
294	0.599	45	0.264	828	1.91
295	0.574	46	0.262	793	1.80
296	0.555	49	0.256	767	1.63
297	0.634	36	0.283	1187	2.59
298	0.627	41	0.271	1175	2.21
299	0.596	44	0.265	1116	1.99
300	0.574	46	0.261	1073	1.81
301	0.708	35	0.283	1560	3.10
302	0.658	39	0.273	1450	2.59
303	0.608	40	0.271	1339	2.33
304	0.593	44	0.262	1307	2.08

TABLE - A1-20

LIQUID SYSTEM = ETHYL-ALCOHOL

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP.(T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
305	100 (99.225)	5°	40.0	43.5	3.5	58.0	71	10.15
306	200 (198.45)		40.0	42.0	2.0	55.5	70	11.40
307	300 (297.70)		40.2	41.5	1.3	53.5	68	12.40
308	400 (396.90)		40.2	41.2	1.0	51.0	66	13.65
309	100 (99.225)	10°	40.0	44.5	4.5	60.0	71	9.15
310	200 (198.45)		40.0	42.4	2.4	58.0	70	10.15
311	300 (297.70)		40.2	41.8	1.6	55.5	70	11.40
312	400 (396.90)		40.0	41.3	1.3	53.5	69	12.40
313	100 (99.225)	15°	40.0	45.0	5.0	64.0	73	7.15
314	200 (198.45)		40.5	43.3	2.8	62.0	72	8.15
315	300 (297.7)		40.2	42.2	2.0	60.0	71	9.15
316	400 (396.90)		40.0	41.5	1.5	58.0	70	10.15
317	100 (99.225)	19°	40.0	46.2	6.2	66.0	74	6.15
318	200 (198.45)		40.0	43.2	3.2	64.0	73	7.15
319	300 (297.70)		40.0	42.4	2.4	62.0	72	8.15
320	400 (396.90)		40.2	42.0	1.8	60.0	72	9.15

TABLE - A1-20 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. Kcal/hr.m^2.°C	$\bar{h}_{ic}$ Theo. Kcal/hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
305	1.757	369	347	2.48	2446	1814
306	1.909	400	397	2.69	2361	1762
307	1.980	416	417	2.80	2257	1725
308	2.102	441	436	2.97	2174	1684
309	2.170	456	446	2.18	2386	1859
310	2.323	488	476	2.34	2302	1811
311	2.583	542	476	2.60	2277	1759
312	2.736	575	516	2.75	2221	1723
313	2.539	533	496	1.87	2619	1964
314	2.778	583	556	2.05	2513	1901
315	2.949	619	596	2.17	2377	1846
316	3.042	639	596	2.24	2212	1800
317	2.933	616	615	1.83	2970	2056
318	3.236	680	635	2.02	2820	1961
319	3.480	731	715	2.17	2659	1898
320	3.480	731	715	2.17	2368	1844

TABLE - A1-20 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
305	0.635	42	0.268	617	2.00
306	0.614	46	0.261	596	1.78
307	0.586	48	0.257	569	1.64
308	0.565	50	0.253	548	1.49
309	0.620	37	0.282	857	2.43
310	0.598	39	0.276	827	2.19
311	0.591	44	0.266	818	1.95
312	0.577	47	0.260	798	1.79
313	0.681	32	0.295	1273	3.14
314	0.653	35	0.286	1222	2.75
315	0.618	37	0.281	1156	2.45
316	0.575	38	0.278	1075	2.21
317	0.772	31	0.295	1700	3.85
318	0.732	34	0.286	1614	3.32
319	0.691	37	0.278	1522	2.91
320	0.615	37	0.278	1355	2.59

TABLE - A1-21

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
321	100 (98.807)	5°	49.8	52.8	3.0	62.0	72	8.15
322	200 (197.614)		50.0	51.5	1.5	60.0	71	9.15
323	300 (296.421)		50.0	51.2	1.2	58.0	70	10.15
324	400 (395.228)		50.2	51.1	0.9	55.5	69	11.40
325	100 (98.807)	10°	49.5	53.0	3.5	64.0	73	7.15
326	200 (197.614)		50.0	51.8	1.8	62.0	73	8.15
327	300 (296.421)		50.0	51.4	1.4	60.0	72	9.15
328	400 (395.228)		50.2	51.4	1.2	58.0	70	10.15
329	100 (98.807)	15°	50.0	54.5	4.5	68.0	74	5.15
330	200 (197.614)		50.0	52.4	2.4	66.0	74	6.15
331	300 (296.421)		50.0	51.6	1.6	64.0	73	7.15
332	400 (395.228)		50.0	51.4	1.4	62.0	72	8.15
333	100 (98.807)	19°	50.5	55.0	4.5	70.0	75	4.15
334	200 (197.614)		50.0	52.6	2.6	68.0	75	5.15
335	300 (296.421)		50.0	51.8	1.8	66.0	74	6.15
336	400 (395.228)		50.2	51.7	1.5	64.0	72	7.15

TABLE - A1-21 (contd.)

Serial No.	Average Condensate Rate	Heat - Released by vapour	Heat - Received by Coolant	Heat - Flux ($\bar{q}$) $X10^{-4}$	$\bar{h}_{ic}$ Expt.	$\bar{h}_{ic}$ Theo.
	Kg/hr	Kcal/hr	Kcal/hr	Kcal/hr.m^2	$\frac{Kcal}{hr.m^2.°C}$	$\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
321	1.476	310	296	2.09	2559	1916
322	1.596	335	296	2.25	2464	1862
323	1.701	357	355	2.40	2366	1814
324	1.701	357	355	2.40	2107	1762
325	1.752	368	346	1.76	2598	1977
326	1.959	411	356	1.97	2415	1913
327	2.149	451	415	2.16	2360	1859
328	2.288	481	474	2.30	2269	1811
329	2.171	456	445	1.60	3111	2132
330	2.263	475	474	1.67	2714	2039
331	2.480	521	474	1.83	2560	1964
332	2.641	555	553	1.95	2392	1901
333	2.131	448	445	1.33	3200	2247
334	2.468	519	514	1.54	2988	2129
335	2.708	568	534	1.68	2738	2036
336	2.869	603	593	1.79	2500	1961

TABLE - A1-21 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
321	0.665	35	0.285	645	2.50
322	0.640	38	0.278	621	2.22
323	0.615	41	0.271	597	2.00
324	0.548	41	0.271	531	1.78
325	0.675	30	0.302	933	3.11
326	0.627	33	0.292	867	2.72
327	0.613	37	0.282	847	2.43
328	0.589	39	0.277	815	2.19
329	0.808	27	0.312	1513	4.35
330	0.705	28	0.308	1319	3.65
331	0.665	31	0.298	1245	3.13
332	0.621	33	0.292	1163	2.75
333	0.832	22	0.331	1832	5.71
334	0.776	26	0.313	1710	4.61
335	0.711	29	0.302	1567	3.85
336	0.649	30	0.298	1431	3.31

TABLE - A1-22

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
337	100(99.568)		29.5	32.7	3.2	51.0	67.	12.875
338	200(199.136)		30.0	31.6	1.6	49.0	66	13.875
339	300(298.70)	5°	30.0	31.2	1.2	47.0	64	14.875
340	400(398.27)		30.0	30.9	0.9	45.0	63	15.875
341	100(99.568)		30.0	34.0	4.0	54.5	68	11.125
342	200(199.136)		30.2	32.3	2.1	51.0	67	12.875
343	300(298.70)	10°	30.0	31.5	1.5	49.0	65	13.875
344	400(398.27)		30.5	31.8	1.3	47.0	65	14.875
345	100(99.568)		29.0	33.9	4.9	58.0	69	9.375
346	200(199.136)		30.0	32.7	2.7	54.5	68	11.125
347	300(298.70)	15°	30.0	32.2	2.2	49.0	66	13.875
348	400(398.27)		30.2	32.0	1.8	47.0	65	14.875
349	100(99.568)		30.0	35.5	5.5	60.0	70	8.375
350	200(199.136)		29.8	32.8	3.0	56.5	69	10.125
351	300(298.70)	19°	30.0	32.4	2.4	52.0	67	12.375
352	400(398.27)		30.2	32.1	1.9	49.0	66	13.875

TABLE - A1-22 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. Kcal / hr.m^2. °C	$\bar{h}_{ic}$ Theo. Kcal / hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
337	7.13	337	332	2.27	1761	1741
338	7.48	353	319	2.37	1712	1709
339	7.61	360	358	2.42	1628	1680
340	7.61	360	358	2.42	1528	1653
341	9.00	425	398	2.04	1829	1803
342	9.65	456	418	2.18	1695	1738
343	10.3	486	477	2.33	1677	1706
344	11.01	520	518	2.49	1674	1676
345	11.34	535	488	1.88	2005	1869
346	12.32	582	538	2.05	1831	1791
347	15.05	691	657	2.43	1749	1695
348	15.89	750	717	2.63	1772	1666
349	12.67	598	548	1.77	2116	1920
350	14.00	661	598	1.97	1936	1832
351	16.43	776	717	2.30	1859	1742
352	17.18	811	756	2.40	1733	1693

TABLE - A1-22 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{f_c}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
337	0.288	143	0.178	411	1.25
338	0.280	150	0.175	400	1.16
339	0.266	152	0.175	380	1.08
340	0.250	152	0.175	356	1.01
341	0.299	128	0.186	608	1.58
342	0.277	137	0.182	564	1.37
343	0.275	146	0.178	558	1.27
344	0.274	157	0.174	557	1.18
345	0.328	118	0.191	902	1.89
346	0.299	128	0.186	824	1.59
347	0.286	156	0.174	787	1.28
348	0.290	165	0.171	798	1.19
349	0.347	112	0.192	1121	2.24
350	0.317	124	0.186	1026	1.85
351	0.304	145	0.176	985	1.52
352	0.284	152	0.174	919	1.35

TABLE - A1-23

LIQUID SYSTEM = CARBON-TETRA-CHLORIDE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
353	100 (99.225)	5°	40.4	43.0	2.6	58.0	69	9.375
354	200 (198.45)		40.0	41.4	1.4	54.5	68	11.125
355	300 (297.70)		40.2	41.2	1.0	53.5	67	11.625
356	400 (396.90)		40.0	40.8	0.8	51.0	66	12.875
357	100 (99.225)	10°	39.8	43.1	3.3	60.0	70	8.375
358	200 (198.45)		40.0	41.9	1.9	55.5	69	10.625
359	300 (297.70)		40.0	41.4	1.4	53.5	67	11.625
360	400 (396.90)		40.2	41.3	1.1	51.0	66	12.875
361	100 (99.225)	15°	40.0	44.3	4.3	63.0	72	6.875
362	200 (198.45)		40.2	42.8	2.6	60.0	70	8.375
363	300 (297.70)		40.0	42.0	2.0	55.5	69	10.625
364	400 (396.90)		40.5	41.9	1.4	53.5	67	11.625
365	100 (99.225)	19°	40.0	44.5	4.5	64.0	73	6.375
366	200 (198.45)		40.0	42.7	2.7	61.0	71	7.875
367	300 (297.70)		40.5	42.7	2.2	58.0	70	9.375
368	400 (396.90)		40.5	42.2	1.7	55.5	68	10.625

TABLE - A1-23 (contd.)

Serial No.	Average Condensate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{ic_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{ic}$ Theo. $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
353	5.47	259	258	1.74	1859	1885
354	6.37	301	278	2.03	1820	1806
355	6.63	313	298	2.11	1811	1786
356	6.79	320	317	2.15	1672	1741
357	7.56	357	327	1.71	2041	1935
358	8.10	382	377	1.83	1721	1824
359	8.96	423	417	2.03	1742	1783
360	9.53	450	437	2.15	1673	1738
361	9.15	431	426	1.51	2202	2020
362	11.01	520	516	1.83	2181	1923
363	13.42	634	596	2.23	2096	1812
364	13.56	640	556	2.25	1934	1772
365	10.125	478	447	1.42	2229	2056
366	12.32	582	536	1.72	2191	1950
367	14.35	678	655	2.01	2144	1867
368	14.92	704	674	2.09	1964	1810

TABLE - A1 -23 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
353	0.304	109	0.195	434	1.72
354	0.298	127	0.186	425	1.45
355	0.297	133	0.183	423	1.38
356	0.274	136	0.182	391	1.25
357	0.334	108	0.197	679	2.10
358	0.282	115	0.193	572	1.66
359	0.285	128	0.186	579	1.52
360	0.274	136	0.182	566	1.37
361	0.361	95	0.205	991	2.58
362	0.357	114	0.192	982	2.12
363	0.343	139	0.181	943	1.67
364	0.317	141	0.180	871	1.53
365	0.365	90	0.207	1181	2.95
366	0.358	109	0.194	1161	2.38
367	0.351	127	0.184	1136	2.00
368	0.322	132	0.182	1041	1.77

TABLE - A1-24

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1') °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
369	100 (98.807)	5°	49.8	51.8	2.0	64.0	72	6.375
370	200 (197.614)		50.0	51.1	1.1	62.0	72	7.550
371	300 (296.421)		50.0	50.8	0.8	61.0	70	7.875
372	400 (395.228)		50.0	50.6	0.6	60.0	70	8.375
373	100 (98.807)	10°	50.0	52.4	2.4	65.0	73	5.875
374	200 (197.614)		50.2	51.6	1.4	63.0	71	6.875
375	300 (296.421)		50.0	51.0	1.0	62.0	71	7.550
376	400 (395.228)		50.4	51.2	0.8	60.0	70	8.375
377	100 (98.807)	15°	49.2	53.0	3.8	65.0	73	5.875
378	200 (197.614)		50.0	52.0	2.0	63.0	72	6.875
379	300 (296.421)		50.0	51.5	1.5	62.0	72	7.550
380	400 (395.228)		50.2	51.4	1.2	60.0	70	8.375
381	100 (98.807)	19°	50.0	53.8	3.8	68.0	74	4.375
382	200 (197.614)		50.2	52.4	2.2	65.0	73	5.875
383	300 (296.421)		50.2	51.8	1.6	64.0	71	6.375
384	400 (395.228)		50.0	51.2	1.2	63.0	71	6.875

TABLE - A1-24 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{ic}$ Expt. $\frac{Kcal}{hr.m^2.\ °C}$	$\bar{h}_{ic_{Theo}}$ $\frac{Kcal}{hr.m^2.\ °C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
369	4.45	210	198	1.41	2216	2076
370	5•01	237	217	1;59	2112	1990
371	5.21	246	237	1.66	2025	1969
372	5.22	247	237	1.66	1985	1939
373	5.67	268	237	1.28	2184	2115
374	6.40	303	277	1.45	2110	2033
375	7.08	335	297	1.60	2124	1986
376	7.08	335	316	1.60	1915	1936
377	7.99	378	375	1.33	2267	2101
378	8.40	397	395	1.39	2029	2021
379	9.56	452	445	1.56	2103	1974
380	10.125	478	474	1.68	2005	1923
381	8.01	378	376	1.12	2561	2259
382	9.45	446	435	1.32	2250	2098
383	10.03	474	474	1.40	2204	2056
384	10.03	474	474	1.40	2044	2017

TABLE - A1-24 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
,369	0.363	89	0.209	518	2.53
370	0.346	100	0.201	493	2.13
371	0.332	104	0.198	473	2.04
372	0.325	104	0.198	464	1.92
373	0.358	81	0.216	726	2.99
374	0.346	91	0.208	702	2.56
375	0.348	101	0.202	706	2.33
376	0.314	101	0.202	634	2.10
377	0.371	83	0.214	1020	3.03
378	0.332	87	0.211	913	2.59
379	0.344	99	0.202	947	2.36
380	0.328	105	0.198	903	2.12
381	0.419	71	0.223	1357	4.29
382	0.369	84	0.212	1193	3.19
383	0.361	89	0.207	1168	2.94
384	0.335	89	0.207	1083	2.73

TABLE - A1-25

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING - CONVERGING CONE SECTION

VAPOUR TEMP. (T_V) = 100°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_W)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_V - T_W}{2}$
			°C			°C	°C	°C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
385	100 (99.568)		29.5	56.0	26.5	80.0	91.0	10.0
386	200 (199.136)		30.0	44.0	14.0	78.0	90.0	11.0
387	300 (298.70)	5°	31.0	40.5	9.5	76.0	89.0	12.0
388	400 (398.27)		30.0	38.0	8.0	72.0	86.0	14.0
389	100 (99.568)		30.0	58.7	28.7	86.0	93.5	7.0
390	200 (199.136)		31.0	47.4	16.4	82.0	91.0	9.0
391	300 (298.70)	10°	30.0	42.0	12.0	78.0	89.0	11.0
392	400 (398.27)		30.0	39.3	9.3	76.5	88.0	11.75
393	100 (99.568)		30.0	64.0	34.0	88.5	94.0	5.75
394	200 (199.136)		30.0	48.5	18,5	86.0	93.5	7.00
395	300 (298.70)	15°	30.0	42.6	12.6	84.0	92.0	8.00
396	400 (398.27)		30.0	40.0	10.0	83.0	91.0	8.55
397	100 (99.568)		29.5	64.5	35.0	90.0	95.5	5.00
398	200 (199.136)		30.0	49.0	19.0	88.5	94.5	5.75
399	300 (298.70)	19°	30.0	43.8	13.8	86.0	92.5	7.00
400	400 (398.27)		30.4	40.9	10.5	84.0	91.5	8.00

TABLE - A1-25 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ Kcal/hr.m^2.°C	$\bar{h}_{idc_{Theo.}}$ Kcal/hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
385	4.971	2680	2638	9.02	9017	7847
386	5.251	2830	2787	9.52	8656	7662
387	5.391	2903	2837	9.76	8139	7497
388	6.216	3350	3186	11.27	8051	7214
389	5.399	2910	2858	6.96	9952	8565
390	6.359	3427	3266	8.20	9119	8043
391	7.048	3800	3591	9.09	8271	7650
392	7.340	3956	3704	9.47	8061	7485
393	6.330	3410	3385	6.00	10419	8939
394	6.873	3703	3684	6.50	9293	8510
395	7.077	3814	3764	6.70	8371	8230
396	7.435	4007	3983	7.04	8284	8106
397	6.666	3592	3485	5.32	10649	9243
398	7.151	3853	3783	5.71	9933	8925
399	7.753	4178	4122	6.19	8847	8498
400	8.519	4590	4182	6.30	8505	8219

TABLE - A1-25 (contd.)

