BOILING OF AQUEOUS SUGAR SOLUTIONS IN A SINGLE TUBE NATURAL CIRCULATION EVAPORATOR

A THESIS
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By

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CERTIFICATE

Certified that the thesis entitled "Boiling of Aqueous Sugar Solutions in a Single Tube Natural Circulation Evaporator" Which is being submitted by Shri ASHOK KUMAR DODEJA in partial fulfilment of the requirements for the award of the degree of MASTER OF ENGINEERING IN CHEMICAL ENGINEERING (Plant and Equipment Design) at the Univeristy of Roorkee, is a record of candidate's own work carried out by him under the supervision and guidance of the undersigned. The matter embodied in this thesis has not been submitted for the award of any other degree.

This is further certified that the candidate has worked for a period of about eight months from January 1977 for preparing this thesis at Univeristy of Roorkee, Roorkee.

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ABSTRACT

The present work embodies an experimental investigation of heat transfer in single vertical tube closed loop natural circulation evaporator for water and sugar solution systems at atmospheric pressure. The vertical tube (test-section) was a copper tube of 940 mm length and 19.02mm inside diameter. The heating of the tube was done by electrical heating element of nicrome wire (22 gauge) wound externally to provide constant uniform flux. The wall and liquid temperatures were measured by means of copper-constantan thermocouples fixed at an interval of 100mm length of test section. The heat flux values ranged from 7.108×10^3 to 24.878×10^3 W/m^2. Five concentrations 9.90, 19.45, 27.40, 43.60 and 57.50 wt% of sugar solutions were used in the study.

The experimental results showed the expected trends qualitatively and quantitatively.

The local values of heat transfer coefficient were correlated by following equations.

Natural convection zone of water:-

$$N_u = 8.10 \times 10^5 \left[G_r \, P_r \right]^{-0.624}$$

with a maximum deviation of ± 20%.

Boiling zone of water:-

$$\frac{h_B}{h_C} = 7.5 \times 10^{-5} \left[\frac{\rho_L}{\rho_V} \, \frac{C_P}{\lambda} \, \frac{d}{z_s} \, (\Delta t)_{sub} \right]^{-0.091} \left[\frac{t_L}{t_W} \, \frac{z}{d} \right]^{(2.79)}$$

with a maximum deviation of ± 30%.

Natural convection zone of 19.45 and 27.40 wt% sugar:-

$$N_u = 0.913 (G_r \, P_r) \quad (P_r)^{-0.26}$$

with a maximum deviation of ± 20%.

Natural convection zone of 43.60 and 57.50 wt% sugar:-

$$N_u = 9.6334 \left(G_r \, P_r \right)^{-.0033} \left(P_r \right)^{0.17}$$

with a maximum deviation of ± 20%.

Boiling zone of sugar solutions :-

$$\frac{h_B}{h_C} = 2.25\times10^{-4}\left[\frac{\rho_L\, C_P\, d\, (\Delta t)_{sub}}{\rho_V\, z_s}\right]^{-1.386}\left[\frac{t_L}{t_W}\,\frac{z}{d}\right]^{2.93}$$

with a maximum deviation of ± 30%.

The physical properties used in the correlations of boiling heat transfer were taken at the saturation temperatures of the test liquids while for natural convection region they were taken at arithematic of wall and liquid temperatures corresponding to the location under consideration.

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NOMENCLATURE

A	Surface area	m^2
C	Specific heat	$\mathrm{kcal}/_{\mathrm{kg}}\,^{\circ}\mathrm{C}$
d	Inside diameter of tube	m
g	Acceleration due to gravity	$\mathrm{m}/_{\mathrm{hr}^2}$
G	Mass velocity	$\mathrm{kg}/_{\mathrm{m}^2\text{-}\mathrm{hr}}$
h	Heat transfer coefficient	$\mathrm{W}/_{\mathrm{m}^2\text{-}\mathrm{hr}}$
k	Thermal conductivety	$\mathrm{kcal}/_{\mathrm{m}\text{-}\mathrm{hr}\text{-}^{\circ}\mathrm{C}}$
q	Heat flux	$\mathrm{W}/_{\mathrm{m}^2}$
Q	Heat input	W
t	Temperature	°C
Δt	Temperature difference $(t_w - t_l)$	°C
ΔT	Temperature drop across the liquid flim	°C
$(\Delta t)_{sub}$	Degree of subcooling at inlet	°C
V	Velocity	m/hr
X	Two -phase mixture quality	
Z	Distance along the test section from the bottom	m
z_s	Distance along the test section from the bottom to a point corresponding to saturation boiling.	m

GREEK LETTERS:-

β	Coefficient of thermal expension	$1/_{^{\circ}\mathrm{C}}$
λ	Latent heat of vaporization	$\mathrm{kcal}/_{\mathrm{kg}}$
μ	Viscosity	$\mathrm{kg}/_{\mathrm{m}\text{-}\mathrm{hr}}$
ρ	Density	$\mathrm{Kg}/_{\mathrm{m}^3}$
σ	Surface tension	$\mathrm{m}/_{\mathrm{hr}}$
ν	Kinematic viscosity	$\mathrm{m}^2/_{\mathrm{hr}}$

SUBSCRIPTS:-

av	Average	out	outlet
B	Boiling	pred	Predicted

c	Convective	s	Saturated
exptl	Experimental	T	Total
in	Input	TP	Two-phase
L	Liquid	V	Vapor
mix	Mixture	W	Wall
z	Distance		

DIMENSIONLESS GROUPS

G_r	Grashof number	=	$\frac{gd^3 \beta t}{\nu^2}$
N_u	Nusselt number	=	hd/k
P_r	Prandtl number	=	$C\mu/k$
R_e	Reynolds number	=	$dV\rho/\mu$
X_{tt}	Lockhart-Martinellii Parameter	=	$\left(\frac{1-x}{x}\right)^{0.9} \left(\rho_V/\rho_L\right)^{0.5} \left(\mu_L/\mu_V\right)$

CHAPTER 1

INTRODUCTION

Natural circulation evaporators are commonly used in chemical processing industries to concentrate solutions of non-volatile solute and volatile solvent.

Natural circulation closed loop evaporator units essentially is a U- tube with movement due to difference in density in hot and cold leg of circulating system. The hot leg in these units is a one pass heat exhang with process fluid to be vaporized in vertical tubes. The prediction of flow and heat transfer rates are two main requirements for the design of these equipments. Hydrodynamics and heat transfer phenomenon in natural circulation boiling of liquids is complicated owing to the presence to two phases.

A number of experimental studies have been carried out during last few years. It has been observed that most of studies has been done on pure liquids, a few studies is carried out on salt solutions. Therefore, it is important to obtain experimental data on natural convective boiling of industrial important salt solution. However it is felt that expertimental data on aqueous sugar solutions will prove useful to add new important results for sugar industries.

In view of the foregoing the study was taken with the following objectives :-

1. To collect expertmental data on natural convective boiling of water and sugar solutions at various concentrations at low & moderate heat fluxes and one atmospheric pressure.
2. To develope generalised correlations for heat transfer during natural, subcooled & saturated boiling of water and sugar solutions.

CHAPTER 2

REVIEW OF LITERATURE

The major published work on the heat transfer in natural circulation vertical tube evaporator is on pure liquids and their mixtures. Quite a few informations is reported about the solutions. Most of the studies conducted are experimental in nature and have resulted in a number of emperical and semi-emperical correclations, some work had also been done to give theoretical approach to justify the developed correlations. A brief review is presented below :-

Piret and Isbin (1) investigated six systems comprising of water, carbon tetrachloride, Isopropyl and n-butyl alchols, 35 wt% and 50 wt% $K_2 CO_3$ aqueous solutions at atmospheric pressure in a 1-inch nominal copper tube with 46.5-inch heated length. They used a modified Dittus Boelter equation to correlate the inside heat transfer coefficient. The equation is -

$$\frac{h_{av} d}{k_L} = 0.0086 \left[\frac{d v_m \rho_L}{\mu_L}\right]^{0.8} \left[\frac{C_L \mu_L}{k_L}\right]^{0.6} \left[\frac{\sigma_{H_2O}}{\sigma_L}\right]^{0.33} \qquad (2.1)$$

Where V_m is log mean liquid vapor assuming a homogeneous flow and is given as :-

$$(V_{mix\ out} - V_{L\ in}) / l_n (V_{mix\ out} - V_{L\ in}) \qquad (2.2)$$

Guerrieri and Talty(2) studied heat transfer to boiling of cyclohexane, methyl alchohal, benzene, pentene, and heptane in two differen single brass tube natural circulation vertical boiler. The tubes were light oil heated and were of dimensions 0.75-inch by 6 ft long and 1 inch inside diameter by 6.5 ft long respectively. Wall and boiling luquid temperature were measured at 6 inch intervals along the tubes. Local heat transfer coefficient and the amount of vapor generated in each 6 inch section were calculated. Heat flux ranged from 2170 to 17400 Btu/hr-ft^2 vapor quality of tube outlet from 2.8.to 11.6 wt%, boiling film temperature drop ranged from 6.1 °F to 24.2 °F. The heat tansfer in the system occurred simultaneously by convection and by nucleate boiling. The authors presented an analysis which accounts for both processes and based on this analysis, correlated their heat transfer results in the form of local two phase convective heat transfer coefficient to the liquid phase coefficient as a function of the rec iprocal Lockhart-Martinelli parameter.

$$\frac{h_{TP}}{h_L} = 3.4 \left(1/X_{tt}\right)^{0.45} \qquad (2.3)$$

Where h_L is the coefficient calculated from Dittus-Boelter equation for liquid flowing alone.

$$h_L = .023\left(\frac{k_L}{d}\right)\left(\frac{d\, G_T(1-X)}{\mu_L}\right)^{0.8}\left(\frac{c_L \mu_L}{k_L}\right)^{0.4} \qquad (2.4)$$

where $G_T(1-X)$ expresses the point mass velocity of the unvaporized part of the stream.

Dengler and Addoms(3) investigated the mechanism of boiling and heat transfer during vaporisation of water in a vertical copper tube of 1- inch inside diameter by 20 ft long. The principal conclusions drawn are as fallows :-

1. The mechanism of heat transfer during vaporisation in tubes is primarily convective. Nucleate boiling is dominant only under conditions of low liquid velocity and is gradually suppressed by the effect of vapor induced forced flow.

2. Operating variables exert independent and often opposing effects on each of these mechanisms :-

(i) Increase in pressure may increase the heat transfer coefficient by its effect on nucleate boiling or decrease them in the range of two phase convection by rising the average fluid density and thereby lowering the velocity.

(ii) Increase in temperature difference promotes nucleate boiling but has no direct effect on the convective coefficients.

(iii) Increase in total mass throughput increases the convective heat transfer but decreases the nucleate boiling heat transfer by lowering the available effective temperature driving force for nucleation.

They arrieved at a correlation similar to that of Guerrieri and Talty

$$\frac{h_{TP}}{h_L} = 3.5\left(\frac{1}{X_{tt}}\right)^{0.5} \qquad (2.5)$$

where h_L is the coefficient calculated from Dittus-Boelter equation for the total flow.

$$h_L = .023\left(\frac{k_L}{d}\right)\left(\frac{4G_T}{\pi D \mu_L}\right)^{0.8}\left(\frac{c_L \mu_L}{k_L}\right)^{0.4} \qquad (2.6)$$

Johnson (4) measured circulation rates and overall temperature driving forces for a 15-inch shell reboiler containg 96.1-inch; 12 gauge, 8 ft long tubes. One tube was equipped with a temperature probe to obtain local boiling stream temperatures. The systems investigated were water and a hydrocarbon having a normal boiling point of 80.8 °C. Circulation rates were estimated by Kern's method. The Lockhart Martinelli parameter is used to calculate friction and expansion losses for the two phase zone. Overall coefficients, driving forces, fluxes, flow, and vaporisation rates are tabulated. Typical temperature profiles were studied for six runs on hydrocarbon.

Lee et al (5) used a reboiler consisting of seven tubes in a bundle. The tubes were of 1-inch outside diameter 14 gauge, 10 ft long admiralty metal. They investigated seven liquids and presented data for a pressure range of 2 to 120 psia. Overall coefficient were presented as functions of overall temperature differences. The average inside film coefficient and the maximum flux were presented in terms of dimensionless groups. The maximum flux was found for each fluid and corresponding system pressure. For heat fluxes above the maximum flux, vapor lock occurred. Recommendations include a maximum overall coefficent of 500 Btu/hr and the need for giving particular attention to reboiler entrance and exit piping.

Ladiev (6) carried out experiments in three tubes 1500 mm long and 24,28 and 52mm diameter. Special attention was given to maintain indentical conditions in all tubes. Experimental results on heat transfer aspect were correlated graphically for water at atmospheric pressure. The apparent liquid level in the tube had a significant effect on the rate of heat transfer. It was found to decreased with an increased in the level. The sharp reduction in the rate of heat transfer at boiling in vertical tubes with inherent circulation was found to be associated with inadequate flow in upper part of the tube. Increase in tube diameter has no significant effect on heat transfer rate, but circulation velocity certainly got decreased. Under vacuum, the general characterstics regarding heat transfer was retained but its absolute level was lower than that at atmospheric pressure.

Claire (7) obtained fundamental information regarding heat transfer in a pilot evaporation plant. The coefficient of heat transfer for brass and stainless steel tubes of different dimensions 2-inch outside diameter, 6 ft length and 1.5 inch outside diameter, 4 ft length were respectively compared. Tests with water boiling atmospheric indicated that the heat transfer was at maximum when the level of water was maintained at about 1/3 of the tube height. For clean tubes the value of U is always higher for brass tubes than for stainless steel tubes.

Bergles et al (8) carried out experiments on water at low pressure (below 100 psia) to investigate the effects of tubes lengths, inlet temperature, tube diameter and pressure of the critical heat flux. The results were related to instabilities of the slug flow region. Critical heat fluxes for water are normally considered to start around 0.4×10^{6} Btu/hr-ft^{2}, however the authors have shown values of half this amount under low pressure conditions.

Beaver and Hughmark (9) conducted experiments on twelve fluids in a ¾ inch by 8 ft long carbon steel tube, heated electrically, to investigate the reliability of using developed correlations in vacuum conditions. It was concluded that for wall minus saturation temperature differences less than 15°F, single phase coefficients dominate and can be predicted by a modified Dittus Boelter equation (Sieder Tate modification)

$$N_u = .023 (R_e)^{0.8} (P_r)^{0.4} (\mu_L / \mu_w)^{0.14} \qquad (2.7)$$

For temperature differences greater than 15 F, nucleation sets in and local inside coefficients can be predicted by existing two phase correlations.

Shellene et al (10) carried out plant tests on a highly instrumented 110 ft^{2} reboiler having a 14-inch shell, containing 70, ¾ inch 13 gauge, 3, 7/8 inch, 12 gauge, 8 ft long tubes. The reboiler was operated in conjunction with an existing distillation column and, except for instrumentation, was identical to a typical commercial unit. Steam was used as the heat source and process fluids were benzene, water, isopropyl alchol, methyl ethyl ketone, glycerine and various aqueous solutions of the alcohol, ketone, and glycerine. The process pressure was slightly above atmospheric. Of particular interest, in the work, was the expoloration of the onset of unstable operation. It was found that the addition of flow resistance to the inlet line extended the stable operating range, and the allowable pressure drop across the tubes decreases as the heat flux increases.

Adding flow resistance to the vapor return line results in a decreased in the maximum flux. The authors recommend to keep the return flow area at-least equal to the total flow area to the tubes. Maximum heat fluxes were tabulated for the various fluids with accompanying temperature differences and percent vaporisation. Other data were presented as heat flux versus log-mean temperature difference and mass velocity versus pressure drop.

Takeda et al (11) measured the axil distribution of vapor hold-up and boiling heat transfer coefficient in the annuli of a natural circulation vertical tube evaporator for pure liquids (water, methanol). Vapor hold up in the annuli was expressed in terms of heat transfer rate and length of boiling region. The effect of vapor holup on the boiling heat transfer coefficient was analysed to give an empirical diamensionless equation.

Calus et al (12) carried out experimental work on single tube natural circulation reboiler to investigate heat transfer to single component liquids. One copper and one stainless steel tube each equipped with a six compartments steam jacket were used. All the tubes had nominal dimensions of 0.5-inch diameter and 48 inch height. Three workers worked independently to obtain the length mean and local point coefficients of heat transfer for water, isopropanol, n-propanol and their azeotropes. Two phase heat transfer coefficients for all liquids were correlated by a single equation.

$$\frac{h_{TP}}{h_L} = 0.065 \left(\frac{1}{X_{tt}}\right)\left(\frac{T_s}{\Delta T_f}\right)\left(\frac{\sigma_{H_2O}}{\sigma_L}\right)^{0.9} \qquad (2.8)$$

All the length mean coefficients from runs with a two-phase mixture quality between 0.02 and 0.68 at the tube exit are correlated well within + 20% accuracy limits. The same equation correlates local heat transfer coefficients from runs with a quality in the range of 12% to 50% within ± 30% accuracy limits. The dimension less groups ($T_s/\Delta T_f$) and σ_{H_2O}/σ_L are powerful correlating factors. The latter group accounts for the difference in the nucleating properties for various single component liquids and its correlating effectiveness indicated that nucleation was one of the important mechanism present in flow boiling.

Skoczylas el al Inzynierial (13) studied heat transfer thin film boiling of aqueous sugar solutions in a SAMBAY type evaporator. Measurement were carried out at atmospheric pressure and reduced pressure of 380mm H_g with rotor revolutions 6600, 14100 rpm. The heat transfer coefficients were also calculated by a derived boiling of one component liquids. The mean absolute error amounts of 13.7%.

