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**LA THÈSE A ÉTÉ
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AN ANALYSIS OF REWETTING PHENOMENA
IN VERTICAL CHANNELS

by

Andrew Ki-Hong Kim

A thesis submitted to the School of Graduate Studies in
partial fulfillment of the requirements of the degree of

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A B S T R A C T

An analytical model based on different flow regimes involved in the rewetting process of bottom flooding mode was established to predict the temperature-time trace curve at various axial locations of the vertical circular channel under the bottom flooding conditions, thus predicting the rewetting rate of hot vertical channel.

Also an empirical correlation of "apparent rewetting temperature" using dimensional analysis was attempted for vertical circular channels under bottom flooding conditions. The variables used in the analysis were K , ρ , C_p , d , D , T_w , T_c , G , Z , T_q , T_s , and . The effect of each variable on "rewetting temperature" is critically discussed. From the comparison of correlated T_q with measured values, it has shown that the correlated values fall within about $\pm 10\%$ of the experimental values.

In order to predict the temperature-time trace curve of rewetting process by computational method, the rewetting process was divided into six heat transfer regions namely, SINGLE PHASE STEAM, DISPERSED FLOW, INVERTED ANNULAR FLOW, TRANSITION BOILING REGION, NUCLEATE BOILING REGION and

SINGLE PHASE LIQUID REGION. In each heat transfer region, an equation to give appropriate values of heat transfer coefficient h was selected from various sources or modified.

In the computation, small elements were considered and in each element, coolant water is heated and its temperature and vapor quality as well as the wall temperature are calculated using appropriate heat transfer equations according to the flow region.

Computed temperature-time trace curves obtained using experimental "rewetting temperature" and correlated "rewetting temperature" were compared to those obtained from experiments and the results show very close resemblance among the three.

Rewetting velocity u_q is obtained from two temperature-time trace curves at different locations of the test tube by knowing the "rewetting time" and the location of the temperature-time trace curve, because rewetting velocity u_q is the propagation rate of the rewetting front along the test section in the direction of the coolant water flow. The result of rewetting velocity u_q shows that predicted rewetting velocity (using correlated T_q) mostly fall within $\pm 20\%$ of the experimental rewetting velocity.

Also the convective heat transfer coefficient h was calculated from experimental temperature-time trace curve and compared with h values used in analytical computation. The comparison shows that analytical h value for NUCLEATE BOILING REGION and SINGLE PHASE LIQUID REGION is higher than experimental h value and that analytical h value for TRANSITION BOILING REGION is lower than the experimental h value. Thus it suggests that the heat transfer correlations for various flow regimes used in the analysis is still to be explored or critically reexamined for the rewetting phenomena.

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N O M E N C L A T U R E

Ad	Channel cross-sectional area, cm^2
Cp	Specific heat of pipe material, $\text{J/Kg}^\circ\text{K}$
Cp _l	Specific heat of water, $\text{J/Kg}^\circ\text{K}$
d	Pipe wall thickness, m
D	Inside diameter of test tube, m
De	Wetted parameter, cm
f	Function
F	Reynolds number factor, Re/Re_1
G	Mass flow velocity, $\text{Kg}/\text{m}^2\cdot\text{s}$
gc	Gravitational constant
h	Heat transfer coefficient, $\text{cal}/\text{s cm}^2\text{K}$, $\text{w}/\text{m}^2\text{K}$
H _{fg}	Latent heat of vaporization, cal/g
K	Thermal conductivity of test tube, $\text{w}/\text{m}^\circ\text{K}$
K _l	Thermal conductivity of water, $\text{w}/\text{m}^\circ\text{K}$
K _{vw}	Thermal conductivity of vapour at wall temp. ($\text{cal}/\text{cm}\cdot\text{s C}$)
l	Length of element, cm
L	Distance from the rewetting front, cm
Nd	Droplet flux, $\#/\text{s}$
p	System pressure, psia
P	Power input per unit volume (heat generation rate), w/m^3
Pr	Prandtl number
ΔP	Difference in vapour pressure corresponding to ΔT, psf
q	Heat flux, w/m^2
q _{crit}	Critical heat flux, $\text{Btu}/\text{hr}\cdot\text{ft}^2$, w/m^2

q_{FB}	Film boiling heat flux, w/m^2
q_{NB}	Nucleate boiling heat flux, w/m^2
q_{TB}	Transition boiling heat flux, w/m^2
Re	Reynolds number
S	Slip ratio
S_p	Suppression factor, $(\Delta T_e/\Delta T)^{0.99}$
t	Time, sec.
T	Temperature, °K
T_{qs}	Temperature difference between rewetting temperature and saturation temperature, °C
T_{sc}	Temperature difference between coolant temperature and saturation temperature, °C
T_{ws}	Temperature difference between wall temperature and water saturation temperature, °C
ΔT	Superheat, $T_w - T_s$
U_q	Rewetting velocity, mm/s, m/s
V_L	Liquid droplet velocity, cm/s
Z	Axial distance from the inlet, m
α	Void fraction
δ	Droplet diameter, cm
ϵ	Emissivity
ϵ_{vc}	Emissivity of vapour at vapour temperature
ϵ_{vw}	Emissivity of vapour at wall temperature
μ	Viscosity, g/cm·s (poise)
μ_{vc}	Dynamic viscosity of vapour at vapour temperature

μ_{vw}	Dynamic viscosity of vapour at wall temperature
$\pi_1 \sim \pi_7$	Non-dimensional parameters
ρ	Mass density, Kg/m
ρ_{vw}	Density of vapour at wall temperature
σ	Stefan-Boltzman constant, $\frac{\text{cal}}{\text{cm}^2 \cdot \text{s} \cdot \text{K}^4}$
σ_s	Vapour-liquid surface tension, dynes/cm
x	Vapour quality

Superscripts

C	Convection
R	Radiation

Subscripts

c	Coolant
g	Value for vapour
l	Value for liquid
q	Rewetting
s	Value at saturation
v	Value for vapour
VL	From vapour to liquid
w	Value at wall condition
WL	From wall to liquid
WV	From wall to vapour

CHAPTER 1

INTRODUCTION

Rewetting refers to the phenomenon of establishment of liquid coolant contact with very hot solid surfaces. Recently, the interest in the problem has been intensified because the rewetting phenomenon is of particular importance during the reflooding phase of emergency core cooling conditions in the "Loss -of- Coolant Accidents" (LOCA) of water - cooled power reactors.

In a postulated "Loss -of- Coolant Accident" (LOCA), the reactor core may develop heat which would overheat the fuel elements beyond the safe design temperature limit before an alternative emergency cooling system could restore coolant flow. The surface temperature of the fuel bundles can be higher than the "Rewetting Temperature", and the heat transfer mechanism involved becomes quite complicated.

A considerable number of works on the rewetting subject have been published during the last decade, and recent reviews on both theoretical and experimental studies are made available by Butterworth et. al. [1] and Sawan et. al. [2]. Experimental studies were done by various workers using different geometries. Andreoni and Courtaud [3], Duffey and Porthouse [4],

Case et al. [5] and Piggott and Porthouse [6] used an annulus with a hot central tube. Campanile and Pozzi [7], Riedle and Winkler [8], Thompson [9] and Cadek et al. [10] used a rod bundle while Riedle and Winkler [8], and Martini and Premoli [11] used a single tube with wetting inside.

The prediction of the rewetting-front velocity by the various models based on one, two and three-dimensional analytical [4,12,13,16,17,18,19,20] and numerical studies [14,15] still requires the knowledge of the "rewetting temperature" and heat transfer coefficient distribution near the front. The "rewetting temperature" is also required to predict the temperature-time trace using a system approach. An incorrectly assumed value of "rewetting temperature" could lead to serious errors in the final analysis of the problem. However, very little work has been carried out to measure or predict the values of the "rewetting temperature", and both dry-region and wet-region heat transfer coefficients, directly. To make the theoretical approaches amenable to mathematical treatment, these three parameters are usually assumed or deduced from the steady-state pool boiling curves and most workers [1,2,4,7,8] assumed the "rewetting temperature" to be constant.

In the present thesis, the apparent "rewetting temperature" is empirically correlated from experimental data using

dimensional analysis for vertical circular channels under the bottom flooding conditions, and convective heat transfer coefficient h was calculated from experimental temperature-time trace curve and compared with the analytical values of h used in the computation.

A computer program was established to predict the temperature-time trace curve at various axial locations of the vertical circular channels under the bottom flooding conditions by dividing the rewetting process into six heat transfer regions and using appropriate heat transfer coefficient h for each heat transfer region. Computed temperature-time trace curves obtained using experimental "apparent rewetting temperature" and correlated "apparent rewetting temperature" were compared with experimental temperature-time trace curves.

Rewetting velocity u_q is obtained from two temperature-time trace curves at different locations of the test tube by knowing the "rewetting time" and the location of the temperature-time trace curve, because rewetting velocity u_q is the propagation rate of the rewetting front along the test section in the direction of the coolant water flow, defined as $u_{ij} = \frac{\Delta Z_{ij}}{\Delta t_{ij}}$ where u_{ij} is the rate of the rewetting front propagation between two particular thermocouples of interest, ΔZ_{ij} is the

axial distance between the thermocouples, and Δt_{ij} is the time interval taken between the two rewetting fronts.

CHAPTER 2

ANALYSIS

2.1 PHYSICAL MODEL

A simple model has been developed to predict the temperature-time curve of the test tube, starting from the time at which coolant, with a known flow rate, reaches test tube bottom, and terminating when the test tube is completely quenched. This will present a temperature-time trace curve at various axial locations of the vertical circular channel under the bottom flooding conditions.

There are many computer codes available, and review of the papers [21,22] show that their prediction of rewetting process is not closely compatible to experimental results. It was noted that use of incorrectly assumed value of rewetting temperature seems to be one of the main sources of error in their analysis. Martini and Premoli [21] used a constant value of 420°C as the prefixed rewetting temperature while Clements et al. [22] used 280°C . The present model was mainly proposed to check the validity of the concept of apparent rewetting temperature, T_q , that has been correlated, and also to see the importance of including appropriate

heat transfer coefficients for various flow regimes with the apparent rewetting temperature in the analysis.

The rewetting velocity u_q is obtained from two temperature-time trace curves at different locations of the test tube by knowing the "rewetting time" and the location of the temperature-time trace curve, because the rewetting velocity u_q is the propagation rate of the rewetting front along the test section in the direction of the coolant water flow.

In the computation, test tube is subdivided in many axial nodes. The thermal transient of each node is calculated from the energy balance made. Wall temperature, when reaches a value of the apparent rewetting temperature, T_q , prefixed in input and when vapor quality is less than 0.7, drops very sharply to near saturation temperature by nucleate boiling and transition boiling heat transfer.

In connection with adopted heat transfer scheme, a satisfactory evaluation of heat transfer coefficient and rewetting temperature is required, since they strongly affect rewetting front velocity. The values of the apparent rewetting temperature used in this model are either derived from experiments, or calculated from rewetting temperature correlation equation obtained from our experiment.

The computer program was written in FORTRAN-IV language for IBM-360/computer. This model is applicable to the emergency core cooling system which injects cold coolant to core bottom in the final phase of the loss of coolant accident.

2.2 REWETTING TEMPERATURE, T_q

2.2.1; DEFINITION OF REWETTING TEMPERATURE

The "rewetting temperature" is the maximum temperature where the actual contact between the coolant liquid and the surface is possible and the wet region heat transfer process starts. The "rewetting temperature" is also known as calefaction, sputtering, quenching temperature or the temperature corresponding to the minimum boiling heat flux. However, it is extremely difficult to determine the exact "rewetting temperature" in a given temperature-time trace.

For present study, the apparent rewetting temperature was found by taking the intersection of the gradients of the trace before the point where there is a rapid fall in temperature and at the steepest portion of the fall as shown in Fig. 2.1. This technique is also used by other workers [6,17,21], and corresponds to the definition of "upstream wall temperature" unaffected by axial conduction as used by Yu.

et al. [23]. The apparent rewetting temperature so defined cannot be the Leidenfrost temperature (the minimum heated surface temperature required to just sustain the stable film boiling process). Therefore, the prediction of the "rewetting temperature" based on hydrodynamic theories such as proposed by Berenson [24], and on thermodynamic theory such as proposed by Spiegler et al. [25], cannot be applicable for the "apparent rewetting temperature".

Literature survey showed that most workers assumed the "rewetting temperature" to be constant. For example, for the "rewetting temperature" of water, T_q , Yamanouchi [17], chose 150°C, Dua and Tien [26], 260°C, Yu et al. [23], 80°C above the saturation temperature of the coolant and Martini et al. [21], 420°C, Cadek et al. [10], reported values between 284 and 541°C from their experimental observation. However, it was shown by Lee et al. [27], that "rewetting temperature" is not constant and depends on the physical properties of the test section and coolant flow condition.

2.2.2: DIMENSIONAL ANALYSIS

To correlate the "apparent rewetting temperature", a dimensional analysis was used.

The rewetting velocity, u_q , and the "effective heat transfer coefficient", h , in the rewetting region are not independent variables, and it is assumed that their effect on the "apparent rewetting temperature", T_q , is negligible. Therefore, in the present study, the following 12 variables, C_p , D , G , K , P , T_c , T_q , T_s , T_w , Z , d and ρ are essential to the problem and have some effect on T_q . This is deduced from experimental program of Lee and Chen [27] as well as from those of other investigators [6,17,21].

The coolant saturation temperature, T_s , has a constant value throughout the experiment, because all the experimental result used in the correlation were obtained at the system pressure of about one atmosphere. Therefore, three new variables, $T_{sc} = T_s - T_c$, $T_{ws} = T_w - T_s$, and $T_{qs} = T_q - T_s$ were introduced to replace four temperatures, T_w , T_c , T_q and T_s .

These 11 variables must have a functional relationship

$$F(C_p, D, G, K, P, T_{qs}, T_{ws}, Z, d, T_{sc}, \rho) = 0 \quad (2.1)$$

Using Buckingham's π theorem with K , d , T_{ws} and G as the primary variables and employing four primary dimensions of mass, length, time and temperature, the following seven non-dimensional parameters were obtained from Eq. (2.1) :

$$f(\pi_1, \pi_2, \pi_3, \dots, \pi_7) = 0 \quad (2.2)$$

where

$$\begin{aligned}\pi_1 &= \frac{T_{qs}}{T_{ws}} & \pi_2 &= \frac{T_{sc}}{T_{wc}} & \pi_3 &= \frac{Cp \cdot G \cdot d}{K} \\ \pi_4 &= \frac{K \cdot \rho^2 \cdot T_{ws}}{d \cdot G^3} & \pi_5 &= \frac{P \cdot d^2}{K \cdot T_{ws}} & \pi_6 &= \frac{D}{d} \\ \pi_7 &= \frac{Z}{d}\end{aligned}$$

The functional relationship among these seven dimensionless parameters was found by Multiple Linear Regression Analysis to 470 experimental data points obtained from four different test sections.

The physical characteristics of the 4 test sections and the ranges of experimental parameters are listed in Tables 1 and 2 respectively.

2.3 FLOW REGIMES

In order to predict the temperature-time trace curve of rewetting process by analysis, the rewetting process was divided into six heat transfer regions as shown in Fig. 2.2. The six heat transfer regions are;

- 1 : SINGLE STEAM PHASE REGION
- 2 : DISPERSED FLOW REGION
- 3 : INVERTED ANNULAR FLOW REGION
- 4 : TRANSITION BOILING REGION
- 5 : NUCLEATE BOILING REGION
- 6 : SINGLE LIQUID FLOW REGION

In each heat transfer region, an equation to give appropriate values of heat transfer coefficient, h , was selected from various sources or modified.