Serial No.	$\bar{h}_1 \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
385	0.306	199	0.160	497	2.57
386	0.294	210	0.157	477	2.34
387	0.276	216	0.156	449	2.14
388	0.273	249	0.148	444	1.84
389	0.338	154	0.175	781	4.01
390	0.309	181	0.166	716	3.12
391	0.281	200	0.160	650	2.56
392	0.274	209	0.158	633	2.34
393	0.354	132	0.184	1108	4.94
394	0.315	143	0.179	988	4.06
395	0.285	147	0.177	890	3.55
396	0.281	155	0.174	880	3.34
397	0.361	117	0.189	1333	5.96
398	0.337	126	0.185	1243	5.18
399	0.300	136	0.180	1107	4.26
400	0.289	150	0.174	1065	3.72

TABLE - A1-26

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
401	100 (99.225)	5°	39.5	65.5	26.0	82.0	92	9.0
402	200 (198.45)		40.0	53.0	13.0	81.0	92	9.5
403	300 (297.70)		40.0	48.7	8.7	79.0	91	10.5
404	400 (396.90)		40.5	47.5	7.0	77.0	91	11.5
405	100 (99.225)	10°	40.0	63.4	23.4	89.0	94.5	5.5
406	200 (198.45)		40.0	51.9	11.9	88.0	94.5	6.0
407	300 (297.70)		40.5	48.8	8.3	86.0	93.0	7.0
408	400 (396.90)		40.0	46.6	6.6	85.0	92.5	7.5
409	100 (99.225)	15°	40.5	66.7	26.2	92.0	96.0	4.0
410	200 (198.45)		40.0	53.6	13.6	91.0	95.5	4.5
411	300 (297.70)		40.5	51.2	10.7	89.0	94.5	5.5
412	400 (396.90)		40.0	48.6	8.6	87.0	93.0	7.0
413	100 (99.225)	19°	40.0	66.4	26.4	94.0	97.0	3.0
414	200 (198.45)		40.0	53.9	13.9	92.0	96.0	4.0
415	300 (297.70)		40.0	51.0	11.0	90.0	95.5	5.0
416	400 (396.90)		40.0	48.9	8.9	89.0	95.0	5.5

TABLE - A1-26 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
401	4.971	2680	2638	9.02	9017	7847
402	5.251	3830	2787	9.52	8656	7662
403	5.391	2903	2837	9.76	8139	7497
404	6.216	3350	3186	11.27	8051	7214
405	4.434	2390	2321	5.72	10405	9097
406	4.472	2410	2362	5.77	9617	8902
407	4.775	2573	2473	6.16	8810	8565
408.	5.000	2695	2619	6.45	8612	8419
409	4.876	2627	2600	4.62	11538	9783
410	5.044	2718	2698	4.78	10611	9504
411	4.765	3239	3185	5.69	10346	9039
412	6.389	3440	3413	6.04	8633	8510
413	4.894	2637	2619	3.90	13030	10502
414	5.520	2975	2758	4.41	11025	9774
415	6.509	3508	3275	5.21	10400	9243
416	5.635	3575	3532	5.30	9635	9025

TABLE - A1-26 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{f_c}{Re_f^{\,1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
401	0.306	196	0.161	497	2.57
402	0.294	196	0.161	477	2.34
403	0.276	197	0.160	449	2.14
404	0.273	208	0.158	444	1.84
405	0.353	126	0.187	817	5.11
406	0.326	127	0.186	755	4.68
407	0.299	136	0.182	692	4.02
408	0.292	143	0.179	676	3.74
409	0.392	102	0.200	1226	7.09
410	0.360	105	0.198	1129	6.31
411	0.351	99	0.202	1100	5.16
412	0.293	133	0.183	918	4.06
413	0.356	86	0.206	1631	9.93
414	0.374	97	0.200	1380	7.45
415	0.353	114	0.188	1302	5.96
416	0.327	117	0.186	1206	5.42

TABLE - A1-27

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
417	100 (98.807)	5°	49.5	70.0	20.5	88.0	95.0	6.0
418	200 (197.614)		50.0	60.5	10.5	86.0	94.0	7.0
419	300 (296.421)		50.0	57.5	7.5	84.0	93.0	8.0
420	400 (395.228)		50.2	56.2	6.0	82.0	91.0	9.0
421	100 (98.807)	10°	50.0	73.6	23.6	91.0	95.5	4.5
422	200 (197.614)		50.0	62.6	12.6	90.0	95.0	5.0
423	300 (296.421)		51.0	59.5	8.5	88.0	94.5	6.0
424	400 (395.228)		50.0	56.6	6.6	86.0	93.0	6.5
425	100 (98.807)	15°	50.0	73.8	23.8	94.0	97.0	3.0
426	200 (197.614)		50.0	63.0	13.0	93.0	96.5	3.5
427	300 (296.421)		50.0	58.7	8.7	92.0	96.0	4.0
428	400 (395.228)		50.0	56.6	6.6	91.0	95.0	4.5
429	100 (98.806)	19°	49.5	73.5	24.0	95.5	98.5	2.25
430	200 (197.614)		50.0	63.6	13.6	94.0	97.5	3.0
431	300 (296.421)		50.2	59.4	9.2	93.0	97.0	3.5
432	400 (395.228)		50.0	57.2	7.2	92.0	96.0	4.0

TABLE - A1-27 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
417	3.859	2079	2023	6.99	11659	8916
418	3.958	2133	2075	7.18	10253	8579
419	4.362	2350	2223	7.91	9884	8297
420	4.472	2410	2371	8.10	9010	8056
421	4.452	2400	2332	5.75	12769	9566
422	4.631	2496	2490	5.98	11970	9317
423	4.745	2557	2519	6.12	10203	8902
424	4.921	2652	2608	6.35	9768	8725
425	4.372	2356	2351	4.14	13797	10518
426	4.876	2628	2569	4.62	13191	10120
427	4.943	2663	2578	4.68	11696	9788
428	4.982	2685	2608	4.72	10482	9504
429	4.423	2384	2371	3.53	15707	11286
430	5.036	2714	2688	4.02	13411	10502
431	5.308	2860	2727	4.24	12113	10105
432	5.308	2860	2846	4.24	10598	9773

TABLE - A1-27 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
417	0.396	154	0.174	643	4.29
418	0.348	158	0.173	566	3.67
419	0.335	174	0.167	545	3.21
420	0.306	179	0.166	497	2.86
421	0.433	127	0.187	1003	6.25
422	0.406	132	0.184	940	5.62
423	0.346	135	0.183	801	4.68
424	0.332	140	0.181	767	4.32
425	0.468	91	0.208	1467	9.46
426	0.448	101	0.201	1402	8.11
427	0.397	103	0.199	1243	7.09
428	0.356	104	0.198	1113	6.31
429	0.533	78	0.217	1966	13.24
430	0.455	88	0.208	1679	9.93
431	0.411	93	0.205	1516	8.51
432	0.359	93	0.205	1327	7.45

TABLE - A1-28

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
433	100(99.568)	5°	29.5	34.5	5.0	58.0	70	9.55
434	200(199.136)		30.0	32.7	2.7	55.5	68	10.80
435	300(298.70)		30.0	32.0	2.0	51.0	66	13.05
436	400(398.27)		30.0	31.5	1.5	49.0	65	14.05
437	100(99.568)	10°	29.8	36.8	7.0	61.0	71	8.05
438	200(199.136)		30.0	33.8	3.8	59.0	69.5	9.05
439	300(298.70)		30.0	32.6	2.6	55.5	68	10.8
440	400(398.27)		30.2	32.4	2.2	53.5	67	11.8
441	100(99.568)	15°	30.5	38.5	8.0	64.0	72	6.55
442	200(199.136)		30.0	34.1	4.1	62.0	71	7.55
443	300(298.70)		30.0	33.0	3.0	60.0	69	8.55
444	400(398.27)		30.0	32.4	2.4	57.5	68	7.80
445	100(99.568)	19°	29.5	38.3	8.8	65.0	73	6.05
446	200(199.136)		30.0	34.9	4.9	63.0	72	7.05
447	300(298.70)		30.0	33.8	3.8	61.0	71	8.05
448	400(398.27)		30.2	33.2	3.0	59.0	70	9.05

TABLE - A1-28 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
433	9.55	521	498	1.75	1832	1765
434	10.80	541	537	1.82	1685	1711
435	13.05	603	597	2.03	1554	1632
436	14.05	603	597	2.03	1444	1603
437	8.217	705	697	1.68	2096	1839
438	8.874	760	757	1.82	2010	1786
439	9.310	797	776	1.91	1767	1709
440	10.240	877	876	2.10	1779	1671
441	9.576	820	796	1.44	2199	1924
442	10.310	884	816	1.55	2057	1857
443	11.00	943	896	1.66	1937	1800
444	11.68	1000	955	1.76	1792	1740
445	11.09	946	876	1.40	2317	1959
446	11.68	1000	978	1.48	2103	1886
447	13.31	1140	1135	1.69	2099	1825
448	14.16	1213	1194	1.80	1986	1772

TABLE - A1-28 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{N_1}$)	Condensation Number (C_v) X10^{-13}
(0)	(13)	(14)	(15)	(16)	(17)
433	0.249	396	0.127	398	1.59
434	0.229	448	0.122	366	1.40
435	0.211	541	0.114	338	1.16
436	0.196	583	0.112	314	1.08
437	0.285	243	0.151	648	2.06
438	0.273	262	0.147	622	1.83
439	0.240	276	0.144	547	1.53
440	0.242	303	0.140	550	1.40
441	0.299	207	0.158	921	2.55
442	0.279	223	0.154	861	2.22
443	0.263	238	0.151	811	1.96
444	0.244	252	0.148	751	1.70
445	0.315	202	0.157	1142	2.90
446	0.285	213	0.155	1037	2.49
447	0.285	242	0.148	1035	2.18
448	0.270	258	0.146	979	1.94

TABLE - A1-29

LIQUID SYSTEM = ETHYL-ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_V) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) (deg)	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
449	100 (99.225)	5°	40.0	44.0	4.0	62.0	71	7.55
450	200 (198.45)		40.5	42.8	2.3	60.0	70	8.55
451	300 (297.70)		40.0	41.6	1.6	58.0	70	9.55
452	400 (396.90)		40.2	41.6	1.4	53.5	66.5	11.80
453	100 (99.225)	10°	40.0	45.0	5.0	66.0	73	5.55
454	200 (198.45)		40.0	43.0	3.0	64.0	72	6.55
455	300 (297.70)		40.0	42.0	2.0	63.0	70	7.05
456	400 (396.90)		40.0	42.0	2.0	59.0	69	9.05
457	100 (99.225)	15°	39.5	46.1	6.6	68	74	4.55
458	200 (198.45)		40.0	43.4	3.4	66.0	72	5.55
459	300 (297.70)		40.0	42.6	2.6	65.0	72	6.05
460	400 (396.90)		40.0	42.1	2.1	62.0	71	7.55
461	100 (99.225)	19°	40.2	47.0	6.8	70.0	74	3.55
462	200 (198.45)		40.0	43.7	3.7	68.0	73	4.55
463	300 (297.70)		40.0	43.2	3.2	66.0	72	5.55
464	400 (396.90)		40.0	42.6	2.6	64.0	71	6.55

TABLE - A1 -29 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
449	7.55	421	397	1.42	1876	1872
450	8.55	471	456	1.58	1853	1814
451	9.55	513	476	1.73	1807	1765
452	11.80	561	555	1.89	1603	1674
453	6.28	537	496	1.23	2316	2018
454	7.12	610	595	1.46	2229	1936
455	7.39	633	595	1.52	2149	1901
456	9.374	801	794	1.92	2119	1786
457	7.65	655	654	1.15	2529	2107
458	8.53	731	675	1.28	2314	2006
459	9.055	776	774	1.36	2253	1963
460	9.788	839	834	1.47	1952	1857
461	7.92	679	675	1.00	2835	2239
462	9.44	810	734	1.20	2639	2105
463	11.28	966	952	1.43	2580	2002
464	12.32	1056	1032	1.56	2389	1921

TABLE - A1-29 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt No. $(\bar{Nu})$	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
449	0.255	313	0.137	407	2.00
450	0.252	355	0.132	403	1.77
451	0.245	396	0.127	392	1.59
452	0.218	490	0.118	348	1.28
453	0.315	186	0.164	717	2.98
454	0.303	211	0.158	690	2.53
455	0.292	219	0.156	665	2.35
456	0.288	277	0.144	656	1.83
457	0.344	165	0.170	1059	3.68
458	0.315	184	0.164	969	3.01
459	0.306	196	0.161	944	2.76
460	0.265	211	0.157	818	2.22
461	0.385	146	0.176	1398	4.95
462	0.359	172	0.167	1301	3.86
463	0.350	205	0.157	1273	3.16
464	0.325	224	0.153	1178	2.68

TABLE - A1-30

LIQUID SYSTEM = ETHYL-ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_v - T_w}{2}$
			°C			°C	°C	°C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
465	100(98.807)	5°	50.0	53.1	3.1	67.0	72.0	5.05
466	200(197.614)		50.4	52.0	1.6	65.0	71.0	6.05
467	300(296.421)		50.0	51.2	1.2	63.0	70.0	7.05
468	400(395.228)		50.0	51.0	1.0	62.0	70.0	7.55
469	100(98.807)	10°	49.5	53.3	3.8	69.0	74	4.05
470	200(197.614)		50.0	52.0	2.0	68.0	73	4.55
471	300(296.421)		50.0	51.8	1.8	66.0	72	5.55
472	400(395.228)		50.0	51.4	1.4	65.0	71	6.05
473	100(98.807)	15°	50.0	55.1	5.1	71.0	75	3.05
474	200(197.614)		50.2	53.1	2.9	69.0	74	4.05
475	300(296.421)		50.2	52.6	2.4	66.0	72	5.55
476	400(395.228)		50.0	52.0	2.0	65.0	72	6.05
477	100(98.807)	19°	50.2	55.5	5.3	72.0	75	2.55
478	200(197.614)		50.5	53.5	3.0	70.0	74	3.55
479	300(296.421)		50.0	52.4	2.4	68.0	73	4.55
480	400(395.228)		50.0	52.0	2.0	67.0	72	5.05

TABLE - A1-30 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ Kcal/hr.m^2.°C	$\bar{h}_{idc_{Theo.}}$ Kcal/hr.m^2.°C
(0)	(7)	(8)	(9)	(10)	(11)	(12)
465	5.05	331	306	1.11	2212	2070
466	6.05	345	316	1.16	1924	1978
467	7.05	381	355	1.28	1818	1904
468	7.55	403	395	1.36	1796	1872
469	4.805	411	375	0.98	2436	2184
470	5.367	461	395	1.10	2425	2121
471	6.250	535	533	1.28	2307	2018
472	6.557	561	553	1.34	2220	1975
473	6.220	533	504	0.94	3070	2329
474	7.230	620	573	1.09	2689	2170
475	8.574	760	711	1.33	2405	2005
476	9.240	792	750	1.39	2299	1963
477	6.930	594	523	0.88	3453	2432
478	8.217	704	592	1.04	2940	2239
479	8.870	760	711	1.12	2476	2105
480	9.240	792	750	1.17	2324	2050

TABLE - A1-30 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
465	0.301	209	0.157	481	2.54
466	0.261	251	0.148	418	2.50
467	0.247	293	0.141	395	2.15
468	0.244	313	0.137	390	2.00
469	0.331	142	0.179	754	4.09
470	0.329	159	0.173	750	3.64
471	0.314	185	0.165	714	2.98
472	0.302	194	0.162	687	2.74
473	0.417	135	0.182	1286	5.49
474	0.365	156	0.174	1126	4.13
475	0.327	192	0.162	1007	3.13
476	0.313	200	0.160	963	2.76
477	0.469	127	0.184	1703	6.88
478	0.399	150	0.174	1450	4.95
479	0.337	161	0.170	1221	3.86
480	0.316	168	0.168	1146	3.48

TABLE - A1-31

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
481	100 (99.568)	5°	29.5	36.3	6.8	53.5	68	12.40
482	200 (199.136)		30.2	33.8	3.6	51.0	67	13.65
483	300 (298.70)		30.0	32.4	2.4	49.5	66	14.40
484	400 (398.27)		30.0	31.8	1.8	47.5	65	15.40
485	100 (99.568)	10°	30.0	37.3	7.3	60.0	70	9.15
486	200 (199.136)		30.5	34.5	4.0	58.0	69	10.15
487	300 (298.70)		30.0	32.8	2.8	56.0	69	11.15
488	400 (398.27)		30.0	32.1	2.1	54.0	67.5	12.15
489	100 (99.568)	15°	29.5	37.5	8.0	64.0	72	7.15
490	200 (199.136)		30.0	34.5	4.5	62.0	71	8.15
491	300 (298.70)		30.0	33.2	8.2	60.0	70	9.15
492	400 (398.27)		30.0	32.4	2.4	58.0	70	10.15
493	100 (99.568)	19°	29.5	38.0	8.5	66.0	74	6.15
494	200 (199.136)		30.0	35.0	5.0	64.0	72	7.15
495	300 (298.70)		30.0	33.4	3.4	62.0	72	8.15
496	400 (398.27)		30.2	32.7	2.5	60.0	71	9.15

TABLE - A1-31 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
481	3.246	681	677	2.29	1847	1450
482	3.435	721	717	2.43	1777	1416
483	3.503	735	717	2.47	1717	1397
484	3.562	748	717	2.52	1635	1374
485	3.469	728	727	1.74	1905	1562
486	3.818	801	796	1.92	1889	1522
487	4.006	841	836	2.01	1805	1487
488	4.006	841	836	2.01	1657	1455
489	3.805	799	797	1.41	1963	1651
490	4.281	899	896	1.58	1937	1598
491	3.194	959	956	1.68	1841	1552
492	4.568	959	956	1.68	1659	1513
493	4.052	850	846	1.26	2049	1712
494	4.808	1010	996	1.50	2092	1649
495	4.924	1034	1015	1.53	1881	1595
496	4.924	1034	996	1.53	1675	1550.