Veneraki (14) investigated the effect of orientation of heat transfer surface on boiling of water and sugar solutions. The heat transfer surface was kept in vertical and horizontal positions. The pressure was changed from 0.1 atmosphere to 1 atmosphere and heat flux ranged from 4×10^3 to 55×10^3 $Kcal/_{m^2}$-hr. Maximum sugar content in the solution was corresponding to 60 Brix. The heat transfer surface was cylindrical made of brass pipe. Based on experimental data a correclation was recommended in the following form:-

$$h = C_1 \, q^n \, p^m \qquad (2.9)$$

The exponent of p, m was found to be dependent on nature of boiling liquid ~~and~~ the orientation of heating surface. The value of m was found to be greater in vertical position than in horizontal. Accordingly the heat transfer coefficent for a vertical orientation of heating surface was higher than that for horizontal position by 20% at high heat fluxes values. At low heat flux values the difference was 40%. Reduction in pressure from 1-atmosphere to 0.1 atmosphere increased the difference between vertical and horizontal position heat transfer coefficient by 25% to 30%.

CHAPTER 3

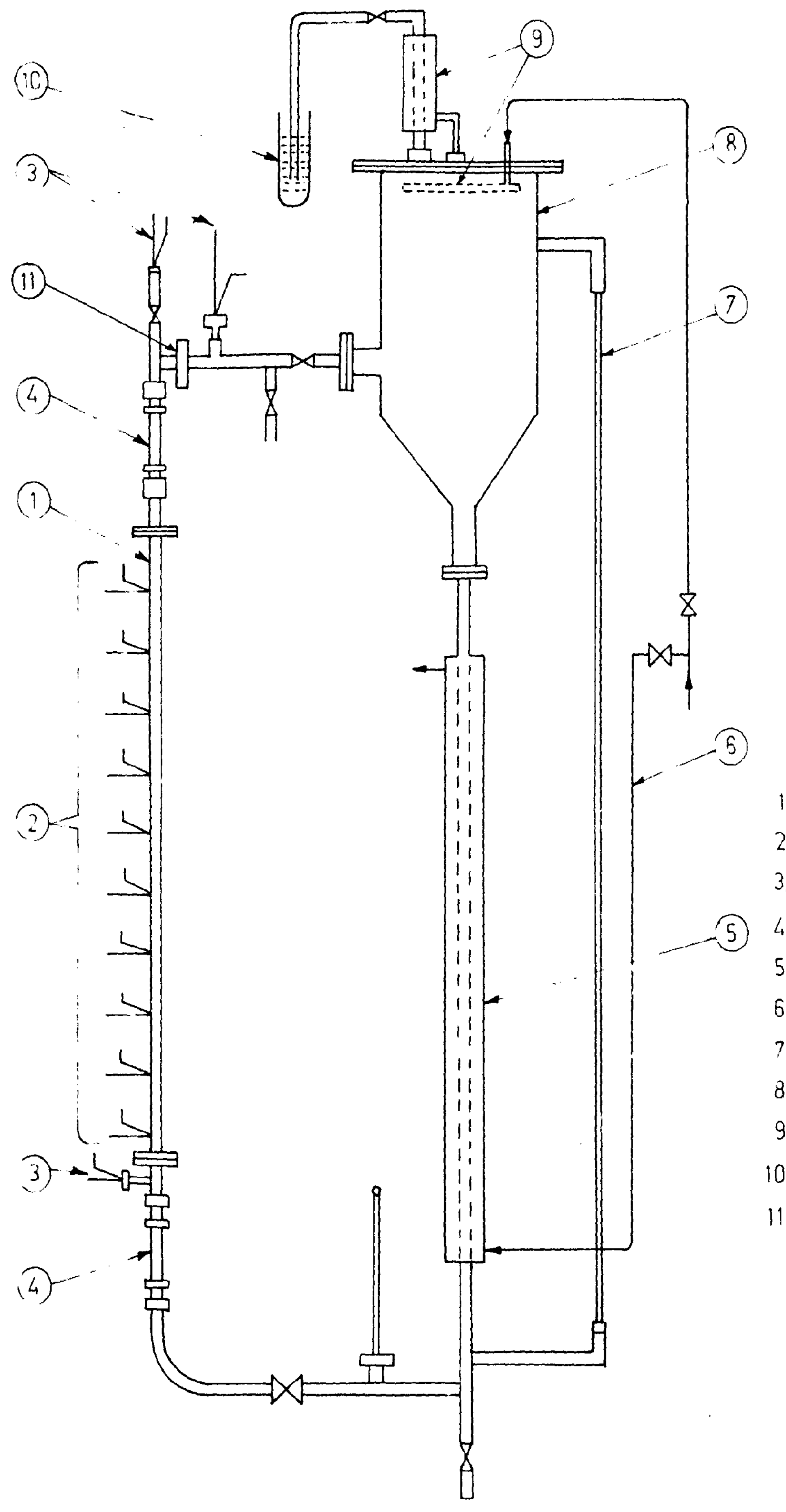
EXPERIMENTAL INVESTIGATION

3.1 EXPERIMENTAL SET-UP

The test apparatus used for the present investigation was a closed loop single vertical tube natural circulation evaporator. The unit mainly consisted of a test section, condensers separator and a cooling jacket.

A sketch of set up is shown in fig 3.1. The loop consisted of two vertical tubes connected by horizontal pipe at the bottom and a vessel at the top. The heated vertical tube (hot leg) is a copper tube of 19.02mm inside diameter, 1.58mm wall thickness and 940mm heated length. The upper and lower ends of test section were connected to two sections of glass tubes of 19mm inside diameter through gland and nut arragement. The glass tubes were carefully aligned with test section to make the axes of all three tubes coincide. These glass tube section were provided for visual observations of flowing liquid and ensuring the absence of any distrubance caused by escape of air bubbles in the loop.

The test section was wound externally with nichrome-wire element (22 gauge) in porcelain beads in two sections, each section of length 470mm, over the tube. The electrical power to heaters was supplied through a stabilised A.C. source. It was regulated by means of auto transformers and measured by calibrated wattmeters. Since one of the objective in present investigation was to study the variation of local heat transfer coefficient all along the length, it became necessary to obtain wall and liquid temperature distribution along the test section at differen t imposed heat fluxs. The wall temperatures were measured by ten copper-constantan thermocouples whose beads were imbedded in the tube wall thickness at locations 100mm apart starting from lower end of heated length. In order to get the bulk temperatures of heated liquid, a cup mixing device was providing just before the exit thermocouple probe. The inlet



1. test section
2. wall thermocouple
3. liquid thermocouple
4. glass tubes
5. cooling jacket
6. cooling water line
7. liquid level indicator
8. separator
9. condensors
10. air vent
11. cup mixing device

Fig. 3.1 Schematic diagram of experimental setup and exit temperatures were measured by means of copper constantan thermocouple probe inserted in the tube. All the wall and liquid thermocouples were connected to vernier type d.c. potentiometer through a cold junction and twelve point selctor switch. The potentiometer was provided with a sensitive spot reflecting galvanometer which made it possible to measure e.m.f. upto 0.001 millivolt with an accuracy of 0.01%.

The exit end of loop after cup mixing device was connected to cylindrical vessel of 282mm inside diameter and 425 mm length with a conical bottom and flanged top cover. This vessel was used to separate vapor from the liquid emerging out of the test section. The conical bottom of vessel was connected to the inlet end of loop through a jacketed vertical tube of same inside diameter as the test section. The cooling water to the jacket was supplied from a constant level tank. The inlet temperature of the test liquid was controlled by regulating the water flow rate to the jacket.

The ascending vapors from separator were condensed by two surface condensers and returned to the circulating liquid. The internal condensor consisted of a spiral coil made of 12.5mm inside diameter copper tube. The coil was kept just below the top of separator. The inlet and outlet ends of coil were taken out through the cover. A jacketed copper tube 20mm inside diameter served as external condenser. The external condenser in series with internal condenser was fitted to the top of the separtor. Water was used as cooling medium and was supplied to condensers from a constant level tank. The air went arrangement at free end of jacketed tube was helpful in visual observation of removal of last traces of dissolved air from test liquids.

3.2 EXPERIMENTAL PROCEDURE

Performance of all the wall and liquid thermocouple installed in position was examined. Water was filled upto a level in the separator to ensure complete submergence of circulation loop. A mercory thermometer was placed in the water. The liquid was then heated by an immersion heater in separator and was circulated through the test section. The liquid temperatore was raised to a particular value and was maintained by means of cooling jacket. Sufficient time was given to ensure thermal equilibrium between test liquid and tube wall and readings of thermocouples and thermometer was recorded. Similar runs were taken at different liquid temperature and reported in tabular form. Calibration of wattmeter performed earlier by R. Prasad (16) were checked .

After the assembly of the experimental set-up, the circulation loop was flushed with water to clean it thoroughly. The electrical and thermocouple connections were made and performance of measurement system was examined. Preliminary runs were taken with forced convection to check the heat balance and reliability of experimental data. When the wall temperatures ceased to change further with time under identical conditions, the main experiment began. Two runs were also taken for pure natural circulation (without boiling) for water at fluxes of 3.554×10^{3} and $5.331 \times 10^{3} W/m^{2}$. The heat transfer coefficent calculated were compared with the predicted values of heat transfer, coefficients given by equation (16,17) and good agreements was found.

During the start up actual run, the test liquid was boiled for about half an hour to drive out any air entrapped in circulation system. This was an important precaution particularly for initial runs with test liquid as dry test surface always entraps a very thin film of air. This air on heating takes the shape of tiny air bubbles which leave the surface on further heating and join the liquid. Thus there sets in, microconvection near the heat transfer surface in addition to those due to density differences. When the release of air bubbles ceased, cooling water to jacket was allowed to flow and desired heat flux adjusted. The system was then left to attain equilibrium conditions. The readings of wattmeters, wall thermocouples and liquid thermocouple were recorded.

CHAPTER 4

RESULTS AND DISCUSSION

The experimental program of the present investigation was carried out with a view to study the heat transfer characteristics during boiling of water and aqueous sugar solutions at atmospheric pressure in a natural circulation single tube evaporator. The experimental measurements included the wall and liquid temperature varations along the evaporator tube length for various values of heat fluxes, degree of subcooling and sugar concentrations. The ranges of the parameters studied are reported in Table 4.1

Table 4.1 Range of Experimental Parameters

System	Concentration wt% sugar	Inlet degree of subcooling, °C	Heat flux W/m^2
Water	0.00	24-60	7.108×10^3-24.878×10^3
Aqueous Sugar solutions	9.90	18.75-55.00	7.108×10^3-24.878×10^3
	19.40	20.00-52.00	
	27.45	17.00-53.00	
	43.60	10.50-46.35	
	57.50	16.20-48.65	

In this chapter, from the wall and liquid temperature distribution along the tube length, the probable mechanisms of heat transfer are discussed. The local values of heat transfer coefficients are computed and their variation along the length of test section examined under the influence of various parameter affecting them.

4.1 VARIATION OF WALL AND LIQUID TEMPERATURES ALONG THE TUBE LENGTH:-

Figures 4.1 and 4.2 show the typical variation of t_W and t_L with z for water. In Figure 4.1 the experimental runs with almost same degree of subcooling have been plotted for various heat fluxes as parameter while the effect of inlet degree of subcooling has been shown in Figure 4.2 for the same value of

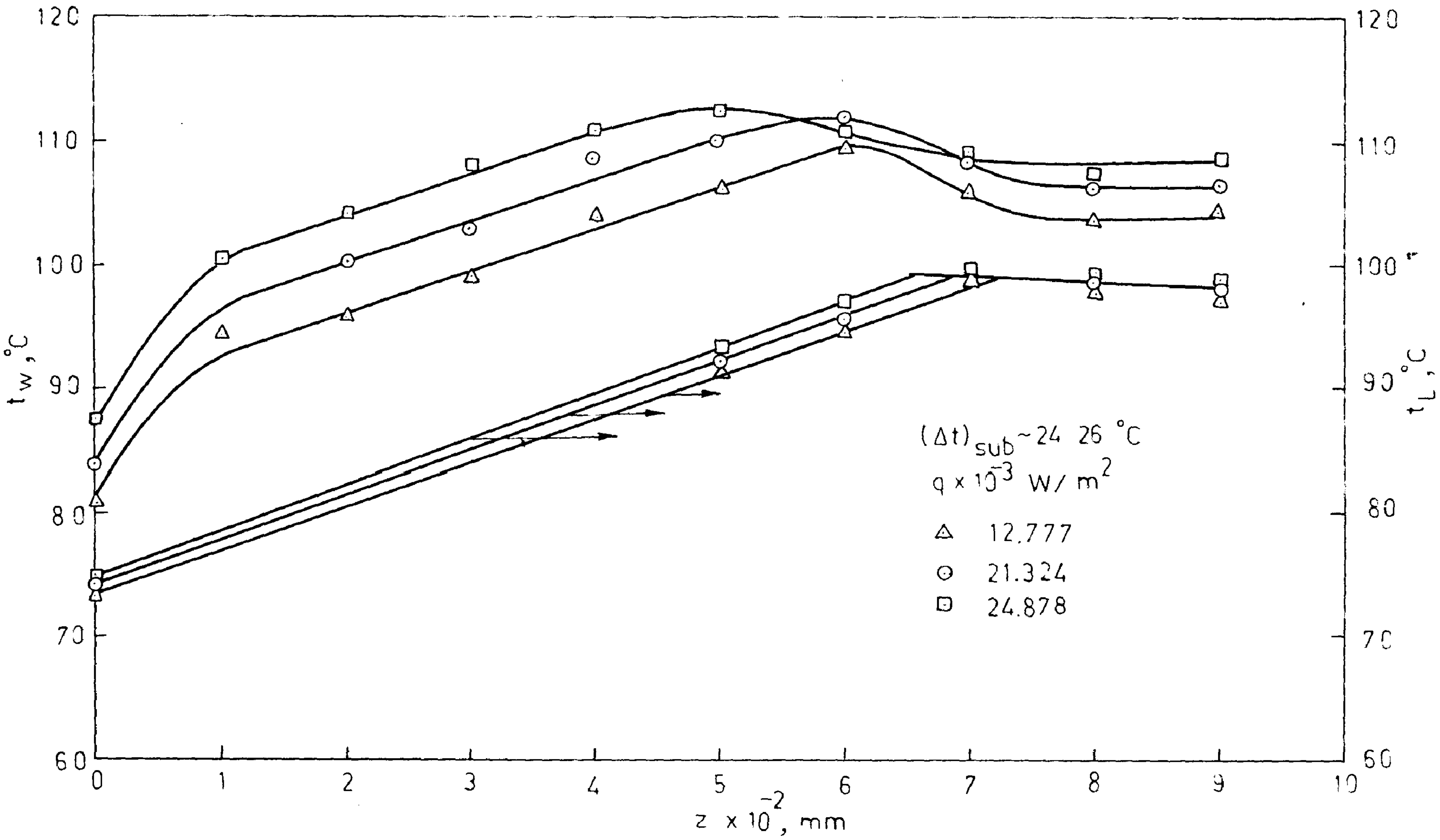


Fig. 4.1 Variation of wall and liquid temperatures along the tube length for constant $(\Delta t)_{sub}$ and various q for water

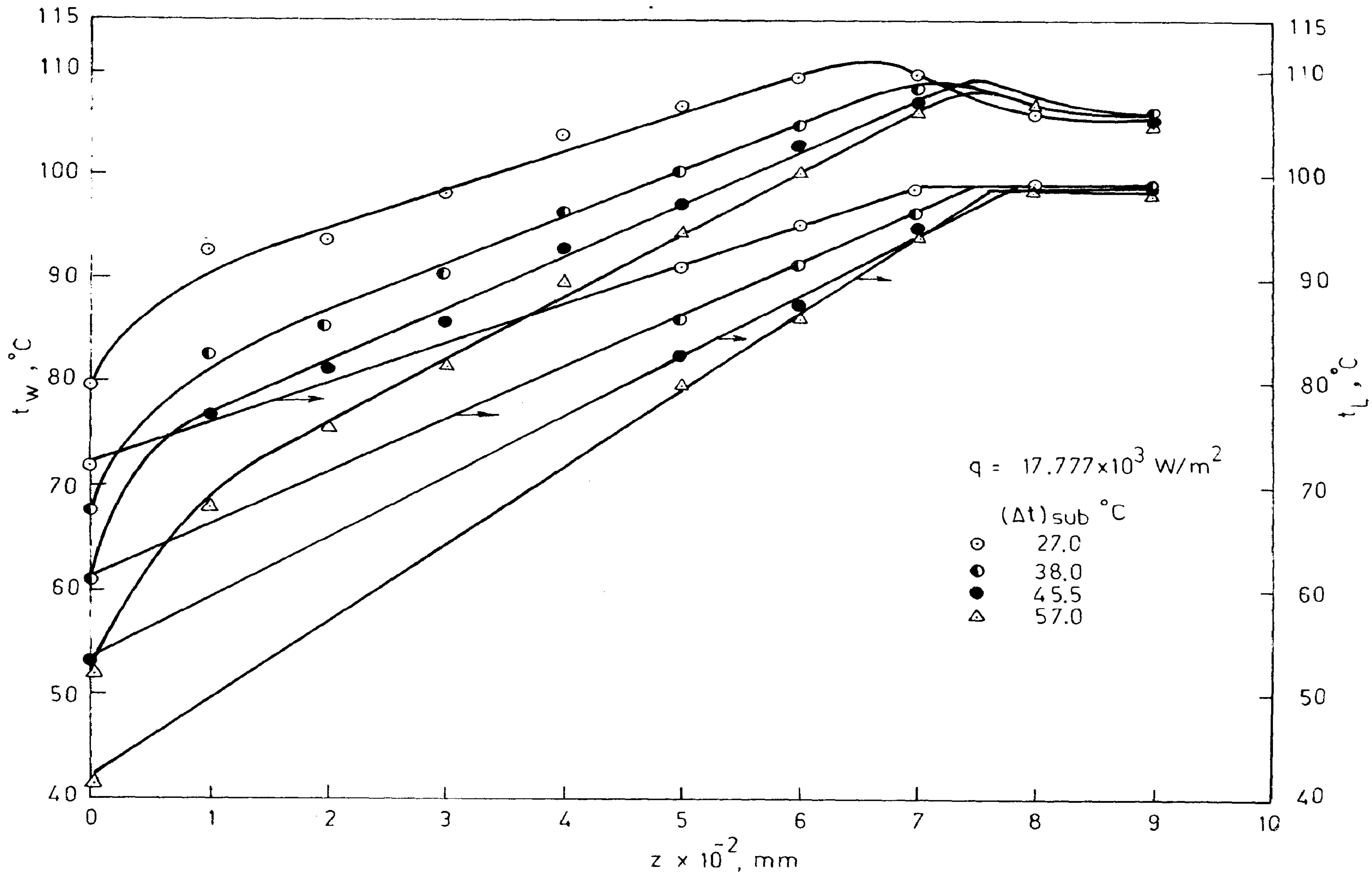


Fig.4.2 Variation of wall and liquid temperatures along the tube length for constant q and various $(\Delta t)_{sub}$ for water

heat flux. From these figures the following characteristic features may be noted.

a. The wall temperature increase aburptly with decaying rates which eventually become constant at a distance of about 100mm from the lower end of the test section. Beyond this length of the tube the temperatures rise linearly with the distance along the tube is upto certain points around which they attain the maximum values and startdecreasing slowly tending to become almost insensitive to the distance in the section near upper end of the tube.

b. The liquid bulk temperature increase linearly with the distance along the tube length upto certain points beyond which they seem to become almost constant. The value of the above temperatures correspond to the saturation temperature of the liquid.

c. The nature of temperature distribution curves is similar at all heat fluxes but with the rise in the value of heat flux, the curves are shifted to higher values of wall temperatures. Thus shifting is more pronounced in the linear portions of the curves than others and thus the points of maximum wall temperatures are moved to lower values of z.

d. When the degree of subcooling of liquid entering the test section is increased keeping heat flux constant, the liquid as well as wall temperature distribution curves except their almost horizontal portions are shifted to lower values of temperatures. The slopes of linear portions of these curves also have a tendency to increase with the rise in $(\Delta t)_{sub}$ showing that the wall and liquid temperatures increase with faster rates as the liquid entering the tube has lower temperatures.