2.3.1 SINGLE LIQUID FLOW REGION

This region is nothing more than a liquid forced-convection flow. The conventional convective equations can be used for predicting heat transfer behavior.

Convection heat transfer coefficient h varies depending on the flow conditions, which can be characterized by the Reynolds number Re . In a long tube, the flow could be assumed laminar when the Reynolds number is below about 2000. In the range of Reynolds number between 2000 and 10,000, the transition from laminar to turbulent flow is thought to be taken place. The flow in this regime is called transitional. At a Reynolds number of about 10,000 or over, the flow can be assumed to be fully turbulent in practice [29].

For laminar flow, convection heat transfer coefficient h is given as [30]

$$h = \frac{48}{11} \frac{K_L}{D} \quad (2.3)$$

For fully developed turbulent flow in smooth tubes for heating,

the following heat transfer coefficient is recommended by Dittus and Boelter [31]

$$h = 0.023 \frac{K_l}{D} Re^{0.8} Pr^{0.4} \quad (2.4)$$

A linear interpolation between laminar and turbulent heat transfer coefficient expression is made to give heat transfer coefficient for transitional flow where Re is between 2000 and 10,000. The heat transfer rate per unit surface area of tube, q_{wl} , is $q_{wl} = h (T_w - T_l)$, where T_w is wall temperature and T_l is water temperature. All the properties used in the equations are evaluated at the water temperature.

2.3.2 NUCLEATE BOILING REGION

This region (region 5 of Fig. 2.2) is also called saturated forced convection boiling region where heat transfer due to convection and nucleated boiling occurs simultaneously. Convection boiling is defined as being the addition of heat to a flowing liquid in such a way that progressive vaporization along a tube occurs and two phases of saturated water and steam coexist. Nucleate boiling is characterized by the formation of bubbles from fixed sites randomly distributed on the surface. Actually, the term nucleate boiling is derived from the fact that bubbles are formed at the nucleation sites. The presence of nucleate boiling gives rise to a considerable increase in the rate of heat transfer from the hot surface.

compared to the rate for normal single-phase mechanism of heat transfer.

One of the most successful of many proposed correlation for nucleate boiling heat transfer coefficient is that of Chen [32] based on the superposition principle where as the total heat flux is the sum of contribution from convection and bubble nucleation. The correlation developed by Chen [32] is at present considered by many authors [33,34] to be the best available for the saturated forced convective boiling regions and is recommended for use with all single component non-metallic fluids. The Chen correlation was shown to fit a wide range of data with fair accuracy. The range of data include those for water in the pressure range of 1 to 35 atm. with liquid flow velocities up to 14.7 ft/sec. (4.41 m/s) heat flux up to 760,000 Btu/hr.ft² (2,395,885 watt/m²) and quality up to 71%. Other fluids included are methanol-cyclohexane, pentane, heptane, and benzene, under 1 atm. pressure, liquid velocities of .1 to 3 ft/sec, a heat flux range of 2,000 to 17,000 Btu/hr.ft² (6,305 to 53,592 watt/m²) and quality up to 12% [33].

The correlation was proposed to account for both the microconvection due to boiling and macroconvection due to flow.

$$h = h_{mic} + h_{mac} \quad (2.5)$$

where

$$h_{mic} = 0.00122 \frac{K_l^{0.79} C_{p_l}^{0.45} \rho_l^{0.49} g_c^{0.25}}{\sigma_s^{0.5} \mu_l^{0.24} H_{fg}^{0.24} \rho_v^{0.24}} (\Delta T)^{0.24} (\Delta P)^{0.75} S_p \quad (2.6)$$

$$h_{mac} = 0.023 (Re_l)^{0.8} (Pr_l)^{0.4} \frac{K_l}{D} F \quad (2.7)$$

Two-dimensionless functions, suppression ratio S_p and convection enhancement factor F , are to account for the suppression of bubble growth due to flow and for the increase in convective turbulence due to the presence of vapor. S_p and F were obtained as functions of a two-phase Reynolds number and the Martinelli parameter, respectively, and are defined as

$$S_p = \left(\frac{\Delta T_e}{\Delta T} \right)^{0.99} \quad F = \left(\frac{Re}{Re_l} \right)^{0.8}$$

Where $Re = \frac{\rho \cdot V \cdot D}{\mu}$ is effective Reynolds number for two-phase fluid and Re_l is Reynolds number for liquid fraction.

ΔT_e is effective superheat with flow and ΔT is the difference in the wall and saturation temperature. S_p and F could also be obtained from graph as in Fig. 2.3 and Fig. 2.4.

ΔP is the saturation pressure difference corresponding to the difference in saturation temperature ΔT .

2.3.3 CRITICAL HEAT FLUX

When the heat flux applied to a surface in contact with a boiling liquid is progressively increased, a point is

reached at which the continuous contact between the liquid and the surface is lost. This heat flux is called burnout heat flux or critical heat flux. When this 'boiling crisis' takes place the heat transfer coefficient is greatly reduced because of the low thermal conductivity of the vapour intervening between the liquid and the surface. This phenomenon can be most easily demonstrated in pool-boiling conditions using an electrically heated wire, and the consequence is the dramatic melting, or burnout, of the heater. A reduction of the heat transfer coefficient also occurs under forced-convective conditions if the heat flux is increased beyond a critical value. The term burnout in forced convective boiling is mostly used to describe the boiling crisis associated with the reduction of heat transfer coefficient and changing the heat transfer mode to transition boiling from nucleate boiling. The critical heat flux point is shown in flow diagram (Fig. 2.2) as C.H.F. between transition boiling region and nucleate boiling region.

The technical importance of the critical heat flux condition has led to the development of a bewildering variety of correlations. Milioti [35] catalogues fifty-nine correlations and detailed comparisons of a wide range of correlations reveal considerable differences. Since most correlations are based on empirical data their validity

can be assumed only when the conditions are within the range of the experimental conditions, and there are only few which falls within the experimental conditions of the present experimental range (i.e. single tube geometry, 1 atm. pressure, and maximum mass velocity of 40 g/cm² s). The most general of the correlations suggested to date is that by Macbeth [36] who compiled data from world-wide sources and arrived at a set of correlations for water boiling inside round tubes and rectangular channels of uniform heat flux for the pressure groups of 560, 1,000, 1,550 and 2,000 psia. Macbeth correlation is based on a hypothesis proposed and tested by Barnett [37]. This hypothesis, the 'local conditions hypothesis' suggests that the critical heat flux is solely a function of the mass quality at the point of the overheating. Macbeth had suggested that it was desirable to divide the critical heat flux data into two regions depending upon the mass velocity magnitude and different correlations were proposed according to the velocity magnitude. For the low-velocity regime, a nearly linear relationship between q_{crit} and quality X exists, and the correlation is independent of pressure.

$$q_{crit} = 0.00633 H_{fg} D^{-0.1} \left(\frac{G}{10^6}\right)^{0.51} (1-X) \cdot 10^6 \quad (2.8)$$

for round tubes, where q_{crit} is in (Btu/hr·ft²), and G in (lb/ft²·hr), D in (in.) and H_{fg} in (Btu/lb). The present experimental condition is very close to the applicable region

for this correlation, but the boundary is not very clear in the low pressure region. To check its validity, q_{crit} of Macbeth correlation was calculated using typical parameter values and compared with our experimental q_{crit} values obtained. Macbeth correlation gives q_{crit} value of 2.43×10^6 Btu/hr.ft² (7.66×10^6 watt/m²) for typical value of G (20 g/cm²s), D (1.39 cm) and quality X (0.01), whereas our experimental q_{crit} value is around 0.65×10^6 Btu/hr.ft² (2.05×10^6 watt/m²). This indicates that maybe our experimental velocity range is too high to use the Macbeth correlation of low-velocity regime. Analysis of data in the high mass velocity region is difficult because the pressure effect is more complex than in the low mass velocity region. However, the range of system pressure in Macbeth correlation for high velocity regime is too high for our experimental conditions which is 1 atm. pressure, and neither low-velocity regime nor high velocity regime Macbeth correlation could be used for this analysis.

An alternative to the Macbeth correlation which has the advantage of being continuous with respect to the variable 'system pressure' with apparently little or no loss of accuracy is the correlation proposed by Biasi et al. [38] :

$$q_{crit} = \frac{1.883 \times 10^3}{D^n \cdot G^{1/2}} \cdot \left[\frac{f(p)}{G^{1/2}} - X \right] \quad (2.9)$$

for the low quality region, and

$$q_{crit} = \frac{1.78 \times 10^3 \cdot h(p)}{D^n \cdot G^{0.6}} [1 - X] \quad (2.10)$$

for the high quality region, where $n = 0.4$ for $D \geq 1$ cm,
 $n = 0.6$ for $D < 1$ cm and

$$f(p) = 0.7249 + 0.099 \cdot p \cdot \exp(-0.032p) \quad (2.11)$$

$$h(p) = -1.159 + 0.149 \cdot p \cdot \exp(-0.019p) + \frac{8.99p}{10+p^2} \quad (2.12)$$

The correlation is evaluated in C.G.S. units and valid over the range of variables, which includes our experimental conditions except system pressure. The recommended system pressure for the correlation is from 2.7 bar to 140 bar, whereas our experiment was carried out at 1 atm. pressure. Since the system pressure for the experiment is close to the recommended pressure range for this correlation than other correlations, q_{crit} value was evaluated to check its validity, extrapolating the system pressure to 1 bar, and the value of the calculated q_{crit} is about 3 times our experimental q_{crit} value. This shows that this correlation could not be used outside its recommended system pressure range.

As a last resort, to get the reasonable critical heat flux correlation, earlier form of Macbeth correlation was studied [34]. In the earlier Macbeth correlation, for the high velocity regime, the correlation for a round tube is in the form,

$$q_{crit} \times 10^6 = A - C \cdot D \cdot (G \times 10^{-6}) H_{fg} \cdot \chi / 4 \quad (2.13)$$

where $A = y_0 \cdot D^{y_1} \cdot (G \times 10^{-6})^{y_2}$ and $C = y_3 \cdot D^{y_4} \cdot (G \times 10^{-6})^{y_5}$

The computer optimized values of y_0 to y_5 which were obtained from the high velocity critical heat flux region in the pressure range of 15 psia are $y_0 = 1.12$, $y_1 = -0.211$, $y_2 = 0.324$, $y_3 = 0.0010$, $y_4 = -1.4$ and $y_5 = -1.05$. This correlation gives q_{crit} value of 0.68×10^6 Btu/hr.ft² (2.14×10^6 watt/m²) for mass velocity of 20 g/cm².s, which is very close to the experimental q_{crit} value of 0.65×10^6 Btu/hr.ft² (2.05×10^6 watt/m²). Check with other mass velocities also confirmed the closeness of calculated q_{crit} value to the experimental. So, this earlier Macbeth correlation which present more complex high-velocity regime Macbeth correlation is modified from, and is given in Eqn. (2.13) is used in this study. The unit of q_{crit} is in (Btu/hr.ft²), mass velocity G in (lb/ft².hr), tube diameter D in (in.), and latent heat of vaporization H_{fg} in (Btu/lb).

2.3.4. TRANSITION BOILING REGION

The critical heat flux condition results in a drop of one or two orders of magnitude in the value of heat transfer coefficient as compared with the values obtained in the nucleate boiling. This decrease is a consequence of the fact that the heat transfer surface is no longer completely

wetted by the liquid phase but wetted by intermittent contact of large liquid droplets carried in the vapour stream. This heat transfer regime is called transition boiling region (Region 4 of Fig. 2.2) and a convenient way of treating transition boiling is to consider that, at any moment, the surface is partially wetted and partially covered by vapour. The total heat flux is then $q_{tot} = q_{wet} \cdot a_{wet} + q_{dry} \cdot a_{dry}$ with $a_{dry} + a_{wet} = 1$. The difficulty is to find proper expressions for the area functions a_{wet} and a_{dry} . One possibility is to work from the film boiling end, and Tong and Young [39] proposed an empirical correlation in the form of $q_{tot} = q_{TB} + q_{FB}$ where q_{TB} is the transition boiling heat flux which equals the measured total heat flux minus the film boiling heat flux, in Btu/hr.ft² (or, watt/m²) and q_{FB} is the film boiling heat flux, in Btu/hr.ft² (or, watt/m²).

Tong and Young observed that the heat transfer coefficient immediately after boiling crisis consists of two components : nucleate-boiling component and film-boiling component. The magnitude of nucleate-boiling component depends strongly on wall superheat, but weakly on mass velocity. The film boiling component is of convective nature and it can be evaluated by Dittus-Boelter type equation. Tong and Young noting that the chance of liquid droplets hitting the wall in a transition boiling is large at a low

quality and at a high steam production rate, derived following correlation empirically.

$$q_{TB} = q_{NB} \exp \left[-0.01 \cdot \left(\frac{x^{2/3}}{dx/dl} \right) \cdot \left(\frac{\Delta T_{sat}}{100} \right)^{1+0.0016 \Delta T_{sat}} \right] \quad (2.14)$$

where q_{NB} is the nucleate boiling heat flux or the measured total heat flux based on the instantaneous local conditions, in Btu/hr·ft², ΔT_{sat} is the wall superheat (i.e. $\Delta T_{sat} = T_{wall} - T_{sat}$) in °F, and l is the channel length in inches.

In correlating film boiling heat transfer coefficient, the Dittus-Boelter equation was developed for single phase convection. The film boiling heat transfer coefficient proposed by Tong is

$$h_{FB} = 0.023 \left[\frac{K_v}{De} \cdot \left(\frac{G \cdot De}{\mu} \right)_v^{0.8} \left\{ \frac{x \cdot \rho_v / \rho_l + S(1-x) \rho_v / \rho_l}{x + S(1-x)} \right\}^{0.8} \cdot \left(\frac{\mu C_p}{K} \right)_v^{0.4} \right] T_v \quad (2.15)$$

where De is wetted parameter and S is slip ratio.

Slip ratio S is the ratio of the gas phase velocity to the liquid phase velocity, (V_g/V_f), and could be expressed as following in terms of the mass quality and the void fraction

$$S = \frac{\rho_l}{\rho_g} \cdot \left(\frac{x}{1-x} \right) \cdot \left(\frac{1-\alpha}{\alpha} \right)$$

Combining the above two equations (Eqns. (2.14), (2.15)), the complete equation of the transition boiling

heat transfer correlation becomes,

$$q_{tot} = q_{NB} \exp \left[-0.01 \left(\frac{\chi^{2/3}}{dx/dl} \right) \left(\frac{\Delta T_{sat}}{100} \right)^{1+0.0016\Delta T_{sat}} \right] \\ + 0.023 (T_w - T_v) \left[\frac{K_v}{De} \left(\frac{G De}{\mu} \right)^{0.8} \right. \\ \left. \left(\frac{\chi \rho_v/\rho_g + S(1-\chi) \rho_v/\rho_l}{\chi + S(1-\chi)} \right)^{0.8} \left(\frac{\mu C_p}{K} \right)_v^{0.4} \right] T_v \quad (2.16)$$

2.3.5 INVERTED ANNULAR FLOW REGION

Inverted annular flow shown as Region 3 in Fig. 2.2, or sometimes called film boiling is characterized by a thin vapor film covering the heating surface and liquid in the centre core, thus would appear to be an inverse of annular flow. Heat transfer by direct contact of the liquid core with the wall is not expected because the wall temperature is high enough (above rewetting temp.) to forbid the wetting of a surface.