TABLE - A1-31 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
481	0.479	78	0.218	466	1.20
482	0.462	82	0.215	448	1.08
483	0.446	84	0.213	433	1.03
484	0.425	85	0.212	412	0.96
485	0.495	59	0.241	684	1.94
486	0.490	65	0.233	678	1.76
487	0.469	68	0.229	648	1.59
488	0.431	68	0.229	595	1.46
489	0.510	47	0.259	954	2.51
490	0.503	53	0.249	942	2.20
491	0.478	40	0.273	895	1.96
492	0.431	57	0.243	807	1.77
493	0.532	43	0.264	1173	3.06
494	0.544	51	0.249	1198	2.64
495	0.488	52	0.248	1077	2.31
496	0.435	52	0.248	959	2.06

TABLE - A1-32

LIQUID SYSTEM = ETHYL ALCOHOL

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
497	100(99.225)		40.0	45.5	5.5	58.0	70	10.15
498	200(198.45)	5°	40.0	43.0	3.0	56.0	69	11.15
499	300(297.70)		40.0	42.2	2.2	54.0	68	12.15
500	400(396.90)		40.5	42.3	1.8	51.0	66	13.65
501	100(99.225)		40.0	46.5	6.5	64.0	72	7.15
502	200(198.45)	10°	40.5	44.0	3.5	62.0	71	8.15
503	300(297.70)		40.0	42.4	2.4	60.0	70	9.15
504	400(396.90)		40.0	42.2	2.2	58.0	69	10.15
505	100(99.225)		40.0	46.8	6.8	68.0	73	5.15
506	200(198.45)	15°	40.5	44.3	3.8	66.0	72.5	6.15
507	300(297.70))		40.0	42.8	2.8	64.0	72	7.15
508	400(396.90)		40.0	42.4	2.4	62.0	72	8.15
509	100(99.225)		39.5	46.5	7.0	69.0	75	4.65
510	200(198.45)	19°	40.0	44.1	4.1	68.0	75	5.15
511	300(297.70)		40.0	43.1	3.1	66.0	73	6.15
512	400(396.90)		40.0	42.5	2.5	64.0	72	7.15

TABLE - A1-32 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
497	2.838	596	546	2.05	1976	1525
498	3.050	641	595	2.16	1934	1489
499	3.150	661	655	2.24	1830	1458
500	3.446	724	715	2.44	1785	1416
501	3.122	655	645	1.57	2193	1662
502	3.338	701	695	1.68	2059	1608
503	3.436	721	715	1.73	1886	1562
504	4.173	876	873	2.09	2066	1522
505	3.236	679	675	1.19	2339	1792
506	3.598	755	754	1.33	2157	1714
507	3.991	838	834	1.47	2059	1651
508	4.556	957	952	1.68	2063	1598
509	3.574	751	694	1.11	2394	1836
510	3.903	820	813	1.22	2360	1789
511	4.460	937	923	1.39	2258	1712
512	4.731	993	992	1.47	2058	1649

TABLE - A1-32 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3} \right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
497	0.514	68	0.228	498	1.46
498	0.502	73	0.223	488	1.32
499	0.476	76	0.220	462	1.21
500	0.463	83	0.214	450	1.08
501	0.569	53	0.249	787	2.49
502	0.535	57	0.244	739	2.18
503	0.490	58	0.242	667	1.94
504	0.537	71	0.226	742	1.75
505	0.607	40	0.274	1237	3.48
506	0.560	45	0.264	1049	2.92
507	0.535	50	0.254	1001	2.51
508	0.536	57	0.244	1003	2.20
509	0.622	38	0.279	1370	4.05
510	0.613	41	0.272	1351	3.66
511	0.586	47	0.260	1293	3.06
512	0.534	50	0.254	1178	2.64

TABLE - A1-33

LIQUID SYSTEM = ETHY ALCOHOL

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 78.3°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_v - T_w}{2}$
				°C		°C	°C	°C
(0)	(1)	(1)		(3)		(4)	(5)	(6)
513	100 (98.807)	5°	50.0	54.2	4.2	65.0	72	6.65
514	200 (197.614)		50.0	52.4	2.4	63.0	71	7.65
515	300 (296.421)		50.0	51.8	1.8	60.0	69	9.15
516	400 (395.228)		50.0	51.4	1.4	59.0	68.5	9.65
517	100 (98.807)	10°	50.0	54.8	4.8	68.0	74	5.15
518	200 (197.614)		50.0	53.4	3.4	64.0	73	7.15
519	300 (296.421)		49.8	52.2	2.4	62.0	71	8.15
520	400 (395.228)		50.0	52.0	2.0	60.0	70	9.15
521	100 (98.807)	15°	50.0	56.0	6.0	69.0	74	4.65
522	200 (197.614)		51.0	54.6	3.6	67.0	73	5.65
523	300 (296.421)		50.0	52.8	2.8	65.0	72	6.65
524	400 (395.228)		50.0	52.1	2.1	64.0	72	7.15
525	100 (98.807)	19°	50.0	57.0	7.0	70.0	75	4.15
526	200 (197.614)		50.5	54.5	4.0	69.0	74	4.65
527	300 (296.421)		50.0	53.0	3.0	67.0	73	5.65
528	400 (395.228)		50.0	52.3	2.3	66.0	73	6.15

TABLE - A1-33 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
513	1.999	420	415	1.41	2125	1695
514	2.263	475	474	1.60	2089	1636
515	2.564	539	533	1.81	1982	1565
516	2.681	563	553	1.89	1963	1544
517	2.396	503	474	1.20	2338	1804
518	3.205	673	672	1.61	2253	1662
519	3.381	710	711	1.70	2088	1608
520	3.778	793	791	1.90	2072	1562
521	3.05	641	592	1.13	2421	1838
522	3.598	756	711	1.33	2351	1751
523	4.163	875	830	1.54	2311	1681
524	4.297	902	869	1.59	2216	1651
525	3.527	741	691	1.09	2649	1889
526	3.764	791	790	1.17	2521	1836
527	4.281	898	889	1.33	2356	1749
528	4.387	921	909	1.36	2220	1712

TABLE - A1-33 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v) $X10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
513	0.552	48	0.257	536	2.22
514	0.543	54	0.247	527	1.93
515	0.515	62	0.236	500	1.62
516	0.510	64	0.233	495	1.53
517	0.607	41	0.272	840	3.45
518	0.585	54	0.248	809	2.49
519	0.543	57	0.244	750	2.18
520	0.538	64	0.234	744	1.94
521	0.629	38	0.278	1177	3.86
522	0.611	45	0.263	1143	3.18
523	0.601	52	0.251	1124	2.70
524	0.575	54	0.247	1077	2.51
525	0.688	37	0.278	1515	4.54
526	0.655	40	0.271	1443	4.05
527	0.612	45	0.261	1349	3.33
528	0.577	46	0.258	1271	3.07

TABLE - A1-34

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
529	100(99.568)	5°	30.0	35.8	5.8	51.0	67	12.875
530	200(199.136)		30.0	33.0	3.0	49.0	66	13.875
531	300(298.70)		29.5	31.5	2.0	47.0	64	14.875
532	400(398.27)		30.0	31.6	1.6	45.0	63	15.875
533	100(99.568)	10°	30.0	37.0	7.0	57.5	69	9.625
534	200(199.136)		29.8	33.4	3.6	55.5	68	10.625
535	300(298.70)		30.0	32.4	2.4	53.5	67	11.625
536	400(398.27)		30.2	32.2	2.0	50.0	65	13.375
537	100(99.568)	15°	29.0	37.0	8.0	60.5	70	8.125
538	200(199.136)		30.0	34.1	4.1	59.0	70	8.875
539	300(298.70)		30.0	32.8	2.8	56.5	68	10.125
540	400(398.27)		30.5	32.7	2.2	55.0	67	10.875
541	100(99.568)	19°	30.0	39.2	9.2	62.5	71	7.125
542	200(199.136)		29.5	34.2	4.7	60.5	70	8.125
543	300(298.70)		30.0	33.4	3.4	59.0	70	8.875
544	400(398.27)		30.0	32.6	2.6	56.5	69	10.125

TABLE - A1-34 (contd.)

Serial No.	Average Conden-sate Rate	Heat - Released by vapour	Heat - Received by Coolant	Heat - Flux ($\bar{q}$) X10^{-4}	$\bar{h}_{idc_{Expt.}}$	$\bar{h}_{idc_{Theo.}}$
	kg/hr	Kcal/hr	Kcal/hr	Kcal/hr.m^2	$\frac{Kcal}{hr.m.^2\ °C}$	$\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
529	12.60	594	578	2.00	1552	1464
530	12.81	605	597	2.03	1464	1436
531	13.18	622	597	2.09	1404	1412
532	13.58	641	637	2.15	1360	1389
533	14.82	701	697	1.68	1743	1572
534	16.20	765	717	1.83	1726	1533
535	16.80	793	717	1.90	1633	1499
536	18.14	857	796	2.05	1530	1448
537	18.30	864	797	1.52	1868	1628
538	18.90	892	816	1.57	1765	1593
539	19.55	921	836	1.62	1598	1542
540	20.07	951	876	1.67	1536	1515
541	21.00	991	916	1.47	2061	1680
542	22.23	1050	936	1.56	1916	1626
543	22.68	1071	1016	1.59	1789	1591
544	22.68	1071	1036	1.59	1568	1540

TABLE - A1-34 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\bar{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
529	0.254	252	0.147	362	1.00
530	0.239	256	0.147	342	0.92
531	0.230	264	0.146	328	0.87
532	0.223	272	0.144	318	0.81
533	0.286	211	0.157	580	1.46
534	0.283	231	0.153	574	1.33
535	0.267	239	0.151	543	1.21
536	0.251	258	0.147	509	1.05
537	0.306	190	0.163	841	1.80
538	0.289	196	0.161	795	1.65
539	0.262	203	0.159	719	1.45
540	0.252	209	0.158	692	1.35
541	0.338	186	0.162	1092	2.10
542	0.314	197	0.159	1015	1.84
543	0.293	201	0.158	948	1.69
544	0.257	201	0.158	831	1.48

TABLE - A1-35

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
545	100 (99.225)	5°	39.5	43.5	4.0	60.0	70	8.375
546	200 (198.45)		40.0	42.2	2.2	58.0	69	9.375
547	300 (297.70)		40.0	41.6	1.6	55.5	68	10.625
548	400 (396.90)		40.0	41.4	1.4	51.0	65	12.875
549	100 (99.225)	10°	40.0	44.4	4.4	64.0	72	6.375
550	200 (198.45)		40.0	42.6	2.6	63.0	71	6.875
551	300 (297.70)		40.0	42.0	2.0	61.0	70	7.875
552	400 (396.90)		40.5	42.1	1.6	59.0	69	8.875
553	100 (99.225)	15°	40.0	45.6	5.6	66.5	73	5.125
554	200 (198.45)		40.0	43.2	3.2	65.0	72	5.875
555	300 (297.70)		40.0	42.4	2.4	62.5	71	7.125
556	400 (396.90)		40.4	42.5	2.1	60.5	69	8.125
557	100 (99.225)	19°	40.0	46.5	6.5	68.0	73	4.375
558	200 (198.45)		40.2	43.8	3.6	66.0	72	5.375
559	300 (297.70)		40.0	42.7	2.7	65.0	72	5.875
560	400 (396.90)		40.2	42.5	2.3	62.5	71.5	7.125

TABLE - A1-35 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat-Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.\ °C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.\ °C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
545	8.590	406	397	1.37	1631	1630
546	9.260	437	436	1.47	1568	1584
547	10.310	487	476	1.64	1545	1536
548	11.870	561	556	1.89	1463	1465
549	10.690	505	437	1.21	1896	1742
550	11.450	541	516	1.29	1880	1710
551	12.600	595	595	1.43	1812	1652
552	13.660	645	635	1.54	1740	1604
553	12.880	609	559	1.07	2087	1828
554	14.350	681	635	1.19	2036	1767
555	16.680	785	715	1.38	1936	1683
556	17.710	858	833	1.51	1855	1629
557	14.350	678	645	1.00	2297	1899
558	16.200	765	714	1.13	2109	1804
559	17.580	830	804	1.23	2096	1764
560	19.890	940	913	1.40	1955	1681

TABLE - A1-35 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right)^{1/3}$	Re_f	$Re'_f = \frac{fc}{Re_f^{1/3}}$	Mean Nusselt Number ($\overline{Nu}$)	Condensation Number (C_v)
(0)	(13)	(14)	(15)	(16)	(17)
545	0.267	172	0.168	381	1.54
546	0.257	185	0.164	366	1.37
547	0.253	206	0.158	361	1.21
548	0.239	237	0.151	342	1.00
549	0.310	152	0.175	631	2.21
550	0.308	163	0.171	625	2.05
551	0.297	179	0.167	603	1.79
552	0.285	194	0.162	579	1.59
553	0.342	134	0.183	940	2.86
554	0.334	149	0.176	917	2.50
555	0.317	173	0.168	872	2.06
556	0.304	184	0.164	835	1.69
557	0.376	127	0.184	1217	3.43
558	0.346	143	0.177	1118	2.79
559	0.343	156	0.172	1111	2.55
560	0.320	176	0.165	1036	2.10

TABLE A1-36

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = DIVERGING-CONVERGING CONE SECTION

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr. (Kg/hr)	Cone Angle (θ) deg.	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
561	100 (98.807)	5°	50.0	53.5	3.5	63.0	70.	6.875
562	200 (197.614)		50.2	52.1	1.9	61.0	69.	7.875
563	300 (296.421)		50.2	51.6	1.4	58.0	67.5	9.375
564	400 (395.228)		50.0	51.1	1.1	55.5	66.	10.625
565	100 (98.807)	10°	50.0	53.8	3.8	67.0	73	4.875
566	200 (197.614)		50.2	52.1	1.9	66.0	72	5.375
567	300 (296.421)		50.0	51.6	1.6	63.0	71	6.875
568	400 (395.228)		50.0	51.3	1.3	62.0	70	7.375
569	100 (98.807)	15°	50.0	54.8	4.8	69.0	74	3.875
570	200 (197.614)		50.0	52.5	2.5	68.0	73	4.375
571	300 (296.421)		50.0	51.8	1.8	66.5	73	5.125
572	400 (395.228)		50.0	51.5	1.5	65.0	72	5.875
573	100 (98.807)	19°	50.0	55.8	5.8	70.0	74	3.375
574	200 (197.614)		50.2	53.4	3.2	69.0	74	3.875
575	300 (296.421)		50.0	52.2	2.2	68.0	73	4.375
576	400 (395.228)		50.0	51.7	1.7	66.5	72	5.125

TABLE - A1-36 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idc_{Expt.}}$ $\frac{Kcal}{hr.m^2.\ °C}$	$\bar{h}_{idc_{Theo.}}$ $\frac{Kcal}{hr.m^2.\ °C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
561	8.070	381	346	1.28	1865	1712
562	8.310	392	375	1.32	1674	1655
563	9.140	432	415	1.45	1546	1584
564	9.610	454	434	1.53	1435	1536
565	8.930	421	375	1.00	2067	1863
566	8.930	421	375	1.00	1875	1818
567	10.900	515	474	1.23	1793	1710
568	11.340	536	514	1.28	1740	1680
569	10.500	495	475	0.87	2244	1960
570	11.280	533	494	0.94	2140	1902
571	12.190	574	533	1.00	1968	1828
572	13.180	621	593	1.09	1857	1767
573	12.890	609	573	0.90	2675	2026
574	14.175	669	632	0.99	2559	1957
575	14.630	691	652	1.02	2341	1899
576	14.920	705	672	1.05	2039	1825

TABLE - A1-36 (contd.)

Serial No.	$\bar{h}_i \left(\frac{\mu_f^2}{\rho_f^2\, g\, k_f^3}\right)^{1/3}$	Re_f	$Re_f' = \frac{f_c}{Re_f^{1/3}}$	Mean Nusselt Number ($\bar{Nu}$)	Condensation Number (C_v) $\times 10^{-13}$
(0)	(13)	(14)	(15)	(16)	(17)
561	0.306	161	0.171	436	1.87
562	0.274	166	0.170	391	1.64
563	0.253	183	0.164	361	1.37
564	0.235	192	0.162	335	1.21
565	0.338	127	0.187	687	2.89
566	0.307	127	0.187	624	2.62
567	0.294	155	0.175	596	2.05
568	0.285	161	0.172	579	1.91
569	0.367	109	0.195	1010	3.78
570	0.351	117	0.191	963	3.35
571	0.322	127	0.186	886	2.86
572	0.304	137	0.181	836	2.50
573	0.438	114	0.191	1418	4.44
574	0.419	125	0.185	1356	3.87
575	0.383	129	0.184	1241	3.43
576	0.334	132	0.182	1081	2.92

TABLE - A1-37

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_v) = 100°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
577	100(99.568)		30.0	46.5	16.5	78.0	91	11.0
578	200(199.136)	0.020884	30.2	38.7	8.5	76.0	90	12.0
579	300(298.70)	(10°)	30.1	36.1	6.0	75.0	89	12.5
580	400(398.27)		30.0	34.6	4.6	73.0	88	13.5
581	100(99.568)		30.2	50.0	19.8	82.0	93	9.0
582	200(199.136)	0.028460	30.0	40.4	10.4	80.0	92	10.0
583	300(298.70)	(15°)	30.0	37.4	7.4	78.0	91	11.0
584	400(398.27)		30.5	36.4	5.9	76.0	90	12.0
585	100(99.568)		30.0	52.5	22.5	84.0	93	8.0
586	200(199.136)	0.0337297	30.0	41.8	11.8	82.0	93	9.0
587	300(298.70)	(19°)	30.5	38.7	8.2	80.0	92	10.0
588	400(398.27)		30.2	36.4	6.2	78.0	92	11.0

TABLE - A1-37 (contd.)

Sl. No.	Average Condensate Rate kg/hr	Heat Released by Vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy\,Expt.}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy\,Theo.}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div.	Con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
577	3.080	1660	1643	7.95	7226	9043	70	52
578	3.320	1788	1693	8.56	7135	8849	57	47
579	3.360	1810	1792	8.67	6933	8759	41	38
580	3.520	1897	1832	9.08	6728	8592	44	42
581	3.729	2010	1971	7.06	7847	9508	62	48
582	3.876	2089	2071	7.34	7340	9261	54	48
583	4.182	2254	2210	7.92	7187	9043	43	42
584	4.423	2381	2350	8.37	6972	8848	44	42
585	4.157	2241	2240	6.64	8305	9793	63	47
586	4.373	2357	2350	6.99	7764	9508	52	46
587	4.569	2461	2449	7.30	7296	9261	49	43
588	4.815	2594	2469	7.69	6991	9043	47	42

TABLE - A1-38

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_v) = 100°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
589	100(99.225)	0.020884 (10°)	40.2	53.7	13.5	83.0	92.0	8.5
590	200(198.45)		40.2	47.1	6.9	81.0	91.0	9.5
591	300(297.70)		40.0	45.0	5.0	80.0	90.5	10.0
592	400(396.90)		40.5	44.6	4.1	78.0	89.0	11.0
593	100(99.225)	0.02846 (15°)	41.0	55.4	14.4	88.0	93.0	6.0
594	200(198.45)		40.5	48.0	7.5	86.0	92.5	7.0
595	300(297.70)		40.5	45.9	5.4	84.0	92.0	8.0
596	400(396.90)		40.0	44.4	4.4	82.0	91.0	9.0
597	100(99.225)	0.0337297 (19°)	40.0	57.5	17.5	88.0	94	6.0
598	200(198.45)		40.5	50.5	10.0	86.0	93	7.0
599	300(297.70)		40.2	47.4	7.2	84.0	92	8.0
600	400(396.90)		40.0	45.6	5.6	82.0	92	9.0

TABLE - A1-38 (contd.)