The above observed characteric features of wall and liquid temperature variations along the tube length seem to suggest the existance of three distinct regions along the test section associated with definite mechanisms of heat transfer. The lower most extending upto a distance of about 100mm from the end corresponds to the entrance effect. The non linear variation of wall temperature in this region may be attributed to the fast development of thermal boundary layer. The section of the tube for which both wall and liquid temperature variations are represented by almost parallel straight lines, corresponds to the second region. The heat transfer in the region seems to take place by natural convection without boiling. The points beyond which the wall temperaturestart decreasing mark the begining of the third region. The wall temperatures at these points are much higher than the saturation temperature of the liquid although the liquid core temperature is less than its saturation value. The bubbles nucleate at the superheated surface and grow to a small size till they extend to the colder liquid where the condensation leading to their collapse takes place. Thus the boiling process, through initiated, get confined to the heat transfer surface creating some additional turbulance. The rate of heat transfer is enhanced resulting in the decrease in wall temperature. As the liquid moves of its temperature also increases linearly and the bubbles nucleate and grow to bigger sizes making the boiling process increasingly effective. This process of surface boiling continues along the tube length till the liquid temperature attains its saturation value and fully developed nucleate boiling sets in. The vapor exists as second phase and its quantity increases along the tube length The heat transfer in the third region is affected by the combined action of both natural convection and nucleate boiling. As the quantity of vapor increases along the tube length, and therefore the fluid velocity, the nucleation effect diminishes and the convective heat transfer predominates. The mutual interaction of the two mechanisms seem to counteract each other and thus the wall temperatures in fully developed boiling regions become insensitive to distance.

At higher values of heat flux the wall temperatures also become higher so as to compensate the increased rate of heat transfer. The increase of the tempratures in boiling regions is less then those in natural convective regions, presumably because the effect of heat flux on natural convective heat transfer is only through the rise in t_W whereas the nucleate boiling heat transfer is very strongly influenced due to a large number of nuclei for bubble formation becoming active at higher heat flux values.

The effect of inlet liquid temperature on the axial wall and liquid temperature distributions may be explained with the help of following equations :-

$$q = mC \frac{dt_L}{dz} = h\ (t_W - t_L) \qquad (4.1)$$

The slope, $\frac{dt_L}{dz}$, of liquid temperature curves increased as the liquid mass flow rate, m, decreases if the specific heat C, remains unchanged at a given constant heat flux. When the liquid enters the tube at a lower temperature, the induced flow rate, m, of the liquid decreases while the specific heat, C, remains almost unchaged. Thus the slope $\frac{dt_L}{dz}$ of liquid temperature curves increases for a given constant heatflux making the lines steeper as seen in Fig. 4.2.

Figures 4.3 and 4.4 represent the typical plots of t_W and t_L versus z with various values of heat flux and inlet degrees of subcooling as parameters for aqueous sugar solutions of 9.90 and 27.45 wt% concentrations respectively. The exclusive effect of heat flux and degree of subcooling could not be shown for sugar solutions as was done for water. However, it may be seen from the figures that the characteristic nature of wall and liquid temperature varitions along the test section length is essentially similar to those for water. The curves for higher heat flux and lower degree of subcooling are shifted relatively more to higher values to t_W than those for water . The spread between the adjacent curves, particularly in matural convective and surface boiling regions, is also observed to be more for sugar solutions.

The effect of sugar concentration on temperature distribution curves is shown in Figures 4.5 and 4.6. Figure 4.5 represents the data at high inlet degree of subcooling and low heat flux value while in Figure 4.6 the data at low degree of subcooling and high heat flux values have been plotted.

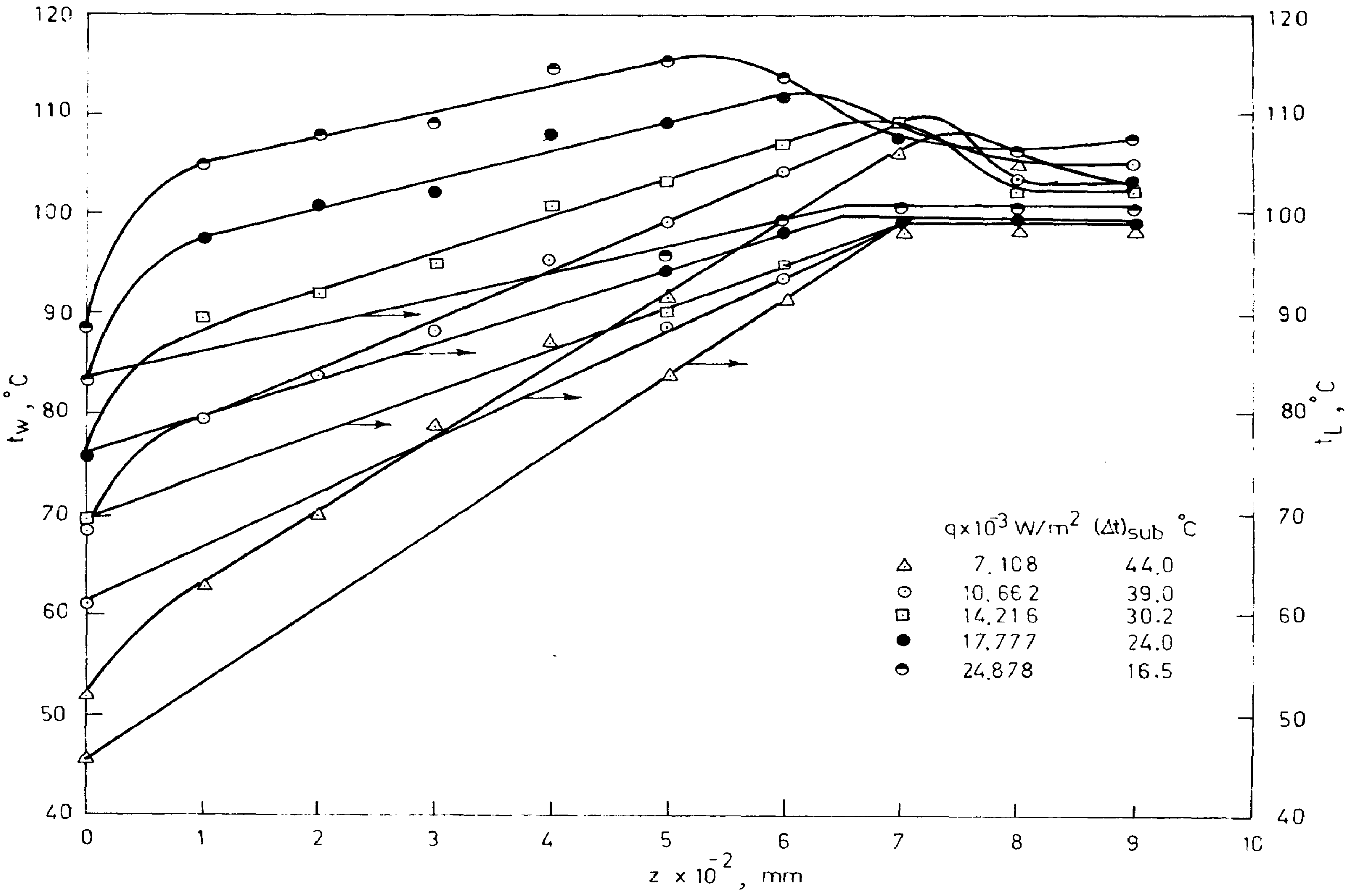


Fig. 4.3 Variation of liquid and wall temperatures along the tube length for various q and $(\Delta t)_{sub}$ for 9.90 wt. % sugar solution

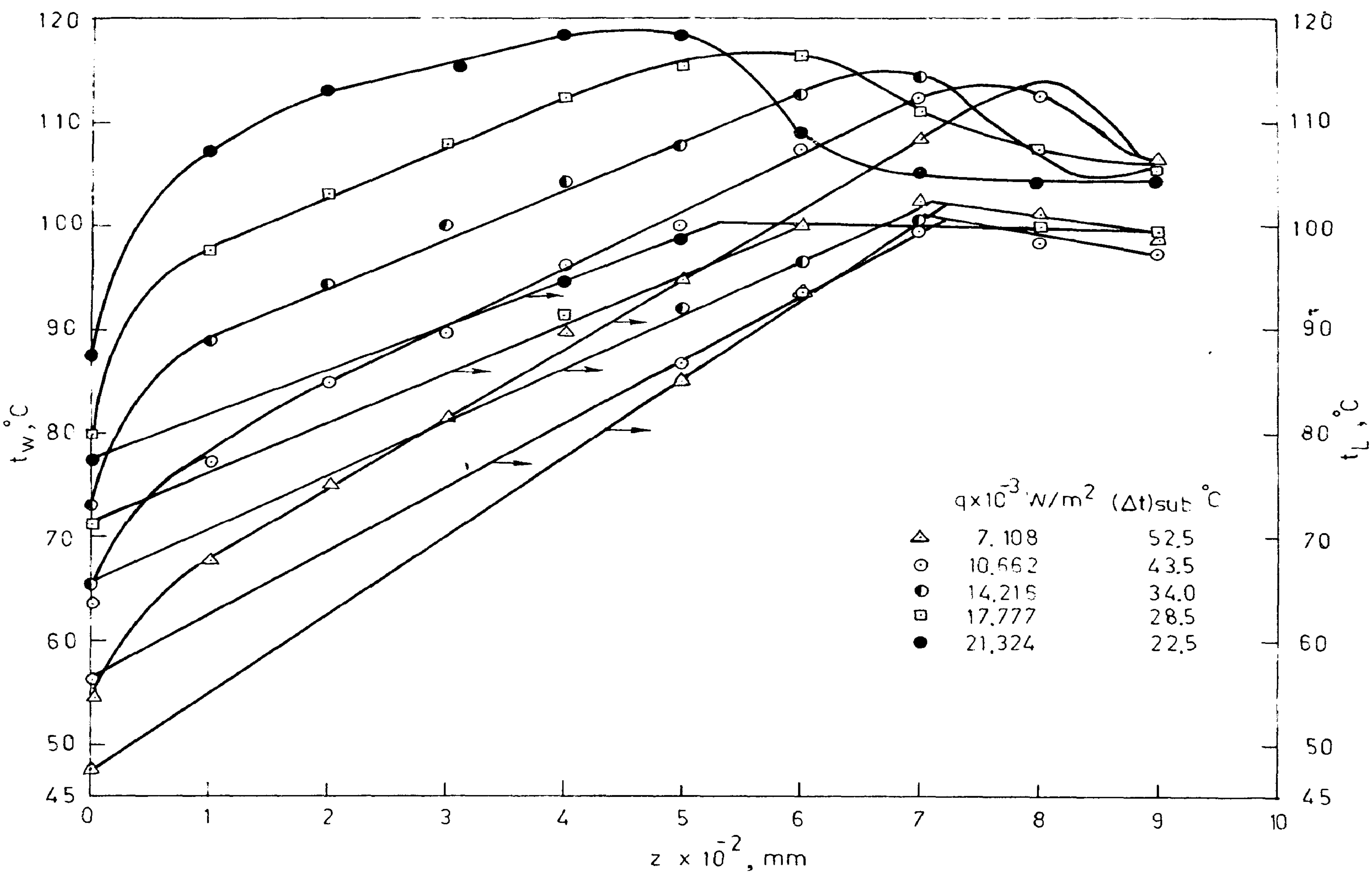


Fig.4.4 Variation of wall and liquid temperatures along the tube length for various q and $(\Delta t)_{sub}$ for 27.45 wt.% sugar solution

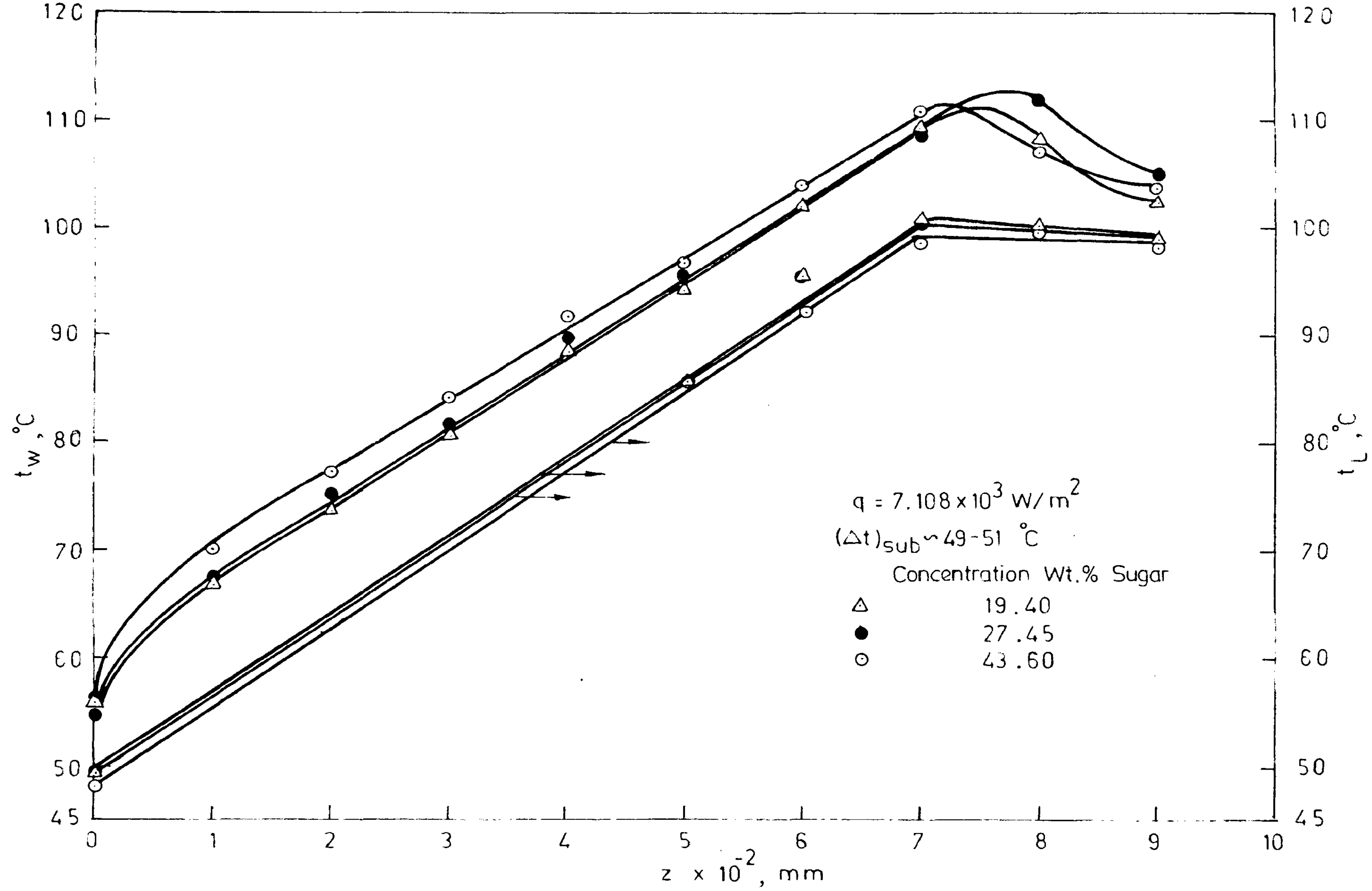


Fig 4.5 Variation of wall and liquid temperatures along the tube length for constant q and constant $(\Delta t)_{sub}$ for various sugar solutions