The vapour-liquid interface is not smooth but irregular. These irregularities occur at random locations but appear to retain their identity to some degree as they pass up the tube with velocities of the same order as that of the liquid core. The vapour in the film adjacent to the heating surface would appear to travel at much greater speeds. The heat is transferred from the wall to the vapour and

subsequently from the vapour to the wavy liquid core. Heat transfer across the wavy vapour-liquid interface takes place by forced convective evaporation. This mode of heat transfer is much more efficient than the single phase convective heat transfer between smooth wall and vapour, hence it is assumed that the bulk of the vapor is at or close to the liquid core temperature. Since the vapour has a low thermal conductivity relative to the liquid, film boiling requires a large temperature difference between the solid surface and liquid.

For simplicity, only heat transfer between wall and liquid core is considered. In this region, heat transfers between wall and vapour, vapour and liquid core are neglected because of small amount of vapour volume and vapour temperature being close to the liquid core temperature. Due to the high wall temperature encountered in this zone, heat transfer by radiation to liquid core is accounted for.

Radiative heat transfer to liquid core is taken as concentric cylinders and radiative heat flux is derived to as

$$q_{WL}^R = \frac{\alpha \sqrt{1-\alpha} (T_w^4 - T_l^4)}{\frac{1}{\epsilon_l} + \left(\frac{1}{\epsilon_w} - 1\right) \sqrt{1-\alpha}} \quad (2.17)$$

where σ is Stefan-Boltzman constant, α is void fraction, ϵ_l is emissivity of water and ϵ_w is emissivity of tube wall material (see Appendix 1 for derivation),

For convective heat transfer between wall and liquid core, Clement et al. [22] used Bromley type equation of stable film boiling on a vertical tube for their development of reflood codes FLIRA [22] and PSCHIT [22] and the same equation is adopted in this study, after a critical evaluation of the equation was made. The Bromley type equation is recommended for low flow and high quality inverted annular flows. The convective heat flux from wall to liquid core of Bromley type is

$$q_{WL}^C = 0.943 \left[\frac{K_v^3 \cdot \rho_v \cdot (\rho_l - \rho_v) \cdot g \cdot \lambda'}{\mu_v \cdot (T_w - T_l) \cdot L} \right]^{\frac{1}{4}} \cdot (T_w - T_l) \quad (2.18)$$

where $\lambda' = H_{fg} \left[1 + \frac{0.4 \Delta T \cdot C_{p_v}}{H_{fg}} \right]$, C_{p_v} is the specific heat of the saturated vapour, ΔT is wall superheat, and L is the distance from the rewetting front. So, total heat flux from wall to liquid core is $q = q_{WL}^R + q_{WL}^C$

2.3.6 DISPERSED FLOW

The flow pattern in this region (Region 2 of Fig. 2.2) is characterized by dispersed liquid droplets flowing in a continuous vapor medium. It is usually encountered at void

fractions well above 80% and is often referred to as the liquid-deficient regime [40,41]. This kind of flow regime usually appears downstream of inverted-annular flow regime, when the liquid core disappears as a result of evaporation or droplet entrainment.

Heat transfer in this flow regime is generally poor but still sufficient to help reduce the fuel rod temperature down to rewetting temperature during reflooding. Heat transfer by direct contact of the droplets with the wall is not possible because the wall temperature is higher than the rewetting temperature. Dispersed-Flow heat transfer can be predicted from empirical correlations or from semi-analytical models based on consideration of the heat transfer mechanisms between the wall, the vapor, and the droplets.

The semi-analytical mechanistic models attempt to evaluate separately the various modes of heat transfer. The wall is cooled by radiation, by forced convection to the vapor, and by interaction with the droplets. Such models require, specification of a number of flow parameters, e.g., droplet diameters and velocities that are not readily available but must be adjusted to fit the data.

Experimental evidence [42] of high vapor superheat

exists. The vapor temperature is controlled both by wall-to-vapor and vapor— liquid heat transfer, therefore thermal non-equilibrium may be present.

It is assumed for this study that droplets are spherical with uniform distribution in a cross section of the channel. At any time the droplet flux $N_d = 6 \frac{G(1-\chi)Ad}{\rho_l \pi \delta^3}$ is conserved and the number of droplets per unit of volume is given as $\frac{6(1-\chi)}{\pi \delta^3}$ (see Appendix 2 for detail)

Three types of heat transfer are considered in this region. They are convective heat transfer, radiative heat transfer, and heat transfer between phases. Convective and radiative heat transfers are between wall and liquid, and wall and vapour. Heat transfer between phases is for vapour-liquid interphase.

For convective heat transfer between wall and vapour, Clements et al. [12] used Colburn equation. In the laminar vapor flow ($Re_v < 2,000$), the heat flux is given as

$$q_{wv}^C = 3.656 \frac{K_v}{D} (T_w - T_v) \quad (2.19)$$

If the vapor flow is turbulent ($Re \geq 10,000$). then the heat

flux from the wall to vapour is given as

$$q_{wv}^C = 0.023 \frac{K_v}{D} Re_v^{0.8} Pr_v^{0.33} \left[\frac{\mu_{vc}}{\mu_{vw}} \right]^{0.467} (T_w - T_v) \quad (2.20)$$

where Re_v is vapour Reynolds number, ($Re_v = \frac{G \cdot \chi \cdot D}{\mu_v}$)
 Pr_v is vapour Prandtl number, ($Pr_v = \frac{Cp_v \mu_v}{K_v}$), μ_{vc} is dynamic viscosity of vapour at vapour temperature, and μ_{vw} is dynamic viscosity of vapour at wall temperature.

For $2,000 \leq Re_v < 10,000$, linear interpolation between above two equations is used.

Forslund [43] has proposed an expression for convective heat transfer between wall and liquid droplets which does not strike the walls.

Forslund's equation is;

$$q_{WL}^C = K' \cdot \frac{\pi}{4} \left[\frac{6 \cdot G \cdot (1-\chi)}{\pi \cdot v_l \cdot \rho_l} \right]^{2/3} \left\{ \frac{K_{vw}^3 \cdot \lambda'' \cdot g \cdot \rho_l \cdot \rho_{vw}}{(T_w - T_s) \mu_{vw} \cdot \sqrt[3]{\frac{\pi}{6}} \cdot \delta} \right\}^{1/4} \cdot (T_w - T_l) \quad (2.21)$$

where $\lambda'' = H_{fg} \left[1 + \frac{7}{20} \frac{Cp_v \cdot (T_w - T_s)}{H_{fg}} \right]^{-3}$, v_l is liquid droplet velocity, δ is droplet diameter and K' is an arbitrary constant.

Forslund used K' value of 0.2 for nitrogen, while Hynek [44] used $K = 1$ for water. So, in this study, Forslund equation is used with $K = 1$ for convective heat flux between wall and liquid droplets.

Radiative heat fluxes from wall to vapour and liquid droplets are considered. Heat flux equations for radiation between wall and vapour, and wall and liquid droplets are approximate solutions with a few general assumptions because of complications involved for exact solutions.

It is assumed that wall is opaque, diffusely emitting and reflecting, and also that droplets are spherical and their temperature, diameter and distribution are uniform over a cross section of the channel. The approximate solution of radiation between wall and droplets is obtained by multiplying the number of droplets in the unit volume to the radiative heat flux from wall to one droplet, with the assumption of vapour having transmissivity of 1, and no interaction between droplets. The radiation heat flux from wall to liquid droplets is derived as

$$q_{WL}^R = \frac{\sigma(A_2/A_c)(T_w^4 - T_d^4)}{\frac{1}{\epsilon_d} + (\frac{1}{\epsilon_w} - 1)(\frac{A_2}{A_c})} \times \frac{6(1-\alpha)}{\pi \delta^3} \times (Ad \cdot l) \quad (2.22)$$

where A_2 is the surface area of the droplet ($A_2 = \pi \delta^2$),
 A_c is the inside surface area of the tube ($A_c = \pi \cdot D \cdot l$)
 and Ad is the cross section area of the tube ($Ad = \pi \cdot D^2/4$).

For the radiation heat flux between wall and vapour, vapour is assumed to have uniform temperature over a cross section, emission is taken as isotropic and scattering is neglected, while a box model enables to account for the

spectral absorption. Heat flux between vapour at temperature T_v and the walls of a black-body container at T_w is given as

$$q_{wv}^R = \sigma \cdot (\epsilon_{vw} T_w^4 - \epsilon_{vc} T_v^4) \quad (2.23)$$

where ϵ_{vw} is the emissivity of vapour evaluated at T_w and ϵ_{vc} is emissivity of vapour at vapour temperature T_v . Kreith [29] gives an approximate solution for non-black body container whose emissivity is larger than 0.7 as

$$q_{wv}^R = \sigma (\epsilon_{vw} T_w^4 - \epsilon_{vc} T_v^4) \frac{(\epsilon_w + 1)}{2} \quad (2.24)$$

where ϵ_w is the emissivity of the enclosure surface.

In this study, even though the emissivity of wall material is less than 0.7 (around 0.4 ~ 0.7) [29], Kreith's approximation is still used, because of lack of information on radiation between vapour and Grey walls, and also its simplicity.

The heat transfer between phases by convection is given by Lee - Ryley [45]. The heat flux from vapour to liquid droplet is given as

$$q_{VL}^C = \frac{K_L}{\delta} \left[2 + 0.74 Re_d^{1/2} Pr_v^{1/3} \right] (T_v - T_s) \quad (2.25)$$

where Re_d is Reynolds number of liquid droplet.

$$(Re_d = \frac{G \cdot (1-\chi) \cdot \delta}{\mu_l (1-\alpha)})$$

Therefore, the total heat flux from wall to liquid droplet is

$$q_{WL}^T = q_{WL}^C + q_{WL}^R \quad (2.26)$$

where q_{WL}^C and q_{WL}^R are given in Eqns (2.21) and (2.22) respectively while the total heat flux from wall to vapour is

$$q_{WV}^T = q_{WV}^C + q_{WV}^R \quad (2.27)$$

where q_{WV}^C is given Eqns. (2.19) and (2.20) and

$$q_{WV}^R = \sigma (\epsilon_{vw} T_w^4 - \epsilon_{vc} T_v^4) \cdot \frac{(\epsilon_w + 1)}{2} \cdot \alpha \quad (2.28)$$

2.3.7 SINGLE STEAM PHASE REGION

In this region, only single phase steam flows in the channel, so this is just a vapour forced convection flow.

Because of high temperature difference between wall and vapour, radiative heat transfer as well as convective heat transfer is considered in this region. For radiation heat flux from wall to vapour, same type of equation as in dispersed flow is used with the absence of liquid droplets. The radiation heat flux from wall to vapour is [29] already given by Eqns. (2.24)

Convection heat transfer coefficient h varies depending on the flow condition. In laminar flow where Reynolds number is less than 2,000, convection heat transfer coefficient h is given as

$$h = \frac{48}{11} \frac{K_v}{D} \quad \text{for constant heat flux.}$$

For fully developed turbulent flow ($Re > 10,000$), Dittus and Boelter [31] equation gives $h = 0.023 \frac{K_v}{D} Re_v^{0.8} Pr_v^{0.4}$. A linear interpolation between laminar and turbulent heat transfer coefficient expression is made to give heat transfer coefficient for transitional flow where Re is between 2,000 and 10,000. The convective heat flux from wall to vapour

$$\text{is } q_{WV}^C = h(T_w - T_v)$$

The total heat flux from wall to vapour is

$$q_{WV}^T = q_{WV}^R + q_{WV}^C \quad (2.29)$$

2.3.8 VOID FRACTION

The void fraction is commonly taken as the fraction of volume occupied by the gas phase. The one-dimensional flow assumption means that the void fraction α is the fraction of flow area occupied by the gas phase.

$$\alpha = \frac{A_g}{A} \quad \text{and } 1 - \alpha = A_l/A \quad (2.30)$$

where A_g and A_l are the cross-sectional areas occupied

by the gas and liquid phases respectively and A is the total cross-sectional area of the channel.

In boiling and condensation, it is convenient to use the fraction of the total mass flow which is composed of vapour or liquid. Thus, the mass quality χ is defined as $\chi = \frac{W_g}{W}$, $1 - \chi = \frac{W_l}{W}$ where W_g and W_l are the mass flow rate of the gas and liquid phases respectively and W is the total mass flow rate of liquid and gas.

One relationship which is useful for two phase flows expresses the ratio of the gas phase velocity to the liquid phase velocity in terms of the mass quality.

$$S = \frac{U_g}{U_l} = \left(\frac{\chi}{1-\chi} \right) \left(\frac{1-\alpha}{\alpha} \right) \cdot \frac{\rho_l}{\rho_g} \quad (2.31)$$

where S is often referred to as the slip ratio.

It is usually greater than unity which means U_g is usually greater than U_l , except homogeneous case in which slip ratio S equals 1.

In this study, the mass quality χ is calculated with the heat flux from the wall to liquid, but void fraction α is an unknown value unless it could be calculated from a relationship which involves mass quality χ and void fraction α .

The simplest relationship will be Eqn. (2.31) with S equal to 1.

$$\alpha = \frac{1}{1 + \frac{\rho_g}{\rho_l} \left(\frac{1-\chi}{\chi} \right)} \quad (2.32)$$

But this is only applicable to homogeneous flow, and would not be useful for general purpose.

In the last few years, Bankoff [46] has suggested that the slip ratio was equal to

$$S = \frac{1 - \alpha}{K' - \alpha} \quad (2.33)$$

where K' was an empirical constant.

For vertical flow of steam-water mixtures, K' was taken to be a function only of pressure ; $K' = 0.71 + 0.0001 \cdot p$

where p is the fluid pressure in psia. For 1 atm. pressure ($p = 14.7$ psia) K' becomes 0.71147. Substituting K' value into Eqn. (2.33) and combining with Eqn. (2.31), the relationship between void fraction α and mass quality χ is

$$\alpha = \frac{\frac{\rho_l}{\rho_g} \left(\frac{\chi}{1-\chi} \right) 0.71147}{1 + \frac{\rho_l}{\rho_g} \left(\frac{\chi}{1-\chi} \right)} \quad (2.34)$$

Simple check reveals that either this equation is not suitable for low pressures or not applicable to wide ranges of mass quality χ . Eqn. (2.34) gives $\alpha = 0$ when $\chi = 0$,

but when χ becomes close to 1, α becomes 0.71147, which indicates for large mass quality χ , α value tends to be wrong.

Baker [47] has given an empirical equation to allow for the influence of mass velocity on the void fraction in vertical up flow ;

$$\chi = \frac{\alpha^2 (Y^{1/3} - 1) + \alpha}{Y - \alpha(Y - Y^{1/3})} \quad (2.35)$$

where $Y = 0.021 \left(\frac{\rho_l}{\rho_g}\right) G^{0.686}$ and G is the mass velocity in $\text{Kg/m}^2 \cdot \text{S}$. This equation is valid over the range $7.5 < Y < 300$ and $G < 950 \text{ kg/m}^2 \text{ S}$.

Checking with various flow rates, this equation gives reasonable value of α with quality χ . Also it shows that void fraction α increases as mass velocity increases for a given mass quality χ . However, Y values for G values of 100, 200, and 400 $\text{Kg/m}^2 \cdot \text{S}$ (experimental values) are 793, 1,275 and 2,052 which are over the range of Y for Baker's equation, eventhough G values are well within range.