Sl. No.	Average Condensate Rate	Heat Released by vapour	Heat Received by Coolant	Heat Flux ($\bar{q}$) X10^{-4}	$\bar{h}_{icy_{Expt.}}$	$\bar{h}_{icy_{Theo.}}$	Percent Enhancement (Experimental)	
	Kg/hr	Kcal/hr	Kcal/hr	Kcal/ hr.m^2	$\frac{Kcal}{hr.m^2.°C}$	$\frac{Kcal}{hr.m^2.°C}$	div.	Con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
589	2.65	1430	1339	6.84	8055	9645	64	58
590	2.75	1481	1369	7.09	7465	9380	71	67
591	2.80	1512	1488	7.24	7240	9261	74	66
592	3.04	1638	1627	7.84	7130	9043	70	67
593	2.695	1452	1429	5.10	8503	10523	74	73
594	2.875	1549	1488	5.44	7775	10125	76	64
595	3.179	1714	1608	6.02	7537	9793	68	65
596	3.503	1888	1746	6.63	7379	9508	68	66
597	3.303	1779	1736	5.27	8790	10523	67	66.5
598	3.709	1999	1985	5.92	8466	10124	65	65
599	3.989	2149	2143	6.37	7964	9792	67	66.5
600	4.246	2289	2223	6.78	7540	9508	73	72.5

TABLE - A1-39

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_V) = 100°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
601	100 (98.807)	0.020884 (10°)	50.0	62.5	12.5	88.0	95.5	6.0
602	200 (197.614)		50.0	56.8	6.8	86.0	92.5	7.0
603	300 (296.421)		50.4	55.0	4.6	84.0	92.0	8.0
604	400 (395.228)		50.2	53.6	3.4	82.0	92.0	9.0
605	100 (98.807)	0.02846 (15°)	50.0	63.6	13.6	92.0	95.5	4.0
606	200 (197.614)		50.0	57.0	7.0	90.0	95.0	5.0
607	300 (296.421)		50.5	55.5	5.0	88.0	94.0	6.0
608	400 (395.228)		50.2	54.2	4.0	86.0	94.0	7.0
609	100 (98.807)	0.0337297 (19°)	50.0	67.5	17.5	92.0	96	4.0
610	200 (197.614)		50.0	59.1	9.1	89.0	95	5.5
611	300 (296.421)		50.2	56.8	6.6	88.0	95	6.0
612	400 (395.228)		50.0	55.2	5.2	86.0	94	7.0

TABLE - A1-39 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div.	Con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
601	2.300	1240	1235	5.94	9896	10522	54	49
602	2.565	1383	1343	6.62	9460	10125	48	44
603	2.594	1398	1363	6.69	8366	9793	53	43
604	2.594	1398	1344	6.69	7438	9508	61	61
605	2.509	1352	1344	4.75	11876	11645	37	26
606	2.654	1430	1383	5.02	10049	11013	44	36
607	2.760	1487	1482	5.22	8708	10523	57	41
608	3.136	1690	1581	5.94	8488	10125	46	39
609	3.224	1737	1729	5.15	12874	11645	36	24
610	3.521	1897	1798	5.62	10226	10754	58	44
611	3.812	2054	1956	6.09	10149	10523	38	30.5
612	3.921	2113	2055	6.26	8949	10125	46	37

TABLE - A1-40

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_v) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Tempt. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
613	100(99.568)	0.020884 (10°)	30.4	34.2	3.8	52.0	67	12.55
614	200(199.136)		30.0	31.9	1.9	51.0	65	13.05
615	300(298.70)		30.2	31.5	1.3	47.5	64	14.80
616	400(398.27)		30.2	31.2	1.0	45.0	62.5	16.05
617	100(99.568)	0.02846 (15°)	30.4	36.0	5.6	58.0	70	9.55
618	200(199.136)		30.5	33.5	3.0	55.5	68	10.80
619	300(298.70)		30.0	32.0	2.0	52.5	67	12.30
620	400(398.27)		30.0	31.6	1.6	49.5	66	13.80
621	100(99.568)	0.0337297 (19°)	30.5	36.7	6.2	60.0	70	8.55
622	200(199.136)		30.0	33.2	3.2	58.0	69	9.55
623	300(298.70)		30.2	32.4	2.2	55.0	67	11.05
624	400(398.27)		30.0	31.8	1.8	52.5	66	12.30

TABLE - A1-40 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div.	Con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
613	4.35	382	378	1.83	1457	1946	61	58
614	4.35	382	378	1.83	1405	1927	62	58
615	4.56	402	388	1.92	1300	1867	71	64
616	4.56	402	398	1.92	1199	1830	82	82
617	6.384	561	558	1.97	2064	2083	16	12
618	6.843	602	597	2.11	1958	2020	19	13
619	6.872	605	517	2.13	1728	1956	32	25
620	7.280	641	637	2.25	1632	1900	37	29
621	7.043	620	617	1.84	2150	2142	19	14.5
622	7.306	643	637	1.91	1996	2083	21	13.0
623	7.517	661	657	1.96	1773	2009	30	23.0
624	8.182	720	717	2.13	1735	1955	32	21.0

TABLE - A1-41

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_v) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
625	100 (99.225)	0.020884 (10°)	40.0	43.3	3.3	57.5	69	9.8
626	200 (198.45)		40.0	41.9	1.9	55.5	68	10.8
627	300 (297.70)		40.2	41.6	1.4	52.0	67	12.55
628	400 (396.90)		40.0	41.2	1.2	49.5	64	13.80
629	100 (99.225)	0.02846 (15°)	40.5	44.7	4.2	63.0	70.0	7.05
630	200 (198.45)		40.0	42.5	2.5	60.0	68.5	8.55
631	300 (297.70)		40.0	41.6	1.6	58.0	68.0	9.55
632	400 (396.90)		40.2	41.5	1.3	55.5	66.5	10.8
633	100 (99.225)	0.0337297 (19°)	40.0	45.4	5.4	63.0	72	7.05
634	200 (198.45)		40.0	43.0	3.0	60.0	71	8.55
635	300 (297.70)		40.2	42.2	2.0	58.0	70	9.55
636	400 (396.90)		40.2	41.9	1.7	56.5	69	10.30

TABLE - A1-41 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div.	Con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
625	3.880	341	327	1.63	1666	2070	64	44
626	4.320	380	377	1.82	1685	2020	54	39
627	4.800	422	416	2.02	1610	1946	57	37
628	4.980	476	437	2.28	1523	1900	60	41
629	5.012	441	417	1.55	2198	2248	29	11
630	5.684	501	496	1.76	2059	2142	34	15
631	5.468	481	476	1.69	1770	2083	45	28
632	6.039	531	516	1.86	1728	2020	47	24
633	6.148	541	536	1.60	2275	2247	32	12
634	6.821	601	595	1.78	2084	2142	38	20
635	6.968	613	595	1.82	1903	2083	45	21
636	7.281	641	635	1.90	1845	2044	37	15

TABLE - A1-42

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_v) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
637	100(98.807)	0.020884 (10°)	50.0	52.6	2.6	61.5	71	7.80
638	200(197.614)		50.5	52.0	1.5	60.0	70	8.55
639	300(296.421)		50.2	51.3	1.1	57.5	69	9.80
640	400(395.228)		50.0	50.9	0.9	55.5	67	10.80
641	100(98.807)	0.02846 (15°)	50.0	53.4	3.4	66.0	72.5	5.55
642	200(197.614)		50.0	52.0	2.0	64.0	71.5	6.55
643	300(296.421)		50.2	51.6	1.4	62.0	71	7.55
644	400(395.228)		50.5	51.6	1.1	60.0	70	8.55
645	100(98.807)	0.0337297 (19°)	50.0	54.6	4.6	66.0	72.5	5,55
646	200(197.614)		50.2	52.7	2.5	64.0	71.5	6,55
647	300(296.421)		50.2	52.1	1.9	62.0	71	7,55
648	400(395.228)		50.0	51.5	1.5	60.0	70	8.55

TABLE - A1-42 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div.	con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
637	3.240	285	257	1.36	1749	2191	58	38
638	3.390	299	296	1.43	1674	2142	61	54
639	3.70	326	326	1.56	1593	2070	71	55
640	4.05	357	355	1.71	1583	2020	62	45
641	4.101	361	336	1.27	2285	2386	52	52
642	4.563	401	395	1.41	2151	2289	55	39
643	4.771	420	415	1.47	1955	2269	60	29
644	5.012	441	435	1.55	1812	2142	64	33
645	5.684	500	455	1.48	2670	2386	41	41
646	6.148	541	494	1.60	2449	2289	45	36
647	6.680	588	563	1.74	2309	2209	39	31
648	6.750	594	592	1.76	2060	2142	51	41

TABLE - A1-43

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_V) = 76.75°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
649	100(99.568)	0.020884 (10°)	30.0	33.8	3.8	51.0	66	12.875
650	200(199.136)		30.2	32.4	2.2	47.5	64	14.625
651	300(298.70)		30.0	31.5	1.5	45.0	62	15.875
652	400(398.27)		30.0	31.2	1.2	43.0	61	16.875
653	100(99.568)	0.02846 (15°)	30.0	34.6	4.6	55.5	68	10.625
654	200(199.136)		30.5	32.9	2.4	53.5	67	11.625
655	300(298.70)		30.0	32.0	2.0	48.0	64	14.375
656	400(398.27)		30.2	31.8	1.6	45.5	62	15.625
657	100(99.568)	0.0337297 (19°)	30.0	35.4	5.4	56.5	68	10.125
658	200(199.136)		30.0	33.0	3.0	53.5	67	11.625
659	300(298.70)		30.5	32.6	2.1	51.5	65	12.625
660	400(398.27)		30.0	31.7	1.7	49.5	64	13.625

TABLE - A1-43 (contd.)

Sl. No.	Average Condensate Rate kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ Kcal/hr.m^2.°C	$\bar{h}_{icy_{Theo.}}$ Kcal/hr.m^2.°C	Percent Enhancement (Experimental) div.	con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
649	8.490	401	378	1.92	1491	1727	29	19
650	9.45	446	438	2.13	1460	1673	30	16
651	9.86	465	448	2.23	1402	1639	32	20
652	10.17	480	478	2.29	1362	1614	19	10
653	9.692	458	458	1.61	1515	1812	42	32
654	10.500	496	478	1.74	1499	1772	33	22
655	12.741	601	597	2.11	1469	1680	33	19
656	13.663	645	637	2.27	1450	1646	34	22
657	11.571	546	538	1.62	1599	1834	39	32
658	12.741	602	597	1.78	1535	1772	38	26
659	13.662	645	627	1.91	1515	1736	34	23
660	14.354	678	677	2.01	1475	1703	33	18

TABLE - A1-44

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_V) = 76.75°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equivalent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
661	100(99.225)	0.020884 (10°)	40.2	43.6	3.4	55.5	67	10.625
662	200(198.45)		40.0	41.9	1.9	51.0	65	12.875
663	300(297.70)		40.0	41.3	1.3	49.5	64	13.625
664	400(396.90)		40.5	41.5	1.0	47.5	63	14.625
665	100(99.225)	0.02846 (15°)	40.0	43.8	3.8	61.0	70	7.875
666	200(198.45)		40.0	42.1	2.1	58.0	69	9.375
667	300(297.70)		40.2	41.7	1.5	55.5	68	10.625
668	400(396.90)		40.2	41.4	1.2	53.5	66	11.625
669	100(99.225)	0.0337297 (19°)	40.0	44.7	4.7	62.0	70	7.375
670	200(198.45)		40.2	42.9	2.7	60.0	69	8.375
671	300(297.70)		40.5	42.4	1.9	58.0	68	9.375
672	400(396.90)		40.5	42.0	1.5	56.5	68	10.125

TABLE - A1-44 (contd.)

Sl. No.	Average Conden-sate Rate kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experi-mental) div.	con..
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
661	7.32	345	337	1.65	1554	1812	39	24
662	8.43	398	377	1.91	1480	1727	31	16
663	8.43	398	387	1.91	1398	1703	38	25
664	8.69	410	397	1.96	1342	1673	38	25
665	8.400	397	377	1.39	1771	1953	30	24
666	9.00	425	417	1.49	1592	1870	37	37
667	9.861	466	447	1.64	1541	1812	40	36
668	10.598	501	476	1.76	1514	1772	32	28
669	10.800	510	466	1.51	2050	1986	13	9
670	11.571	546	536	1.62	1933	1923	17	13
671	12.600	595	566	1.76	1885	1870	19	14
672	13.186	623	595	1.85	1824	1834	16	8

TABLE - A1-45

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_V) = 76.75°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equi-valent to Diverging or Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_W) °C	Avg. Conden-sate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
673	100 (98.807)	0.020884 (10°)	50.0	52.5	2.5	61.0	70	7.875
674	200 (197.614)		50.0	51.3	1.3	60.0	69	8.375
675	300 (296.421)		50.2	51.2	1.0	57.5	68	9.625
676	400 (395.228)		50.4	51.2	0.8	55.5	68	10.625
677	100 (98.807)	0.02846 (15°)	50.0	53.0	3.0	66.0	72	5.375
678	200 (197.614)		50.5	52.0	1.5	64.0	71	6.375
679	300 (296.421)		50.5	51.5	1.0	62.0	70	7.375
680	400 (395.228)		50.0	50.8	0.8	61.0	70	7.875
681	100 (98.807)	0.0337297 (19°)	50.5	54.1	3.6	67.0	73	4.875
682	200 (197.614)		50.0	52.0	2.0	65.0	72	5.875
683	300 (296.421)		50.0	51.4	1.4	64.0	71	6.375
684	400 (395.228)		50.2	51.4	1.2	62.0	70	7.375

TABLE - A1-45 (contd.)

Sl. No.	Average Conden-sate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy\,Expt.}$ Kcal/hr.m^2.°C	$\bar{h}_{icy\,Theo.}$ Kcal/hr.m^2.°C	Percent Enhancement (Experi-mental) div.	con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	
673	5.670	268	247	1.28	1630	1953	40	34
674	5.940	280	257	1.34	1600	1923	35	32
675	6.670	316	297	1.51	1567	1858	36	36
676	6.670	316	316	1.51	1419	1812	44	35
677	6.265	296	296	1.04	1935	2149	26	17
678	6.462	305	296	1.07	1681	2059	40	21
679	7.043	333	296	1.17	1586	1986	39	33
680	7.436	351	316	1.23	1566	1953	39	28
681	7.560	357	356	1.06	2171	2202	27	18
682	8.494	401	395	1.19	2024	2102	19	11
683	8.894	420	415	1.25	1953	2059	20	13
684	10.062	475	474	1.41	1909	1986	18	7

TABLE - A1-46

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_V) = 100°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area Equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
685	100(99.568)		30.0	53.0	23.0	80.0	91	10.0
686	200(199.136)		30.2	42.0	11.8	78.0	90	11.0
687	300(298.70)	0.020884x2 (10°)	30.5	38.6	8.1	76.0	89	12.0
688	400(398.27)		30.0	36.1	6.1	74.0	88	13.0
689	100(99.568)		30.0	59.0	29.0	82.0	93	9.0
690	200(199.136)		30.0	44.8	14.8	80.0	92	10.0
691	300(298.70)	0.02846x2 (15°)	30.2	40.6	10.4	78.0	91	11.0
692	400(398.27)		30.2	38.3	8.1	76.0	90	12.0
693	100(99.568)		31.0	61.0	30.0	84.0	93	8.0
694	200(199.136)		30.5	46.0	15.5	82.0	92	9.0
695	300(298.70)	0.0337297x2 (19°)	30.5	41.5	11.0	80.0	91	10.0
696	400(398.27)		30.0	38.6	8.6	78.0	90	11.0

TABLE - A1-46 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ Kcal/hr.m^2.°C	$\bar{h}_{icy_{Theo.}}$ Kcal/hr.m^2.°C	Percent Enhancement (Experimental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
685	4.313	2324	2290	5.56	5564	7788	79
686	4.451	2399	2350	5.74	5221	7604	75
687	4.600	2479	2419	5.93	4945	7441	67
688	4.600	2479	2429	5.93	4565	7293	77
689	5.433	2928	2887	5.14	5687	7996	83
690	5.726	3086	2947	5.42	5395	7768	72
691	5.984	3206	3107	5.63	5120	7604	64
692	6.273	3380	3226	5.94	4948	7441	67
693	5.750	3099	2987	4.59	5742	8235	85
694	6.161	3320	3087	4.92	5468	7996	92
695	6.572	3541	3286	5.25	5249	7788	69
696	6.900	3718	3425	5.51	5011	7604	70

TABLE - A1-47

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_v) = 100°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp (T_w) °C	Avg. Condensate Temp °C	$\Delta T_f = \frac{T_v - T_2}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
697	100(99.225)	0.020884x2 (10°)	40.0	57.0	17.0	86.0	93.5	7.0
698	200(198.45)		40.0	48.9	8.9	85.0	93.0	7.5
699	300(297.70)		40.2	47.1	6.9	82.0	91.5	9.0
700	400(396.90)		40.0	45.8	5.8	80.0	91.0	10.5
701	100(99.225)	0.02846x2 (15°)	40.0	60.0	20.0	88.0	95.0	6.0
702	200(198.45)		40.0	50.5	10.5	86.0	94.0	7.0
703	300(297.70)		40.0	47.8	7.8	85.0	93.5	7.5
704	400(396.90)		40.2	46.4	6.2	83.0	92.5	8.5
705	100(99.225)	0.0337297x2 (19°)	40.5	64.5	24.0	89.0	95.0	5.5
706	200(198.45)		40.0	53.0	13.0	87.5	94.5	6.25
707	300(297.70)		40.0	49.0	9.0	85.0	93.0	7.5
708	400(396.90)		40.2	47.7	7.5	83.0	92.0	8.5

TABLE - A1-47 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ Kcal/hr.m^2.°C	$\bar{h}_{icy_{Theo.}}$ Kcal/hr.m^2.°C	Percent Enhancement (Experimental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
697	3.039	1690	1687	4.05	5780	8514	80
698	3.080	1770	1766	4.24	5650	8368	70
699	3.224	2065	2054	4.94	5493	7995	60
700	3.317	2324	2302	5.56	5299	7788	63
701	3.771	2032	1985	3.57	5950	8849	94
702	4.259	2295	2084	4.03	5760	8514	84
703	4.452	2399	2322	4.21	5620	8368	84
704	4.792	2582	2461	4.54	5337	8111	62
705	4.631	2496	2381	3.70	6727	9043	94
706	4.929	2656	2580	3.94	6299	8759	75
707	5.308	2860	2679	4.24	5652	8368	84
708	5.750	3099	2979	4.59	5405	8111	78

TABLE - A1-48

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_V) = 100°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o)	ΔT	Avg. Wall Temp (T_W) °C	Avg. Condensate Temp °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
709	100(98.807)		50.0	66.5	16.5	90.0	96.0	5.0
710	200(197.614)	0.020884x2 (10°)	50.2	58.6	8.4	89.0	95.5	5.5
711	300(296.421)		50.4	56.2	5.8	86.0	93.0	7.0
712	400(395.228)		50.5	55.1	4.6	85.0	92.5	7.5
713	100(98.807)		50.5	68.5	18.0	92.0	96.5	4.0
714	200(197.614)	0.02846x2 (15°)	50.2	60.0	9.8	90.0	95.0	5.0
715	300(296.421)		50.0	56.5	6.5	89.0	94.5	5.5
716	400(395.228)		50.0	55.5	5.5	87.5	93.5	6.25
717	100(98.807)		50.5	71.5	21.0	92.0	96.0	4.0
718	200(197.614)	0.0337297x2 (19°)	50.2	62.2	12.0	90.0	96.0	5.0
719	300(296.421)		50.0	58.2	8.2	89.0	95.5	5.5
720	400(395.228)		50.0	56.5	6.5	87.5	94.5	6.25

TABLE - A1-48 (contd.)

Sl. No.	Average Conden-sate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experi-mental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
709	3.136	1638	1630	3.92	7843	9261	63
710	3.286	1660	1659	3.97	7226	9043	66
711	3.834	1738	1719	4.16	5944	8514	72
712	4.313	1787	1779	4.28	5704	8368	71
713	3.383	1823	1779	3.20	8007	9793	72
714	3.898	2101	1937	3.69	7382	9261	79
715	3.898	2101	1927	3.69	6711	9043	74
716	4.182	2254	2174	3.95	6336	8759	65
717	4.083	2200	2075	3.26	8153	9793	93
718	4.600	2479	2371	3.67	7350	9261	83
719	4.759	2564	2431	3.80	6911	9043	75
720	5.000	2695	2569	3.99	6392	8759	66

TABLE - A1-49

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_V) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_W) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
721	100 (99.568)		30.0	35.4	5.4	55.5	66.5	10.8
722	200 (199.136)		30.0	32.9	2.9	53.5	65.5	11.8
723	300 (298.70)	0.020884x2 (10°)	30.2	32.4	2.2	49.5	62.5	13.8
724	400 (398.27)		30.0	31.7	1.7	47.5	62.5	14.8
725	100 (99.568)		30.8	38.3	7.5	58.0	69.0	9.55
726	200 (199.136)		30.5	34.3	3.8	55.5	67.5	10.8
727	300 (298.70)	0.02846x2 (15°)	30.0	32.7	2.7	53.5	66.5	11.8
728	400 (398.27)		30.0	32.1	2.1	51.5	65.5	12.8
729	100 (99.568)		29.8	38.3	8.5	59.0	70.0	9.05
730	200 (199.136)		30.0	34.5	4.5	56.5	69.0	10.30
731	300 (298.70)	0.0337297x2 (19°)	30.2	33.4	3.2	53.5	67.0	11.80
732	400 (398.27)		30.0	32.5	2.5	51.5	66.0	12.80

TABLE - A1-49 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
721	6.17	543	537	1.30	1204	1699	74
722	6.61	582	578	1.39	1180	1661	70
723	7.49	660	657	1.58	1145	1598	54
724	7.71	679	677	1.63	1098	1570	62
725	8.526	750	747	1.31	1380	1752	59
726	8.640	760	757	1.34	1236	1699	66
727	9.191	809	806	1.42	1204	1662	61
728	9.529	839	836	1.47	1152	1628	56
729	9.969	877	816	1.30	1437	1775	61
730	10.623	935	896	1.38	1346	1719	56
731	11.368	1000	956	1.48	1256	1662	67
732	11.676	1027	996	1.52	1189	1628	67

TABLE - A1-50

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_v) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging - Converging Cone Section (for θ)	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
733	100 (99.225)	0.020884x2 (10°)	40.0	44.6	4.6	61.5	69.5	7.8
734	200 (198.45)		40.5	42.9	2.4	59.5	68.5	8.8
735	300 (297.70)		40.2	41.9	1.7	57.5	67.5	9.8
736	400 (396.90)		40.0	41.5	1.5	55.5	66.5	10.8
737	100 (99.225)	0.02846x2 (15°)	40.5	46.7	6.2	64.0	71.5	6.55
738	200 (198.45)		40.2	43.5	3.3	62.0	70.5	7.55
739	300 (297.70)		40.0	42.2	2.2	59.0	69.0	9.05
740	400 (396.90)		40.0	41.7	1.7	56.5	68.0	10.30
741	100 (99.225)	0.0337297x2 (19°)	40.5	48.0	7.5	64.0	71.0	6.55
742	200 (198.45)		40.2	44.2	4.0	62.0	70.5	7.55
743	300 (297.70)		40.0	42.8	2.8	59.0	69.0	9.05
744	400 (396.90)		40.0	42.1	2.1	58.0	69.0	9.55

TABLE - A1-50 (Contd.)