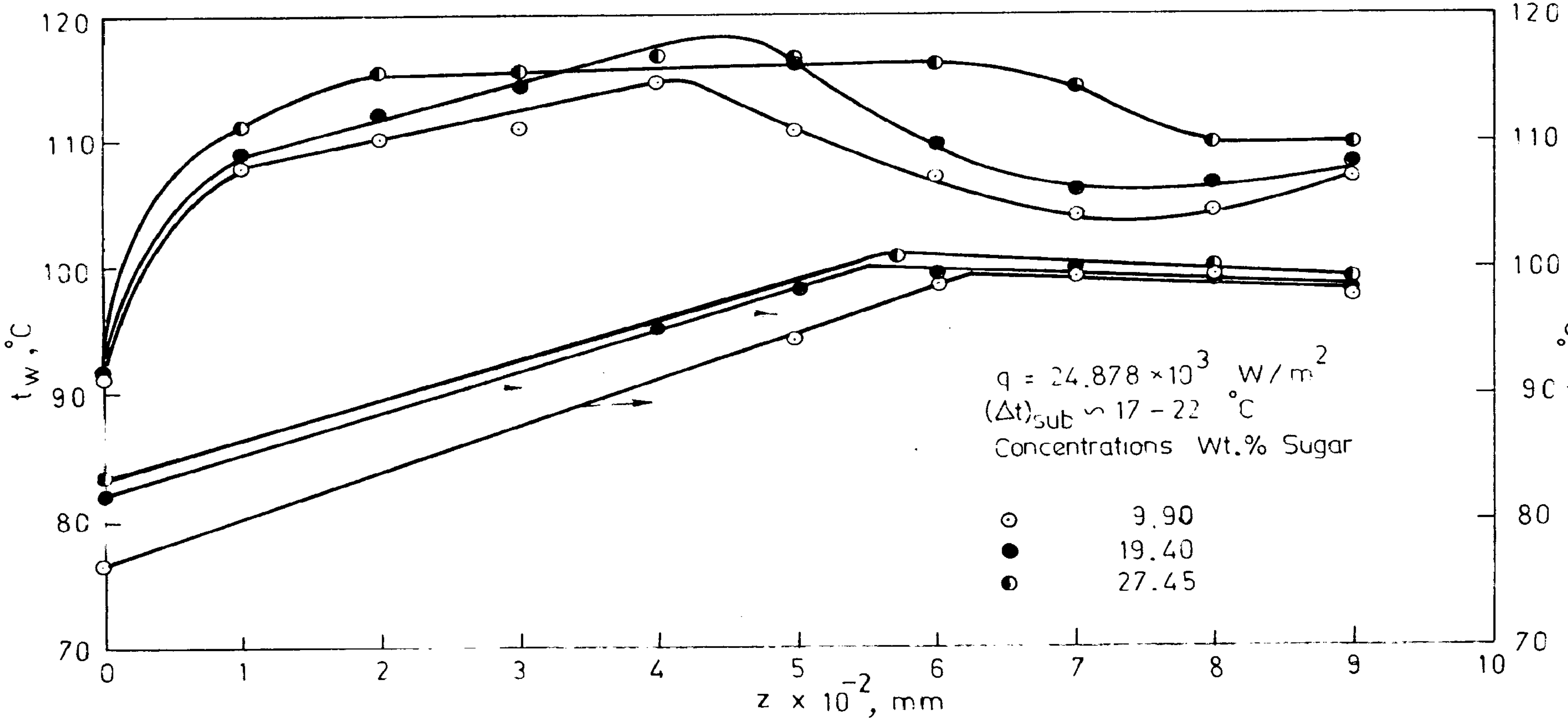


Fig. 4.6 Variation of wall and liquid temperatures along tube length for constant q and constant $(\Delta t)_{sub}$ for various concentrations of sugar solutions

The concentration of the solutions forms the parameter in both the figures. The nature of curves is almost similar at all concentration but the wall temperature curves are moved to higher values of t_W as the concentration is increased. The curves of Figure 4.5 also indicated that the natural convective heat transfer mechanism extends over the major portion of the test section where as boiling process seems to be predominant for the conditions of Figures 4.6.

4.2 VARIATION OF HEAT TRANSFER COEFFICIENT ALONG THE TUBE LENGTH:-

The local values of heat transfer coefficient along the tube length were computed from the measured values of wall and liquid tempretures along the test section at uniform heat flux conditions of by using Eq.(4.1). The experimental values of small h so obtained have been plotted in Figures 4.7 and 4.8 with q and $(\Delta t)_{sub}$ as parameters respectively for water. The data points of entrance section have not been included in these figures.

The heat transfer coefficient increases almost linearly with z at a slow rate upto certain position of the tube beyond which it rises with a progressively enhancing rate. The straight line portions of the curves represent the region of single phase natural convective heat transfer and remaining portions indicate the nucleate boiling region. The increasing high rate of heat transfer nucleate boiling region is attributed to the turbulence created by nucleation and growth of bubbles at reducing local degree of subcooling along the heating surface. The number of active centres for bubble formation depends strongly upon heat flux and as it is raised to higher value, the number of nucleation sites multiplies resulting in more turbulance at the heating surface. This explains the shifting of curves in Figures 4.7 to higher values of h as q is increased. The increase in h with q for single phase natural convection region may be attributed to the rise in the wall temperature. The effect of increasing the value of $(\Delta t)_{sub}$ at test section inlet is to shift the curves to lower values of h. The decrease in heat transfer coefficient with increase in the inlet degree of subcooling during natural convection

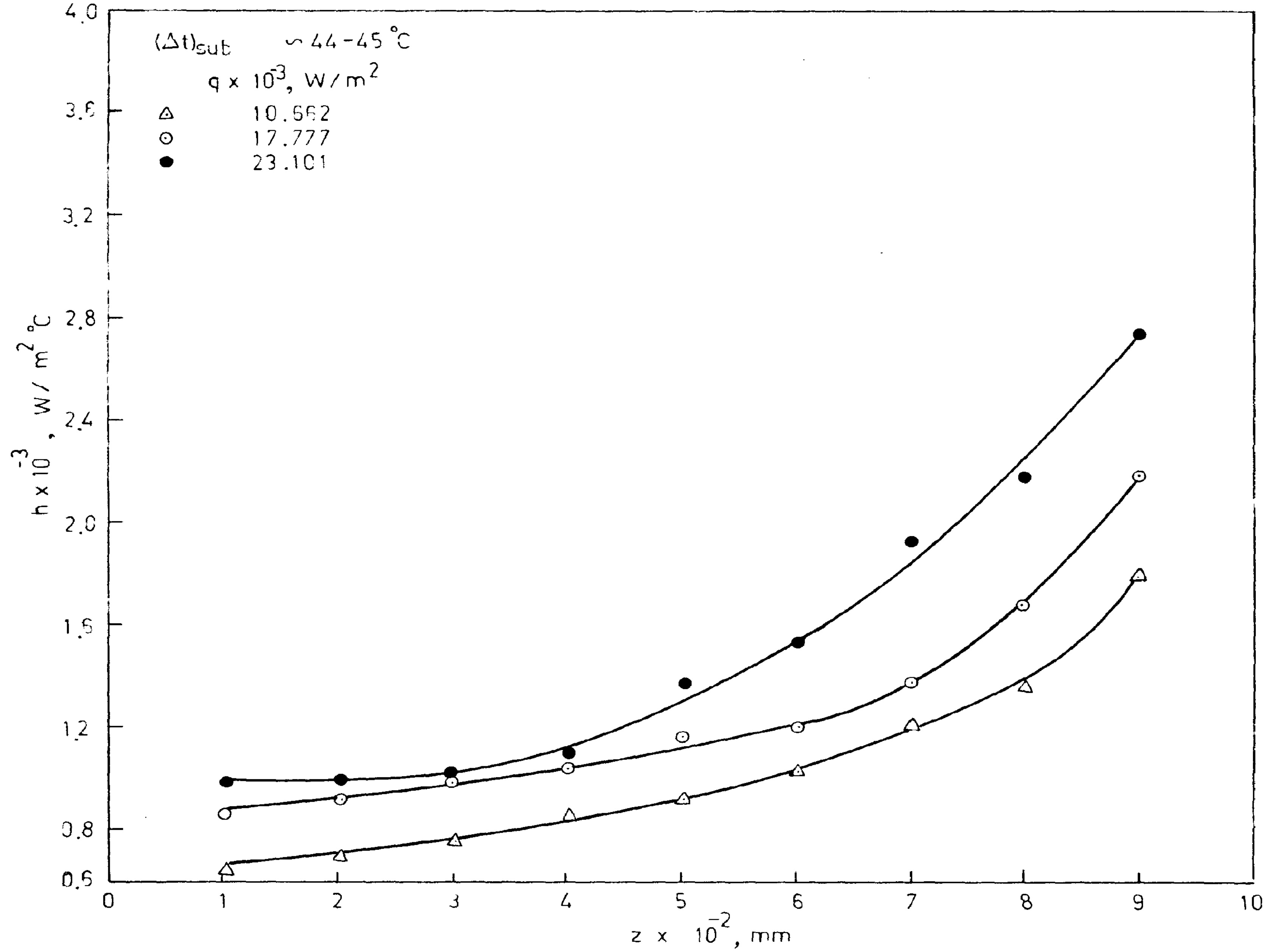


Fig 4.7 Variation of h with z for a constant $(\Delta t)_{sub}$ and various q for water

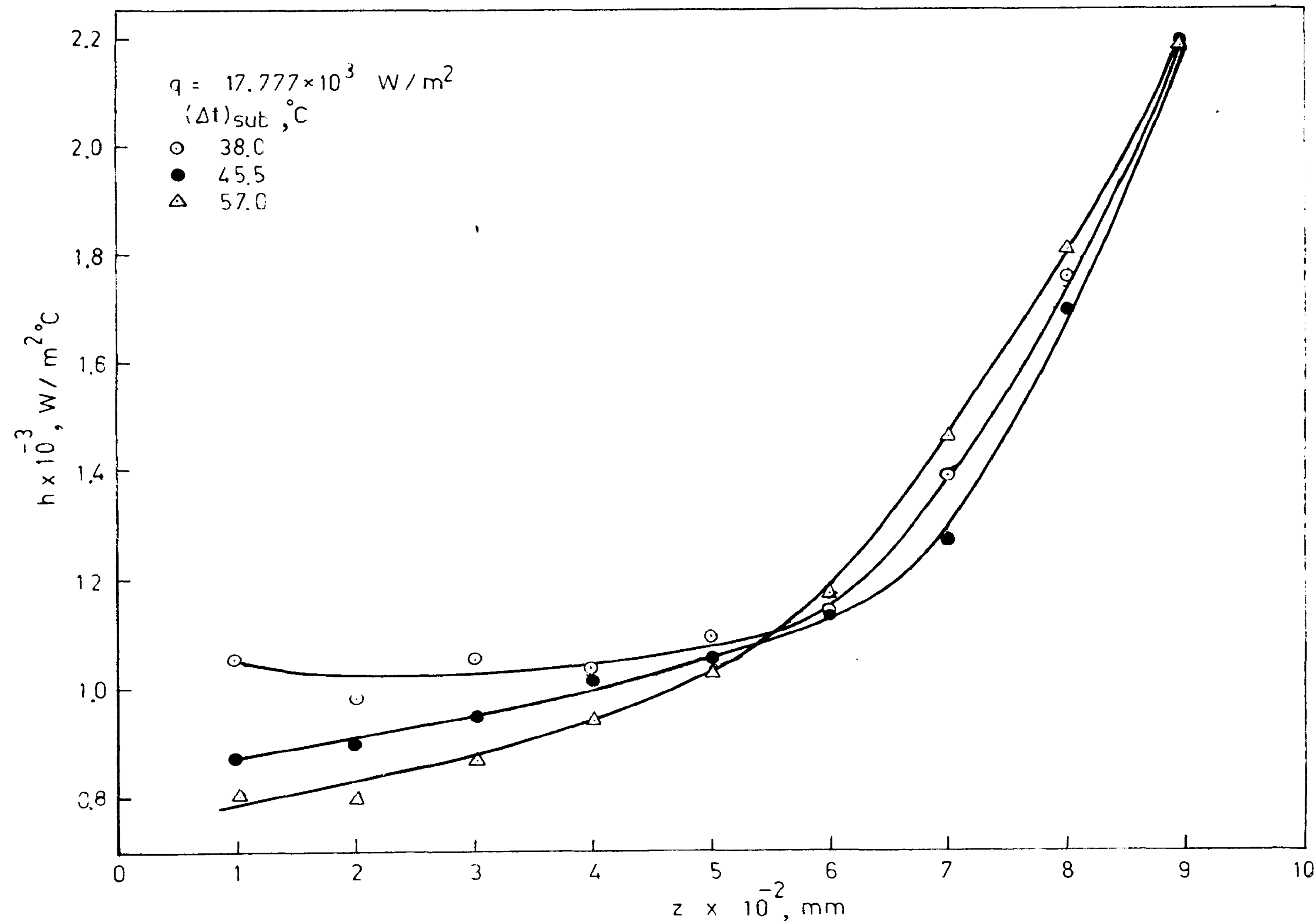


Fig.4.8 Variation of h with z for a constant q and various $(\Delta t)_{sub}$ for water

seems to be due to the reduction in wall temperature in contact with colder liquid, while in nucleate boiling it is the resultant of convection and bubble formation which may be suppressed by the colder liquid.

The representative plots of h versus z for aqueous solutions have been shown in Figures 4.9 and 4.10. The data points have been selected for various concentrations at the same values of heat flux and approximately constant inlet degree of subcooling in both the figures so as to show the exclusive effect of concentration of heat transfer coefficient. It is observed from the figures that the variation of h with z is similar to that of water by the addition of sugar in water lowers the values of heat transfer coefficient at almost all levels of the test section. The curves, however, tend to coverage somewhere inbetween the natural convective and fully developed saturated boiling regions of the tube. At low heat flux and high degree of subcooling this condition seem to be reached near the exit and all the tube as illustrated by Figure 4.9. From Figure 4.10 it is observed that this region of almost same values of h at all concenttations is shifted to a lower level of the tube as the heat flux is raised and inlet degree of subcoling is lowered. The decrease in heat transfer coefficient with increase in the sugar concertration of soution during natural covection may be due to the changes in physical properties of solutions. In the boiling section of the tube, the retarded rate of bubble formation and growth on the heating surface may be responsible for lower heat transfer coefficients at higher concentrations of solutions. As the boiling solution moves upward with a net amount of vapor generation, it becomes more concentrated and the heat transfer coefficient increases with a decreasing rate eventually passing through a maximum value for each concentration and finally decreases. This seems to be the probable explanation for the characteristic behaviour of curves in Figure 4.10.

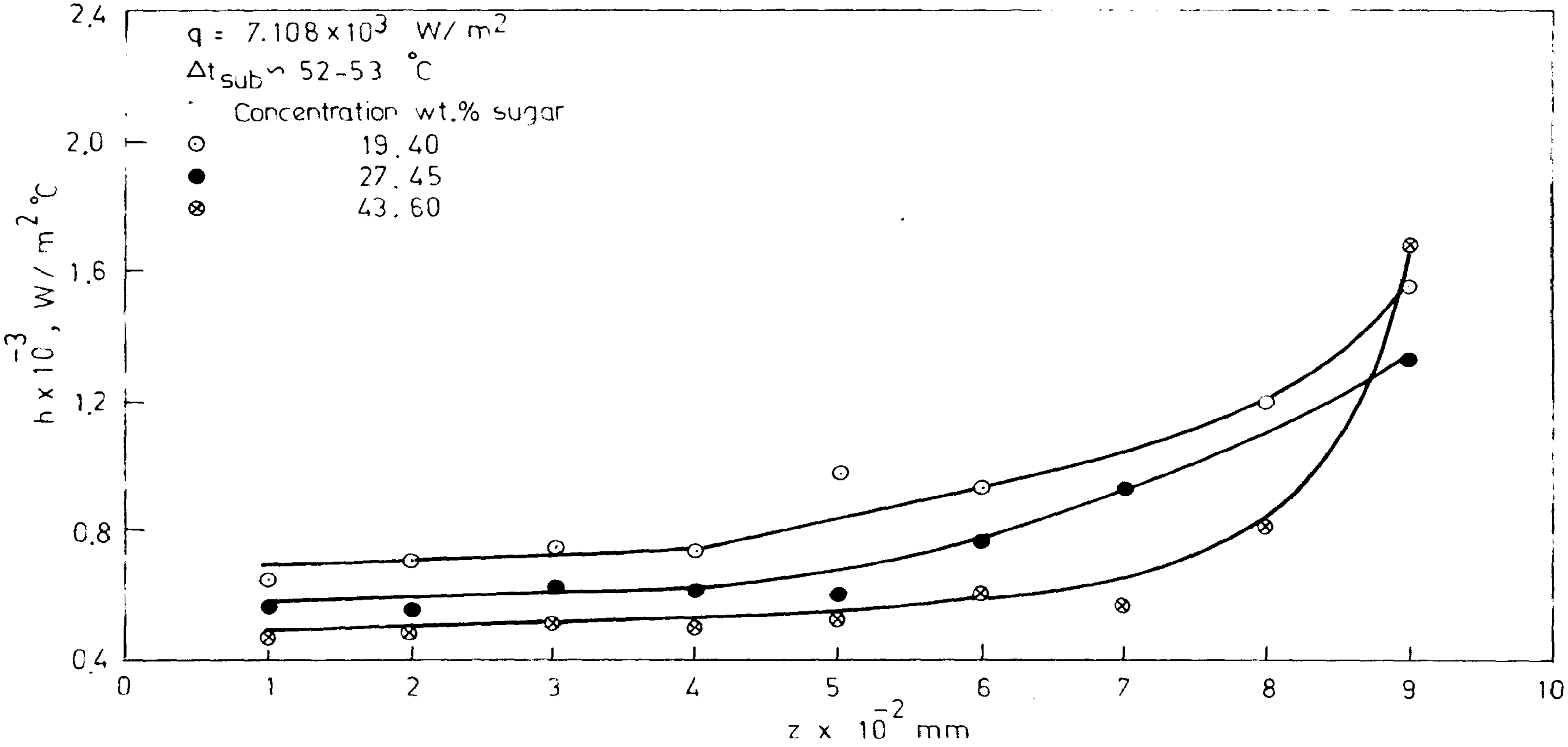


Fig. 4.9 Variation of h with z for constant q and $(\Delta t)_{sub}$ for various concentrations of sugar solutions