There are only two variables which affect Y value for a given G value, namely ρ_l and ρ_g . ρ_l is almost independent of pressure but ρ_g depends very much on temperature and pressure, and if ρ_g is considerably large, then Y value will be small for given G value.

Thus, this equation would probably be better applicable to very low mass velocity or very high pressure.

Martinelli-Nelson [48] assumed that a thermodynamic equilibrium exists at all points in the flow and Lockhart-Martinelli [49] correlation was arbitrarily applied to atmospheric pressure steam-water flow. As a result, void fraction α as a function of mass quality and absolute pressure of steam-water flow was given in graphical form. In this study, this graph by Martinelli-Nelson was used for relationship between void fraction α and quality χ . This graph shows quality down to 0.006 only, so lower quality values were reasonably extrapolated.

23 points were obtained from the graph and those 23 pairs of void fraction and quality are listed in table 3.

In the graph, the mass quality χ is in log scale and void fraction α is in plain scale, so noting this, proper interpolation between data was used to get void fraction α from given mass quality χ .

CHAPTER 3

COMPUTATION

3.1 REWETTING TEMPERATURE, T_q , CORRELATION

Using Buckingham's π theorem on the 11 variables, seven non-dimensional parameters were obtained. The functional relationship among these seven dimensionless parameters ($\pi_1, \pi_2, \pi_3, \dots, \pi_7$) is

$$\pi_1 = E \cdot \pi_2^{A2} \cdot \pi_3^{A3} \cdot \pi_4^{A4} \cdot \pi_5^{A5} \cdot \pi_6^{A6} \cdot \pi_7^{A7} \quad (3.1)$$

E, A2, A3, A4, A5, A6 and A7 are constant numbers and they are determined by Multiple Linear Regression Analysis. Eq. (3.1) is not a linear equation, but it could be transformed into linear equation, if one take logarithm on both side of the equation. Thus the equation (3.1) becomes

$$\begin{aligned} \ln(\pi_1) = \ln(E) + A2 \ln(\pi_2) + A3 \ln(\pi_3) + A4 \ln(\pi_4) \\ + A5 \ln(\pi_5) + A6 \ln(\pi_6) + A7 \ln(\pi_7) \end{aligned} \quad (3.2)$$

and Eq. (3.2) is a function which is linear in the coefficients. For simplicity, Eq. (3.2) could be written as ;

$$\begin{aligned} Y = A1 + A2 \cdot X2 + A3 \cdot X3 + A4 \cdot X4 + A5 \cdot X5 \\ + A6 \cdot X6 + A7 \cdot X7 \end{aligned} \quad (3.3)$$

where X2, X3, --- , X7 are independent variables and Y is dependent variable.

The method of least squares [50] is built on the hypothesis that the optimum description of a set of data is one which

minimizes the weighted sum of squares of deviations of the data Y_i from the fitting function Y . With this definition, the weighted sum of squares of deviation, χ^2 (chi-square), becomes

$$\chi^2 = \sum (Y_i - A_1 - A_2 \cdot X_2 - A_3 \cdot X_3 - A_4 \cdot X_4 - A_5 \cdot X_5 - A_6 \cdot X_6 - A_7 \cdot X_7)^2 \quad (3.4)$$

The least-square-fit is obtained when the weighted sum of squares of deviation, χ^2 (chi-square), is minimum.

The coefficients for which χ^2 is a minimum are determined by setting the derivatives of χ^2 with respect to each of the 7 coefficients ($A_1, A_2, \dots, A_7$) equal to 0.

This yields 7 simultaneous equations such as

$$\begin{aligned} \sum Y_i &= A_1 \cdot N + A_2 \cdot \sum X_2 + A_3 \cdot \sum X_3 + \dots + A_7 \cdot \sum X_7 \\ \sum Y_i X_j &= A_1 \cdot \sum X_j + A_2 \cdot \sum X_2 \cdot X_j + A_3 \cdot \sum X_3 \cdot X_j + \dots + A_7 \cdot \sum X_7 \cdot X_j \end{aligned} \quad (3.5)$$

Solving these 7 simultaneous equations, one could obtain the coefficients which would give the least-square-fit.

The linear-correlation coefficient between the j th variable X_j and the dependent variable Y is given as [50]

$$r_{jy} = \frac{S_{jy}^2}{S_j S_y} \quad (3.6)$$

where $S_{jy}^2 = \frac{1}{N-1} \sum [(X_j - \bar{X}_j)(Y_i - \bar{Y})]$

$$S_j^2 = \frac{1}{N-1} \sum (X_j - \bar{X}_j)^2, \quad S_y^2 = \frac{1}{N-1} \sum (Y_i - \bar{Y})^2$$

N is the number of points and $\bar{Y}$ and $\bar{X}_j$ are averages of Y_i and the functions X_j over the N observations.

3.2 TEMPERATURE-TIME TRACE CURVE

The main objective of this computation is selecting appropriate heat transfer coefficient h from 6 heat transfer regions, thus with prefixed rewetting temperature, T_q , predicting the temperature-time trace curve of a single vertical tube along its entire axial positions.

From trial computations, as will be discussed, it seems that using the element time interval of about 0.1 sec. is most optimal considering the convergence and the computational time. The element time interval is the time taken by the elemental volume of water to pass that certain element completely. For mass velocity G of 20 g/cm²s or larger, this element time interval would be obtained with element length, DX , of about 2 cm, while with mass velocity G of 10 g/cm²s, the element length is 1 cm. When the length DX of larger than 2 cm was used in the computation for mass velocity G of 20 g/cm²s, the instability in heat transfer between wall and coolant gave unreasonable values (i.e. wall temperature dropping to lower than coolant temperature), While a smaller length of DX (i.e. 1 cm, 0.5 cm) takes too much computational time (up to CPU of 40 min. on IBM 360) with a minimal difference in the results. The difference between the results using element length, DX , of 2 cm and 1 cm is less than 1 %, but the difference in computational time is

more than 300 %, and using element length of 0.5 cm gives less than 1 % improvement in result over 1 cm element length with more than double the computational time. When mass velocity G is $10 \text{ g/cm}^2 \text{ s}$, using element length, DX , of larger than 1 cm (i.e. 2 cm) gives unreasonable value in temperature-time trace, while smaller length of DX (i.e. 0.5 cm) takes more than double the computational time with less than 1 % difference in results. This shows that the element time interval of about 0.1 sec. (i.e. element length, DX , of 2 cm for $G \geq 20 \text{ g/cm}^2 \text{ s}$ and DX of 1 cm for $G=10 \text{ g/cm}^2 \text{ s}$) is good value to choose, considering the accuracy of the results and the computational time.

The initial wall temperature distribution of the computational model was chosen to be as close to the experimental initial wall temperature distribution as possible. The experimental initial wall temperature distribution is not uniform, but is different along the axial distance. To get the same wall temperature profile for the computational model as the experimental temperature profile, 4 axial points along the tube were chosen. Two of these points were chosen near each end and the other two were around the middle about 1 m apart. At those 4 points, the initial wall temperature T_w and the experimental rewetting temperature T_q were obtained. The initial wall temperature distribution of the computational model is calculated by interpolating along the channel between the 4 experimental wall temperatures at the specific axial points.

The experimental rewetting temperature distribution along the axial distance is also calculated using the same method (interpolation between the 4 rewetting temperatures), whereas the correlated rewetting temperature distribution along the channel is calculated using the proposed empirical correlation given by Eq.(5.2).

In the computation, each element is taken as a control volume, and heat balance between wall and coolant water is established in that control volume using appropriate heat transfer equation according to the heat transfer region.

At the inlet of the channel, when the coolant water is introduced to the first element, the flow is single phase liquid flow. If the wall temperature is less than the rewetting temperature, then the heat transfer region is assumed to be either the Nucleate Boiling Region or the Wet Region according to the wall temperature. If the wall temperature is higher than the ~~rewetting~~ temperature, normally the flow is assumed to be one of the inverted annular flow, dispersed flow, or single phase steam flow. However, when the mass quality is 0, none of the three regions (Inverted Annular Flow, Dispersed Flow, Single Phase Steam Flow) has physical meaning. So, when the wall temperature is higher than the rewetting temperature, and the mass quality χ is 0 (it occurs usually at the inlet of the channel), the heat transfer mode is

assumed to be nucleate boiling without the microconvection effect due to boiling. In other words, only macroconvection occurs due to flow, and the heat transfer coefficient would be

$$h_{\text{mac}} = 0.023 (Re_\ell)^{0.8} (Pr_\ell)^{0.4} \frac{K_\ell}{D} F \quad (3.7)$$

This choice is based on the assumption that because of high wall temperature, and no actual contact between wall and liquid, boiling is not possible and convection due to flow is the only heat transfer mechanism. Also, this gives better result compared to various other trials.

Besides this special case, normally in each element, the heat transfer between the wall and either water or vapour, depends on the heat transfer region which is assigned to each element. The particular heat transfer region is determined by the wall temperature, and the flow conditions in the element. In this study as stated in chapter 2, six different heat transfer regions are assumed.

When the mass quality X of the flow is 1 (i.e. single phase steam flow), the heat transfer equation of the Single Steam Phase Region is used to calculate heat transfer from the wall to the vapour. Dispersed Flow is assumed to exist when the mass quality of the flow is larger than 0.05 and less than 1.

Inverted Annular Flow Region is selected when the wall temperature is greater than the rewetting temperature and the mass quality is less than 0.05. The vapour quality χ is important factor in distinguishing the Inverted Annular Flow Region from other regions. Here, it was assumed that if vapour quality χ is 0, then because of no void fraction, the Inverted Annular Flow Region does not exist.

If vapour quality is larger than 0.7, again it was assumed that the vapour quality is too large to support Nucleate Boiling or Transition Boiling. Transition Boiling starts when the wall temperature is below the rewetting temperature and the heat flux in the Nucleate Boiling Region is larger than the critical heat flux. If the wall temperature is less than the rewetting temperature and vapour quality is less than 0.7, then Nucleate Boiling Region is selected.

Here, it is assumed that if the vapour quality is larger than 0.7, then even though the wall temperature is lower than the rewetting temperature, the Nucleate boiling can not be sustained because of high void fraction. And heat transfer mode should be changed to Dispersed Flow Region.

Single Phase Liquid Region exists only when the wall temperature is less than 115°C for the present study and the water temperature is less than 100°C . In the Nucleate Boiling Region, if the heat flux from the wall to coolant liquid is greater

than q_{crit} (critical heat flux) then the heat transfer mode is changed from Nucleate Boiling Region to Transition Boiling Region and heat flux is calculated again using heat transfer coefficient from Transition Boiling Region.

In each element, coolant water was heated and its new temperature and vapour quality as well as the new wall temperature were calculated using appropriate heat transfer equations according to the flow region, and then flow moves to the next element. The heat flux from the wall to the coolant was calculated, and as the first attempt, the heat flux was used to increase the water temperature until it reaches 100°C before any vapour was generated, and the wall temperature was decreased accordingly by the amount of heat flux lost. After the water temperature reaching 100°C , the mass quality χ was increased by the heat flux. However, this causes a problem such that controlling the rewetting process by the rewetting temperature is not possible, because single liquid flow with mass quality zero reaches certain elements before its wall temperature drops to the rewetting temperature. The liquid flow with mass quality zero decreases the wall temperature substantially compared to the inverted annular flow does, and the process resembles the rewetting when the wall temperature is higher than the rewetting temperature. This produces a large error in final results.

After serious consideration, some modifications were made in such a way that after the water temperature reaching certain level, the vapour starts to form and the water temperature and the mass quality increases simultaneously. This idea is based on the assumption that there exist temperature gradient across the water body. In other words, the water temperature immediately adjacent to the wall or vapour phase is 100°C , but the water temperature inside the liquid core is not necessarily 100°C , and in this study, the water temperature means the average water temperature. From trial computation, using average water temperature of 70°C seem to give good result. So, in the computation, water temperature was increased to 70°C without any nucleation of vapour using calculated heat flux, and after water temperature reaching 70°C , the water temperature and the mass quality (i.e. amount of vapour formed) were increased simultaneously until the water temperature reaches 100°C using part of the heat fluxes from wall to liquid for each process according to the region.

The computation shows that in Nucleate Boiling Region, using 67 % of the heat flux to increase the water temperature while remaining 33 % is used to increase the mass quality gives better result compared to other trials. In Transition Boiling Region, half of the heat flux is used

for the water temperature and the other half for the mass quality. In the Inverted Annular Flow Region, it is little more complicated than the other regions. The minimum water temperature before nucleation of any vapour, is set to 70°C. Having the water temperature of 70°C or more, if the mass quality is less than 0.001, then the mass quality is increased to 0.001 without further increase in the water temperature. When the mass quality is less than 0.03 but more than 0.001, with the water temperature between 70°C and 100°C, half of the heat flux is used to increase the water temperature. and the other half is used to increase the mass quality χ . If the mass quality is larger than 0.03, then the water temperature is increased to 100°C before any increase in the mass quality. In all cases, once the water temperature reaches 100°C, the mass quality χ is the only variable which would be increased.

As stated before, in the dispersed flow, it is assumed that the droplet flux $N_d = 6 \frac{G(1 - \chi) \cdot A_d}{\rho_l \pi \cdot \delta^3}$ is conserved. So, there is direct relationship between the droplet diameter and the mass quality. Since the droplet temperature is assumed to be 100°C, the convective heat flux from wall to liquid droplet is used to increase the mass quality χ . After calculating mass quality, new droplet diameter is calculated using droplet flux equation with new mass quality χ .

$$\delta = \left(\frac{6 G(1-X) Ad}{Nd \cdot \rho_L \cdot \pi} \right)^{1/3} \quad (3.8)$$

Besides the convection and radiation, conduction heat transfer in the tube wall is also considered in the heat flux computation. Since this model is for vertical bottom flooding, the circumferential temperature variations are neglected because of symmetry. Also the tube wall is very thin, and the Biot number is smaller than one, so the radial temperature gradients are assumed to be negligible. Under these conditions, the temperature distribution in the wall is one-dimensional, and only axial conduction heat transfer in the tube wall is used for the heat flux computation.

During the initial rewetting process, steam would flow through the most part of the hot tube, until the coolant water flow (either in inverted annular flow or dispersed flow form) reaches that point, and the dry wall temperature would not remain same as the initial temperature during that time. The wall temperature would drop by the convection heat transfer to steam eventhough it would be small. This is called the precursory cooling and this precursory cooling effect is also included in the computation.

At first the actual convection heat transfer coefficient to steam was used with varying Reynolds number, Prandtle number, and vapour temperatures. The Reynolds number at

certain element depend on the coolant flow rate, and also depend on the amount of vapour formed during the rewetting process in each elements proceeding that certain element. Vapour temperature was calculated in each elements using the heat fluxes from the wall to the vapour. Including this type of precursory cooling in the program, takes too much computational time. So, approximate precursory cooling which uses very little computational time is devised from modifying the actual precursory cooling model. First modification was made assuming that the vapour flow is always turbulent. The wall temperature drop due to the modified precursory cooling is

$$\Delta T_w = \frac{0.023}{4} \cdot \frac{K_v \cdot (Re_v)^{0.8}}{C_{p_w} \cdot \rho_w (D+\delta) \delta} \cdot (T_w - 100.) \cdot \delta \quad (3.9)$$

where δ is the time taken for the coolant water flow to reach that particular element, and Re_v is vapour Reynolds number ($Re_v = \frac{G \cdot D \cdot \chi}{\mu_v}$). Here, quality χ is assumed to be average 0.02 because the exit quality of the first water element is average about 0.04, and absolute viscosity of steam at 100°C is $1.29 \times 10^{-4} \text{ g/cm}\cdot\text{s}$, so $Re_v = \frac{G \cdot D \cdot 0.02}{1.29 \times 10^{-4}}$. Vapour Prandtle number, Pr_v , at 100°C is close to 1, so $(Pr_v)^{0.4}$ term in the turbulent convection heat transfer equation can be dropped. Vapour temperature is assumed to be at constant 100°C .