Sl. No.	Average Conden-sate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\ddot{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent enhancement (Experi-mental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
733	5.22	460	456	1.10	1412	1842	64
734	5.84	513	476	1.23	1395	1788	60
735	6.29	553	506	1.32	1351	1740	62
736	6.82	600	595	1.44	1330	1699	59
737	7.043	620	615	1.09	1663	1925	52
738	7.448	655	655	1.15	1524	1958	52
739	7.807	687	655	1.21	1334	1775	69
740	8.100	713	675	1.25	1216	1719	61
741	8.640	760	744	1.13	1720	1925	65
742	9.127	803	794	1.19	1577	1859	67
743	9.529	839	834	1.24	1374	1775	83
744	9.529	839	834	1.24	1302	1752	84

TABLE - A1-51

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_v) = 77.1°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
745	100 (98.807)	0.02088 4x2 (10°)	50.0	54.0	4.0	66.0	72.0	5.55
746	200 (197.614)		50.2	52.2	2.0	65.0	71.5	6.05
747	300 (296.421)		50.0	51.5	1.5	64.0	71.0	6.55
748	400 (395.228)		50.0	51.2	1.2	61.5	69.0	7.80
749	100 (98.807)	0.02846x2 (15°)	50.0	55.0	5.0	68.0	73.5	5.5
750	200 (197.614)		50.0	52.6	2.6	66.0	72.5	6.5
751	300 (296.421)		50.0	52.0	2.0	65.0	72.5	7.5
752	400 (395.228)		50.2	51.8	1.6	63.0	71.0	8.0
753	100 (98.807)	0.0337297x2 (19°)	50.0	56.2	6.2	68.0	73.5	4.55
754	200 (197.614)		49.8	52.9	3.1	67.0	73.5	5.05
755	300 (296.421)		50.0	52.1	2.1	66.0	73.0	5.55
756	400 (395.228)		50.0	51.6	1.6	65.0	73.0	6.05

TABLE - A1-51 (contd.)

Sl. No.	Average Conden-sate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experi-mental) div.-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
745	4.62	407	395	0.97	1755	2006	39
746	4.80	423	395	1.01	1674	1964	45
747	5.10	449	445	1.07	1637	1925	41
748	5.40	475	474	1.14	1458	1843	52
749	5.635	496	494	0.87	1915	2109	60
750	6.231	548	514	0.96	1735	2006	55
751	6.750	594	593	1.04	1725	1964	39
752	7.200	634	632	1.11	1580	1890	46
753	7.043	620	613	0.92	2020	2109	71
754	7.043	620	613	0.92	1820	2054	62
755	7.448	655	622	0.97	1750	2006	42
756	7.448	655	632	0.97	1605	1964	45

TABLE - A1-52

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_V) = 76.75°C

Sl. No.	Coolant Rate Lit/hr (kg/hr)	Area equivalent to Diverging-Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_W)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_V - T_W}{2}$
			°C			°C	°C	C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
757	100 (99.568)		30.2	35.0	4.8	53.5	66	11.625
758	200 (199.136)		30.0	32.7	2.7	50.0	64	13.375
759	300 (298.70)	0.020884x2 (10°)	30.0	31.9	1.9	48.5	64	14.125
760	400 (398.27)		30.0	31.4	1.4	47.5	63	14.625
761	100 (99.568)		30.0	37.0	7.0	56.5	68.5	10.125
762	200 (199.136)		30.0	33.6	3.6	54.5	67.5	11.125
763	300 (298.70)	0.02846x2 (15°)	30.2	32.8	2.6	51.5	65	12.625
764	400 (398.27)		30.0	32.1	2.1	49.5	65	13.625
765	100 (99.568)		29.5	37.7	8.2	58.0	69.0	9.375
766	200 (199.136)		30.0	34.3	4.3	56.0	68.0	10.375
767	300 (298.70)	0.0337297x2 (19°)	30.0	33.0	3.0	54.5	67.5	11.125
768	400 (398.27)		30.2	32.6	2.4	52.5	66.5	12.125

TABLE - A1-52 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{\bar{h}}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{\bar{h}}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
757	11.01	520	478	1.24	1071	1490	63
758	12.06	570	538	1.36	1020	1439	69
759	12.60	595	568	1.42	1009	1419	62
760	12.74	601	597	1.44	983	1407	56
761	15.12	714	697	1.25	1239	1543	51
762	15.53	734	717	1.29	1159	1507	52
763	16.68	788	777	1.38	1097	1460	46
764	17.72	837	836	1.47	1079	1432	43
765	17.45	824	816	1.22	1303	1626	58
766	18.59	878	856	1.30	1254	1533	53
767	19.55	923	896	1.37	1230	1507	46
768	20.62	974	956	1.44	1190	1475	32

TABLE - A1-53

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP (T_v) = 76.75°C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging + Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
769	100 (99.225)	0.020884x2 (10°)	40.2	44.3	4.1	63.0	70.0	6.875
770	200 (198.45)		40.5	43.0	2.5	58.5	67.5	9.125
771	300 (297.70)		40.0	41.7	1.7	57.5	66.5	9.625
772	400 (396.90)		40.0	41.4	1.4	55.5	65.5	10.625
773	100 (99.225)	0.02846x2 (15°)	40.0	45.6	5.6	64.0	71.0	6.375
774	200 (198.45)		40.2	43.2	3.0	62.0	70.0	7.375
775	300 (297.70)		40.0	42.2	2.2	60.0	69.0	8.375
776	400 (396.90)		40.0	41.7	1.7	58.0	68.0	9.375
777	100 (99.225)	0.0337297x2 (19°)	40.0	46.5	6.5	65.0	71.0	5.875
778	200 (198.45)		40.0	43.4	3.4	63.0	70.0	6.875
779	300 (297.70)		40.2	42.6	2.4	61.0	69.0	7.875
780	400 (396.90)		40.0	42.1	2.1	59.0	68.0	8.875

TABLE - A1-53 (contd.)

Sl. No.	Average Condensate Rate Kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
769	8.929	421	406	1.00	1466	1699	29
770	10.800	510	496	1.22	1338	1583	41
771	11.230	530	506	1.26	1318	1562	38
772	11.930	564	555	1.35	1270	1524	37
773	12.060	570	556	1.00	1571	1732	24
774	12.600	595	595	1.04	1417	1670	44
775	14.175	670	655	1.18	1406	1617	38
776	16.200	714	675	1.25	1338	1572	39
777	14.175	669	645	0.99	1688	1767	36
778	15.120	714	675	1.06	1540	1699	37
779	16.200	765	715	1.13	1440	1643	46
780	18.000	850	833	1.26	1420	1594	38

TABLE - A1-54

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = STRAIGHT UNIFORM TUBE

VAPOUR TEMP. (T_V) = 76.75 °C

Sl. No.	Coolant Rate Lit/hr (Kg/hr)	Area equivalent to Diverging - Converging Cone Section m^2 (For θ)	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
781	100(98.807)	0.020884x2 (10°)	50.0	53.5	3.5	66.0	71.5	5.375
782	200(197.614)		50.2	52.0	1.8	64.0	70.5	6.375
783	300(296.421)		50.2	51.6	1.4	62.0	69.0	7.375
784	400(395.228)		50.0	51.1	1.1	61.0	69.0	7.875
785	100(98.807)	0.02846x2 (15°)	50.5	54.3	3.8	68.0	73.0	4.375
786	200(197.614)		50.2	52.4	2.2	66.0	72.0	5.375
787	300(296.421)		50.0	51.6	1.6	64.0	71.0	6.375
788	400(395.228)		50.0	51.4	1.4	62.0	70.0	7.375
789	100(98.807)	0.0337297x2 (19°)	49.8	55.2	5.4	68.0	73.0	4.375
790	200(197.614)		50.0	52.8	2.8	66.0	72.0	5.375
791	300(296.421)		50.0	52.0	2.0	65.0	71.5	5.875
792	400(395.228)		50.0	51.6	1.6	63.0	70.0	6.875

TABLE - A1-54 (contd.)

Sl. No.	Average Condensate Rate kg/hr	Heat Released by vapour Kcal/hr	Heat Received by Coolant Kcal/hr	Heat Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{icy}$ Expt. $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{icy_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$	Percent Enhancement (Experimental) div-con.
(0)	(7)	(8)	(9)	(10)	(11)	(12)	(13)
781	7.820	369	345	0.88	1644	1807	26
782	7.980	377	355	0.90	1416	1732	33
783	8.790	415	415	0.99	1347	1670	33
784	9.290	439	435	1.05	1334	1643	31
785	8.723	412	376	0.72	1654	1903	36
786	9.610	454	435	0.79	1484	1807	44
787	10.800	510	474	0.89	1405	1732	40
788	12.060	570	553	1.00	1358	1670	37
789	11.450	541	534	0.80	1833	1903	46
790	11.940	564	553	0.84	1555	1807	65
791	12.600	595	593	0.88	1501	1767	56
792	13.833	653	632	0.97	1408	1699	45

TABLE - A1-55

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate lit/hr (kg/hr)	No. of Diverging-Converging Units	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
793	100(99.568)	One	30.3	39.3	9.0	81.0	92	9.5
794	200(199.136)		30.0	35.0	5.0	77.0	90	11.5
795	300(298.70)		30.0	33.4	3.4	74.0	89	13.0
796	400(398.27)		30.2	32.9	2.7	71.5	87	14.25
797	100(99.568)	Two	30.0	49.0	19.0	80.0	92	10.0
798	200(199.136)		30.5	40.5	10.0	77.0	90	11.5
799	300(298.70)		30.5	37.4	6.9	74.0	89	13.0
800	400(398.27)		30.0	35.4	5.4	71.5	87	14.25
801	100(99.568)	Three	30.5	57.0	26.5	81.0	92.5	9.5
802	200(199.136)		30.0	44.5	14.5	78.0	90	11.0
803	300(298.70)		30.0	39.7	9.7	76.0	89	12.0
804	400(398.27)		30.0	37.3	7.3	74.0	87	13.0
805	100(99.568)	Four	30.0	62.0	32.0	82.0	92	9.0
806	200(199.136)		30.5	47.7	17.2	80.0	91	10.0
807	300(298.70)		30.0	41.5	11.5	78.0	89.5	11.0
808	400(398.27)		30.5	39.5	9.0	76.0	88.0	12.0

TABLE - A1-55 (contd.)

Serial No.	Average Conden-sate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat-Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.\,°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
793	1.669	916	896	9.53	10032	10233
794	1.885	1016	996	10.57	9192	9756
795	1.927	1038	1016	10.80	8307	9462
796	2.066	1112	1075	11.56	8119	9247
797	3.556	1917	1892	9.97	9973	10103
798	3.750	2021	1991	10.51	9143	9756
799	3.830	2066	2061	10.75	8267	9462
800	4.059	2187	2150	11.38	7984	9247
801	5.036	2714	2639	9.41	9908	10233
802	5.391	2905	2887	10.07	9159	9865
803	5.391	2905	2897	10.07	8395	9653
804	5.476	2951	2907	10.23	7873	9462
805	6.389	3443	3186	8.95	9951	10373
806	6.509	3508	3425	9.12	9125	10103
807	6.509	3508	3435	9.12	8296	9865
808	6.765	3648	3584	9.48	7903	9653

TABLE - A1-56

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate lit/hr (kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_v - T_w}{2}$
				°C		°C	°C	°C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
809	100(99.225)		40.0	48.1	8.1	84.0	94	8.0
810	200(198.45)		40.2	44.5	4.3	82.0	92	9.0
811	300(297.70)	One	40.5	43.5	3.0	79.0	91	10.5
812	400(396.90)		40.0	42.4	2.4	75.0	90	12.5
813	100(99.225)		40.5	57.5	17.0	83.0	93	8.5
814	200(198.45)		40.0	48.9	8.9	81.0	92	9.5
815	300(297.70)	Two	40.0	46.2	6.2	79.0	91	10.5
816	400(396.90)		40.2	45.0	4.8	76.0	90	12.0
817	100(99.225)		40.0	63.0	23.0	84.0	93	8.0
818	200(198.45)		40.2	52.2	12.0	82.0	92	9.0
819	300(297.70)	Three	40.0	48.0	8.0	81.0	91	9.5
820	400(396.90)		40.5	46.5	6.0	79.0	91	10.5
821	100(99.225)		40.0	67.5	27.5	86.0	94	7.0
822	200(198.45)		40.0	54.8	14.8	84.0	92	8.0
823	300(297.70)	Four	40.5	51.0	10.5	82.0	92	9.0
824	400(396.90)		40.5	48.5	8.0	80.0	91	10.0

TABLE - A1-56 (contd.)

Serial No.	Average Conden-sate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) $X10^{-4}$ Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.\ ^\circ C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.\ ^\circ C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
809	1.513	815	804	8.48	10599	10683
810	1.605	864	853	9.00	9988	10373
811	1.725	930	893	9.67	9215	9981
812	1.906	1027	953	10.68	8548	9555
813	3.194	1722	1687	8.90	10539	10522
814	3.317	1788	1766	9.30	9791	10233
815	3.520	1897	1846	9.86	9399	9981
816	3.670	1978	1905	10.30	8575	9653
817	4.423	2384	2282	8.26	10335	10683
818	4.662	2512	2381	8.71	9680	10373
819	4.726	2547	2382	8.83	9298	10233
820	4.726	2547	2382	8.83	8413	9981
821	5.227	2817	2729	7.33	10468	11045
822	5.564	2999	2937	7.80	9751	10683
823	5.948	3206	3126	8.34	9266	10373
824	6.053	3262	3175	8.48	8485	10103

TABLE - A1-57

LIQUID SYSTEM = WATER

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 100°C

Serial No.	Coolant Rate lit/hr (kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
825	100(98.807)	One	50.5	58.1	7.6	88.0	95	6.0
826	200(197.614)		50.0	53.9	3.9	86.0	94.5	7.0
827	300(296.421)		50.2	52.8	2.6	84.0	92	8.0
828	400(395.228)		50.0	52.1	2.1	82.0	92	9.0
829	100(98.807)	Two	50.8	65.3	14.5	88.0	94	6.0
830	200(197.614)		50.5	58.7	8.2	85.0	93	7.5
831	300(296.421)		50.0	55.6	5.6	83.0	93	8.5
832	400(395.228)		50.0	54.2	4.2	82.0	92	9.0
833	100(98.807)	Three	50.0	68.6	18.6	90.0	95	5.0
834	200(197.614)		50.0	60.0	10.0	87.5	94.5	8.25
835	300(296.421)		51.0	58.0	7.0	86.0	93	7.0
836	400(395.228)		50.2	55.6	5.4	84.0	92	8.0
837	100(98.807)	Four	50.0	74.0	24.0	90.0	96	5.0
838	200(197.614)		50.0	62.8	12.8	88.0	95	6.0
839	300(296.421)		50.5	59.0	8.5	87.5	94.5	6.5
840	400(395.228)		50.5	57.0	6.5	86.0	94	7.0

TABLE - A1-57 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
825	1.396	753	751	7.83	13058	11479
826	1.468	791	771	8.20	11757	11045
827	1.468	791	771	8.22	10287	10683
828	1.582	852	830	8.86	9849	10373
829	2.782	1499	1433	7.80	12997	11479
830	3.053	1646	1620	8.56	11417	10856
831	3.136	1690	1660	8.80	10343	10522
832	3.136	1690	1660	8.80	9768	10373
833	3.450	1859	1838	6.45	12894	12015
834	3.876	2089	1976	7.24	11592	11363
835	3.876	2089	2075	7.24	10350	11045
836	4.207	2267	2134	7.86	9828	10683
837	4.600	2479	2371	6.45	12896	12015
838	4.825	2600	2529	6.76	11271	11479
839	4.825	2600	2549	6.76	10404	11252
840	4.929	2656	2569	6.91	9869	11045

TABLE - A1-58

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_V) = 77.1°C

Serial No.	Coolant Rate lit/hr (kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o)	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
841	100 (99.568)	One	30.5	33.3	2.8	52.5	67	12.30
842	200 (199.136)		30.0	31.4	1.4	50.5	65	13.30
843	300 (298.70)		30.0	31.0	1.0	48.5	64	14.30
844	400 (398.27)		30.2	31.0	0.8	45.5	63	15.80
845	100 (99.568)	Two	30.2	34.8	4.6	56.5	69	10.30
846	200 (199.136)		30.5	33.2	2.7	52.5	67	12.30
847	300 (298.70)		30.0	31.9	1.9	50.5	65	13.30
848	400 (398.27)		30.5	32.0	1.5	48.5	64	14.30
849	100 (99.568)	Three	30.0	36.8	6.8	56.5	69	10.3
850	200 (199.136)		30.0	33.8	3.8	53.5	67	11.8
851	300 (298.70)		30.5	33.1	2.6	51.5	65	12.8
852	400 (398.27)		30.5	32.6	2.1	49.5	64	13.8
853	100 (99.568)	Four	30.0	39.3	9.3	56.5	68	10.3
854	200 (199.136)		30.5	35.6	5.1	53.5	66	11.8
855	300 (298.70)		30.2	33.8	3.6	51.5	65	12.8
856	400 (398.27)		30.5	33.4	2.9	49.5	64	13.8

TABLE - A1-58 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
841	3.176	280	279	2.91	2368	2133
842	3.306	291	279	3.03	2276	2092
843	3.521	310	299	3.22	2255	2054
844	3.811	335	319	3.48	2206	2004
845	5.226	460	458	2.40	2323	2230
846	6.113	538	537	2.80	2275	2133
847	6.480	572	568	2.97	2237	2092
848	6.894	606	597	3.15	2205	2054
849	7.900	695	677	2.41	2340	2230
850	8.756	771	757	2.67	2266	2155
851	9.127	803	777	2.78	2176	2112
852	9.818	864	836	3.00	2171	2073
853	10.623	935	926	2.43	2361	2230
854	11.571	1018	1016	2.65	2244	2155
855	12.461	1097	1075	2.85	2229	2112
856	13.224	1161	1155	3.02	2188	2073