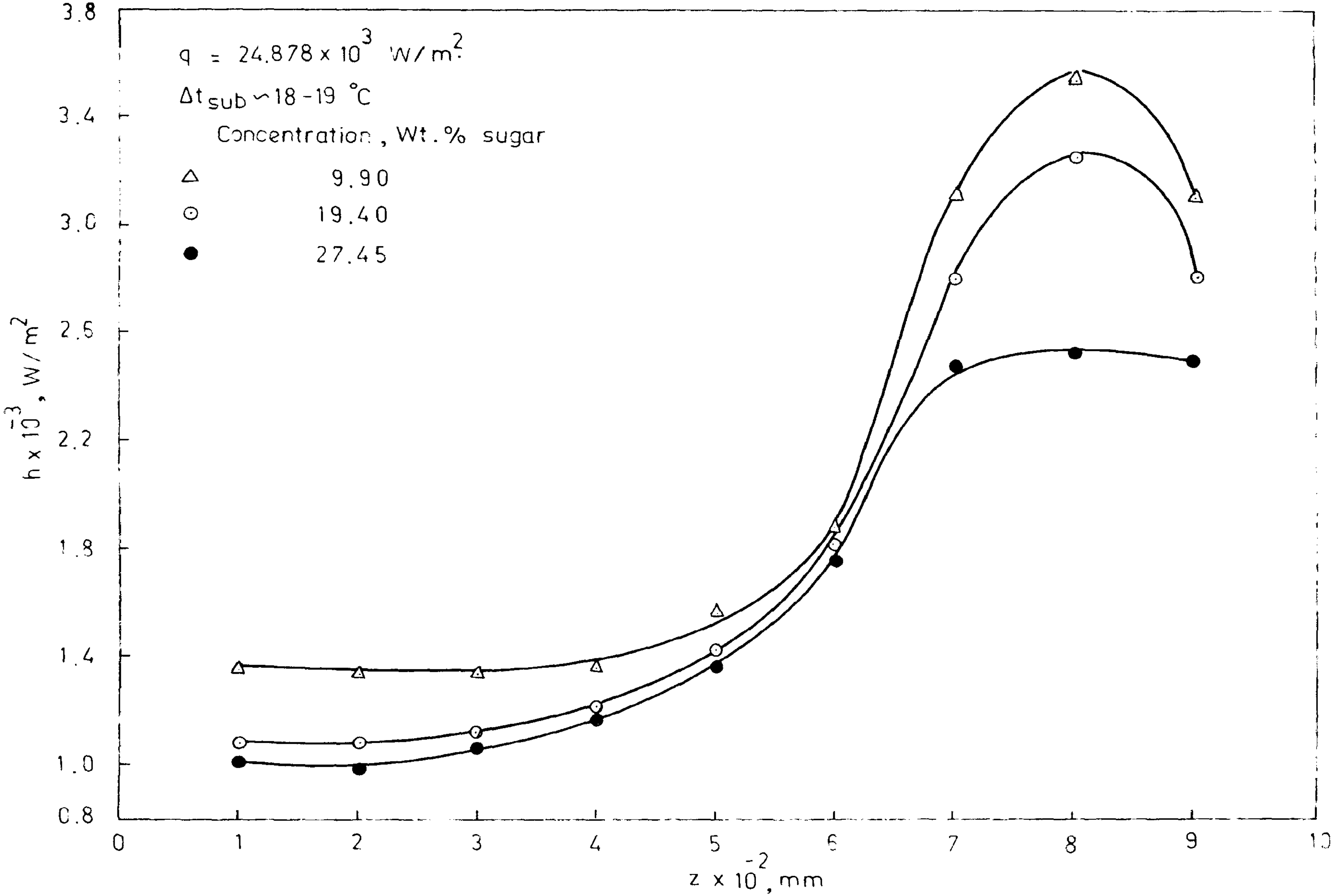


Fig.4.10 Variation of h along z with constant q and constant $(\Delta t)_{sub}$ for various concentrations of sugar solutions

4.3 GENERAL CORRELATIONS:-

The results of the previous sections have clearly indicated the existance of two distinct regions where heat transfer takes place by the following modes.

a. Single phase natural convection
b. Natural circulation boiling

The region of non-boiling natural convective heat transfer starts from the lower most and of the tube and extends upto a point where the wall superheat attains a value sufficient enough for the effective surface boiling. The nucleate boiling takes place in the remaining portion of the tube. Since the machanisms of heat transfer process in the above mentioned two regions are different it was thought desirable to develop the correlations for them separately.

43.1 Single Phase Natural Convection

The natural convective heat transfer coefficient depends upon the temperature difference between heat transfer surface and the liquid and its physical properies. The correlations for computing heat transfer coefficients, therefore, have generally been developed in terms of Nusselt, and Prandtl numbers.

The experimental data points for water and aqueous sugar solution in the present study were found to be best correlated in the following functional form :-

$$Nu = C_1 (G_r \; P_r)^{n_1} (P_r)^{n_2} \qquad (4.2)$$

The dimensionless groups in Eq (4.2) were computed at various thermocouple positions along the tube and the physical properties were taken at arithmatic average of corresponding wall and liquid temperatures. The exponents n_1 and n_2 and constant C_1 for Eq(4.2) were determined by regression analysis. A least square curve fitting program was prepared and the computations were done on a digital computer, I.B.M 1620.

The values of n_1, n_2 and C_1 for water and sugar solution have been given in Table 4.2. A comparison between the experimental N_u and those predicted by the proposed equations have been shown in Fig 4.11. Almost all the data points for water and sugar solutions lie within a maximum error of ± 20%.

Table 4.2 Values of C_1, n_1 and n_2 in Eq 4.2

System	C_1	n_1	n_2
Water	8.10×10^{5}	-0.624	0
19.4 and 27.45 wt% sugar solution	0.913	0.15	0.26
43.60 and 57.50 wt% sugar solution	9.6334	-0.0033	0.17

4.3.2 Natural Circulation Boiling

The heat transfer coefficient for boiling region depends upon the additional turbulence created by the bubble formation and their dynamics over natural convective flow. It was, therefore, decided to use a rates of these transfer coefficients with those of single phase natural convection for developing a suitable correlation. The parameters influencing the value of h_B/h_C were grouped together and a correlation in the following functional forms, as suggested by earlier workers (16) for water glycerine solutions, was obtained.

$$\frac{h_B}{h_C} = C_2\left(\frac{\rho_L}{\rho_V}\,\frac{C\,d}{z_s}\,(\Delta t)_{sub}\right)^{n3}\left(\frac{t_L}{t_W}\,\frac{z}{d}\right)^{n_4} \qquad (4.3)$$

The values of n_3, n_4 and C_2 were determined by the regression analysis as indicated in the previous section. Using the values the Eq(4.3) assumes the following form :-

$$\frac{h_b}{h_C} = 7.5\ \mathrm{x}10^{-5}\left(\frac{\rho_L}{\rho_V}\,C\,\frac{z_s}{d}\,(\Delta t)_{sub}\right)^{-0.90}\left(\frac{t_L}{t_W}\,\frac{z}{d}\right)^{2.79}$$

For sugar solutions:- (4.4)

$$\frac{h_B}{h_C} = 2.25\mathrm{x}10^{-4}\left(\frac{\rho_L}{\rho_V}\,C\,\frac{z_s}{d}\,(\Delta t)_{sub}\right)^{-1.386}\left(\frac{t_L}{t_W}\,\frac{z}{d}\right)^{2.93} \qquad (4.5)$$

Figure 4.12 shows a comparison between the experimental values of h_B/h_C and those predicted by above quations. Almost all the data points of surface as well as saturated boiling of water

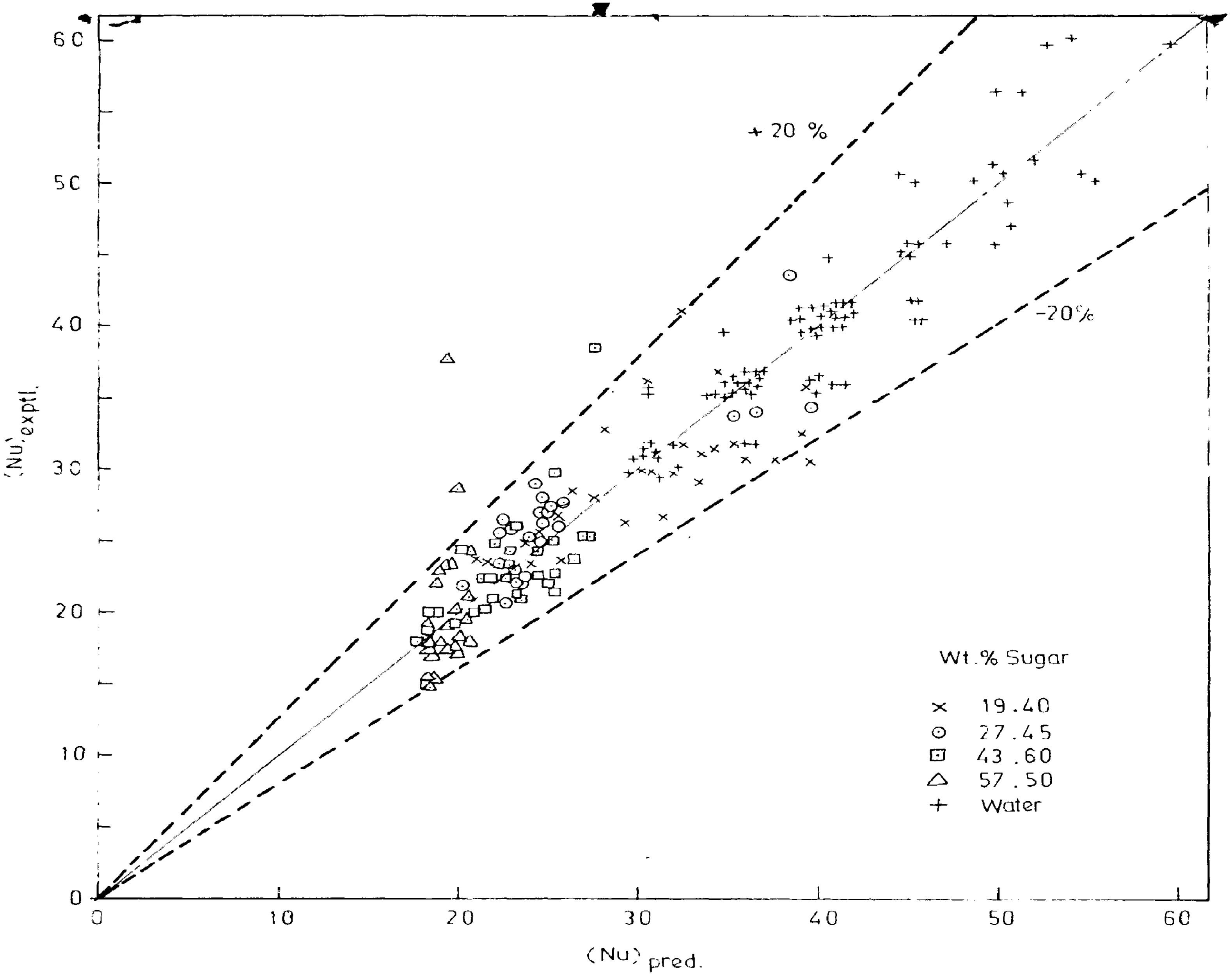


Fig. 4.11 Comparison between experimental Nusselt numbers and those predicted by eqn. (4.2)

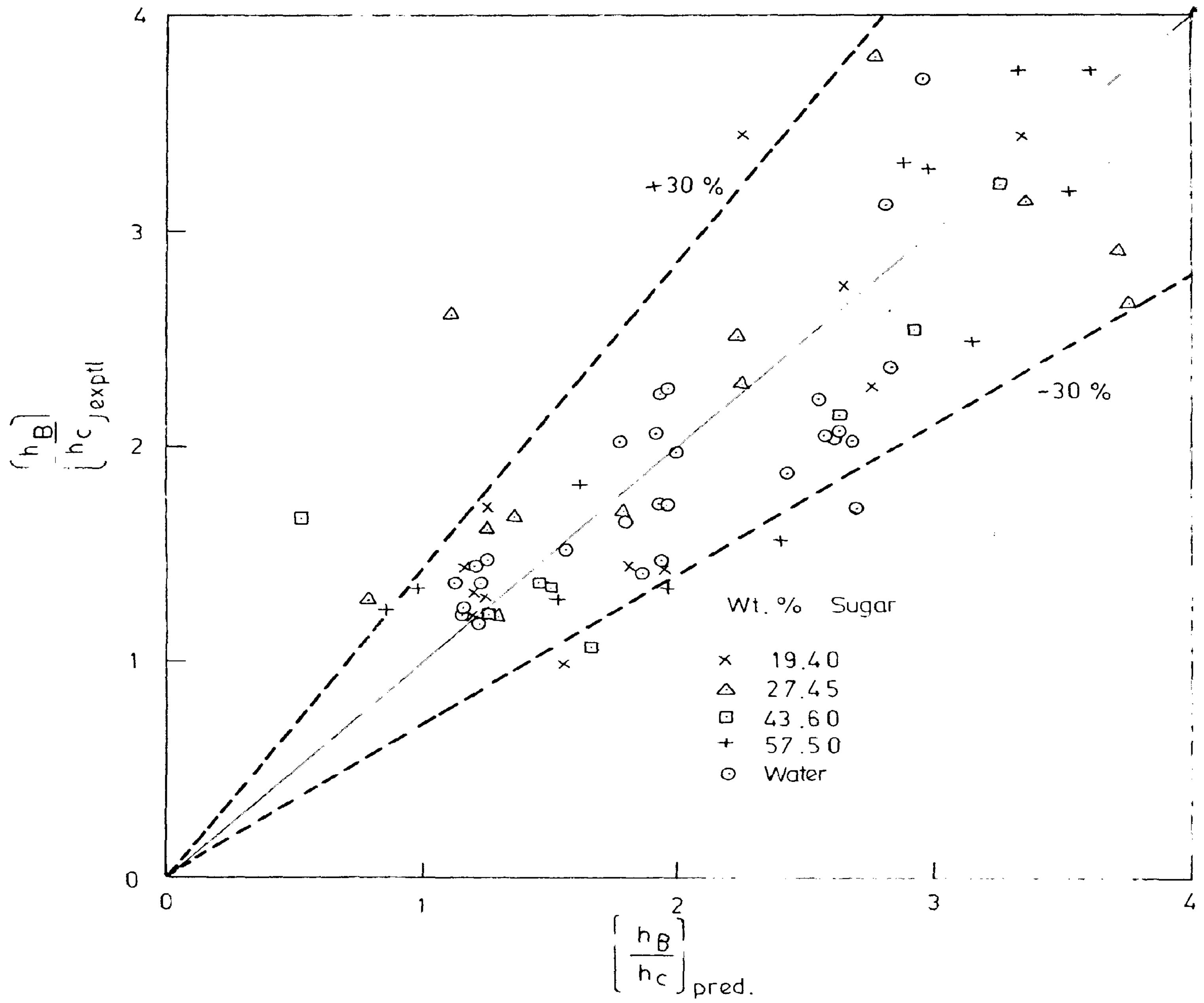


Fig. 4.12 Comparison between experimental $\frac{h_B}{h_c}$ and those predicted by eqn (4.3)

and aueous sugar solutions show a reasonably good agreement with a maximum deviation of about ± 30%.

The physical properties for Eq (4.3) of water and sugar solutions were taken at their respective saturation temperatures.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

1. Based on wall and liquid temperature variations along the tube at uniform heatflux, it is concluded that there exist distinct regions over different lengths in which heat transfer takes place by natural convection, sugar boiling and fully developed saturated boiling for water and aqueous sugar solutions entering a vertical tube of an evaporator. The lengths of the tube corresponding to the regions are strongly influences by the heatflux, inlet degree of subcooling and sugar concentrations in the solutions.

2. The heat transfer coefficient remains almost same or increases very slowly with distance along the tube length in natural convective region except near entrance. Beyond the onset of boiling, transfer coefficients rise sharply along the tube length. The values of h are observed to increase with q while they decrease with inlet degree of subcooling.

3. The increases in the sugar concentration of solutions affects the heat tansfer coefficient adversily in all the regions. Near the exit end of the tube during saturated boiling region, the transfer coefficients attain maximum values and then decrease.

4. The experimental data of single phase natural convective region, excluding entrance, have been correlated by the following equation with a maximum deviation of about ± 20%.

For Water:-

$$N_u = 8.10 \times 10^5 (P_r \; G_r)^{-0.624}$$

19.4 and 27.45 wt% sugar solutions:-

$$N_u = 0.913 \; (G_r \; P_r)^{0.15} \; (P_r)^{-0.26}$$

43.6 and 57.50 wt% sugar solutions:-

$$N_u = 9.6334 \; (G_r \; P_r)^{-0.0033} (P_r)^{0.17}$$

5. The heat transfer coefficients for subcooled and saturated boiling of water and sugar solutions wer best represented by the following equations:-

For Water:-

$$\frac{h_B}{h_C} = 7.5\text{x}10^{-5} \left(\frac{\rho_L}{\rho_V} \underline{C} \frac{z_s}{d} (\Delta t)_{sub} \right)^{-0.09} \left(\frac{t_L}{t_W} \frac{z}{d} \right)^{2,79}$$

For Sugar Solutions :-

$$\frac{h_B}{h_C} = 2.25\text{x}10^{-4} \left(\frac{\rho_L}{\rho_V} \underline{C} \frac{z_s}{d} (\Delta t)_{sub} \right)^{-1.386} \left(\frac{t_L}{t_W} \frac{z}{d} \right)^{2.93}$$

The maximum deviations between the experimental values of h and those predicted by the above equations were found to be about ± 30%.

The investigation may be extended to more industrially important aqueous salt soulutions over wide range of concentrations, heatflux and degree of subcooling. The experimentals may be carried out at higher pressures and vacuum to develop generalized correlations useful for the design of multiple effect evaporators.