However, the result showed that this modified

precursory cooling equation gives too small wall temperature drop compared to the actual convection heat transfer to the steam flow. When the coefficient of Eq.(3.9) was increased from 0.0056 to 0.01 as eq.(3.9) is strictly applicable for fully developed flow, this produces an improvement in the results. So using this modified precursory cooling, one would get the same results as using actual precursory cooling process with a lot less computational time. The wall temperature drop using this modified precursory cooling until the water coolant flow reach a particular element is

$$\Delta T_w = 0.01 \frac{K_v (Re_v)^{0.8}}{C_{p_w} \rho_w (D+\delta) \cdot \delta} (T_w - 100.) \delta t \quad (3.10)$$

In this study, it is assumed that the rewetting front propagates in the direction of the coolant flow and in one direction only. Consequently, the occurrence of sudden wetted patches along the hot tube wall surface is not permitted before the rewetting front reached that point.

In the computation, the amount of water temperature, wall temperature and mass quality variation is calculated as

$$\Delta T_l = \frac{Q_c}{G (1-\chi) A_d C_{p_l}} \quad (3.11)$$

$$\Delta T_w = \frac{Q \cdot \Delta t}{C_{p_w} Vol_w \rho_w} \quad (3.12)$$

$$\Delta \chi = \frac{Q_c}{H_{fg}} \quad (3.13)$$

where Q_c is the total heat transferred from wall to liquid in the element and Q is the total heat lost by the wall in the element. A_d is the cross sectional area of the channel, Vol_w is the volume of the tube wall in the element and Δt is the time taken for the coolant water flow to pass the element.

Many thermodynamic properties used in the computation are function of temperature. Those are density, viscosity and thermal conductivity of water and vapour, specific energy of vapour and emissivity of tube and vapour. In the computation, those values are all calculated for each temperature variance using subroutines.

For a reference, program flow chart for this computation is shown in Fig.3.1 and FORTRAN IV computer program (Appendix 3) is also attached for future reference.

3.3 CONVECTIVE HEAT TRANSFER COEFFICIENT

During the reflooding period, several heat-transfer modes occur successively at a given location, resulting in a continuous change of the surface heat flux. Parameters, which can influence the heat-transfer rate include coolant flow rate, coolant inlet subcooling, initial wall temperature, wall thickness, wall material and axial location.

From an energy balance for the length ΔZ of the test section, one obtains

$$\begin{aligned} (\rho C_p)_w \frac{\pi}{4} (D_o^2 - D_i^2) \Delta Z \frac{dT_w}{dt} &= \dot{q} \frac{\pi}{4} (D_o^2 - D_i^2) \Delta Z \\ &- q_c \cdot \pi \cdot D_i \cdot \Delta Z - q_a \frac{\pi}{4} (D_o^2 - D_i^2) - (q_{ra} \pi D_i \Delta Z) \\ &- (q_{1s} \pi D_o \Delta Z) \end{aligned} \quad (3.14)$$

where $\dot{q}$ is power generation rate, q_c is convective heat flux, q_a is axial heat flux, q_{ra} is radiation heat flux and q_{1s} is the heat loss to the surroundings.

Due to the finite number of thermocouples installed on a test section, it is impossible to calculate the axial temperature gradient for $\Delta Z \rightarrow 0$. However, a reasonably accurate axial temperature gradient can be obtained, if the temperature-time gradient is used as ;

$$q_a = K \frac{dT}{dZ} = K \frac{dT}{dt} \frac{dt}{dZ} = K \frac{1}{U} \frac{dT}{dt} \quad (3.15)$$

Since temperature-time trace and local rewetting velocity are known from the experimental data, the axial heat flux can then be calculated.

The influence of radiation on surface heat flux becomes significant only at very high wall temperature.

In the transition boiling region, it only accounts for less than 1 % of the total heat flux. Therefore, it is neglected in the present study.

The heat flux transferred to the coolant from the wall can now be derived from Eq.(3.14) as ;

$$q_c(t) = \frac{D_o^2 - D_i^2}{4D_i} \left(\dot{q} - (\rho C_p)_w \frac{dT_w}{dt} - \frac{K}{U^2} \frac{d^2 T_w}{dt^2} \right) - q_{ls} \frac{D_o}{D_i} \quad (3.16)$$

In the analysis the power generation rate is 0, and the heat loss from the test section to the surrounding is relatively small because the test section was well insulated.

Using the finite-difference approximation, the surface heat flux at a given location can be evaluated from the recorded temperature-time trace curve. The term $\left(\frac{K}{U^2} \frac{d^2 T_w}{dt^2} \right)$ expresses the contribution of axial conduction to the surface heat flux based on the assumption of one-dimensional conduction.

In order to determine the heat transfer coefficient defined as $h(t) = \frac{q_c(t)}{T_{wi} - T_s}$ the inside surface temperature T_{wi} must be known whereas the outside surface temperature was measured in experiment. The inverse heat transfer problem could be used to calculate the inside surface temperature from the measured outside surface temperature.

However, this process is very cumbersome and since the tube wall is very thin, and the Biot number is smaller than one, the radial temperature gradient is assumed to be negligible. The calculated inside surface temperature using the inverse heat transfer problem shows usually less than 1 % difference from the measured outside surface temperature so for this study the outside surface temperature is used for the inside surface temperature.

CHAPTER 4

EXPERIMENT

4.1 APPARATUS

The main experimental loop consisted of two sections, one fixed on the laboratory floor and the other on a pivotable mounting which allows experiments to be carried out with test sections at any angle between horizontal and vertical. The general layout of the major components are shown schematically in Fig. 4.1.

The fixed section consists of a water demineralizer, capable of treating 0.4 m^3 per hour, a preheater of 0.2 m^3 capacity, a main supply boiler operating at water temperatures up to 200°C and at pressure up to 21 bars, a flow circulator and two flow rate transducers mounted in parallel. Each transducer has a two-way valve mounted in series with it, so that either transducer can be selected depending on the flow rate to be measured. One transducer covers the lower end of the flow rate (6 to 60 g/s), and the other the higher end (35 to 350 g/s). The movable section consists of a by-pass circuit, the test section, a pair of quick acting valves to divert the coolant flow to or away from the test section, an exhaust condenser system for retaining and

condensing the steam which is produced in the test section, and a reverse tank for the coolant. All these components are supported on a mounting, consisting of a pair of vertical members with a frame pivotally mounted between them.

The pivoting frame clamps onto the condenser and the reverse tank. Lengths of flexible hose (heavy duty Aeroquip rubber hose) are used to connect the water leads to the movable unit. All pipings and fittings for the entire apparatus were made of stainless steel to avoid unwanted contamination.

The physical characteristics of the four test-sections used in the study are summarized in Table 1.

Each test-section was instrumented with about 40 chromel-alumel thermocouples, spot-welded onto the outside wall surface at 10 to 12 different axial positions, and 2 to 4 were around the circumference at a given axial location. The test-sections were heated by an a.c. power source capable of 30 KVA at up to 1500 amps. The terminal clamps, made of copper, were attached to both ends of the test-section, and Fiberfrax was used for the insulation. Pressure transducers were connected with the test-sections for inlet and outlet pressure measurements. The test-sections were washed with distilled water followed by acetone rinses before they were installed in the apparatus.

4.2. EXPERIMENTAL PROCEDURE AND DATA DEDUCTION

All tests were conducted at atmospheric pressure..

Using either the preheater and/or the main supply boiler, the water temperature was set at a given degree of subcooling. The power to the test-section was turned on and the wall temperature was brought up to the desired value by increasing the power stepwise. Experiments were performed by diverting the water of a given flow rate, flowing at steady state condition in the bypass circuit, to the test section while the power was either maintained or switched off depending on the selection of the test parameters.

The signals from the thermocouples, from the transducers for flow rate and pressures, and power input to the test-section were monitored continually on a digital high speed data acquisition system together with multi-channel recorders. The variation of the water flow rate during the course of experiment was usually very small, under $\pm 4\%$ of the designated flow rate. The temperature-time trace curves for several axial locations were produced by the multi-channel recorder which is connected to certain thermocouples. The rewetting velocities and the rewetting temperatures were obtained from the temperature-time traces.

CHAPTER 5

RESULT AND DISCUSSION

5.1 T_q correlation

To determine the individual influence of each π -term, the linear-correlation coefficient was obtained for all the non-dimensional parameters.

The linear-correlation coefficient r_j is defined as a measure of the degree of linear-correlation between the j th parameter X_j and the dependent variable Y . The value of r ranges from 0, when there is no correlation, to 1 when there is complete correlation. The linear-correlation coefficient r is useful for testing whether one particular parameter should be included in the theoretical function to which the data are fitted.

The linear correlation coefficients for π_2 to π_7 were : 0.420, 0.392, 0.425, 0.046, 0.0055 and 0.492, respectively. This indicates that the terms π_5 and π_6 do not affect the final result as much as the other variables ; i.e. the variables P and D do not affect the apparent rewetting temperature, T_q , significantly.

The effect of power generation rate P during the rewetting velocity has been predicted by Duncan and Leonard [28] who extended the one-dimensional solution of Yamanouchi [17] to a more general case by the inclusion of a uniform heat generation term. The numerical solution of the two-dimensional model by Thompson [15] also confirmed the effect.

However, from the above analysis, it can be said that the power generation rate itself could not have affected the apparent rewetting temperature T_q , nor the heat transfer coefficient h significantly, but is an independent parameter which affects the rewetting velocity. Obviously, the tube diameter D could not affect T_q very much, and this seems reasonable. Therefore, the final correlation for Eq. (2.2) is

$$\pi_1 = E(\pi_2)^{A_2}(\pi_3)^{A_3}(\pi_4)^{A_4}(\pi_7)^{A_7} \quad (5.1)$$

where $E = 19.51$

$$A_2 = 0.107$$

$$A_3 = -0.162$$

$$A_4 = -0.099$$

$$A_7 = -0.163$$

or

$$T_q = 19.51 \times T_{ws} \times \left(\frac{T_{sc}}{T_{ws}}\right)^{0.107} \times \left(\frac{Cp \cdot G \cdot d}{K}\right)^{-0.162} \times \left(\frac{K \cdot \rho^2 \cdot T_{ws}}{d \cdot G^3}\right)^{-0.0989} \times \left(\frac{z}{d}\right)^{-0.163} + T_s \quad (5.2)$$

The comparison of T_q predicted by Eq. (5.2) with measured values is shown in Fig. 5.1. It can be seen that the correlated values fall within $\pm 10\%$ of the experimental.

The reduced chi-square (χ^2_r) is the chi-square value divided by the number of degrees of freedom, and is a convenient measure of fit. If the function used is a perfect fit, then the reduced chi-square should be zero.

For the above functional relationship, the reduced chi-square is 0.013.

5.2 TEMPERATURE-TIME TRACE CURVES (T-T PLOTS)

The result of the T-T plot computation shows that in all cases the predicted rewetting time is about 5~10 sec. longer than the experimental rewetting time at all axial locations. This is probably due to the end effect.

In the experiment, end section is more susceptible to heat loss to the surroundings. In the computation, this effect is not taken into consideration, and wall temperature of the end section would be higher than the experimental temperature, thus delaying the rewetting of the surface. Also, at the inlet of the test section, it is not very clear what kind of heat transfer mechanism is involved when coolant liquid

is first introduced to the very hot surface.

The initial time difference between the computed rewetting time and the experimental rewetting time caused by end effect could become large depending on the computation method used. Therefore, to offset this end effect in the computation and also to obtain reasonably good result, various modifications were made. One of the modifications worth mentioning is using Nucleate Boiling heat transfer when quality X is 0 regardless of the wall temperature based on the assumption that Nucleate Boiling heat transfer occurs at the inlet of the test section when water is first introduced to the hot tube. This seems to decrease the initial time difference caused by end effect between the predicted and experimental rewetting time substantially. However, this gives problem of controlling rewetting process by rewetting temperature and the rewetting velocities obtained using this modification are much faster compared to the experimental ones.

Another modification tried was to increase the heat transfer coefficient in the Inverted Annular Region. The heat transfer coefficient, h , was arbitrarily increased by 10 %, 30 %, and 50 %. This seems to shorten the initial time difference between the computed rewetting time and

the experimental rewetting time, but this also results in increasing the rewetting velocity along the tube, thus giving larger error in the final result. After considerable amount of trial and error, it was decided that the computation method described in chapter 3 (3.2 temperature-time trace curve computation) gives best result. Also, to prevent this end effect error in the final result, the second thermocouple position (about 150 cm from the inlet) instead of the inlet of the test tube is taken as the reference point.

The examples of the computation for temperature-time trace curve is shown in Fig. 5.2 to Fig. 5.10, and it shows mostly close resemblance between the experimental temperature-time trace curves and the computed ones. However, Fig. 5.8 and Fig 5.9 shows a large difference in rewetting time between the experimental T-T plot and the computed one. A close look at these two figures reveals that the rewetting temperature used in the computation is different from the experimental one. A better result could have been obtained if the rewetting temperature used in the computation would have been closer to the experimental rewetting temperature. So, this demonstrates the important role the rewetting temperature plays in the computation of rewetting velocity.

To further prove this point, several other values of rewetting temperature claimed by various authors were used in the computation. Yamanouchi [17] used 150°C as the rewetting temperature in his computation, while Yu et al. chose 180°C . Rewetting temperature of Dua and Tien 260°C , and that of Martini is 420°C . Fig. 5.11 shows the comparison of the experimental temperature-time trace curve with the computed one using Yamanouchi T_q , and using T_q of Yu et al. Fig. 5.12 shows the temperature-time trace curves of experiment and computation using Dua & Tien T_q , and Martini T_q . Fig. 5.11 and Fig. 5.12 shows clearly that better result is obtained by using the apparent rewetting temperature that has been correlated from experiment as given by Eq. (5.2)

It is important to choose appropriate element size in the finite difference method. To show the effect of element size, two sets of temperature-time trace curve was computed using two different element size. The computed T-T plots (temperature-time trace curves) in Fig. 5.5 is computed with element size of 2 cm, while Fig. 5.13 shows the T-T plots which are computed with 1 cm element size. Both figures represent T-T plot for experiment #432 which used mass velocity G of $10 \text{ g/cm}^2\text{s}$. Since mass velocity is slower than others, the element time interval is large

compared to other higher mass velocities ($20 \text{ g/cm}^2 \text{ s}$ and $40 \text{ g/cm}^2 \text{ s}$) for same element size, because the element time interval is the time taken by the elemental volume of water to pass that certain element completely. With mass velocity of $10 \text{ g/cm}^2 \text{ s}$, 2 cm element size represent about 0.2 sec element time interval, and 1 cm element size produce 0.1 sec element time interval. In chapter 3, it was stated that the optimal element time interval is about 0.1 sec.; and if the element time interval is much larger than 0.1 sec then the computation would show instability. This is shown clearly in Fig. 5.5. With element time interval of about 0.2 sec, the computed T-T plot shows the wall temperature drop to below the coolant water temperature after rewetting. Fig. 5.13 with 1 cm element size shows a large improvement in result over Fig. 5.5.