TABLE - A1-59

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr (kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_1)	Outlet (T_o) °C	ΔT	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
857	100 (99.225)		40.0	42.0	2.0	61.0	71	8.05
858	200 (198.45)	One	40.5	41.6	1.1	58.0	69	9.55
859	300 (297.70)		40.5	41.3	0.8	56.5	68	10.30
860	400 (396.90)		40.0	40.6	0.6	55.5	67	10.80
861	100 (99.225)		40.0	43.2	3.2	64.0	72	6.55
862	200 (198.45)	Two	40.5	42.2	1.7	63.0	70	7.05
863	300 (297.70)		40.0	41.3	1.3	61.0	69	8.05
864	400 (396.90)		40.2	41.2	1.0	60.0	68	8.55
865	100 (99.225)		40.0	44.8	4.8	64.0	71	6.55
866	200 (198.45)	Three	40.0	42.5	2.5	63.0	71	7.05
867	300 (297.70)		40.2	42.0	1.8	62.0	70	7.55
868	400 (396.90)		40.0	41.4	1.4	61.0	70	8.05
869	100 (99.225)		40.5	47.0	6.5	64.0	71	6.55
870	200 (198.45)	Four	40.0	43.7	3.7	62.0	71	7.55
871	300 (297.70)		40.0	42.7	2.7	60.0	70	8.55
872	400 (396.90)		40.2	42.3	2.1	59.0	69	9.05

TABLE - A1-59 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
857	2.382	210	198	2.18	2556	2372
858	2.613	230	218	2.39	2560	2273
859	2.842	250	238	2.60	2525	2230
860	2.892	255	238	2.65	2457	2203
861	3.724	328	318	1.70	2605	2497
862	3.904	344	337	1.79	2538	2452
863	4.438	391	387	2.03	2527	2372
864	4.563	401	397	2.09	2440	2336
865	5.491	483	476	1.67	2557	2497
866	5.785	509	496	1.76	2504	2452
867	6.159	542	536	1.88	2490	2410
868	6.416	565	556	2.00	2434	2372
869	7.364	648	645	1.69	2573	2497
870	8.416	741	734	1.93	2553	2410
871	9.257	815	804	2.12	2480	2336
872	9.529	839	833	2.18	2411	2303

TABLE - A1-60

LIQUID SYSTEM = ETHYL ACETATE

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 77.1°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT	Avg. Wall Temp (T_w)	Avg. Condensate Temp.	$\Delta T_f = \frac{T_v - T_w}{2}$
				°C		°C	°C	°C
(0)	(1)	(2)		(3)		(4)	(5)	(6)
873	100(98.807)		50.0	51.6	1.6	65.0	71.0	6.05
874	200(197.614)	One	50.0	50.8	0.8	64.0	70.5	6.55
875	300(296.421)		50.5	51.1	0.6	62.0	70.0	7.55
876	400(395.228)		50.5	51.0	0.5	61.0	70.0	8.05
877	100(98.807)		50.2	52.6	2.4	68.0	72.0	4.55
878	200(197.614)	Two	50.5	51.8	1.3	67.0	71.5	5.05
879	300(296.421)		50.0	51.0	1.0	65.0	71.0	6.05
880	400(395.228)		50.0	50.8	0.8	64.0	71.0	6.55
881	100(98.807)		50.5	54.0	3.5	68.0	72.0	4.55
882	200(197.614)	Three	50.0	51.9	1.9	67.0	71.5	5.05
883	300(296.421)		50.5	51.8	1.3	66.0	71.0	5.55
884	400(395.228)		50.0	51.1	1.1	65.0	71.0	6.05
885	100(98.807)		50.0	54.8	4.8	68.0	72.0	4.55
886	200(197.614)	Four	50.5	53.2	2.7	67.0	71.5	5.05
887	300(296.421)		50.5	52.4	1.9	66.0	71.0	5.55
888	400(395.228)		50.0	51.5	1.5	65.0	70.0	6.05

TABLE - A1-60 (contd.)

Serial No.	Average Condensate Rate kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) x10^{-4} Kcal/hr m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
873	1.873	165	158	1.72	2838	2547
874	1.928	170	158	1.77	2700	2497
875	2.160	190	178	1.97	2618	2410
876	2.250	198	198	2.06	2559	2372
877	2.793	246	237	1.28	2813	2735
878	3.000	264	257	1.37	2720	2665
879	3.446	303	296	1.58	2605	2547
880	3.640	320	316	1.67	2542	2497
881	4.154	366	346	1.26	2790	2735
882	4.30	396	375	1.37	2720	2665
883	4.765	419	385	1.45	2618	2603
884	5.063	446	434	1.55	2557	2547
885	5.586	492	474	1.28	2813	2735
886	6.113	538	534	1.40	2771	2665
887	6.416	565	563	1.47	2648	2603
888	6.750	594	593	1.55	2554	2547

TABLE - A1-61

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate lit/hr (kg/hr)	No. of Diverging -Converging Units	Average Coolant Temperature Inlet (T_i) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
889	100 (99.568)	One	30.5	32.6	2.1	51.5	66	12.625
890	200 (199.136)		30.0	31.1	1.1	49.5	65	13.625
891	300 (298.70)		30.2	31.0	0.8	45.5	63	15.625
892	400 (398.27)		30.0	30.6	0.6	43.0	62	16.875
893	100 (99.568)	Two	30.0	33.7	3.7	54.5	68	11.125
894	200 (199.136)		30.2	32.1	1.9	52.5	66	12.125
895	300 (298.70)		30.0	31.3	1.3	50.5	65	13.125
896	400 (398.27)		30.2	31.2	1.0	49.5	64	13.625
897	100 (99.568)	Three	30.0	35.0	5.0	56.5	67	10.125
898	200 (199.136)		30.5	33.1	2.6	54.5	66	11.125
899	300 (298.70)		30.5	32.3	1.8	52.5	65	12.125
900	400 (398.27)		30.8	32.2	1.4	51.5	64	12.625
901	100 (99.568)	Four	30.5	37.2	6.7	56.5	67	10.125
902	200 (199.136)		30.0	33.6	3.6	54.5	66	11.125
903	300 (298.70)		30.0	32.5	2.5	52.5	65	12.125
904	400 (398.27)		30.5	32.4	1.9	50.5	64	13.125

TABLE - A1-61 (contd.)

Serial No.	Average Condensate Rate	Heat - Released by vapour	Heat - Received by Coolant	Heat - Flux ($\bar{q}$) X10^{-4}	$\bar{h}_{idct_{Expt.}}$	$\bar{h}_{idct_{Theo.}}$
	Kg/hr	Kcal/hr	Kcal/hr	Kcal/hr.m^2	$\frac{Kcal}{hr.m^2.°C}$	$\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
889	4.499	212	209	2.21	1747	1893
890	4.764	223	219	2.32	1702	1858
891	5.062	239	239	2.49	1591	1795
892	5.299	250	239	2.60	1541	1761
893	7.875	372	368	1.94	1740	1954
894	8.462	400	378	2.08	1716	1913
895	8.541	406	388	2.11	1609	1875
896	8.859	418	398	2.17	1596	1858
897	10.698	505	498	1.75	1730	2001
898	11.571	546	518	1.89	1702	1954
899	12.063	570	538	1.98	1630	1913
900	12.063	570	558	1.98	1567	1894
901	14.354	678	667	1.76	1742	2001
902	15.534	734	717	1.91	1716	1955
903	15.971	754	747	1.96	1618	1913
904	16.676	787	757	2.05	1560	1875

TABLE - A1-62

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE
TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE
VAPOUR TEMP. (T_v) = 76.75°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	No.of Diverging -Converging Units	Average Coolant Temperature Inlet (T_i)	Outlet (T_o)	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_v - T_w}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
905	100 (99.225)	One	40.0	41.7	1.7	58.0	68.0	9.375
906	200 (198.45)		40.5	41.4	0.9	55.5	67.0	10.625
907	300 (297.70)		40.2	40.9	0.7	53.5	65.5	11.625
908	400 (396.90)		40.0	40.5	0.5	50.5	65.0	13.125
909	100 (99.225)	Two	40.0	43.0	3.0	60.0	69.0	8.375
910	200 (198.45)		40.0	41.6	1.6	59.0	68.0	8.875
911	300 (297.70)		40.8	41.9	1.1	58.0	66.0	9.375
912	400 (396.90)		40.5	41.4	0.9	55.5	66.0	10.625
913	100 (99.225)	Three	40.5	44.7	4.2	62.0	69.0	7.375
914	200 (198.45)		40.5	42.7	2.2	60.0	68.0	8,375
915	300 (297.70)		40.0	41.6	1.6	59.0	66.5	8.875
916	400 (396.90)		40.5	41.7	1.2	58.0	66.5	9.375
917	100 (99.225)	Four	40.5	46.1	5.6	62.0	69.0	7.375
918	200 (198.45)		40.0	43.1	3.1	60.0	68.0	8.375
919	300 (297.70)		40.2	42.4	2.2	58.0	67.0	9.375
920	400 (396.90)		40.0	41.8	1.8	55.5	66.0	10.625

TABLE - A1-62 (contd.)

Serial No.	Average Condensate Rate	Heat - Released by vapour	Heat - Received by Coolant	Heat - Flux ($\bar{q}$) $X10^{-4}$	$\bar{h}_{idct_{Expt.}}$	$\bar{h}_{idct_{Theo.}}$
	Kg/hr	Kcal/hr	Kcal/hr	$Kcal/hr.m^2$	$\frac{Kcal}{hr.m^2.°C}$	$\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
905	3.779	179	169	1.86	1987	2040
906	4.199	198	179	2.06	1939	1977
907	4.429	209	208	2.17	1871	1933
908	4.805	227	198	2.36	1799	1875
909	6.671	315	298	1.64	1957	2098
910	6.915	327	318	1.70	1917	2068
911	7.087	335	327	1.74	1859	2040
912	7.662	362	357	1.88	1773	1977
913	9.000	425	417	1.47	1999	2166
914	9.776	462	437	1.60	1913	2098
915	10.125	478	476	1.66	1868	2068
916	10.309	487	476	1.69	1802	2040
917	11.812	558	556	1.45	1968	2166
918	13.186	623	615	1.62	1935	2098
919	14.354	678	655	1.76	1881	2040
920	15.534	734	715	1.91	1797	1977

TABLE - A1-63

LIQUID SYSTEM = CARBON - TETRA - CHLORIDE

TYPE OF CONDENSER = DIVERGING-CONVERGING TUBE

VAPOUR TEMP. (T_V) = 76.75°C

Serial No.	Coolant Rate Lit/hr (Kg/hr)	No.of Diverging -Converging Units	Average Coolant Temperature Inlet (T_1) °C	Outlet (T_o) °C	ΔT °C	Avg. Wall Temp. (T_w) °C	Avg. Condensate Temp. °C	$\Delta T_f = \frac{T_V - T_W}{2}$ °C
(0)	(1)	(2)	(3)			(4)	(5)	(6)
921	100 (98.807)	One	50.0	51.4	1.4	62.0	70.0	7.375
922	200 (197.614)		50.0	50.7	0.7	60.0	68.5	8.375
923	300 (296.421)		50.5	51.0	0.5	59.0	67.0	8.875
924	400 (395.228)		50.5	50.9	0.4	56.5	66.5	10.125
925	100 (98.807)	Two	50.0	52.2	2.2	66.0	71.0	5.375
926	200 (197.614)		50.0	51.2	1.2	64.0	68.5	6.375
927	300 (296.421)		50.5	51.4	0.9	62.0	68.0	7.375
928	400 (395.228)		50.5	51.2	0.7	60.0	67.0	8.375
929	100 (98.807)	Three	50.5	54.2	3.7	64.0	70.0	6.375
930	200 (197.614)		50.0	52.0	2.0	63.0	69.0	6.875
931	300 (296.421)		50.0	51.4	1.4	62.0	68.0	7.375
932	400 (395.228)		50.0	51.1	1.1	60.0	67.0	8.375
933	100 (98.807)	Four	50.0	54.7	4.7	65.0	71	5.875
934	200 (197.614)		50.0	52.6	2.6	63.0	70	6.875
935	300 (296.421)		50.5	52.3	1.8	62.0	69	7.375
936	400 (395.228)		50.2	51.6	1.4	61.0	68	7.875

TABLE - A1-63 (contd.)

Serial No.	Average Condensate Rate Kg/hr	Heat - Released by vapour Kcal/hr	Heat - Received by Coolant Kcal/hr	Heat - Flux ($\bar{q}$) X10^{-4} Kcal/hr.m^2	$\bar{h}_{idct_{Expt.}}$ $\frac{Kcal}{hr.m^2.°C}$	$\bar{h}_{idct_{Theo.}}$ $\frac{Kcal}{hr.m^2.°C}$
(0)	(7)	(8)	(9)	(10)	(11)	(12)
921	3.115	147	138	1.53	2073	2166
922	3.335	158	138	1.64	1962	2098
923	3.478	164	148	1.71	1923	2068
924	3.635	172	158	1.79	1767	2001
925	4.609	218	217	1.13	2110	2344
926	5.250	248	237	1.29	2024	2246
927	5.727	271	267	1.41	1911	2166
928	6.231	294	277	1.53	1826	2098
929	8.100	382	366	1.32	2078	2246
930	8.463	400	395	1.39	2018	2204
931	8.859	418	415	1.45	1966	2166
932	9.295	439	435	1.52	1818	2098
933	9.947	470	464	1.22	2081	2293
934	11.009	520	514	1.35	1967	2204
935	11.812	558	534	1.45	1968	2166
936	11.812	558	553	1.45	1843	2130

APPENDIX - II.

Table - II-1 Dimensions of Diverging/Converging Cone Sections (Material : Brass)

Nos.	r_1	r_2	r_S	H	t	A_i	A_o	Θ
1.	0.01	0.02250	0.081	0.144	0.0015	0.014860	0.016231	5
2.	0.01	0.03575	0.081	0.144	0.0015	0.020884	0.022253	10
3.	0.01	0.04975	0.081	0.144	0.0015	0.028460	0.029866	15
4.	0.01	0.06000	0.081	0.144	0.0015	0.0337297	0.035175	19

Table - II-2 Dimensions of Diverging - Converging (Combined) Cone Sections (Material:Brass)

Nos.	r_1	r_2	r_S	H	t	A_i	A_o	Θ
1.	0.01	0.02250	0.081	0.288	0.0015	0.02972	0.032462	5
2.	0.01	0.03575	0.081	0.288	0.0015	0.041768	0.044506	10
3.	0.01	0.04975	0.081	0.288	0.0015	0.05692	0.059732	15
4.	0.01	0.06000	0.081	0.288	0.0015	0.0674594	0.070350	19

Table - II-3 Dimensions of Uniform Cylindrical Tube Sections (Material : Brass)

Nos.	Area Equivalent to	r_e	r_S	H	t	A_i	A_o
1.	10°Div./Con. Cone	0.02300	0.081	0.144	0.0015	0.020884	0.022253
2.	10°Div.-Con. Cone	0.02300	0.081	0.288	0.0015	0.041768	0.044506
3.	15°Div./Con. Cone	0.03150	0.081	0.144	0.0015	0.028460	0.029866
4.	15°Div.-Con. Cone	0.03150	0.081	0.288	0.0015	0.05692	0.059732
5.	19°Div./Con. Cone	0.03730	0.081	0.144	0.0015	0.0337297	0.035175
6.	19°Div.-Con. Cone	0.03730	0.081	0.288	0.0015	0.0674594	0.070350

Table - II-4 Dimensions of Diverging - Converging Tubes (Material : Copper)

Nos.	No. of Div.-Con Unit	r_1	r_2	r_s	H	t	A_i	θ
1.	One	0.01	0.02	0.0305	0.10	0.00081	9.6114×10^{-3}	11.31
2.	Two	0.01	0.02	0.0305	0.20	0.00081	18.2228×10^{-3}	11.31
3.	Three	0.01	0.02	0.0305	0.30	0.00081	28.8342×10^{-3}	11.31
4.	Four	0.01	0.02	0.0305	0.40	0.00081	38.4456×10^{-3}	11.31

APPENDIX - III.

Table - III-1, Physical, Thermodynamic and Transport Properties of Test Fluids

Liquids	Boiling point (°C)	Density (gms/cm^3)	Specific Heat (Cals/gms.°C)	Latent Heat (Cals/gms.)	Thermal conductivity (Cals/Sec.Cm°C)	Viscosity (gms/Cm.Sec
Water (Distilled)	100	0.95838	1.00000	538.88	1.6430×10^{-3}	0.0027
Ethyl Alcohol	78.3	0.73720	0.77000	210.00	0.3593×10^{-3}	0.0045
Ethyl Acetate	77.1	0.9000	0.45700	88.00	0.4170×10^{-3}	0.0026
Carbon-Tetra-Chloride	76.75	1.5750	0.22000	47.22	0.3880×10^{-3}	0.0054

APPENDIX - IV.

Solution of Equations Involved in Mathematical Analyses.