APPENDIX I

EXPERIMENTAL DATA

Table I.1	Experimental Data of Heat Transfer to Water.
Table I.2	Experimental Data of Heat Transfer to Various Sugar Solutions.

Table I.1 EXPERIMENTAL DATA OF HEAT TRANSFER TO WATER

$z\times10^{-2}$ mm	Temperature, °C		$z\times10^{-2}$ mm	Temperature, °C	
	t_W	t_L		t_W	t_L
$q=7.108\times10^{3}$ W/m^2					
Run No W.1			Run No W.4		
0	48.85	42.00	0	62.05	55.40
1	59.05	-	1	74.60	-
2	65.40	-	2	78.10	-
3	74.30	-	3	83.35	-
4	83.20	-	4	89.75	-
5	87.55	81.25	5	93.90	84.40
6	96.25	90.20	6	98.90	88.50
7	103.70	99.35	7	103.40	95.70
8	101.20	98.05	8	102.60	97.20
9	101.20	98.05	9	101.05	96.80
$q=8.885\times10^{3}$ W/m^2					
Run No W.2			Run No W-5		
0	48.10	42.00	0	46.80	39.00
1	59.50	-	1	59.75	-
2	66.80	-	2	66.80	-
3	75.00	-	3	74.65	-
4	82.30	-	4	81.75	-
5	88.65	81.95	5	88.25	75.99
6	96.15	84.80	6	94.55	83.90
7	103.15	97.00	7	101.40	90.80
8	101.60	98.30	8	102.90	96.70
9	101.60	97.85	9	101.20	95.65
Run No W.3 $q=10.662\times10^{3}$ W/m^2			Run No W.6		
0	55.60	49.10	0	52.80	45.25
1	67.40	-	1	65.50	-
2	73.70	-	2	73.30	-
3	79.50	-	3	80.70	-
4	86.70	-	4	89.10	-
5	90.70	81.45	5	94.00	85.30
6	97.65	88.00	6	101.40	92.80
7	102.60	94.60	7	107.60	101.40
8	102.00	96.95	8	103.90	99.75
9	100.60	96.30	9	103.70	98.80

$zx10^{-2}$ mm	Temperature, °C		$zx10^{-2}$ mm	Temeperature, °C	
	t_W	t_L		t_W	t_L
$q=12.439x10^3$ $W/_m2$					
Run No W.7			Run No W.10		
0	53.60	45.00	0	58.85	51.40
1	67.40	-	1	72.90	-
2	74.30	-	2	78.60	-
3	79.95	-	3	83.20	-
4	86.95	-	4	91.05	-
5	92.00	83.25	5	95.15	83.60
6	98.70	92.20	6	100.55	89.90
7	105.05	99.55	7	106.55	97.30
8	104.00	98.70	8	107.35	99.45
9	102.80	98.50	9	104.85	98.95
$q=14.216x10^3$ $W/_m2$			Run No W.11		
Run No. W.8					
0	65.35	58.50	0	65.50	57.70
1	79.00	-	1	78.30	-
2	82.05	-	2	81.35	-
3	88.20	-	3	88.40	-
4	94.95	-	4	95.00	-
5	98.40	87.15	5	99.60	86.80
6	103.50	91.30	6	106.00	94.00
7	107.60	97.80	7	108.10	99.50
8	105.70	99.50	8	103.70	98.80
9.	104.80	98.90	9	104.50	98.60
Run No W.9			Run No W.12		
0	53.75	44.90	0	72.30	65.80
1	67.50	-	1	84.60	-
2	73.30	-	2	87.95	-
3	79.30	-	3	92.40	-
4	86.75	-	4	98.25	-
5	91.35	80.50	5	101.00	88.10
6	98.50	86.60	6	105.10	92.70
7	104.70	96.65	7	107.85	96.80
8	106.85	99.20	8	105.30	98.55
9	104.25	98.55	9	103.60	97.45

$z \times 10^{-2}$ mm	Temperature, °C		$z \times 10^{-2}$ mm	TEemperature, °C	
	t_W	t_L		t_W	t_L
$q = 17.777$ W/m^2					
Run No W. 13			Run NoW16		
0	71.65	63.95	0	61.40	53.65
1	86.65	-	1	77.00	-
2	89.65		2	81.40	-
3	94.50	-	3	86.00	-
4	99.90	-	4	93.40	-
5	102.95	90.25	5	98.00	82.75
6	106.85	93.05	6	102.70	88.60
7	109.35	97.85	7	107.35	95.20
8	104.70	98.50	8	107.35	99.70
9	104.10	97.90	9	105.65	99.30
Run No W.14			Run No W.17		
0	68.05	61.00	0	52.45	42.00
1	82.95	-	1	68.60	-
2	85.60	-	2	75.95	-
3	90.60	-	3	81.80	-
4	96.90	-	4	89.90	-
5	100.95	86.70	5	94.75	79.80
6	105.25	91.95	6	100.50	86.75
7	108.20	96.70	7	106.80	95.25
8	106.70	99.90	8	107.65	98.90
9	105.80	99.25	9	106.20	98.85
Run No W.15			Run No W.18		
0	54.00	45.90	0	79.90	72,30
1	70.05	-	1	93.00	-
2	76.35	-	2	94.05	-
3	81.20	-	3	98.70	-
4	88.65	-	4	104.20	-
5	93.50	80.80	5	106.30	91.80
6	100.85	87,35	6	109.50	95.30
7	105.95	95.45	7	109.80	99.20
8	108.90	99.90	8	105.20	99.20
9	105.90	99.30	9	105.70	97.90

$z\mathrm{x}10^{-2}$ mm	Temperature, °C t_W	t_L	$z\mathrm{x}10^{-2}$ mm	TEMPerature, °C t_W	t_L
Run No W.19			Run No. W.22		
0	75.10	66.50	0	84.05	75.75
1	89.80	-	1	99.30	-
2	91.70	-	2	100.55	-
3	97.10	-	3	103.50	-
4	103.10	-	4	108.90	-
5	105.80	90.30	5	110.30	92.90
6	110.00	94.00	6	111.90	96.20
7	111.60	99.30	7	109.20	99.30
8	106.30	99.30	8	106.80	99.30
9.	106.30	99.30	9	106.85	98.50
Run No W.20	$q : 19.547\ W/_m{}^2$		Run No W.23		
0	55.30	45.24	0	68.10	58.05
1	73.10	-	1	85.00	-
2	78.70	-	2	85.90	-
3	82.90	-	3	92.95	-
4	91.85	-	4	99.80	-
5	96.20	89.00	5	102.50	87.30
6	102.30	86.10	6	107.20	92.50
7	108.35	92.60	7	109.90	99.20
8	107.35	99.50	8	106.40	99.00
9	105.35	97.50	9	106.85	98.35
Run No. W.21 $q = 21.324\mathrm{x}10^3\ W/_m{}^2$			Run No. W.2 $q = 23.101\mathrm{x}10^3\ W/_m{}^2$		
0	57.70	47.55	0	55.25	44.10
1	76.15	-	1	75.00	-
2	79.50	-	2	80.55	-
3	85.30	-	3	85.20	-
4	93.00	-	4	91.75	-
5	97.70	82.15	5	98.90	
6	102.85	88.70	6	103.55	87.45
7	109.25	95.65	7	108.30	96.20
8	109.45	99.50	8	111.80	99.85
9	106.80	98.55	9	107.45	99.00

Run No W.25

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	82.25	74.45
1	99.15	-
2	99.70	-
3	103.20	-
4.	109.10	-
5.	110.55	93.45
6	112.50	94.70
7	109.60	100.70
8	109.65	99.05
9	107.35	99.05

$q = 24.878 \times 10^3$ W/m^2

Run No W. 26

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	56.00	44.25
1	76.80	-
2	81.80	-
3	87.30	-
4	94.00	-
5	99.20	81.20
6	104.40	88.90
7	109.90	97.25
8	111.90	98.30
9	108.70	98.30

Run No W.27

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	77.90	67.90
1	[illegible]	-
2	94.40	-
3	100.50	-
4	106.35	-
5	109.40	88.70
6.	111.90	94.50
7	109.50	99.70
8.	106.25	98.40
9	107.85	97.65

Run No. W.28

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	92.60	85.85
1	106.15	-
2	111.90	-
3	113.15	-
4	113.85	-
5	114.00	96.15
6	108.60	99.10
7	106.50	99.10
8	107.10	99.75
9	109.20	99.25

Run No. W. 29

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	77.50	68.50
1	95.80	-
2	97.50	-
3	100.50	-
4	107.75	-
5	109.60	90.05
6	110.75	94.00
7	109.30	97.45
8	108.90	99.60
9	108.75	98.75

Run No. W.30

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
0	68.40	58.60
1	87.20	-
2	90.00	-
3	94.60	-
4	101.50	-
5	104.20	86.20
6	109.30	90.80
7	110.60	95.20
8	109.55	99.45
9	108.30	98.60

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
Run No W.31		
0	62.05	50.30
1	81.50	-
2	85.20	-
3	89.60	-
4	96.15	-
5	100.55	82.40
6	105.40	87.60
7	11[illegible].10	94.15
8	110.80	98.70
9	108.40	98.70

$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L
Run No W.32		
0	83.60	75.05
1	100.75	-
2	104.35	-
3	104.35	-
4	111.10	-
5	112.40	93.60
6	111.05	97.10
7	108.80	99.45
8	107.20	99.45
9	108.50	98.85

Natural Circulation (Non-Boiling)

$z \times 10^{-2}$ mm	Temperature, °C		$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L		t_W	t_L
$q = 3.554 \times 10^3$ W/$_m$^{2}			$q = 5.331 \times 10^3$ W/$_m$^{2}		
Run No W.33			Run No. W. 34		
0	40.20	36.30	0	54.30	49.75
1	45.70	-	1	60.70	-
2	47.35	-	2	62.30	-
3	48.60	-	3	65.00	-
4	49.35	-	4	68.35	-
5	50.95	45.30	5	69.70	63.55
6	53.05	47.10	6	71.90	65.85
7	54.70	48.70	7	74.30	68.30
8	57.20	50.50	8	77.45	71.00
9	58.65	52.35	9	79.90	73.05

Table I.2 EXPERIMENTAL DATA OF HEAT TRANSFER TO WARIOUS SUGAR SOLUTIONS

$z\times10^{-2}$	Temperature, °C		$z\times10^{-2}$	Temperature, °C	
mm	t_W	t_L	mm	t_W	t_L
9.90 Wt% Sugar Solution					
$q = 7.108\times10^3 W/m^2$			$q = 17.777\times10^3 W/m^2$		
Run No. S.1			Run No. S.4		
0	52.10	45.55	0	83.30	76.70
1	63.10	-	1	97.30	-
2	70.30	-	2	101.10	-
3	79.45	-	3	102.50	-
4	87.55	-	4	108.45	-
5	91.75	84.80	5	109.15	94.40
6	100.40	93.95	6	111.90	97.65
7	106.10	99.95	7	108.45	99.40
8	106.30	98.90	8	104.70	99.20
9	103.70	98.50	9	104.70	97.70
$q = 10.662\times10^3 W/m^2$			$q = 21.324\times10^3 W/m^2$		
Run No. S. 2			Run No S.5		
0	68.10	61.30	0	88.10	81.25
1	79.90	-	1	105.15	-
2	84.05	-	2	108.35	-
3	88.85	-	3	108.80	-
4	95.40	-	4	114.70	-
5	99.05	88.30	5	115.15	97.65
6	104.40	93.10	6	114.10	106.20
7	108.90	99.35	7	108.25	101.70
8	103.60	99.35	8	106.40	100.20
9	103.10	98.50	9	108.00	99.75
$q = 14.216\times10^3 W/m^2$			$q = 24.878\times10^3 W/m^2$		
Run No S. 3			Run No S.6		
0	76.55	69.85	0	91.75	83.70
1	89.90	-	1	108.05	-
2	92.30	-	2	110.10	-
3	95.80	-	3	111.05	-
4	100.90	-	4	114.70	95.40
5	103.75	90.90	5	110.75	99.70
6	107.00	94.85	6	107.10	100.95
7	108.85	99.60	7	103.95	100.30
8	103.15	98.30	8	106.30	100.05
9	103.15	97.85	9	109.05	99.70

$zx10^{-2}$ mm	Temperature, °C		$zx10^{-2}$ mm	Temperature, °C	
	t_W	t_L		t_W	t_L
19.40 Wt% Sugar Solution					
$q = 7.108x10^3$ W/m^2			$q = 14.216x10^3$ W/m^2		
Run No S. 7			Run No. S.10		
0	56.25	49.30	0	78.20	69.95
1	67.50	-	1	91.80	-
2	74.25	-	2	95.50	-
3	81.35	-	3	98.35	-
4	88.95	-	4	103.35	-
5	94.20	85.75	5	107.50	92.80
6	101.95	95.60	6	111.45	98-05
7	108.70	101.15	7	111.20	102.70
8	106.35	100.45	8	106.20	100.50
9	103.85	99.30	9	106.00	99.85
$q = 8.885x10^3$ W/m^2			$q = 17.777x10^3$ W/m^2		
Run No S. 8			Run No S. 11		
0	57.10	49.60	0	82.90	75.10
1	69.40	-	1	98.45	-
2	76.00	-	2	102.05	-
3	83.15	-	3	104.80	-
4	90.60	-	4	110.95	-
5	94.95	87.40	5	113.70	96.40
6	101.40	94.55	6	110.55	100.85
7	108.70	101.25	7	104.80	99.75
8	109.35	99.75	8	104.80	98.90
9	104.70	99.15	9	105.65	98.25
$q = 10.662x10^3$ W/m^2			$q = 21.224x10^3$ W/m^2		
Run No S. 9			Run No. S. 12		
0	66.00	58.05	0	90.70	81.65
1	78.85	-	1	106.95	-
2	85.30	-	2	110.85	-
3	90.90	-	3	113.40	-
4	98.35	-	3	117.60	95.50
5	101.75	88.50	5	117.60	99.50
6	107.65	95.25	6	111.50	101.35
7	111.15	102.50	7	109.25	101.10
8	105.85	100.35	8	107.40	101.20
9	104.40	98.85	9	108.00	99.75

$z\times10^{-2}$ mm	Temperature, °C t_W	t_L	$z\times10^{-2}$ mm	Temperature, °C t_W	t_L
$q = 24.878\times10^3$ $W/_m{}^2$					
Run No S.13					
0	91.40	82.40			
1	108.80	-			
2	112.40	-			
3	114.50	-			
4	117.50	95.25			
5	116.30	98.50			
6	109.55	99.15			
7	106.00	99.75			
8	106.40	98.60			
9	108.30	98.40			

27.45 Wt% Sugar solution

$q = 7.108\times10^3$ $W/_m{}^2$			$q = 10.662\times10^3$ $W/_m{}^2$		
Run No S.14			Run No. S.16		
0	54.90	47.90	0	64.10	56.50
1	67.55	-	1	77.80	-
2	75.50	-	2	84.85	-
3	81.50	-	3	89.95	-
4	89.50	-	4	96.25	-
5	95.30	85.40	5	100.45	86.80
6	103.40	94.45	6	107.45	94.00
7	109.05	102.70	7	112.25	99.80
8	112.00	101.45	8	112.50	98.45
9	104.80	98.90	9	106.20	98.55
$q= 8.885\times10^3$ $W/_m{}^2$			$q = 12.439\times10^3$ $W/_m{}^2$		
Run No S.15			Run No S. 17		
0	66.10		0	73.55	65.60
1	78.20	-	1	89.20	-
2	85.30	-	2	94.60	-
3	89.60	-	3	99.90	-
4	97.20	-	4	105.20	-
5	101.70	87.50	5	107.95	92.10
6	106.90	93.30	6	113.00	96.50
7	111.75	99.40	7	114.30	100.95
8	115.15	99.20	8	106.50	98.60
9	108.30	99.00	9	104.45	97.90

$q = 17.777 \times 10^3$ W/m^2

Run No S. 18

$z \times 10^{-2}$ mm	Temperature, °C t_W	t_L
0	79.75	71.10
1	97.80	-
2	103.75	-
3	108.25	-
4	112.50	91.70
5	115.25	95.20
6	116.45	100.35
7	111.45	100.35
8	107.40	98.85
9	106.95	98.85

$q = 21.324 \times 10^3$ W/m^2

Run No S. 19

$z \times 10^{-2}$ mm	Temperature, °C t_W	t_L
0	87.60	77.50
1	107.40	-
2	113.50	-
3	113.20	-
4	118.75	94.80
5	118.95	99.45
6	109.05	100.50
7	104.80	99.50
8	105.40	99.30
9	108.00	99.30

$q = 24.878 \times 10^3$ W/m^2

Run No S. 20

$z \times 10^{-2}$ mm	Temperaturem °C t_W	t_L
0	93.00	83.60
1	111.35	-
2	115.25	-
3	115.10	-
4	116.80	-
5	116.20	98.60
6	116.20	100.60
7	114.10	100.10
8	109.70	99.75
9	109.70	99.25

43.60 Wt% Sugar solution

$q = 7.108 \times 10^3$ W/m^2

Run No S. 21

$z \times 10^{-2}$ mm	t_W	t_L
0	56.55	48.15
1	70.40	-
2	77.50	-
3	84.10	-
4	91.75	-
5	96.70	85.65
6	104.05	92.60
7	110.65	98.55
8	107.05	100.05
9	102.70	98.50

$q = 8.885 \times 10^3$ W/m^2

Run No S. 22

$z \times 10^{-2}$ mm	t_W	t_L
0	67.50	59.50
1	81.75	-
2	86.70	-
3	93.40	-
4	99.55	-
5	103.15	87.50
6	108.60	95.80
7	113.90	102.40
8	112.20	102.60
9	105.25	100.00