Computation for other experiments which also has mass velocity of $10 \text{ g/cm}^2 \text{ s}$ shows the same effect. Fig. 5.7 shows the computed T-T plot with 2 cm element size for Experiment #436, while Fig. 5.14 shows the computed T-T plots of 1 cm element size. The computed T-T plots of Fig. 5.14 is very close to experimental one, while Fig 5.7 shows a certain discrepancy between the computed and experimental T-T plots.

However, computation results show that using element time interval of less than 0.1 sec does not improve the result significantly, but increase the computation time markedly. The difference in result between computation with 0.05 sec element time interval and 0.1 sec. element time interval is less than 1 %, while the computation time is more than double. So, this shows the element time interval of about 0.1 sec is optimal element time interval.

Computation for all other T-T plots (Fig. 5.2 to Fig. 5.9) were done with 2 cm element size.

In order to see the effect of axial conduction in the surface heat flux, axial conduction term in heat flux computation was purposely neglected, and the results were compared with the result of computation which include the axial conduction term. Comparison of the results show that results are almost identical and the difference in T-T plots or the rewetting velocities is less than 1 %. This confirms the finding of Lee [27] that the contribution of axial conduction to the surface heat flux based on the one-dimensional data reduction is very small.

5.3 REWETTING VELOCITY, U_q

Rewetting velocity U_q is obtained from two temperature-time trace curves at different locations of the test tube by knowing the "rewetting time" and the location of the temperature-time trace curve, because rewetting velocity U_q is the propagation rate of the rewetting front along the test section in the direction of the coolant water flow.

Fig. 5.15 shows the variation of the rewetting velocities at the different axial locations from the inlet of the tube. The experimental rewetting velocities remain almost same, along the test tube, while the computed rewetting velocities show some fluctuations. However, considering the end effect and other difficulties encountered in computation (assumption of initial wall temperature and rewetting temperature distributions etc.), the result shows reasonable consistency along the test tube.

Fig. 5.16 represents the rewetting velocity U_q versus initial wall temperature T_w . Here, it shows almost same pattern for experimental rewetting velocity and computed rewetting velocity using experimental T_q . (i.e. rewetting velocity decreases as the initial wall temperature increase) However, the computed rewetting velocities using correlated T_q show a large difference from the experimental

rewetting velocities. This is probably due to the difference in rewetting temperatures. The correlated rewetting temperature distribution could vary up to 10 % from the experimental rewetting temperature distribution and this seem to give a large difference in result as shown in Fig. 5.16.

Fig. 5.17 and 5.18 show the relationship between rewetting velocity U_q versus mass velocity G . The experimental rewetting velocities and correlated rewetting velocities all show the same pattern of increasing with increase in mass velocity. However, the computed rewetting velocities with correlated T_q show a large difference in value from the experimental rewetting velocities. Again, the difference in rewetting temperature distribution seems to be the source of this difference.

Fig. 5.19 shows the rewetting velocities versus coolant water temperature. Eventhough there is small discrepancy between the rewetting velocities, general pattern is the decrease of rewetting velocity with increase in coolant water temperature.

Fig. 5.20 shows the accuracy of the computed rewetting velocities compared to the experimental rewetting velocities. This shows about 70 % of the computed velocities fall within ± 15 % line, while about 80 % of the computed velocities

have less than 20 % difference from experimental rewetting velocities.

5.4 HEAT TRANSFER COEFFICIENT, h

Fig. 5.21 to Fig. 5.32 shows the heat transfer coefficient distribution versus the wall temperature. This shows a large discrepancy between the computed and the experimental based on one-dimensional data reduction. Most discrepancies occur at the high heat flux region which is Transition Boiling Region and Nucleate Boiling Region rather than in the low heat flux region (Inverted Annular Region and Single Liquid Region). One explanation for this large differences between the Chen's nucleate boiling and Tong's transition boiling heat transfer coefficients given by Eqs. (2.5, 2.16) with our experimental heat transfer coefficients could be the fact that all the boiling heat flux correlations are based on the steady state condition whereas our experiment is transient.

An attempt was made to improve the computed heat transfer coefficient distribution in the Nucleate and Transition Boiling Region. In Chen's nucleate boiling equation, Eq. (2.6), Δp was defined as the difference in vapour pressure corresponding to ΔT ($T_w - T_s$), thus saturation pressure corresponding to the wall temperature was obtained from

steam table. However, to make the boiling heat transfer coefficient distribution closer to the experimental one, ΔP in Chen's nucleate boiling equation was changed such a way that vapour pressure increases linearly with wall temperature. This modification seems to be somewhat extravagant and physically not correct. However, the result shows a large improvement in the heat transfer coefficient distribution. The results are presented in Fig. 5.33 to Fig. 5.44, and even though the maximum computed heat transfer coefficient in nucleate boiling is more than 1.5 times the maximum experimental value, this shows a large similarities between the computed heat transfer coefficient distributions and the experimental ones.

Surprisingly, this modification gave very little change in the results of the computation of the temperature-time trace curve and the rewetting velocity. The difference in values of the rewetting velocity computed using Chen's nucleate boiling correlation and "modified" Chen's nucleate boiling correlation is less than 3 %. From this, it can be said that the effect of the rewetting temperature in the rewetting analysis is much more dominant than the effect of the boiling heat fluxes behind the rewetting front, because our rewetting temperature depends on most of the physical parameters which affect the rewetting process.

such as flow rate, G , initial wall temperature, T_w , coolant temperature, T_c , conductivity of material, K , heat capacity of material, Cp_w , etc. The rewetting velocity is calculated from the rewetting time which is the time taken for the wall to have actual contact with water, and the boiling heat transfer occurs after the wetting of the wall surface, and probably that is the reason for the small effect of the boiling heat transfer coefficient on the computation of the rewetting velocity.

CHAPTER 6

CONCLUSION

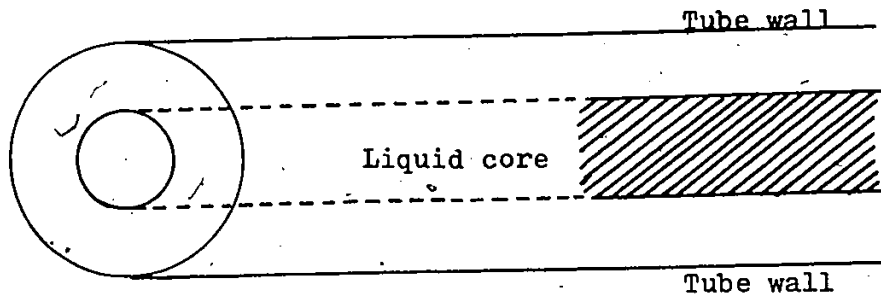
The following conclusions were drawn from the present study of the rewetting phenomena by bottom flooding in vertical channels :

1. In general, a substantial agreement was obtained between the computed results and the experimental results of the particular rewetting processes studied.
2. It is observed that the effect of the boiling heat transfer coefficient in the rewetting is minor compared to the effect of the apparent rewetting temperature, T_q , thus the initial thesis that the effect of the rewetting temperature T_q in the rewetting analysis is much more dominant than other parameters is verified.
3. Both analysis and experiment indicate that for rewetting of a vertical channel by bottom flooding, the rewetting velocities decrease with increase in initial wall temperature and the coolant water temperature, while it increase with increase in mass velocity G .

APPENDIX 1

RADIATION HEAT FLUX

Radiation heat flux in Inverted Annular Flow Region. :
Radiative heat transfer from wall to liquid core is taken
as heat transfer between concentric cylinders.



Now

$$A_l \cdot G_l = \frac{A_l}{A_w} (A_w \cdot J_w) \quad (1)$$

Where

A = Surface area

G = Irradiation = Total radiation incident upon
a surface per unit time and per unit area.

J = Radiosity = Total radiation which leaves
a surface per unit time per unit area.

and subscript l is for liquid core and w is for
tube wall

From Eqn. (1), we obtain

$$G_l = J_w \quad (2)$$

For tube wall surface ;

$$A_w \cdot G_w = A_l \cdot J_l + \left(\frac{A_w - A_l}{A_w} \right) (A_w \cdot J_w) \quad (3)$$

Radiosities of liquid core and tube wall are

$$J_w = W_w + (1 - \alpha_{RW}) \cdot G_w \quad (4)$$

$$J_l = W_l + (1 - \alpha_{RL}) \cdot G_l \quad (5)$$

Where W is total radiation emitted per unit time and per unit surface $(\sigma \cdot T^4)$.

α_{RW} is absorbtivity of tube wall, and

α_{RL} is absorbtivity of liquid core

Eqn. (4), substituting for G_w from Eqn. (3) and rearranging, becomes ;

$$J_w = \frac{A_w \cdot W_w + (1 - \alpha_{RW}) \cdot A_l \cdot W_l}{A_w \alpha_{RW} - \alpha_{RW} \alpha_{RL} \cdot A_l + \alpha_{RL} \cdot A_l} \quad (6)$$

Rearranging Eqn. (5), and substituting for G_l from Eqn. (2) one obtains ;

$$J_l = \frac{A_w \cdot (1 - \alpha_{RL}) \cdot W_w + A_w \alpha_{RW} \cdot W_l + A_l \cdot W_l - \alpha_{RW} \cdot A_l \cdot W_l}{A_w \alpha_{RW} - \alpha_{RW} \alpha_{RL} \cdot A_l + \alpha_{RL} \cdot A_l} \quad (7)$$

The heat flux from tube wall surface to the liquid core per unit wall surface area is ;

$$q_{WL}^R = J_w - G_w = \frac{A_l}{A_w} (J_w - J_l) \quad (8)$$

Substituting for J_w and J_l , Eqn. (8) becomes ;

$$q_{WL}^R = \frac{A_l}{A_w} \cdot \frac{\alpha_{Rl} \cdot W_w - \alpha_{Rw} \cdot W_l}{\alpha_{Rw} - \alpha_{Rl} \cdot \frac{A_l}{A_w} + \alpha_{Rl} \cdot \frac{A_l}{A_w}} \quad (9)$$

By Kirchhoff's law ;

$$\alpha_{Rw} = \epsilon_w \text{ and } \alpha_{Rl} = \epsilon_l$$

Where ϵ_w is emissivity of tube wall and ϵ_l is emissivity of water.

Also $\frac{A_l}{A_w} = \sqrt{1 - \alpha}$ where α is void fraction, $W_w = \sigma \cdot T_w^4$ and $W_l = \sigma \cdot T_l^4$

Therefore, Eqn. (9) becomes ;

$$q_{WL}^R = \sigma \cdot \frac{\sqrt{1 - \alpha} \cdot (T_w^4 - T_l^4)}{\frac{1}{\epsilon_l} + (\frac{1}{\epsilon_w} - 1) \sqrt{1 - \alpha}} \quad (10)$$

APPENDIX 2

NUMBER OF DROPLETS PER UNIT VOLUME

We can express the droplet flux as ;

$$N_d = \frac{\text{Water Mass Flow Rate}}{\text{Each Droplet Mass}} = \frac{G(1-\chi) \cdot A_d}{\frac{4}{3} \pi \cdot \left(\frac{\delta}{2}\right)^3 \cdot \rho_l}$$

$$= 6 \cdot \frac{G(1-\chi) \cdot A_d}{\rho_l \pi \delta^3} \quad (1)$$

Number of droplets per unit volume is calculated by multiplying the droplet flux by unit time then divide by the element volume.

$$N = \frac{N_d \cdot \Delta t}{A_d \cdot dX} = \frac{6 \frac{G(1-\chi) \cdot A_d}{\rho_l \pi \delta^3} \cdot \Delta t}{A_d \cdot dX} = \frac{6 \cdot G(1-\chi) \cdot \Delta t}{\rho_l \cdot \pi \cdot \delta^3 \cdot dX} \quad (2)$$

Where N is the number of droplets per unit volume

Δt is the unit time and dX is the distance

travelled by droplets during time Δt

Therefore, $dX = U \cdot \Delta t$ and U = liquid droplet velocity.

Liquid droplet velocity U is volume flow rate of water divided by the cross sectional area occupied by the water flow.

$$U = \left(\frac{G(1-\chi) \cdot A_d}{\rho_l} \right) / (A_d \cdot (1-\alpha)) = \frac{G(1-\chi)}{\rho_l(1-\alpha)} \quad (3)$$

Substituting for dX , thus introducing Eqn. (3) in Eqn. (2)

we obtain ;

$$N = \frac{6 \cdot G(1-\chi) \cdot \Delta t}{\rho_l \cdot \pi \cdot \delta^3 \cdot U \cdot \Delta t} = \frac{6 \cdot G(1-\chi)}{\rho_l \cdot \pi \cdot \delta^3 \cdot \frac{G(1-\chi)}{\rho_l(1-\alpha)}} = \frac{6(1-\alpha)}{\pi \cdot \delta^3}$$

APPENDIX 3 COMPUTER PROGRAM

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1. C
2. C
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5. C
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7. C
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37. C
38. C

*****
*
* PROGRAM TO COMPUTE THE TEMPERATURE-TIME TRACE CURVE
*
*
* WITH CORRELATED REWETTING TEMPERATURE
*
*****
*
* DIMENSION TWALL( 400),X( 400),DTIME( 400)
* DIMENSION TWATER( 400),TVAPOR( 400)
* DIMENSION TQI(500)
* REAL MYUL,MYUV,KL,KV
* REAL LENGTH
* REAL KW
* D=1.386
* DELTA = 0.102
* Z1 = 53.34
* Z2 = 157.48
* Z3 = 243.84
* Z4 = 274.32
* ASSIGN INITIAL WALL TEMPERATURE AT CERTAIN AXIAL LOCATIONS
* T1 = 614.
* T2 = 600.
* T3 = 539.
* T4 = 520.
* TC = 60.
* G = 20.
* CHOOSE AXIAL LOCATIONS WHERE T-T PLOT IS TO BE COMPUTED AT
* ZK = 157.48
* ZL = 243.84
* KW = 4.48842E-2
* CPW = 0.1274826
* RHOW = 7.86
* EX = 1.
* DEFINE ELEMENT SIZE
* DX = 2.
* K1 = Z1/DX+1.
* K2 = Z2 / DX + 1.
* K3 = Z3/DX + 1.