IV.1 Solution of Equation (3.2.14)

Rewriting equation (3.2.14)

$$\frac{l\,dl}{\delta_d} = \left[\frac{\rho^2 g \lambda \cos\theta}{3\mu k (T_v - T_w)}\right] \cdot \left[d(l\,\delta_d^3)\right] \qquad \ldots(3.2.14)$$

$$= [X] \cdot \left[d(l\,\delta_d^3)\right] \qquad \ldots(3.2.14a)$$

where, $$[X] = \left[\frac{\rho^2 g \lambda \cos\theta}{3\mu k\ (T_v - T_w)}\right]$$

Let, $\vartheta = l\,\delta_d^3$

differentiating, $d\vartheta = d\ (l\,\delta_d^3)$

Again, $\vartheta^{1/3} = l^{1/3} \cdot \delta_d$

From equation (3.2.14a) it follows,

$$\frac{l^{4/3}.dl}{\vartheta^{1/3}} = [X] \cdot d\vartheta$$

B.C. $\quad l = l_1 \quad , \qquad \delta_d = 0, \qquad \vartheta = 0$

$\qquad l = l \quad , \qquad \delta_d = \delta_d \, , \qquad \vartheta = l\,\delta_d^3$

$$\text{Integrating,} \quad \int_{l_1}^{l} l^{4/3}.dl = [X] \, . \int_0^{\vartheta} \vartheta^{1/3} . \, d\vartheta$$

$$\frac{3}{7} \left(l^{7/3} - l_1^{7/3} \right) = [X] \, . \, \frac{3}{4} \left(\vartheta \right)^{4/3}$$

$$\frac{4}{7\,[X]} \left(l^{7/3} - l_1^{7/3} \right) = \left(l\,\delta_d^3 \right)^{4/3} = l^{4/3} . \, \delta_d^4 \, .$$

Therefore,

$$\delta_d^4 = \left[\frac{4}{7\,[X]} \right] . \left[\frac{l^{7/3} - l_1^{7/3}}{l^{4/3}} \right]$$

$$\text{or,} \quad \delta_d = \left[\frac{4}{7\,[X]} \right]^{1/4} . \left[\frac{l^{7/3} - l_1^{7/3}}{l^{4/3}} \right]^{1/4}$$

$$= \left[\frac{4}{7\,[X]} \right]^{1/4} . \left[l - \left(\frac{l_1^7}{l^4} \right)^{1/3} \right]^{1/4}$$

$$= \left[\frac{4}{7\,[X]} \right]^{1/4} . \left[l \left\{ 1 - \left(\frac{l_1}{l} \right)^{7/3} \right\} \right]^{1/4}$$

$$= \left[\frac{4}{7\ [X]\ \sin\theta} \right]^{\frac{1}{4}} \cdot \left[r \left\{ 1 - \left(\frac{r_1}{r}\right)^{7/3} \right\} \right]^{\frac{1}{4}}.$$

(Because, $r = l \sin\theta$)

Substituting for $[X]$,

$$\delta_d = \left[\frac{(4 \times 3)\ \mu k (T_v - T_w)}{7 \rho^2\ g \lambda \cos\theta \sin\theta} \right]^{\frac{1}{4}} \cdot \left[r \left\{ 1 - \left(\frac{r_1}{r}\right)^{7/3} \right\} \right]^{\frac{1}{4}}$$

$$= \left[\frac{24 \mu K (T_v - T_w)}{7 \rho^2\ g \lambda \sin 2\theta} \right]^{\frac{1}{4}} \cdot \left[r \left\{ 1 - \left(\frac{r_1}{r}\right)^{7/3} \right\} \right]^{\frac{1}{4}} \quad \ldots (3.2.15)$$

IV.2. Solution Of Equation (3.2.19)

Rewriting equation (3.2.19) we have,

$$Q_d = \int_{l_1}^{l} h_{id} \cdot (T_v - T_w) \cdot 2\pi\, l\, dl \cdot \sin\theta \quad \ldots (3.2.19)$$

Putting the value of h_{id} from equation (3.2.18), we have,

$$Q_d = \int_{l_1}^{l} \left[\left(\frac{7 \rho^2 g \lambda k^3 \cos\theta}{12 \mu (T_v - T_w)} \right)^{\frac{1}{4}} \cdot 2\pi (T_v - T_w) \cdot \sin\theta \right] \cdot \left[\frac{1}{l \left[1 - \left(\frac{l_1}{l}\right)^{7/3} \right]} \right]^{\frac{1}{4}} \cdot l\, dl$$

$$= B_d \int_{l_1}^{l} \left[\frac{1}{l\left\{1 - \left(\frac{l_1}{l}\right)^{7/3}\right\}} \right]^{\frac{1}{4}} . l dl$$

where,

$$B_d = \left[\frac{7 \rho^2 g k^3 \lambda \cos\theta}{12 \mu (T_v - T_w)} \right]^{\frac{1}{4}} . 2 \quad (T_v - T_w).\sin\theta$$

$$= B_d \int_{l_1}^{l} \frac{l^{4/3} .dl}{(l^{7/3} - l_1^{7/3})^{\frac{1}{4}}}$$

Let, $l^{7/3} - l_1^{7/3} = x$

$$l^{7/3} = x + l_1^{7/3}$$

differentiating, $l^{4/3}.dl = \frac{3}{7} dx$

B.C., $l = l_1$, $x = 0$

$l = l.$, $x = x$

So we have,

$$Q_d = B_d \int_0^x \frac{\frac{3}{7} dx}{x^{\frac{1}{4}}}$$

$$= \frac{3}{7} B_d \int_0^x x^{-\frac{1}{4}} dx = \frac{4}{7} B_d x^{\frac{3}{4}} = \frac{4}{7} B_d (l^{7/3} - l_1^{7/3})^{3/4}.$$

Now putting,

$$r = l \sin\theta$$

$$Q_d = \frac{4}{7} B_d \left[\frac{r}{\sin\theta}\right]^{7/4} \cdot \left[1 - \left(\frac{r_1}{r}\right)^{7/3}\right]^{3/4}$$

Putting the value of B_d,

$$= \frac{8}{7} \pi (T_v - T_w) . \sin\theta \cdot \left[\frac{7 \rho^2 g \lambda k^3 . \cos\theta}{12 \mu (T_v - T_w)}\right]^{\frac{1}{4}} \cdot \left[\frac{r}{\sin\theta}\right]^{7/4}$$

$$\left[1 - \left(\frac{r_1}{r}\right)^{7/3}\right]^{3/4}.$$

Therefore

$$Q_d \Big|_{r=r_2} = \frac{8}{7} \pi (T_v - T_w)^{\frac{3}{4}} . \sin\theta \left[\frac{7 \rho^2 g \lambda k^3 \cos\theta}{12 \mu}\right]^{\frac{1}{4}} .$$

$$\left[\frac{r_2}{\sin\theta}\right]^{7/4} \cdot \left[1 - \left(\frac{r_1}{r_2}\right)^{7/3}\right]^{3/4}.$$

$$= \frac{8}{7} \lambda (T_v - T_w)^{\frac{3}{4}} \cdot \operatorname{cosec}\theta \left[\frac{7\rho^2 g \lambda k^3 \sin 2\theta}{24\mu} \right]^{\frac{1}{4}} \cdot \left[r_2 \right]^{7/4} \cdot \left[1 - \left(\frac{r_1}{r_2} \right)^{7/3} \right]^{3/4} \qquad \ldots (3.2.20)$$

IV.3 Solution Of Equation (3.3.8b).

Rewriting equation (3.3.8b), we have,

$$\frac{r dr}{\delta_c} = \left[\frac{\rho^2 g \lambda \sin 2\theta}{6 \mu k (T_V - T_W)} \right] \cdot \left[d\ (r \delta_c^3) \right] \qquad \ldots (3.3.8b)$$

$$= [Y] \cdot \left[d\ (r \delta_c^3) \right] \qquad \ldots (3.3.8c)$$

where, $[Y] = \left[\frac{\rho^2 g \lambda \sin 2\theta}{6 \mu k (T_V - T_W)} \right]$

Let, $\vartheta = r \delta_c^3$

differentiating, $d\vartheta = d\ (r \delta_c^3)$

Again, $\vartheta^{1/3} = r^{1/3} \cdot \delta_c$

Therefore, $\delta_c = \left(\frac{\vartheta}{r} \right)^{1/3}$

B.C., $r = r_2$, $\delta_c = 0$, $\vartheta = 0$

$r = r$, $\delta_c = \delta_c$, $\vartheta = r \cdot \delta_c^3$

Therefore, from equation (3.3.3c),

$$\frac{r^{4/3}}{\vartheta^{1/3}} \cdot dr = [Y] . d\vartheta .$$

Integrating,

$$-\int_{r_2}^{r} r^{4/3}\, dr = [Y] . \int_{0}^{\vartheta} \vartheta^{1/3}\, d\vartheta$$

(Negative sign has been incorporated since, for converging cone section local radius decreases from top to bottom of the cone)

$$-\left[\frac{3}{7}\left(r^{7/3} - r_2^{7/3}\right)\right] = \frac{3}{4}\,[Y]\,(\vartheta)^{4/3}$$

$$\text{or,} \quad -\left[\frac{4}{7\,[Y]}\right] . \left[r^{7/3} - r_2^{7/3}\right] = \left[r . \delta_c^{3}\right]^{4/3} = r^{4/3} . \delta_c^{4}$$

$$\text{or,} \quad \delta_c = -\left[\frac{4}{7\,[Y]}\right]^{1/4} . \left[\frac{r^{7/3} - r_2^{7/3}}{r^{4/3}}\right]^{1/4}$$

$$= -\left[\frac{4}{7\,[Y]}\right]^{1/4} . \left[r - \frac{r_2^{7/3}}{r^{4/3}}\right]^{1/4}$$

$$= -\left[\frac{4}{7\,[Y]}\right]^{1/4} . \left[r\left\{1 - \left(\frac{r_2}{r}\right)^{7/3}\right\}\right]^{1/4}$$

$$= \left[\frac{4}{7\,[Y]}\right]^{1/4} . \left[r\left\{\left(\frac{r_2}{r}\right)^{7/3} - 1\right\}\right]^{1/4}$$

Substituting for $[Y]$,

$$\delta_c = \left[\frac{24\ K(T_v-T_w)\mu}{7\rho^2\ g\lambda\ \sin 2\theta}\right]^{\frac{1}{4}} \cdot \left[r\left\{\left(\frac{r_2}{r}\right)^{7/3} - 1\right\}\right]^{\frac{1}{4}} \qquad \ldots(3.3.9)$$

IV.4 Solution Of Equation (3.3.13a)

Rewriting equation (3.3.13a) we have,

$$Q_c = -\int_{r_2}^{r} h_{ic}\ (T_v - T_w).\ 2\pi\ r\,dl \qquad \ldots(3.3.13a)$$

$$= -\int_{r_2}^{r} h_{ic}\ (T_v - T_w)\ 2\pi\ r.\ \mathrm{cosec}\,\theta\ dr$$

(Since, $r = l \sin\theta$)

Now putting the value of h_{ic} from equation (3.3.11), we have,

$$Q_c = -\left[\frac{7\rho^2 g\lambda k^3\ \sin 2\theta}{24\mu(T_v-T_w)}\right]^{\frac{1}{4}} \cdot\ (T_v - T_w)\,2\pi\ \cdot\ \mathrm{cosec}\,\theta\ \cdot$$

$$\int_{r_2}^{r}\left[\frac{1}{\left\{r\left(\frac{r_2}{r}\right)^{7/3} - 1\right\}}\right]^{\frac{1}{4}} .r.dr.$$

$$= - B_c \int_{r_2}^{r} \left[\frac{1}{r\left\{\left(\frac{r_2}{r}\right)^{7/3} - 1\right\}} \right]^{\frac{1}{4}} . \, rdr.$$

where, $B_c = \left[\frac{7 \rho^2 g \lambda k^3 \sin 2\theta}{24 \mu (T_v - T_w)} \right]^{\frac{1}{4}} . \quad (T_v - T_w) . 2\pi . \mathrm{cosec}\,\theta .$

Let, $(r_2^{7/3} - r^{7/3}) = x$

differentiating, $- \frac{7}{3} r^{4/3} . \, dr = dx$

$$r^{4/3} . \, dr = - \frac{3}{7} \, dx$$

B.C. $r = r_2$, $x = 0$

$r = r$, $x = x$

Therefore,

$$Q_c = - B_c \int_{r_2}^{r} \frac{r^{4/3} \, dr}{(r_2^{7/3} - r^{7/3})^{\frac{1}{4}}} = B_c \int_0^x \frac{\frac{3}{7} \, dx}{x^{\frac{1}{4}}} .$$

$$= \frac{4}{7} B_c . (r_2^{7/3} - r^{7/3})^{3/4}$$

$$= \frac{4}{7} B_c . \left[r^{7/3} \left\{ \left(\frac{r_2}{r}\right)^{7/3} - 1 \right\} \right]^{3/4}$$

Therefore,

$$Q_c \Big|_{r=r_1} = \frac{8}{7} \pi \left[\frac{7 \rho^2 g \lambda k^3 \sin 2\theta}{24\mu}\right]^{\frac{1}{4}} . (T_v - T_w)^{3/4} . \mathrm{cosec}\, \theta .$$

$$(r_1)^{7/4} . \left[\left(\frac{r_2}{r_1}\right)^{7/3} - 1 \right]^{3/4} \qquad ...(3.3.14)$$

IV.5. Solution Of Equation (3.3.8c)

Rewriting equation (3.3.8c)

$$\frac{rdr}{\delta_c} = [Z] . [d(r \delta_c^3)]$$

where, $[Z] = \left[\frac{\lambda \rho^2 g \sin 2\theta}{6\mu k (T_v - T_w)}\right]$

Let, $r \delta_c^3 = \vartheta$

differentiating,

$$d(r\delta_c^3) = d\vartheta$$

Again, $r^{1/3} \cdot \delta_c = \vartheta^{1/3}$

Therefore, $\delta_c = \left(\frac{\vartheta}{r}\right)^{1/3}$.

Thus, we have,

$$r^{4/3} \cdot dr = [Z] \cdot \vartheta^{1/3} \cdot d\vartheta$$

B.C. $\quad r = r_2$, $\quad \delta = \delta_2$, $\quad \vartheta = r_2 \delta_2^3 = V_2$ (say)

$r = r$, $\quad \delta_c = \delta_c$, $\quad \vartheta = r\,\delta_c^3 = V$ (say)

Integrating

$$-\int_{r_2}^{r} r^{4/3}\, dr = [Z] \int_{V_2}^{V} V^{1/3}\, dv.$$

(Incorporating negative sign in the integration as for a converging cone, local radius decreases from top to bottom of the cone)

$$\frac{3}{7}\left[r_2^{7/3} - r^{7/3}\right] = \frac{3}{4}\,[Z] \cdot \left[V^{4/3} - V_2^{4/3}\right].$$

$$= \frac{3}{4}\,[Z] \cdot \left[(r\,\delta_c^3)^{4/3} - (r_2\,\delta_2^3)^{4/3}\right].$$

or, $\quad \dfrac{4}{7\,[Z]} \cdot (r_2^{7/3} - r^{7/3}) + (r_2\,\delta_2^3)^{4/3} = (r.\,\delta_c^3)^{4/3} = r^{4/3} \cdot \delta_c^4$

$$\text{or,}\quad \delta_c = \left[\frac{4}{7\,[Z]} \cdot \frac{(r_2^{7/3} - r^{7/3})}{r^{4/3}} + \left(\frac{r_2\,\delta_2^3}{r} \right)^{4/3} \right]^{1/4}$$

Substituting for, $[Z]$

$$\delta_c = \left[\frac{24\mu k(T_v - T_w)}{7\lambda\rho^2 g \sin 2\theta} \cdot \frac{(r_2^{7/3} - r^{7/3})}{r^{4/3}} + \left(\frac{r_2\,\delta_2^3}{r} \right)^{4/3} \right]^{1/4}$$

IV.6. Solution Of Equation (3.4.6)

Rewriting equation (3.4.6), we have

$$Q_c = -2\pi k(T_v - T_w).\operatorname{cosec}\theta \int_{r_2}^{r} \left[\frac{4}{7\,[Z]} (r_2^{7/3} - r^{7/3}) + r_2^{4/3} \cdot \delta_2^4 \right]^{-1/4} .r^{4/3} . dr \qquad \ldots(3.4.6)$$

$$\text{Let,}\quad \left[\frac{4}{7\,[Z]} \left\{ r_2^{7/3} - r^{7/3} \right\} + r_2^{4/3}\,\delta_2^4 \right] = x$$

differentiating,

$$-\frac{4}{7\,[Z]} \left[\frac{7}{3}\, r^{4/3} \right] dr = dx$$

$$r^{4/3} . dr = -\frac{3}{4}\,[Z]\,dx$$

B.C. $r = r_2, \quad x = r_2^{4/3} . \delta_2^{4} = x_2$

$$r = r, \quad x = \left[\frac{4}{7\,[Z]}\left(r_2^{7/3} - r^{7/3}\right) + r_2^{4/3} . \delta_2^{4}\right] = x.$$

Therefore, we have from equation (3.4.6),

$$Q_c = -\,2\pi k\,(T_v - T_w).\mathrm{cosec}\,\theta \;.\; \int_{x_2}^{x} x^{-\frac{1}{4}}\left(-\tfrac{3}{4}[Z]\;dx\right) .$$

$$= +\,2\pi k\,(T_v - T_w).\mathrm{cosec}\,\theta \;.\; \frac{3}{4}\,[Z] \int_{x_2}^{x} x^{-\frac{1}{4}} . \,dx.$$

$$= 2\pi k\,(T_v - T_w).\mathrm{cosec}\,\theta \;.\; [Z] \;.\; \left[x^{3/4} - x_2^{3/4}\right] .$$

Substituting for $[Z]$,

$$Q_c = 2\pi k\,(T_v - T_w).\mathrm{cosec}\,\theta \;.\; \left[\frac{\lambda\rho^2 g \sin 2\theta}{6\mu K\,(T_v - T_w)}\right] .$$

$$\left[\left\{\frac{24\mu k \Delta T}{7\lambda\rho^2 g \sin 2\theta}\left(r_2^{7/3} - r^{7/3}\right) + r^{4/3} . \delta_2^{4}\right\}^{3/4} - \left(r_2^{4/3} . \delta_2^{4}\right)^{3/4}\right] .$$

$$Q_c\Big|_{r=r_1} = \frac{2\pi\lambda\rho^2 g\cos\theta}{3\mu}\left[\left\{\frac{24\mu k(T_v-T_w)(r_2^{7/3}-r_1^{7/3})}{7\lambda\rho^2 g\sin 2\theta} + r_2^{4/3}\cdot\delta_2^4\right\}^{3/4} - (r_2^{4/3}\cdot\delta_2^4)^{3/4}\right].$$

Now, $$\delta_2 = \left[\frac{24\mu k(T_v-T_w)}{7\rho^2 g\lambda\sin 2\theta}\right]^{\frac{1}{4}}\cdot\left[r_2\left\{1-\left(\frac{r_1}{r_2}\right)^{7/3}\right\}\right]^{\frac{1}{4}}$$

or, $$\delta_2^4 = \left[\frac{24\mu k(T_v-T_w)}{7\rho^2 g\lambda\sin 2\theta}\right]\cdot\left[r_2\left\{1-\left(\frac{r_1}{r_2}\right)^{7/3}\right\}\right]$$

$$= \left[\frac{24\mu k(T_v-T_w)}{7\rho^2 g\lambda\sin 2\theta}\right]\cdot\left[\frac{r_2^{7/3}-r_1^{7/3}}{r_2^{4/3}}\right]$$

or, $$r_2^{4/3}\cdot\delta_2^4 = \left[\frac{24\mu k(T_v-T_w)(r_2^{7/3}-r_1^{7/3})}{7\rho^2 g\lambda\cdot\sin 2\theta}\right]$$

Therefore,

$$Q_c = \frac{2\pi\rho^2 g\lambda\cos\theta}{3\mu}\cdot\left[(r_2^{4/3}\cdot\delta_2^4 + r_2^{4/3}\cdot\delta_2^4)^{3/4} - (r_2^{4/3}\cdot\delta_2^4)^{\frac{3}{4}}\right]$$

$$= \frac{2\pi\rho^2 g\lambda\cos\theta}{3\mu}\cdot\left[(2r_2^{4/3}\cdot\delta_2^4)^{\frac{3}{4}} - (r_2^{4/3}\cdot\delta_2^4)^{\frac{3}{4}}\right]$$

$$= \frac{2\pi\,\rho^{2}\,g\,\lambda\cos\theta}{3\mu}\cdot\left[(2)^{\frac{3}{4}}-1\right]\cdot\left(r_2^{4/3}\cdot\delta_2^{4}\right)^{\frac{3}{4}}$$

$$= \frac{2\pi\,\rho^{2}\,g\,\lambda\cos\theta}{3\mu}\cdot 0.681\left(r_2^{4/3}\cdot\delta_2^{4}\right)^{\frac{3}{4}}$$

Therefore,

$$Q_c\Big|_{r=r_1} = 3.5937.\cos\theta.\left[\frac{\rho^{2}\,k^{3}\,\lambda\,g\,(T_v-T_w)^{3}.(r_2^{7/3}-r_1^{7/3})^{3}}{\mu\,(\sin 2\theta)^{3}}\right]^{\frac{1}{4}} \qquad \ldots(3.4.7)$$

IV.7. Solution Of Equation (3.5.5).