$zx10^{-2}$ mm	Temperature, °C t_W	t_L
$q = 10.662x10^{3}\ W/_{m}{}^{2}$		
Run No S. 23		
0	73.40	64.05
1	87.90	-
2	93.30	-
3	98.25	-
4	102.70	-
5	106.25	90.04
6	112.40	94.40
7	115.30	101.90
8	111.30	101.90
9	105.20	99.90
$q = 14.216x10^{3}\ W/_{m}{}^{2}$		
Run No S. 24		
0	80.25	71.35
1	96.65	-
2	102.75	-
3	107.20	-
4	111.45	91.15
5	113.35	94.60
6	117.40	99.30
7	117.40	102.50
8	110.70	99.70
9	106.70	99.70

$zx10^{-2}$ mm	Temerature °C t_W	t_L
$q= 17.777x10^{3}\ W/_{m}{}^{2}$		
Run No S. 25		
0	89.60	79.45
1	108.50	-
2	112.45	-
3	112.50	-
4	115.55	94.50
5	118.25	97.10
6	116.20	102.20
7	108.00	101.60
8	105.85	99.70
9	107.10	99.45
$q =21.324x10^{3}\ W/_{m}{}^{2}$		
Run No S. 26		
0	92.50	
1	113.15	-
2	119.20	-
3	119.00	-
4	119.00	100.35
5	116.95	101.25
6	108.55	101.25
7	106.20	100.40
8	107.30	100.00
9	106.65	99.75

57.50 Wt% Sugar Solution

$q = 7.108x10^{3}\ W/_{m}{}^{2}$		
Run No S.27		
0	66.25	55.35
1	80.35	-
2	87.90	-
3	92.90	-
4	99.75	-
5	103.40	85.65
6	110.40	94.20
7	-	101.10
8	105.00	100.40
9	-	99.15

$q = 8.885x1\ 0^{3}\ W/_{m}{}^{2}$		
Run No S. 28		
0	72.05	60.80
1	87.70	-
2	94.15	-
3	99.85	-
4	106.05	-
5	109.25	93.45
6	115.30	94.95
7	120.10	104.50
8	116.80	102.25
9	106.70	99.65

$z \times 10^{-2}$ mm	Temperature, °C		$z \times 10^{-2}$ mm	Temperature, °C	
	t_W	t_L		t_W	t_L
$q = 10.662 \times 10^3$ W/m^2			$q = 21.324 \times 10^3$ W/m^2		
Run No S. 29			Run No S. 31		
0	79.65	67.65	0	99.70	84.05
1	94.90	-	1	122.35	-
2	101.30	-	2	123.95	-
3	106.65	-	3	121.65	-
4	112.50	90.80	4	122.50	99.35
5	113.35	94.30	5	118.50	101.50
6	119.75	98.80	6	112.30	103.90
7	121.45	105.75	7	109.70	102.40
8	116.40	103.20	8	110.50	101.50
9	107.75	100.45	9	111.80	101.50
$q = 14.216 \times 10^3$ W/m^2			$q = 24.878 \times 10^3$ W/m^2		
Run No S. 30			Run No S. 32		
0	86.00	73.40	0	101.70	87.80
1	105.05	-	1	123.90	-
2	112.50	-	2	123.90	-
3	117.50	-	3	120.15	-
4	122.15	95.00	4	121.00	98.00
5	123.60	96.90	5	121.00	99.25
6	124.90	101.40	6	116.45	103.30
7	117.75	103.50	7	110.15	102.50
8	110.30	101.40	8	111.50	102.00
9	108.90	101.40	9	113.40	101.80

APPENDIX II

PHYSICAL PROPERTIES OF TEST LIQUIDS

Table II. 1 THERMAL CONDUCTIVYTY (Kcal/hr-m-°c) OF AQUEOUS SUGAR SOLUTIONS(18)

Temperature °C	Sugar concentration Wt% Sugar in Aqueous Solution 0	10	20	30	40	50	60
0	0.486	0.468	0.434	0.407	0.381	0.355	0.329
10	0.501	0.474	0.447	0.420	0.393	0.336	0.339
20	0.515	0.487	0.460	0.431	0.404	0.376	0.348
30	0.528	0.500	0.471	0.442	0.413	0.386	0.357
40	0.540	0.511	0.482	0.452	0.423	0.394	0.360
50	0.551	0.522	0.492	0.461	0.432	0.402	0.373
60	0.561	0.531	0.501	0.470	0.440	0.410	0.379
70	0.570	0.540	0.509	0.477	0.446	0.416	0.386
80	0.578	0.547	0.516	0.484	0.452	0.422	0.391

Table II.2 SPECIFIC HEAT OF AQUEOUS SUGAR SOLUTIONS (18)

kcal/kg-°C

Temperature °C	Sugar Concentrations Wt% Sugar 0	10	20	30	40	50	65
20	0.9990	0.9428	-	0.8299	-	0.7213	0.6416
99*	1.0077	0.9510	0.8943	0.8381	0.7830	0.7295	-

* The specific heat of sugar solutions at 99°C was estimated by assuming the similar variation with temperature as heat of water.

Table II.3 SPECIFIC GRAVITY OF AQUEOUS SUGAR SOLUTIONS IN $gm/_{ml}$ (18)

Wt% sugar in solution	Temperature, °C 0	10	15	20	25	30	40	50
0	0.9998	0.9997	0.9991	0.9982	0.9970	0.9956	0.9929	0.9881
9	1.0371	1.0359	1.0350	1.0340	1.0327	1.0310	1.1070	1.0230
10	1.0410	1.0401	1.0392	1.0380	1.0360	1.0350	1.0310	1.0270
20	1.0854	1.0823	1.0823	1.0800	1.0790	1.0770	1.0730	1.0680
27	1.1182	1.1158	1.1144	1.1128	1.1111	1.1092	1.1049	1.1000
28	1.1230	1.1205	1.1191	1.1175	1.1157	1.1138	1.0950	1.1047
46	1.2150	1.2110	1.2090	1.2070	1.2050	1.2030	1.1980	1.1930
57	1.2770	1.2730	1.2710	1.2680	1.2660	1.2640	1.2580	1.2530
58	1.2830	1.2790	1.2770	1.2740	1.2720	1.2690	1.2640	1.2590

Tbale II.4 VISCOSITY OF AQUEOUS SUGAR SOLUTIONS IN centi-poise (18)

Wt% Sugar in solution	Temperature, °C								
	0	10	20	30	40	50	60	70	80
20	3.806	2.658	1.957	1.501	1.190	0.970	0.810	0.680	0.590
27	5.577	3.791	2.725	2.048	1.595	1.280	1.050	0.880	0.750
28	5.927	4.012	2.873	2.152	1.671	1.340	1.100	0.920	0.780
43	19.710	12.060	7.922	5.504	4.003	3.020	2.360	1.890	1.540
44	21.910	13.270	8.639	5.958	4.306	3.230	2.510	2.000	1.630
57	133.200	66.860	37.230	22.510	14.540	8.910	7.060	5.220	3.990
58	160.100	78.700	43.100	25.700	16.410	11.100	7.820	5.740	4.350

Table II.5 VOLUME FACTOR FOR THERMAL EXPENSION OF AQUEOUS SUGAR SOLUTIONS 1/°C (18)

Temperature °C	Wt% Sugar Solution 10	20	30	40	50	60
0	0.9969	0.9950	0.9940	0.9930	0.9920	0.9920
10	0.9980	0.9980	0.9970	0.9960	0.9960	0.9960
20	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
30	1.0027	1.0029	1.0030	1.0030	1.0000	1.0030
40	1.0063	1.0063	1.0072	1.0072	1.0072	1.0082
50	1.0106	1.0110	1.0110	1.0110	1.0100	1.0100
60	1.0150	1.0160	1.0160	1.0160	1.0170	1.0170
70	1.0210	1.0210	1.2220	1.0220	1.0190	1.0180
80	1.0270	1.0270	1.0280	1.0280	1.0280	1.0270
90	1.0342	1.0342	1.0342	1.0342	1.0342	1.0360
100	1.0417	1.0417	1.0417	1.0417	1.0400	1.0390

PHYSICAL PROPERTIES OF TEST LIQUIDS:-

During the experimental investigations, a wide variation in wall and liquid temperatures along tube length was found to exist. The physical properties of liquids used were strongly temperature dependent. In order to calculate the physical property at certain location, the airthmatic average of wall and liquid temperature corresponding to the required location was taken. The physical properties which were not available in literature have been taken by extrapolation. A third degree poloynomial was fitted for available data to express the physical properties, except the viscosity of sugar solution, as a function of temperature. The equation is of the following form :-

$$P = A + Bt + Ct^2 + Dt^3$$

The viscosity of sugar solution was represented by a plynomial in exponential form :-

$$P = EXP\ (A + Bt + Ct^2 + Dt^3)$$

where P=Property and t = Temperature, °C

The values of physical constants for water and sugar solutions for different concentrations are given in Table II.6

Table II.6A VALUES OF VARIOUS CONSTANTS

System	Temperature Range - °C	Heat Capacity kcal/kg-°C	Thermal conducity kcal/hr-m-°C	Viscosity kg/hr-m	Various Constants
Water	10-100	1.0000	471.5266×10^{-3}	593.0184×10^{-2}	A
		1.0000	245.1467×10^{-5}	-141.5104×10^{-3}	B
		1.0000	-166.3855×10^{-7}	155.0054×10^{-5}	C
		1.0000	370.2873×10^{-10}	-628.9977×10^{-8}	D
			kcal/hr-m-C	centi-poise	
9.90 Wt% Sugar	10-80	-	459.8811×10^{-3}	-	A
		-	144.3217×10^{-5}	-	B
		-	-357.6192×10^{-8}	-	C
		-	$-105.3501 \times 10^{-10}$	-	D
19.45 wt% sugar	10-80	-	433.8772×10^{-3}	109.4227×10^{-2}	A
		-	136.4845×10^{-5}	-220.6773×10^{-4}	B
		-	-379.1181×10^{-8}	-584.3888×10^{-7}	C
		-	$-541.8113 \times 10^{-11}$	102.1892×10^{-8}	D

Table II.6 (contd..)

System	Temperature Range °C	Coefficient of Thermal Expansion 1/°C	Density gm/cc	Kinematic viscosity m^2/hr	Various Constants
27.45 Wt% Sugar	10.80	994.5314×10^{-3}	111.6018×10^{-2}	-	A
		235.8817×10^{-6}	952.3837×10^{-6}	-	B
		139.0786×10^{-8}	553.8784×10^{-7}	-	C
		118.1173×10^{-10}	587.3118×10^{-9}	-	D
43.60 Wt% Sugar	10-80	992.5391×10^{-3}	121.4665×10^{-2}	-	A
		371.9929×10^{-6}	$-\ 363.9016 \times 10^{-6}$	-	B
		$-\ 139.3396 \times 10^{-8}$	$-\ 417.4133 \times 10^{-9}$	-	C
		288.5400×10^{-10}	$-\ 199.8520 \times 10^{-10}$	-	D
57.50 Wt% Sugar	10-80	993.0273×10^{-3}	128.4237×10^{-2}	-	A
		286.2843×10^{-6}	$-\ 516.4536 \times 10^{-6}$	-	B
		231.0196×10^{-8}	113.4867×10^{-8}	-	C
		$-\ 718.2364 \times 10^{-11}$	$-\ 197.3918 \times 10^{-10}$	-	D

Table III.6B VALUES OF VARIOUS CONSTANTS

System	Temperature Range-°C	Coefficient of Thermal Expansion 1/°C	Density gm/cc	Kinematic viscosity m^2/hr	Various Constants
Water	10-100	-127.2494×10^{-6}	-	592.6088×10^{-5}	A
		196.1620×10^{-7}	-	-141.3311×10^{-6}	B
		-207.7249×10^{-9}	-	156.6153×10^{-8}	C
		998.3920×10^{-12}	-	-644.8154×10^{-11}	D
9.90 Wt% Sugar	10-80	996.7269×10^{-3}	104.2233×10^{-2}	-	A
		878.9942×10^{-7}	-208.7000×10^{-6}	-	B
		386.6359×10^{-8}	-135.6518×10^{-9}	-	C
		-288.0203×10^{-11}	-377.0776×10^{-10}	-	D
19.45 Wt% Sugar	10-80	995.0388×10^{-3}	108.2181×10^{-2}	-	A
		180.2655×10^{-6}	106.9090×10^{-6}	-	B
		278.5425×10^{-8}	-112.4489×10^{-7}	-	C
		-561.9922×10^{-12}	689.2065×10^{-10}	-	D

Table III.6 (contd..)

System	Temperature Range -°C	Heat capacity kcal/kg-°C	Thermal condutivity kcal/hr-m-°C	Viscosity centipoise	Various Constants
27.45 W% Sugar	10-80	-	408.1656×10^{-3}	176.8781×10^{-2}	A
		-	119.8795×10^{-5}	-407.6391×10^{-4}	B
		-	-210.7667×10^{-8}	276.2358×10^{-6}	C
		-	$-130.3613 \times 10^{-10}$	-102.2107×10^{-8}	D
43.60 Wt% Sugar	10-80	-	382.6657×10^{-3}	316.6971×10^{-2}	A
		-	104.7118×10^{-5}	-640.7795×10^{-4}	B
		-	382.0499×10^{-9}	726.0484×10^{-6}	C
		-	$-332.4242 \times 10^{-10}$	-464.6417×10^{-8}	D
53.50 Wt% Sugar	10-80	-	332.5200×10^{-3}	505.5615×10^{-2}	A
		-	659.8311×10^{-6}	-744.3781×10^{-4}	B
		-	480.1901×10^{-8}	518.5761×10^{-6}	C
		-	$-483.4761 \times 10^{-10}$	-185.8846×10^{-8}	D

APPENDIX III

SAMPLE CALCULATIONS

III.1 GENERAL

(a) Calculation of heat transfer surface :-

Inside diameter of test section = 0.01902m

Length of heated section = 0.94000m

Heat transfer area A = x.01902x0.94

$= 0.05618 m^2$

(b) Calculation of heat flux :-

$q = Q/_A = Q/_{.0.05618}\ W/_{m}{}^2$

$= 17.777\ Q.\ W/_{m}{}^2$

where Q is in watts

(c) Calculation of temperature drop across the tube wall of test section:-

The eqution of conductive heat transfer through a cylindrical test surface gives the temperature drop across the wall as below :-

$$t_W = \frac{qd_o}{2k} \ln \frac{d_o}{d_i}$$

d_i = Inside diameter of tube

d_o = I. d+2xWall thickness between inner surface and thermocouple bead.

for No run noW1 for water

$z = 0.70m, t_W = 103.70°C$

at this temperature k, for copper (19) = 322 $kcal/_{m\text{-}hr\text{-}°C}$

$$t_W = \frac{17.777x10^3 x0.86x0.0206}{2x322} \ln \frac{0.0206}{0.01902}$$

$= 0.03875°C$

III. 2 Experimental And Predicted Values of Natural Convective Nusselt Number for Water:-

Run No W.21

$q = 21.324x10^3\ W/_{m}{}^2$ $t_W = 97.50°C$

z= 500 mm $t_L = 82.00°C$

Average temperature= $\frac{97.50+82.00}{2} = 89.75°C$

Physical properties at average temperature:-

$k = 58.5 \times 10^{-2}$ kcal/m-hr-°C

$\rho = 965.3$ kg/m^3

$C_p = 1.005$ kcal/kg-°C

$= 0.326 \times 10^{-6}$ m^2/sec

$= 6.95 \times 10^{-4}$ 1/°C

$$N_{pr} = \frac{C_p \mu}{k} = \frac{1.005 \times 1.001 \times 3.6}{58.5 \times 10} = 10.251$$

$$N_{Gr} = \frac{g D^3 \beta \Delta t}{\nu^2} = \frac{9.81 \times 1 \quad (.019)^2 \times 6.95 \times 10^{-4} \times (92.50-82.00)}{(0.326 \times 10^{-6})^2}$$

$$= 6.82 \times 10^5$$

$$h = \frac{21.324 \times 10^3 \times 0.86}{(92.50-82.00)} = 1183.135 \text{ kcal/m}^2\text{-hr-°C}$$

$$N_{Nu} = \frac{hD}{k} = \frac{1183.135 \times .019}{58.5 \times 10^{-2}} = 41.169$$