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39. K4 = Z4/DX + 1.
40. LENGTH = 350.
41. EL = 0.963
42. KTC = ZK/DX + 1.
43. LTC = ZL/DX + 1.
44. LIMIT = LENGTH/DX
45. TSC = 100.-TC
46. CGS = 4.184E7
47. A1 = (CPW*G*DELTA/KW)**(-0.162)
48. VTIME = 0.
49. UREV = G*DX*0.02/0.0001295
50. VAPORK = 0.0145/241.9
51. DELTW = 0.01 *VAPORK*(VREV**0.8)/(CPW*RHOW*DELTA*(D+DELTA))
52. CALCULATE INITIAL WALL TEMPERATURE DISTRIBUTION AND
53. INITIAL REWETTING TEMPERATURE DISTRIBUTION
54. ALONG THE AXIAL LOCATION OF THE TUBE
55. DO 10 I=1,LIMIT
56. IF (I .LE. K1) TWALL(I) = T1
57. IF (I .GT. K1 .AND. I .LE. K2) TWALL(I) = (T2-T1)/FLOAT(K2-K1)
58. 1 *FLOAT(I-K1) + T1
59. IF (I .GT. K2 .AND. I .LE. K3) TWALL(I) = (T3-T2)/FLOAT(K3-K2)*
60. 1 FLOAT(I-K2) + T2
61. IF (I .GT. K3 .AND. I .LE. K4) TWALL(I) = (T4-T3)/FLOAT(K4-K3)*
62. 1 FLOAT(I-K3) + T3
63. IF (I .GT. K4) TWALL(I) = T4
64. TWS = TWALL(I) - 100.
65. TQI(I) = 19.51 * TWS * (TSC/TWS)**0.107 * A1*(CGS*KW*(RHOW**2)
66. 1 *TWS/(DELTA*(G**3.))**(-0.0989)*((FLOAT(I)-0.5)*DX/DELTA)
67. 2 **(-0.163) + 100.
68. 10 CONTINUE
69. TKTC = TWALL(KTC)
70. TLTC = TWALL(LTC)
71. X(1) = 0.
72. TVAPOR(1) = 0.
73. TIME = 0.
74. TIMEL = 0.
75. DO 100 J=1,2000
76. TL = TC
77. TWATER(1) = TC
78. INV = 12

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79. DELT = 0.
80. JCONT = 0
81. DEL = 0.125
82. X(1) = 0.
83. TV = 100.
84. DO 110 I=1,LIMIT
85. NUC = 13.
86. XN = X(I)
87. RSTM = 1
88. TG = TGI(I)
89. IF (I.EQ. 1) GO TO 1
90. TWN1 = TWALL(I-1)
91. GO TO 2
92. 1 TWN1 = TWALL(I)
93. 2 CONTINUE
94. TWN = TWALL(I)
95. IF (I.EQ. LIMIT) GO TO 3
96. TUN2 = TWALL(I+1)
97. GO TO 4
98. 3 TUN2 = TWALL(I)
99. 4 CONTINUE
100. C
101. C
102. C
103. IF (TWATER(I).GE. 100.) GO TO 20
104. IF (TWALL(I).GE. 115.) GO TO 20
105. CALL WEYREG(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,TL,TW,DT)
106. XN = 0.
107. GO TO 40
108. 20 CONTINUE
109. IF (INV.EQ. 13) GO TO 24
110. IF (TWALL(I).GT. TG) GO TO 21
111. IF (X(I).GT. 0.7) GO TO 23
112. CALL NUCROL(TWN1,TWN,TWN2,TG,G,D,NUC,CPW,RHOW,KW,DX,DELTA,TL,
113. 1 XN,TW,DT)
114. IF (NUC.EQ. 12) GO TO 23
115. TV = 100.
116. GO TO 40
117. 21 IF (XN.EQ. 0.) GO TO 23
118. IF (TWALL(I).GT. TG) GO TO 24

```

CHOOSE APPROPRIATE HEAT TRANSFER REGION

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117. CONTINUE
118. IF (INV .EQ. 13) GO TO 24
119. IF (X(I) .GT. 0.7) GO TO 24
120. XH = X(I)
121. IF (I .EQ. 1) GO TO 7
122. XH1 = X(I-1)
123. GO TO 8
124. XH1 = 0.
125.
126.
127. CONTINUE
128. CALL TRANS(TWN1,TWN,TWN2,TQ,G,D,CPW,RHOW,KW,DX,DELTA,XH1,XN,
129. TW,DT,TL)
130. TV = 100.
131. GO TO 40
132.
133. 24 IF (X(I) .GT. 0.05) GO TO 25
134. CALL INVARU(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,EL,EX,DX,DELTA,XN,TU,
135. 1 DT,JCONT,TL)
136. TV = 100.
137. INV = 13
138. GO TO 40
139.
140. 25 IF (X(I) .GE. 1.0) GO TO 26
141. TV = TVAPOR(I)
142. CALL DISPER(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,EL,EX,XN,
143. 1 TU,TU,DT,DEL)
144. TL = 100.
145. GO TO 40
146.
147. 26 CALL STEAM(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,EL,EX,TW,TU,DT,
148. 1 XN,RCTM,DEL)
149. TL = 100.
150. XN = 1.0
151.
152. 40 CONTINUE
153. TWALL(I) = TW
154. DTIME(I) = DT
155. IF (I .EQ. LIMIT) GO TO 110
156. TWATER(I+1) = TL
157. X(I+1) = XN
158. TVAPOR(I+1) = TV
159. IF .NOT. (J .EQ. 1) GO TO 110
160. VTIME = DTIME(I) + VTIME
161. DELTAX = DELTAX(TWALL(I+1)-100.)*VTIME
162. TWALL(I+1) = TWALL(I+1)-DELTAX
163.

```

```

159. 110 CONTINUE
160. IF ( .NOT. ( J .EQ. 1 ) ) GO TO 50
161. WRITE(6,41)
162. 41 FORMAT(1H1,9X,'TIME INTERVAL',8X,'WALL TEMP.',9X,'WATER TEMP.',
163. 1 10X,'VAPOR QUALITY',8X,'VAPOR TEMP.',////)
164. DO 51 I=1,LIMIT
165. WRITE(6,52) DTIME(I),TWALL(I),TWATER(I),X(I),TVAPOR(I)
166. 52 FORMAT(1X,5F20.5)
167. 51 CONTINUE
168. DO 53 I=1,KTC
169. TIME = TIME + DTIME(I)
170. 53 CONTINUE
171. DO 56 I = 1,LTC
172. TIME = TIME + DTIME(I)
173. 56 CONTINUE
174. WRITE(6,45) ZK,ZL
175. 45 FORMAT(1H1//30X, 'T/C POSITION =',F7.2,44X,'T/C POSITION =',
176. 1 F7.2//)
177. WRITE(6,43) TKTC,TLTC
178. WRITE(6,44) TQI(KTC),TQI(LTC)
179. 43 FORMAT(31X,'INITIAL TWALL =',F6.1,44X,'INITIAL TWALL =',F6.1/)
180. 44 FORMAT(31X,'REWETTING TEMP. =',F6.1,44X,'REWETTING TEMP. =',F6.1
181. 1 ////)
182. WRITE(6,42)
183. 42 FORMAT(5X,'TIME',6X,'WALL TEMP.',6X,'WATER TEMP.',6X,'VAPOR QUALIT
184. 1Y',5X,'*',9X,'TIME',6X,'WALL TEMP.',6X,'WATER TEMP.',6X,
185. 2 'VAPOR QUALITY',//)
186. GO TO 54
187. 50 CONTINUE
188. TIME = TIME + DTIME(KTC)
189. TIME = TIME + DTIME(LTC)
190. 54 CONTINUE
191. WRITE(6,55) TIME,TWALL(KTC),TWATER(KTC),X(KTC),TIME,TWALL(LTC),
192. 1 TWATER(LTC),X(LTC)
193. 55 FORMAT(1X,F9.3,F13.2,F16.2,F19.5,8X,'*',8X,F6.3,5X,F8.2,8X,F8.2,
194. 1 9X,F10.5/)
195. IF (TWALL(LTC) .LE. 100.) GO TO 70
196. 100 CONTINUE
197. 70 CONTINUE
198. STOP
199. END

```

```

200. SUBROUTINE WETREG(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,TL,TW,
201. 1 DT)
202. REAL MYUL,MYUV,KL,KV
203. REAL KW
204. AS = 3.1416*DX*DX
205. AD = 3.1416*DX*2./4.
206. A = (3.1416/4.)*(D+2.*DELTA)**2.-D**2.)
207. VOLW = A*DX
208. CPL = 1.0
209. CALL WATER(TL,RHOL,MYUL,KL)
210. VEL = G/RHOL
211. RE = G*DX/MYUL
212. PR = CPL*MYUL/KL
213. IF (RE .GE. 1000.) GO TO 10
214. IF (RE .LE. 2000.) GO TO 20
215. H = (48./11.)*(KL/D) + (RE-2000.)*(0.023*KL/DX*(10000.**0.8)*
216. 1 (PR**0.4) - (48./11.)*(KL/D))/(10000.-2000.)
217. GO TO 30
218. 10 H = 0.023*KL/DX*(RE**0.8)*(PR**0.4)
219. GO TO 30
220. 20 H = (48./11.)*(KL/D)
221. 30 CONTINUE
222. QC = H*(TWN-TL)*AS
223. Q = QC + KW*(2.*TWN-TWN1-TWN2)/DX
224. DT = DX/VEL
225. DTW = Q*DT/(CPW*VOLW*RHOW)
226. TW = TWN-DTW
227. DTL = QC/(CPL*G*AD)
228. TL = TL+DTL
229. RETURN
230. END

```

SUBROUTINE NUCBOL(TWN1,TWN2,TQ,G,D,NUC,CPW,RHOL,KW,DX,DELTA,

1 TL,XN,TW,DT)

REAL MYUL,MYUV,KL,KV

REAL KW

CALL WATER(TL,RHOL,MYUL,KL)

CPL = 1.0

GC = 1.0

SIGMA = 58.9

HFG = 539.0556

RHOV = 5.9587E-4

TS = 100.

PSS = 14.696*144./0.00209

AS = 3.1416 * D * DX

VOLW = 3.1416*((D/2.+DELTA)**2.-(D/2.))**2.)*DX

A = (3.1416/4.)*((D+2.*DELTA)**2.-D**2.)

AD = 3.1416 * D**2. / 4.

MYUV = 1.29464E-4

HMI = 0.00122*(KL**0.79*CPL**0.45*RHOL**0.49*GC**0.25)/

1 (SIGMA**0.5*MYUL**0.29*HFG**0.24*RHOV**0.24)

PRL = CPL*MYUL/KL

HMA = 0.023*PRL**0.4*KL/D

CALL VOIDF(XN,HMA)

REL = G*(1.-XN)*MYUL*(1.-ALPHA)

XTTF = (XN/(1.-XN))**0.9 * (RHOL/RHOV)**0.5 * (MYUV/MYUL)**0.1

CALL FF(XTTF,F)

RE = REL*F**1.25

CALL SF(RE,S)

ATW = TW

IF (TWN.LE.100.) ATW=100.

CALL SATPRE(ATW,PSW)

HMIC = HMI*(ATW-TS)**0.24*(PSW-PSS)**0.75*8

HMAC = HMA*REL**0.8*F

QC = (HMIC+HMAC)*AS*(ATW-TS)

VEL = G*(1.-XN)/(RHOL*(1.-ALPHA))

DT = DX*RHOL/G

DINCH = D/2.54

GBR = G*(30.48**2)*3600./454.

HFGBR = HFG*1.8

ACCOEF = 1.12*DINCH**(-0.211)*(GBR/1.0E6)**(0.324)

CCCOEF = 0.001*DINCH**(-1.4)*(GBR/1.0E6)**(-1.05)

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271. QCRI = (ACDEF - CCDEF*DINCH*(GBR/1.0E6)*HFGBR*XN/4.)*1.0E6
272. QCRIT = (QCRI/13272.)*AS
273. IF (QC.GT. QCRIT) NUC = 12
274. IF (NUC.EQ. 12) GO TO 30
275. Q = QC + KW*(2.*TWN-TWN1-TWN2)/DX
276. DTW = Q*DT/(CPW*VOLW*RHOW)
277. TW = TWN-DTW
278. IF (TL.LT. 70.) GO TO 3
279. IF (TL.GE. 100.) GO TO 1
280. IF (TL.LT. 100.) GO TO 2
281. NUC = 13
282. 1 DMFG = QC/HFG
    XN = XN+DMFG/(G*AD)
    GO TO 40
283. 2 DMFG = QC*0.33/HFG
    XN = XN+DMFG/(G*AD)
284. DTL = QC*0.67/(G*(1.-XN)*AD*CPL)
    TL = TL+DTL
285. IF (TL.LE. 100.) GO TO 40
286. TLE = TL - 100.
287. QCE = TLE*G*(1.-XN)*AD*CPL
288. DMFG = QCE/HFG
289. XN = XN+DMFG/(G*AD)
290. GO TO 40
291. 3 DTL = QC/(G*(1.-XN)*AD*CPL)
    TL = TL+DTL
292. IF (TL.LE. 100.) GO TO 40
293. TLE = TL-100.
294. QCE = TLE*G*(1.-XN)*AD*CPL
295. DMFG = QCE/HFG
296. XN = XN+DMFG/(G*AD)
297. GO TO 40
298. 40 CONTINUE
299. NUC = 13
300. 30 CONTINUE
301. RETURN
302. END
303.
304.
305.
306.
307.

```

```

309. SUBROUTINE TRANST(TWN1,TWN2,TQ,G,D,CPW,RHOW,KW,DX,DELTA,XN1,
310. 1 XN,TQ,DT,TL)
311. REAL NYUL,MYUV,KL,KV,KW
312. CFL = 1.0
313. CC = 1.0
314. SIGMA = 58.9
315. CALL WATER(TL,RHOL,NYUL,KL)
316. HFO = 537.0353
317. RHOU = 5.7557E-4
318. TQ = 100.
319. PCC = 14.6762144/0.00207.
320. AS = 3.1416 * D * DX
321. A = (3.1416/4.)*((D+2.)*DELTA)**2.-D**2.)
322. VOLU = 3.1416*((D/2.)*DELTA)**2.-(D/2.)***2.)*DX
323. AS = 3.1416 * D**2. / 4.
324. MYUV = 3.27434E-4
325. CPV = 0.131
326. KV = 3.294E-5
327. HPI = 0.00122*(KL**0.79*CFL**0.45*RHOL**0.49*CC**0.25)/
328. 1 (SIGMA**0.5*NYUL**0.29*HFG**0.24*RHOV**0.24)
329. PRL = CFL*NYUL/KL
330. HMG = 0.0234*PRL**0.4*KL/D
331. CALL VOIDF(XN,ALPHA)
332. KCL = 0*(1.-XN)*ED/(NYUL*(1.-ALPHA))
333. XTTF = (XN/(1.-XN))**0.9 * (RHOL/RHOV)**0.5 * (MYUV/NYUL)**0.1
334. CALL FF(XTTF,F)
335. RE = RELREF**1.25
336. CALL SF(RE,S)
337. DXTEL = (XN-XN1)/DX
338. ATW = TUN
339. IF (TWN.GT. TQ) ATW = TQ
340. IF (TWN.LE. 100.) ATW = 100.
341. IF ( ATW .GE. 350. ) ETW = 350.
342. IF ( ATW .LT. 350. ) ETW = ATW
343. CALL SATPRE(ATW,PSW)
344. IF (XN.EQ. 0.)AND. TWN.GT. TQ) PSW=PSS
345. HNIC = HNIC(ATW-TS)**0.24*(PSW-PSS)**0.75*S
346. HNIC = HNIC*REL**0.78*F
347. IF (XN.EQ. 0.) GO TO 104
348. DE = GASORT(1,-ALPHA)
349. GLIP = (RHOL/RHOV)*((1.-ALPHA)/ALPHA)*(XN/(1.-XN))
350. HFB = 0.023*(KV/DE)*(G*XN*DE/NYUV)**0.8*((XN+SLIP*(1.-XN))*
351. 1 (RHOU/RHOL))/(XN+SLIP*(1.-XN))**0.8*(MYUV*CPV/KV)**0.4
352. ANB = (G*IC*HNIC)*(ATW-TS)*AS
353. SFB = HFB*(ATW-TS)
354. IF (DXTEL.EQ. 0.) GO TO 10