Rewriting equation (3.5.5), we have,

$$\bar{h}_{id} = 0.84.\left[\frac{\rho_f^{2}\,g\,\lambda\,k_f^{3}\,\sin 2\theta}{\mu_f\,r_2}\right]^{\frac{1}{4}}\cdot\left[\frac{(1-a^{7/3})^{\frac{3}{4}}}{1-a^{2}}\right].$$

$$\left[\frac{\bar{h}_{id}}{Re_f}\cdot\left(\frac{4H}{\mu_f\,\lambda}\right)\right]^{\frac{1}{4}} \qquad \ldots(3.5.5)$$

$$\text{or,}\quad \bar{h}_{id}^{-3/4} = 0.84.\left[\frac{\rho_f^{2}\,g\,\lambda\,k_f^{3}\,\sin 2\theta}{\mu_f\,r_2}\right]^{\frac{1}{4}}\cdot\left[\frac{(1-a^{7/3})^{\frac{3}{4}}}{(1-a^{2})}\right]\cdot\left[\frac{4H}{Re_f\,\mu_f\,\lambda}\right]^{\frac{1}{4}}$$

$$\bar{h}_{id} = (0.84)^{4/3} \cdot \left[\frac{\rho_f^2\, g\, \lambda\, k_f^3 \sin 2\theta\; 4H}{\mu_f^2\, r_2 \cdot \lambda}\right]^{1/3} \cdot \left[\frac{(1-a^{7/3})}{(1-a^2)^{4/3}}\right] \cdot Re_f^{-1/3}$$

$$= (0.84)^{4/3} \cdot (8)^{1/3} \cdot \left[\frac{\rho_f^2\, g\, k_f^3 \sin\theta \cos\theta\, (r_2 - r_1)}{\mu_f^2\, r_2 \cdot \tan\theta}\right]^{1/3} \cdot$$

$$\left[\frac{(1-a^{7/3})}{(1-a^2)^{4/3}}\right] \cdot Re_f^{-1/3}$$

$$= 1.585. \left[\frac{\rho_f^2\, g\, k_f^3}{\mu_f^2}\right]^{1/3} \cdot \left[\frac{\cos\theta^{2/3} \cdot (1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}}\right] \cdot Re_f^{-1/3}$$

$$\bar{h}_{id} \cdot \left[\frac{\mu_f^2}{\rho_f^2 g k_f^3}\right]^{1/3} = 1.585. \left[\frac{(1-a^{7/3})(1-a)^{1/3}}{(1-a^2)^{4/3}} \cdot \cos\theta^{2/3}\right] \cdot Re_f^{-1/3} \qquad \ldots(3.5.6)$$

$$= 1.585.\; f_c \cdot Re_f^{-1/3} \qquad \ldots(3.5.7)$$

APPENDIX - V.

DIMENSIONAL ANALYSIS

The Condensation heat transfer coefficient in a diverging-converging system has been found to be influenced by a number of factors.

$$\bar{h} = f(\bar{u},\ \rho_f,\ k_f,\ C_p,\ \mu_f,\ \lambda,\ \Delta T_f,\ g\cos\theta,\ D_e,\ \delta,\ L\sin\theta)$$

Dimensions of these factors are given below.

(1)	Average Fluid velocity	... $\bar{u}$ ...	$\frac{L}{\theta}$
(2)	Fluid Density	... ρ_f ...	$\frac{M}{L^3}$
(3)	Thermal Conductivity	... k_f ...	$\frac{Q}{T L \theta}$
(4)	Specific Heat	... C_p ...	$\frac{Q}{M T}$
(5)	Fluid Viscosity	... μ_f ...	$\frac{M}{L\theta}$
(6)	Latent Heat	... λ ...	$\frac{Q}{M}$
(7)	Average Temperature Difference Between Film and Wall	... ΔT_f ..	T
(8)	Acceleration Due to Gravity	... $g\cos\theta$ ..	$\frac{L}{\theta^2}$
(9)	Equivalent Radius	... De ...	L
(10)	Film Thickness	... δ ...	L
(11)	Linear Dimension of Heat Transfer Surface	... $L\sin\theta$ ..	L
Also, (12)	Average Heat Transfer Coefficient	... $\bar{h}$ ...	$\frac{Q}{\theta L^2 T}$

Applying π - Theorem

$$\phi\,(\pi_1, \pi_2, \pi_3 \;\ldots\ldots.) = 0$$

$$\pi = \phi'\,((\bar{h})^a, (\bar{u})^b, \rho_f^c, k_f^d, C_p^e, \mu_f^f, \lambda^g, \Delta T_f^h, g\cos\theta^i, D_e^j,$$
$$\delta^k, L\sin\theta^l) = 1$$

$$\pi = \alpha\left(\frac{Q}{\theta L^2 T}\right)^a \left(\frac{L}{\theta}\right)^b \left(\frac{M}{L^3}\right)^c \left(\frac{Q}{TL\theta}\right)^d \left(\frac{Q}{MT}\right)^e \left(\frac{M}{L\theta}\right)^f \left(\frac{Q}{M}\right)^g$$
$$(T)^h \left(\frac{L}{\theta^2}\right)^i (L)^j (L)^k (L)^l$$

Summing exponents, we have,

$\sum Q$: $a + d + e + g = 0$

$\sum M$: $c - e + f - g = 0$

$\sum L$: $-2a + b - 3c - d - f + i + j + k + L = 0$

$\sum \theta$: $-a - b - d - f - 2i = 0$

$\sum T$: $-a - d - e + h = 0$

π_1, π_2, π_3 ...... may be evaluated by simple algebra,

So by solving, we have,

$a = -i + j + k + L + f - e - g.$

$b = e - f + g - 2i.$

$c = e - f + g.$

$d = i - j - k - l - f$

$h = -g.$

Thus,

$$\phi'\Big[(\bar{h})^{-i+j+k+l+f-e-g}\,(\bar{u})^{e-f+g-2i}\,(\rho_f)^{e-f+g}\,(k_f)^{i-j-k-l-f}\,.\,(C_p)^{e}\,(\mu_f)^{f}\,(\lambda)^{g}\,(\Delta T_f)^{-g}\,(g\cos\theta)^{i}\,(De)^{j}\,(\delta)^{k}\,(L\sin\theta)^{l}\Big]=1.$$

$$\text{or, }\phi'\Big[\{(\bar{h})^{-i}\,(\bar{u})^{-2i}\,(k_f)^{i}\,(g\cos\theta)^{i}\}.\{(\bar{h})^{j}\,(k_f)^{-j}\,(De)^{j}\}\,\{(\bar{h})^{k}\,(k_f)^{-k}\,(\delta)^{k}\}\,\{(\bar{h})^{l}\,(k_f)^{-l}\,(L\sin\theta)^{l}\}\,\{(\bar{h})^{f}\,(\bar{u})^{-f}\,(\rho_f)^{-f}\,(k_f)^{-f}\,(\mu_f)^{f}\}\,\{(\bar{h})^{-e}\,(\bar{u})^{e}(\rho_f)^{e}(C_p)^{e}\}\,\{(\bar{h})^{-g}(\bar{u})^{g}\,(\rho_f)^{g}\,(\lambda)^{g}\,(\Delta T_f)^{-g}\}\Big]=1.$$

$$\text{or, }\phi'\Big[\Big(\frac{k_f\,g\cos\theta}{\bar{h}\,\bar{u}^{2}}\Big)^{i}\Big(\frac{\bar{h}\,De}{k_f}\Big)^{j}\Big(\frac{\bar{h}\delta}{k_f}\Big)^{k}\Big(\frac{\bar{h}\,L\sin\theta}{k_f}\Big)^{l}\Big(\frac{\bar{h}\,\mu_f}{\bar{u}\,\rho_f\,k_f}\Big)^{f}\Big(\frac{\bar{u}\,\rho_f\,C_p}{\bar{h}}\Big)^{e}\Big(\frac{\bar{u}\,\rho_f\,\lambda}{\bar{h}\,\Delta T_f}\Big)^{g}\Big]=1$$

$$\text{or, }\phi'\Big[\Big(\frac{\delta g\cos\theta}{\bar{u}^{2}}\Big)^{i}\Big(\frac{\bar{h}\,De}{k_f}\Big)^{j}\Big(\frac{\bar{h}\,De}{k_f}\Big)^{k}\Big(\frac{\delta}{De}\Big)^{k}\Big(\frac{\bar{h}\,De}{k_f}\Big)^{l}\Big(\frac{L\sin\theta}{De}\Big)^{l}\Big(\frac{\mu_f}{\delta\,\bar{u}\,\rho_f}\Big)^{f}\Big(\frac{\delta\,\bar{u}\,\rho_f}{\mu_f}\Big)^{e}\Big(\frac{C_p\,\mu_f}{k_f}\Big)^{e}\Big(\frac{1}{3}\Big)^{g}\Big(\frac{L^{3}\rho_f^{\,2}\,\lambda\,g\cos\theta}{\mu_f\,k_f\,\Delta T_f}\Big)^{g}\Big(\frac{\delta}{L}\Big)^{g}\Big]=1\qquad\Big[\because\ \bar{u}=\frac{1}{3}\,\frac{\rho_f\,g\cos\theta\,\delta^{2}}{\mu_f}\Big]$$

Therefore,

$$\left(\frac{\bar{h}.De}{k_f}\right) = A\left(\frac{L^3 \rho_f^2 \lambda g \cos\theta}{\mu_f k_f \Delta T_f}\right)^B \left(\frac{g \delta \cos\theta}{\bar{u}^2}\right)^C \left(\frac{\delta \bar{u} \rho_f}{\mu_f}\right)^D \left(\frac{C_p \mu}{k_f}\right)^E$$

$$\left(\frac{\delta}{De}\right)^F \left(\frac{L \sin\theta}{De}\right)^G \left(\frac{\delta}{L}\right)^H \quad \text{........(V-1)}$$

Where, A, B,C....H, are new constants.

Expression (V.1) can be reduced to the following form,

$$\left(\frac{\bar{h}\ De}{k_f}\right) = f\left(\frac{L^3 \rho_f^2 \lambda g \cos\theta}{\mu_f k_f \Delta T_f}\right)^n ,$$

in which, other dimensionless groups have been neglected, since this reduced form is capable of defining the system well.

Here, $\left(\frac{\bar{h}\ De}{k_f}\right) = \overline{Nu}$, Mean Nusselt Number,

and, $\left(\frac{L^3 \rho_f^2 \lambda g \cos\theta}{\mu_f k_f \Delta T_f}\right) = C_v$, Condensation Number.

'f' and 'n' are proportionality constant and exponents respectively, which are evaluated from experimental results.

APPENDIX -VI .

SAMPLE CALCULATIONS

1) Sample Calculation For Heat Balance (Experimental):

Let us consider condensation of water vapour in a diverging cone section. Consider the data from Table AI-1 and Serial No.1.

Condensate Rate	= 3.045 kg/hr.
Latent heat of vaporization of Water	= 538.88 Kcal/kg.
Heat Released by Condensation	$= m\lambda$
	= 3.045 x 538.88
	= 1641 Kcal/hr
Temperature difference of the Coolant, ΔT	= 16.4°C
Coolant Flow Rate = 100 lit/hr	= 99.548 kg/hr at 30°C.
Heat Received by Coolant	= 99.568 x 16.4
	= 1633 Kcal/hr
Taking Sp. heat of Water, C_p	= 1 Kcal/kg.°C.

2) Sample Calculation For Heat Transfer Area, (Inside And Outside)

For the diverging Cone of $\theta = 5°$

From Table AII.1, we have,

$r_1 = 0.01$ m

$r_2 = 0.02250$ m

$H = 0.144$ m

Area of Truncated Cone is given by,

$$A = \pi(r_1+r_2).\sqrt{H^2 + (r_2-r_1)^2}$$

$$= 0.014860 \text{ m}^2$$

Inside Area of the diverging cone section,

$A_i = 0.014860 \text{ m}^2$

And, Outside Area of the diverging Cone Section,

$A_o = 0.016231 \text{ m}^2$

where, the thickness of the plate, t = 0.0015 m.

3) Sample Calculation for ΔT_f.

$$\Delta T_f = T_f - T_w \quad \text{where,} \quad T_f = \frac{T_v+T_w}{2}$$

here, $T_v = 100°C$, $T_w = 80°C$, $T_f = \frac{100 + 80}{2} = 90°C$

$$\Delta T_f = (90-80)°C = 10°C.$$

4) Sample Calculation for $\bar{q}$,

$$\bar{q} = \frac{Q}{A_i}$$

$$= \frac{1641}{0.01486}$$

$$= 11.04 \times 10^4 \quad Kcal/hr.m^2$$

5) Sample Calculation For $\bar{h}_i$,

i) Experimental,

Heat released by vapour $= 1641$ Kcal/kg

$$\Delta T_f = 10°C$$

$$A_i = 0.01486 \ m^2$$

$$\bar{h}_i = Q/(A.\Delta T_f) = 1641/(0.01486 \times 10)$$

$$= 11043 \ Kcal/hr.m^2.°C$$

ii) Theoretical,

Heat transfer coefficient for diverging cone section, is given by, Rewriting equation (3.2.22) ,

$$\bar{h}_{id} = 0.84 \cdot \left[\frac{\rho_f^2 \, g \, \lambda \, k_f^3 \, \mathrm{Sin}2\theta}{\mu_f \ r_2 \ \Delta T_f}\right]^{\frac{1}{4}} \cdot \left[\left(\frac{1}{1-(r_1/r_2)^2}\right)\right] \cdot \left[1-\left(\frac{r_1}{r_2}\right)^{7/3}\right]^{\frac{3}{4}}$$

Taking the physical and transport properties for water from Table AIII-1 of Appendix - III and r_1 and r_2 values from Table AII.1, of Appendix - II,

$$\bar{h}_i = 0.2593476 \ \ \mathrm{Cal/cm^2.\ S.\ ^\circ C}$$

$$= 0.2593476 \times 36000 \ \mathrm{Kcal/hr.m^2.^\circ C}$$

$$\simeq 9336 \ \mathrm{Kcal/hr.m^2.^\circ C.}$$

6) Sample Calculation For $\bar{h}_i . \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3}$ (Experimental).

here,

$$\bar{h}_i = 11043 \ \mathrm{Kcal/hr.m^2.^\circ C.}$$

$$= \frac{11043}{36000} \ \mathrm{Cal/S.Cm^2.^\circ C}$$

$$\mu_f = 0.0027 \ \mathrm{gm/Cm.S.}$$

$$\rho_f = 0.95838 \ \mathrm{gm/cm^3.}$$

$$g = 981 \ \mathrm{cm/S^2}$$

$$k_f = 1.643 \times 10^{-3} \ \mathrm{Cal/S.Cm.^\circ C}$$

$$\bar{h}_i . \left(\frac{\mu_f^2}{\rho_f^2 \, g \, k_f^3}\right)^{1/3} = \frac{11043}{36000} . \left[\frac{(0.0027)^2}{(0.95838)^2 \times 981 \times (1.643 \times 10^{-3})^3}\right]^{1/3} .$$

$$\simeq \ 0.375 \ \text{(Dimensionless).}$$

7) Sample Calculation For Re_f (Experimental):

$$Re_f = \frac{4.G'}{\mu_f} = \frac{4\ W'}{\mu_f.\pi D_e}.$$

$$W' = 3.045\ Kg/hr.$$

$$\mu_f = 0.0027\ gm/cm.S.$$

$$= 0.0027x360\ Kg/m.hr.$$

$$D_e = \frac{r_1 + r_2}{Cos\ \theta} = \frac{0.01+0.02250}{Cos 5} = 0.0326241\ m.$$

$$Re_f = \frac{4x3.045}{(0.0027x360)x\pi\ x\ 0.0326241} \simeq 122.$$

8) Sample Calculation for Re'_f. (Experimental):

$$Re'_f = fc/\ Re_f^{1/3}$$

$Re_f = 122$ as already calculated in item No.(7).

$$fc = \left[\frac{(1-a)^{1/3}.\ (1-a^{7/3})}{(1 - a^2)^{4/3}} .\ Cos\ \theta^{2/3}\right] = 0.9338405$$

$$a = \frac{r_1}{r_2}$$

$$\therefore Re'_f = 0.93384050/(122)^{1/3} \simeq 0.188.$$

9) Calculation For $\bar{N}u$, (Experimental)

$$\bar{N}u = \frac{\bar{h}_i . De}{k_f} .$$

here,

$$\bar{h}_i = 11043 \text{ Kcal/hr.m}^2.°C$$

$$k_f = 1.6430\times10^{-3} \text{Cals/S.cm.°C} = 0.59148 \text{ Kcal/hr.m.°C.}$$

$$D_e = \frac{r_1 + r_2}{\text{Cos } \theta} = 0.0326241 \text{ m.}$$

$$\bar{N}u = \frac{11043 \times 0.0326241}{0.59148} \approx 609.$$

10) Calculation For C_v, (Experimental)

$$C_v = \frac{l^3 \rho_f^2 \lambda g \text{ Cos } \theta}{\mu_f k_f \Delta T_f} , \quad \text{where,} \quad l = \frac{r_2 - r_1}{\text{Sin } \theta} .$$

here,

$$l = 14.34 \text{ cm.}$$

$$\rho_f = 0.95838 \text{ gms/cm}^3$$

$$\lambda = 538.88 \text{ Cal/gm.}$$

$$g = 981 \text{ Cm/S}^2$$

$$\mu_f = 0.0027 \text{ gms/Cm.S.}$$

$$k_f = 1.643 \times 10^{-3} \text{ Cal/Cm.S.°C.}$$

$$\Delta T_f = 10°C$$

$$C_v = \frac{(14.34)^3 \times (0.95838)^2 \times 538.88 \times 981 \times \mathrm{Cos}\ 5}{0.0027 \times 1.643 \times 10^{-3} \times 10}$$

$$\simeq 3.22 \times 10^{13}.$$

For the calculations of parameters in case of converging, diverging-converging cone sections, diverging-converging tube and straight cylindrical tubes, same procedures were adopted as discussed in case of diverging cone system. So sample calculations for these systems (other than diverging system) have been omitted.

NOMENCLATURE

A = Heat Transfer Area, (m^2).

C_p = Specific heat, (Kcal/kg.°C)

D_e = $\left(\frac{r_1+r_2}{\text{Cos } \theta}\right)$, Equivalent diameter; (m)

g = Acceleration due to Gravity; (m/hr^2)

G' = Condensate loading per linear circumferencial length, (kg/hr.m).

$h, \bar{h}$ = Local and average heat transfer coefficient, $(Kcal/hr.m^2.°C)$.

H = Vertical height of diverging or converging segment, (m)

k = Thermal Conductivity (Kcal/hr.m.°C)

l = Slant height, (m)

Q = Heat flow, (Kcal/hr).

r = Local cone radius, (m)

r_e = Equivalent radius, (m)

T = Temperature, (°C)

T_f = $\frac{1}{2}(T_v + T_w)$, Mean film temperature, (°C)

ΔT_f = $(T_f - T_w)$, Temperature difference between film and wall, (°C).

t = Thickness, (m)

$u, \bar{u}$ = Local and average condensate velocity, (m/hr).

W = Condensate flow per unit surface area, $(Kg/hr.m^2)$

W' = Rate of condensate per tube, (Kg/hr).

μ = Viscosity, (Kg./m.hr).

λ = Latent heat of condensation or vaporisation, (Kcal/kg.).

ρ = Density, (Kg/m^3).

δ = Film thickness, (m).

θ = Half-apex angle of cone, (degree).

Subscripts:

c = Converging

d = Diverging

dc = Diverging-Converging

f = Film

i = Inside

o = Outside

s = Shell

Sat. = Saturation

v = Vapour

w = Wall or condensing surface

1 = Smaller side

2 = Bigger side

Dimensionless Groups:

$Re_f = [\frac{4G'}{\mu_f}]$, Condensate Reynolds Number

$\bar{N}u = [\frac{\bar{h}_i \cdot (r_1 + r_2)}{k_f \cos\theta}]$, Mean Nusselt Number

$C_v = [\frac{l^3 \rho_f^2 \lambda \cdot g \cdot \cos\theta}{\mu_f k_f \cdot \Delta T_f}]$, Condensation Number

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