Computed Nusselt number from correlation :-

$$N_u = 8.10 \times 10^5 \, (G_r \, P_r)^{-0.624}$$

$$= 8.10 \times 10^5 \, (6.9949 \times 10^6)^{-0.624}$$

$$= 43.326$$

Thus % error $= \dfrac{43.326-41.169}{41.169} \times 100 = +\,5.2387\%$

III.3 EXPERIMENTAL and Predicted Values of Natural Convective Nusselt Number For Sugar Solution of (19.45 and 27.40) Wt% Sugar

run no S.7

$z = 200$ mm $t_W = 74.25$°C

$Q = 7.108 \times 10^3$ W/m^2 $t_L = 64.40$°C

Average Temperature $= \dfrac{74.25+64.40}{2} = 69.30$°C

Physical properties at this temperature:-

$k = 0.509$ kcal/hr-m°C

$= 1.094\text{x}10^3\ \text{kg}/_{\text{m}}$

$C_p = .8381\ \text{kcal}/_{\text{kg}-°\text{C}}$

$= 0.88$ centi-poise

$= 1.022\ \ 1/_{°\text{C}}$

$$N_{Pr} = \frac{C_p\mu}{k} = \frac{0.8381\text{x}0.88\text{x}3.6}{0.509} = 3.084$$

$$N_{Gr} = \frac{gD^3\beta\Delta t}{\nu^2} = \frac{(1.094\text{x}10^3)^2\text{x}9.81\text{x}(.019)^3\text{x}1.022\text{x}(74.25-64.40)}{(0.88\text{x}3.6)^2}$$

$$= 3.69\ \text{x}\ 10^9$$

$$h = \frac{7.108\text{x}10^3\text{x}0.86}{(74.25-64.40)} = 620.589\ \text{kcal}/_{\text{hr}-\text{m}^2-°\text{C}}$$

$$N_{Nu} = \frac{hD}{k} = \frac{620.589\text{x}0.019}{0.509} = 23.594$$

Computed Nusselt number from Correlation:-

$$N_u = 0.913(Gr\ Pr)^{0.15}(Pr)^{-0.26} = 0.913(1.0019\text{x}10^{10})^{-.15}(3.084)^{-.26}$$

$$= 21.800$$

Thus % error in Nusselt number $= \frac{21.800 - 23.594\text{x}100}{23.594}$

$= -\ 7.5845\%$

III.4 Experimental And Predicted Values of Natural Convective Nusselt Number For 43.60 and 57.50 Wt% Sugar Solutions

run number S.21

$Q = 7.108\text{x}10^3\ \text{W}/_{\text{m}^3}$ $\qquad t_W = 77.40°\text{C}$

$z = 200\text{mm}$ $\qquad t_L = 63.40°\text{C}$

Average Temperature $= \frac{77.40+63.40}{2} = 70.40°\text{C}$

Physical properties at average temperature:-

$k = 0.446\ \text{kcal}/_{\text{m}-\text{hr}-°\text{C}}$

$\rho = 1.187\text{x}10^3\ \text{kg}/_{\text{m}^3}$

$C_p = 0.7830\ \text{kcal}/\text{kg}-°\text{C}$

$\mu = 2.80$ centi-poise

$\beta = 1.022\ \ 1/_{°\text{C}}$

$$N_{Pr} = \frac{C_P \mu}{k} = \frac{0.7830 \times 2.00 \times 3.6}{0.446} = 58.216$$

$$N_{Gr} = \frac{g D^3 \beta \Delta t}{\nu^2} = \frac{(1.187 \times 10^3)^2 \times 9.81 \times (0.019)^3 \times 1.022 \times 14}{(2 \times 3.6)^2}$$

$$= 1.65 \times 10^7$$

$$h = \frac{7.108 \times 10^3 \times 0.86}{14} = 424.50 \quad \text{Kcal}/m^2 \cdot hr \cdot {}^\circ C$$

$$N_{Nu} = \frac{hD}{k} = \frac{424.50 \times 0.019}{0.446} = 17.979$$

Nusselt number calculated from correlation :-

$$N_u = 9.6334\,(Gr\ Pr)^{-.0033}\,(P_r)^{0.17}$$

$$= 9.6334\,(9.6239 \times 10^8)^{-.0033}\,(58.216)^{0.17}$$

$$= 17.864$$

Thus % error in Nusselt number $= \frac{17.864 - 17.979 \times 100}{17.979}$

$= -0.63973\%$

III. 5 Experimental And Predited Values of Heat Transfer Coefficient For Boiling of Water

run no. W.18

$z = 800mm$ $\qquad t_W = 104.70\,{}^\circ C$

$Q = 17.777 \times 10^3\ W/_m{}^2$ $\qquad t_L = 98.50\,{}^\circ C$

$z_S = 700\ mm$ $\qquad (\Delta t)_{sub} = 26.80\,{}^\circ C$

Physical properties at saturation temperature -

$\rho_L = 958.4\ kg/_m{}^3$

$\rho_V = 0.598\ kg/m^3$

$C_p = 1.00$ centi-poise

$\lambda = 539\ kcal/_{kg}$

$$h_B = \frac{17.777 \times 10^3 \times 0.86}{(104.70 - 98.50)} \doteq 2351\ kcal/_{m^2}\text{-hr-}{}^\circ C$$

$(h_C)_{avg} = 1.032 \times 10^3\ kcal/_{m^2} - hr.\,{}^\circ C$

$$\frac{h_B}{h_C} = 2351/1.032\times10^{3} = 2.276$$

$$\left(\frac{t_L}{t_W}\frac{z}{d}\right) = \frac{104.70\times800}{98.00\times19} = 339.511$$

$$\left(\frac{\rho_L}{\rho_V}\frac{C_p d\cdot(\Delta t)_{sub}}{z_s}\right) = \frac{958.40\times1.00\times19\times26.80}{0.598\times539\times700}$$

$$= 2.162$$

Thus heat transfer coefficient calculated from correlation :-

$$\frac{h_B}{h_C} = 7.5\times10^{-5}\left(\frac{\rho_L}{\rho_V}\frac{C_p\; d\; (\Delta t)_{sub}}{z_s}\right)^{-.091}\left(\frac{t_L}{t_W}\frac{z}{d}\right)^{2.79}$$

$$= 7.5\times10^{-5}(2.162)^{-.091}(39.511)^{2.79}$$

$$= 1.9580$$

% error in $\frac{h_B}{h_C} = \frac{1.9580-2.276}{2.276} \times 100$

$= -13.97\%$

III.6 Experimental And Predicted Values of Heat Transfer Coefficient for Boiling of Sugar Solutions

run no. W. S.7

z = 900mm

$Q = 7.108\times10^{3}$ W/m^2

z_s = 700mm (

t_W = 104.00°C

t_L = 100.00°C

$(\Delta t)_{sub}$ = 44.00°C

Physical properties at saturation temperature:-

$\rho_L = 1.0927\times10^{3}$ kg/m^3

$\rho_V = 0.598$ kg/m^3

$\lambda = 539$ kcal/kg

$C_p = 0.8943$ kcal/kg-°C

$$h_B = \frac{7.108\times10^{3}\times0.86}{(104.00-100)} = 1528 \text{ kcal}/{}_{m^2}\text{-hr-}^{\circ}C$$

$$(h_C)_{avg} = 665.00 \text{ kcal}/{}_{m^2}\text{-hr } ^{\circ}C$$

$$\frac{h_B}{h_C} = 1528/665 = 2.295$$

$$\frac{t_L}{t_W}\frac{z}{d} = \frac{104.00\times 900}{100.00\times19} = 45.54$$

$$\frac{\rho_L}{\rho_V} C_p \frac{d(\Delta t)_{sub}}{z_B} = \frac{1092.70\times0.8943\times44}{0.598\times539\times700}$$

$$= 3.62$$

Heat transfer coefficient calculated from correlation:-

$$\frac{h_B}{h_C} = 2.25\times10^{-4}\left(\frac{\rho_L}{\rho_V} C_p \frac{z_s}{d} (\Delta t)_{sub}\right)^{-1.386}\left(\frac{t_L}{t_W}\frac{z}{d}\right)^{2.93}$$

$$= 2.25\times10^{-4}(3.62)^{-1.386}(45.54)^{2.93}$$

$$= 2.7420$$

% error in h_B/h_C $= \frac{2.7420 - 2.2950}{2.2950}$

$= + 19.479\%$

APPENDIX IV

LISTING OF COMPUTER PROGRAMS

```
*************************************************************************
                                                  PROGRAM I
*************************************************************************
C  C  CALCULATION OF DIMENSIONLESS NUMBERS FOR SUGAR
C     D IS CONSTANT
C     N3 IS NUMBER OF SUGAR CONCENTRATIONS
C     CP IS SPECIFIC HEAT
C     QFLUX IS HEAT FLUX
C     TW IS WALL TEMERATURE
C     TL IS LIQUID TEMERATURE
C     AM IS MASS FLOW RATE
C     DELT IS DIFFERENCE OF WALL AND LIQUID TEMPERATURE
C     TM1 IS AVERAGE TEMPERATURE OF WALL AND LIQUID
C     AK IS THERMAL CONDUCTIVITY
C     AMU IS VISCOSITY
C     BETA IS COEFFICIENT OF VOLUME EXPANTION
C     R IS DENSITY
C     H IS HEAT TRANSFER COEFFICIENT
C     ANU IS NUSSELT NUMBER
C     RE IS REYNOLDS NUMBER
C     PR IS PRANDTL NUMBER
C     GR IS GRASHOF NUMBER
      DIMENSION TW(200),TL(200),A(20),B(20),C(20)
      D=4.*.3*.022/(0.0194*0.0194)
      READ,N3
      DO200 MM=1,N3
      READ,CONC
      PUNCH,CONC
      READ,(A(I),B(I),C(I),I=1,4)
      READ,N1,CP
      DO200K=1,N1
      READ,N2,QFLUX
      READ,(TW(J),TL(J),J=1,N2)
      DO 200 J=1,N2
      AM=D*QFLUX/((TL(4)-TL(1))*CP)
      DELT=TW(J)-TL(J)
      TM1=0.5*(TW(J)+TL(J))
      TM2=TM1*TM1
      AK=A(1)+B(1)*TM1+C(1)*TM2
      P=A(2)+B(2)*TM1+C(2)*TM2
      AMU=EXPF(P)*3.6
      BETA=A(3)+B(3)*TM1+C(3)*TM2
      R=(A(4)+B(4)*TM1+C(4)*TM2)*1.00E03
      H=QFLUX/DELT
      RBYMU=R/AMU
      ANU=.0194*H/AK
      RE=.0194*AM/AMU
      PR=CP*AMU/AK
      GR=9.3319E02*BETA*DELT*RBYMU*RBYMU
      PUNCH 202,ANU,RE,PR,GR,H,TW(J),TL(J),QFLUX
  202 FORMAT(8F10.3)
  200 CONTINUE
      STOP
      END
                                             DATA CARD FOLLOWS
*************************************************************************
```

```
******************************************************************
                                             PROGRAM II
******************************************************************
C  C  CALCULATION OF DIMENSIONLESS NUMBERS FOR WATER
C     CP IS SPECIFIC HEAT
C     QFLUX IS HEAT FLUX
C     D IS CONSTANT
C     TW IS WALL TEMERATURE
C     TL IS LIQUID TEMERATURE
C     DELT IS DIFFERENCE OF WALL AND LIQUID TEMPERATURE
C     AM IS MASS FLOW RATE
C     TM1 IS AVERAGE TEMPERATURE OF WALL AND LIQUID
C     AK IS THERMAL CONDUCTIVITY
C     BETA IS COEFFICIENT OF VOLUME EXPANTION
C     H IS HEAT TRANSFER COEFFICIENT
C     ANU IS NUSSELT NUMBER
C     CUBYR IS RATIO OF VISCOSITY AND DENSITY
C     RE IS REYNOLDS NUMBER
C     PR IS PRANDTL NUMBER
C     GR IS GRASHOF NUMBER
      DIMENSION TW(200),TL(200)
      D=4.*.3*.022/(0.0194*0.0194)
      READ,N1,CP
      DO200 K=1,N1
      READ,N2,QFLUX
      READ,(TW(J),TL(J),J=1,N2)
      DO 200 J=1,N2
      AM=D*QFLUX/((TL(4)-TL(1))*CP)
      DELT=TW(J)-TL(J)
      TM1=0.5*(TW(J)+TL(J))
      TM2=TM1*TM1
      AK=471.526E-03+245.146E-05*TM1-166.385E-07*TM2
      AMU=593.010E-02-141.51E-03*TM1+155.005E-05*TM2
      BETA= -127.249E-06+196.162E-07*TM1
      CUBYR=592.608E-05-141.0331E-06*TM1+156.615E-08*TM2
      H=QFLUX/DELT
      ANU=.0194*H/AK
      RE=.0194*AM/AMU
      PR=CP*AMU/AK
      GR=9.3319E02*BETA*DELT/CUBYR/CUBYR
      PUNCH 202,ANU,RE,PR,GR,H,TW(J),TL(J),QFLUX
 202  FORMAT (8F10.3)
 200  CONTINUE
      STOP
      END
                                  DATA CARD FOLLOWS
********************************************************************
```

```
****************************************************************************
                                     PROGRAM III
****************************************************************************
C     LEAST SQUARE METHOD CALCULATION FOR BEST CURVE FITTING
      DIMENSION A(15,15),X(15,200),XB(15),CO(15),Y(200),ER(200)
      READ1,N,M,IM
    1 FORMAT(3I5)
C
C     N IS NUMBER OF VARIABLES
C     M IS NUMBER OF DATA POINTS
C     IM=1 INDICATE THAT RELATIONSHIP IS OF TYPE Y=A*(X**B)*(Z**C)
C     IM=2 INDICATE THAT RELATION IS OF TYPE Y=A+B*X+C*Z
C
C     READING OF DATA POINT X(I,J)
C
    3 FORMAT(6E12.5)
      DO5 J=1,M
      READ3,(X(I,J),I=1,N)
      PRINT 3,(X(I,J),I=1,N)
    5 CONTINUE
C
C     CHECKING THE VALUE OF IM
C     IF IM=1 GOTO STATEMENT NO10
C     IF IM=2 GOTO STATEMENT NO 20
C
      IF(IM-1)10,10,20
C
C     TAKING THE LOGRATHIM OF GIVEN DATA (X(I,J))
C
   10 DO 15 I=1,N
      DO 15 J=1,M
   15 X(I,J)=ALOG(X(I,J))
C
C     MEAN VALUE CALCULATION XB, XB IS MEAN VALUE
C
   20 AM=M
      DO 50 I=1,N
      SUM=0.0
      DO 40 J=1,M
      SUM=SUM+X(I,J)
   40 CONTINUE
      XB(I)=SUM/AM
   50 CONTINUE
      PRINT55
   55 FORMAT( 5X, 11HMEAN VALUES//)
      PRINT3,(XB(I),I=1,N)
C
C     SUBSTRACTING MEAN VALUES XB(I) FROM DATA POINT X(I,J)
C
      DO 60 I=1,N
```

```
      DO 60 J=1,M
   60 X(I,J)=X(I,J)-XB(I)
C
C     COEFFICIENT ARE DENOTED BY A(I,K)
C     GENERATION OF COEFFICIENT OF EQUATION TO BE SOLVED
C
      N1=N-1
      DO 65 I=1,N1
      DO 65 K=I,N
      A(I,K)=0.0
      DO 65 J=1,M
      A(I,K)=A(I,K)+X(I,J)*X(K,J)
   65 CONTINUE
C
C     INVERSION OF MATRIX OF COEFFICIENT ,(A(I,K))
C
      DO 70 I=2,N1
      I1=I-1
      DO 70 J=1,I1
   70 A(I,J)=A(J,I)
      PRINT 71
   71 FORMAT( 5X, 24HCOEFFICIENT OF EQUATIONS//)
      DO 80 I=1,N1
      PRINT3,(A(I,J),J=1,N)
   80 CONTINUE
C
C     CALCULATION OF ROOTS OF EQUATION BY SUBROUTINE SIMEQ
C     EXP-VALUES OF UNKNOWNS ARE CALCULATED POWER OF RELATION
C
      CALL SIMEQ(A,N1,CO)
C
C     DETERMINATION OF POWERS AND CONSTANT OF RELATION
C     POWERS ARE DENOTED BY CO AND CONSTANT BY COI
C
      COI1=XB(N)
      DO 82 K=1,N1
      COI1=COI1-CO(K)*XB(K)
   82 CONTINUE
      IF(IM-1)83,83,84
   83 COI=EXP (COI1)
      GO TO 85
   84 COI=COI1
   85 PRINT 86
   86 FORNAT(5X,22HCONSTANTS OF RELATION//)
      PRINT87,COI
   87 FORMAT(5X,2HA=,F10.4)
      PRINT88,(CO(K),K=1,N1)
   88 FORMAT(2X,5HB(I)=, 8E12.5)
C
C     CHECKING THE RELATION WHICH WE GET THROUGH THIS METHOD
```

```
C     THIS MEANS THAT WE ARE CALCULATING NUSSELT NUMBER BACK
C
      DO90 K=1,N
      DO90 J=1,M
  90  X(K,J)=X(K,J)+XD(K)
      DO100 J=1,M
      Y(J)=CO(1)
      DO100 K=1,N1
      Y(J)=Y(J)+CO(K)*X(K,J)
 100  CONTINUE
      IF(IM-1)105,105,120
 105  DO115 J=1,M
      DO110 I=1,N
 110  X(I,J)=EXP (X(I,J))
      Y(J)=EXP (Y(J))
 115  CONTINUE
C
C     CALCULATION OF ERROR PERCENTAGE WHICH OCCUR IN CALCULATIONP
C     OF NUSSELT NUMBER BACK BY FINAL REATION
C
 120  DO130 I=1,M
      ER(I)=(Y(I)-X(N,I))*100./X(N,I)
 130  CONTINUE
      PRINT131
 131  FORMAT(5X,12HFINAL RESULT//)
      PRINT132
 132  FORMAT( 40H****************************************,1H*,
     139H***************************************)
      PRINT133
 133  FORMAT(1X,10HREYNOLD NO,3X,10HPRANDTL NO,2X,11HR=VISM/VISW,
     12X,10HNUSSELT NO,2X,10HNUSSELT NO,2X,10HERROR PER.)
      PRINT132
      DO 140 J=1,M
      PRINT888,(X(I,J),I=1,N),Y(J),ER(J)
  888 FORMAT(5X,8E12.5)
 140  CONTINUE
      PRINT132
      STOP
      END
      SUBROUTINE  SIMEQ(A,N,X)
      DIMENSION A(15,15),U(15,15),U1(15,15),X(15)
      M=N+1
      DO20 IT=1,N
      JT=1
    1 IF(JT-1)3,3,2
    2 J=IT+1
      I=IT
      GO TO 4
    3 I=IT
      J=IT
```

```
  4  U(I,J)=A(I,J)
     IF(IT-1)7,7,5
  5  M1=IT-1
     DO6K=1,M1
  6  U(I,J)=U(I,J)-U(I,K)*U(K,J)
  7  IF(JT-1)8,8,10
  8  I=I+1
     IF(I-N)4,4,9
  9  JT=2
     GOTO1
 10  U(I,J)=U(I,J)/U(I,I)
     J=J+1
     IF(J-M)4,4,20
 20  CONTINUE
     DO30 I=1,N
     DO30 J=1,M
     IF(I-J)25,27,27
 25  U1(I,J)=U(I,J)
     GOTO30
 27  U1(I,J)=0.0
 30  CONTINUE
     DO35 J=1,N
 35  X(J)=0.0
     N1=N
 40  I=N1
     X(I)=U1(I,M)
     DO45 J=1,N
 45  X(I)=X(I)-U1(I,J)*X(J)
     N1=N1-1
     IF(N1-1)47,40,40
 47  RETURN
     END
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