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354. CHECK=GER*(2./3.)/DXTDL
355. IF (CHECK .GT. 3.E4) GO TO 10
356. GC = GDR*EXP(-0.001/2.54*(XN**2./3.)/DXTDL)*(BTW-TS)*1.8/100.)
357. 1. **((1.+0.0016*(BTW-TS)*1.8))+ QFB*AS
358. GO TO 20
359. 10 GC = (HMC+HMAC)*(ATW-TS)*AS
360. 20 HINCH = D/2.54
361. GER = GC*(30.48**2)*3600./454.
362. HFOGR = HFG*1.8
363. AGGER = 1.12*HINCH*(0.211)*(GER/1.0E6)**(0.324)
364. CCOEF = 0.001*HINCH*(-1.4)*(GER/1.0E6)**(-1.05)
365. GCRI = (ACCOEF-CCOEF)*HINCH*(GER/1.0E6)*HFOGR*(XN/4.)*1.0E6
366. GCRIT = (GCRI/15272.)*%S
367. IF (GC .GT. GCRIT) GC = GCRIT
368. R = GC + KUS*(2.4TWR-TWNI-TWN2)/DX
369. VEL = GC*(1.-XN)/(KICL*(1.-ALPHA))
370. DT = GCRIT/60P*VOLUME*RHOW
371. DTW = GCRIT/60P*VOLUME*RHOW
372. TW = TWNI-DTW
373. IF (TL .LT. 70.) GO TO 3
374. IF (TL .GE. 100.) GO TO 1
375. IF (TL .LT. 100.) GO TO 2
376. DMFG = GC/HFG
377. XN = XN+DMFG/(GXAD)
378. GO TO 40
379. 2 DMFG=GC*0.5/HFG
380. XN = XN+DMFG/(GXAD)
381. DTL = GC*0.5/(GX*(1.-XN)*AD*CP)
382. TL = TL+DTL
383. IF (TL .LE. 100.) GO TO 40
384. TLE = TL - 100.
385. QCE = TLE*GX*(1.-XN)*AD*CP
386. DMFG = QCE/HFG
387. XN = XN+DMFG/(GXAD)
388. GO TO 40
389. 3 DTL = GC/(GX*(1.-XN)*AD*CP)
390. TL = TL+DTL
391. IF (TL .LE. 100.) GO TO 40
392. TLE = TL - 100.
393. QCE = TLE*GX*(1.-XN)*AD*CP
394. DMFG = QCE/HFG
395. XN = XN+DMFG/(GXAD)
396. TL = TL + 100.
397. CONTINUE
398. END

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SUBROUTINE IRVAND(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,EL,EX,DX,DELTA)
1  XN=TW,DT,JCONT,TL)
REAL NYUL,MYUV,KL,KV
REAL KW
RHOL = 0.93768
DPS = 539.0556
RHOU = 5.9587E-4
TS = 100.
CPL = 1.
JCONT = JCONT + 1
AS = 3.1416 * D * DX
A = (3.1416/4.)*(D+2.*DELTA)**2.-D**2.)
VOLW = 3.1416*(D/2.+DELTA)**2.-(D/2.)*2.)*DX
AD = 3.1416*D**2./4.
MYUV = 1.27464E-4
NYUL = 2.682738E-3
CPV = 0.451
KV = 5.974E-5
GRAVD = 32.2*2.54*12.
STERC = 1.3681E-12
TV = 100.
CALL VOIDF(XN,ALPHA)
CALL ENISUT(TWN,EX,TU,EW,EV)
PHIR = STERC*(SGRT(1.-ALPHA)*(TWN+273.)*4.-(TS+273.)*4.))/
1  (1./EL*(1./EW-1.)*SGRT(1.-ALPHA))
RAMDA = HFG * (1./O.4*(TWN-TS)*CPV/HFG)
IF (TWN.LE. TS) PHIC = 0.
IF (TWN.LE. TS) GO TO 11
PHIC = 0.945*(KV**3.*RHOU*(RHOL-RHOW)*GRAVD*(RAMDA/(NYUV*(TWN-TS))*
1  DX*FLUAT(JCONT)))*O.25*(TWN-TS)
11  QC = (PHIR*PHIC)*AS
Q = QC + KW*(2.*TWN-TWN1-TWN2)/DX
VEL = GS*(1.-XN)/(RHOL*(1.-ALPHA))
DT = DX*RHOL/G
DTW = GADT/(CPW*VOLW*RHOW)
TW = TWN-DTW
IF (TL.LT. 70.) GO TO 10
IF (XN.LT. 0.001) GO TO 30
IF (TL.LT. 100. .AND. XN.LE. 0.03) GO TO 40
IF (TL.LT. 100.) GO TO 10

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30 DMFG = QC/HFG
   XN = XN + DMFG/(G*AD)
   GO TO 20
10 DTL = QC/(G*(1.-XN)*AD*CPL)
   TL = TL+DTL
   IF (TL.LE. 100.) GO TO 20
   TLE = TL-100.
   QCE = TLE*G*(1.-XN)*AD*CPL
   DMFG = QCE/HFG
   XN = XN+DMFG/(G*AD)
   TL = 100.
   GO TO 20
40 DTL' = QC*0.5/(G*(1.-XN)*AD*CPL)
   TL = TL+DTL
   DMFG = QC*0.5/HFG
   XN = XN+DMFG/(G*AD)
20 CONTINUE
   RETURN
   END
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459. SUBROUTINE DISPER(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,EL,EX,
460. XN,TW,TU,DT,DEL)
461. REAL MYUL,MYUV,KL,KV
462. REAL MYUVW,KVW
463. REAL KW
464. RHOL = 0.95768
465. MYUL = 2.882738E-3
466. HFG = 539.0556
467. TS = 100.
468. AS = 3.1416*DX
469. A = (3.1416/4.)*(D+2.*DELTA)**2.-D**2.)
470. VOLW = A*DX
471. A2 = 3.1416*(DEL**2.)
472. GRAVC = 32.2*2.54*12.
473. STEBC = 1.3681E-12
474. AD = 3.1416*DX**2./4.
475. CALL VOIDF(XN,ALPHA)
476. DFLUX = 6.*G*(1.-XN)*AD/(RHOL*3.1416*DEL**3.)
477. CALL VAPOR(TU,RHOW,CPW,MYUV,KV)
478. REV = G*XN*D/(MYUV*ALPHA)
479. PRV = CPW*MYUV/KV
480. RED = G*(1.-XN)*DEL/(MYUL*(1.-ALPHA))
481. REL = G*(1.-XN)*D/(MYUL*(1.-ALPHA))
482. PHVIC = (KV/DEL)*(2.+0.74*RED**0.5*PRV***(1./3.))*(TV-TS)
483. CALL VAPOR(TWN,RHOW,CPW,MYUVW,KVW)
484. IF (REV .GE. 10000.) GO TO 11
485. IF (REV .LE. 2000.) GO TO 12
486. H = 3.656*(KV/D)+(REV-2000.)*(0.023*(KV/D)*(10000.**0.8)*(PRV**
487. 1 0.33)*(MYUV/MYUVW)**0.467 - 3.656*(KV/D))/(10000.-2000.)
488. GO TO 13
489. 12 H = 3.656*(KV/D)
490. GO TO 13
491. 11 H = 0.023*(KV/D)*(REV**0.8)*(PRV**0.33)*(MYUV/MYUVW)**0.467
492. 13 CONTINUE
493. PHWVC = H*(TWN-TV)
494. RAMDA = HFG*(1.+0.35*(TWN-TS)*CPW/HFG)**(-3.)
495. VEL = G*(1.-XN)/(RHOL*(1.-ALPHA))
496. PHWLC = (3.1416/4.)*(6.*G*(1.-XN)/(3.1416*VEL*RHOL))***(2./3.)*
497. 1 (KVW**3.*RAMDA*GRAVC*RHOL*RHOW)/(TWN-TS)*MYUVW*(3.1416/6.))**
498. 2 (1./3.)*DEL)**0.25*(TWN-TS)

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499. CALL EMISVT(TWN,EX,TV,EW,EV)
500. CALL EMISVT(TWN,EX,TWN,EW,EV)
501. PHWLR = STEBC*((A2/AS))*((TWN+273.))**4.-(TS+273.))**4.)/(1./EL+
502. 1 (1./EW-1.))*((A2/AS))*((6.*(1.-ALPHA)/(3.1416*DEL**3.)))*(AD*DX)
503. PHWVR = STEBC*(EVW*(TWN+273.))**4.-EV*(TV+273.))**4.)*((EW+1.)/2.))*
504. 1 ALPHA
505. QWVT = (PHWVC+PHWVR)*AS
506. QVIC = PHVIC*(6.*(1.-ALPHA)/DEL)*(AD*DX)
507. IF (QWVT.LE.QVIC) QVIC=QWVT
508. QC = (PHWLC+PHWLR)*AS+QVIC
509. QV = QWVT-QVIC
510. Q = (PHWLC+PHWLR+PHWVC+PHWVR)*AS + KW*AS*(2.*TWN-TWN1-TWN2)/DX
511. DT = DX*RHOL/G
512. DTW = Q*DT/(CPW*VOLW*RHOL)
513. TW = TWN-DTW
514. DMFG = QC/HFG
515. DTV = QV/(CPV*GXN*AD)
516. TV = TV+DTV
517. XN = XN + DMFG/(G*AD)
518. DEL = (6.*G*(1.-XN)*AD/(DFLUX*RHOL*3.1416))**((1./3.))
519. IF (DEL.LE.0.005) DEL = 0.005
520. RETURN
521. END

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522. SUBROUTINE STEAM(TWN1,TWN,TWN2,G,D,CPW,RHOW,KW,DX,DELTA,EL,EX,
523. 1 TW,TU,DT,XN,NSTM,DELT)
524. REAL MYUL,MYUV,KL,KV
525. REAL MYUVW,KVW
526. REAL KW
527. AS = 3.1416*DX
528. A = (3.1416/4.)*((D+2.*DELTA)**2.-D**2.)
529. VOLW = A*DX
530. STEBC = 1.3681E-12
531. AD = 3.1416*DX**2./4.
532. RHOL = 0.95768
533. CALL VAPOR(TV,RHOV,CPV,MYUV,KV)
534. REV = G*DXN /MYUV
535. PRV = CPV*MYUV/KV
536. VELV = G/RHOV
537. DT = DX*RHOV/G
538. IF (REV .GE. 10000.) GO TO 11
539. IF (REV .LE. 2000.) GO TO 12
540. H = (48./11.)*(KV/D) + (REV-2000.)*(0.023*(KV/D)*(10000.**0.8)
541. 1 *(PRV**0.4) - (48./11.)*(KV/D))/(10000.-2000.)
542. GO TO 13
543. 11 H = 0.023*(KV/D)*(REV**0.8)*(PRV**0.4)
544. GO TO 13
545. 12 H = (48./11.)*(KV/D)
546. 13 CONTINUE
547. PHWVC = H*(TWN-TV)
548. CALL EMISVT(TWN,EX,TU,EW,EV)
549. CALL EMISVT(TWN,EX,TWN,EW,EVW)
550. PHWVR = STEBC*(EVW*(TWN+273.))**4.-EV*(TU+273.))**4.)*((EW+1.)/2.)
551. QC = (PHWVC+PHWVR)*AS
552. Q = QC + KW*(2.*TWN-TWN1-TWN2)/DX
553. IF (NSTM .EQ. 1) GO TO 15
554. DTW = Q*DELT/(CPW*VOLW*RHOW)
555. TW = TWN-DTW
556. DTV = QC/(CPV*G*AD)
557. TV = TU+DTV
558. GO TO 17
559. 15 DTW = Q*DT/(CPW*VOLW*RHOW)
560. TW = TWN-DTW
561. DTV = QC/(CPV*G*AD)
562. TV = TU+DTV
563. 17 CONTINUE
564. RETURN
565. END

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566. SUBROUTINE WATER ( TL , RHOL , MYUL , KL )
567. DIMENSION TLD(11),RHOLD(11),MYULD(11),KLD(11)
568. REAL MYUL,MYULD,KL,KLD
569. TL IS TEMPERATURE OF WATER IN DEGREE C
570. RHOL , MYUL , KL IS DENSITY , VISCOSITY AND
571. HEAT CONDUCTIVITY OF WATER , RESPECTIVELY
572. CHANGE TL TO DEGREE F FROM DEGREE C
573. TL = TL * 9. / 5. + 32.
574. DATA TLD/32.,40.,50.,60.,70.,80.,90.,100.,150.,200.,250./
575. DATA RHOLD/62.42,62.43,62.41,62.37,62.3,62.22,62.11,
576. 1 62.0,61.2,60.12,58.8/
577. DATA MYULD/1.20,1.04,0.88,0.76,0.658,0.578,0.514,
578. 1 0.458,0.292,0.205,0.158/
579. DATA KLD/0.319,0.325,0.332,0.340,0.347,0.353,0.359,
580. 1 0.364,0.384,0.394,0.396/
581. DO 10 I=1,11
582. IF(TL.GT. TLD(I)) GO TO 10
583. IF ( TL .LT. TLD(I) ) GO TO 1000
584. RHOL = RHOLD(I-1)+(RHOLD(I) -RHOLD(I-1))/(TLD(I)-TLD(I-1))*
585. 1 (TL-TLD(I-1))
586. MYUL = (MYULD(I-1)+(MYULD(I)-MYULD(I-1))/(TLD(I)-TLD(I-1))*
587. 1 (TL-TLD(I-1))) * 0.001
588. KL = KLD(I-1)+(KLD(I)-KLD(I-1))/(TLD(I)-TLD(I-1))*(TL-TLD(I-1))
589. C NOW CHANGE THE UNITS FROM ENGLISH UNITS INTO C.G.S.
590. C UNITS
591. RHOL = RHOL / 62.43
592. MYUL = MYUL/0.0672
593. KL = KL / 241.9
594. GO TO 11
595. 10 CONTINUE
596. 11 CONTINUE
597. C CHANGE TEMPERATURE BACK TO DEGREE C
598. TL = (TL-32.)*5./9.
599. 1000 RETURN
600. END

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601. SUBROUTINE VOIDF(X,ALPHA)
602. DIMENSION XD(23),ALPHAD(23)
603. C X IS VAPOR QUALITY AND ALPHA IS VOID FRACTION
604. DATA XD/0.,0.00015,0.0003,0.0005,0.001,0.002,0.003,
605. 1 0.004,0.006,0.008,0.01,0.02,0.03,0.04,0.05,0.06,0.1,0.2,0.3,
606. 2 0.4,0.5,0.7,1.0/
607. DATA ALPHAD/0.,0.025,0.09,0.15,0.3,0.41,0.48,0.53,0.59,
608. 1 0.63,0.66,0.75,0.79,0.82,0.84,0.85,0.89,0.93,0.955,0.97,0.98,
609. 2 0.99,1.0/
610. DO 10 I=1,23
611. IF(X.GT. XD(I)) GO TO 5
612. IF (I.EQ. 1) GO TO 20
613. IF (I.EQ. 2) GO TO 19
614. 3 ALPHA = ALPHAD(I-1)+(ALPHAD(I)-ALPHAD(I-1))/(ALOG(XD(I))-
615. 1 ALOG(XD(I-1)))*(ALOG(X)-ALOG(XD(I-1)))
616. GO TO 11
617. 5 IF (I.EQ. 23) GO TO 3
618. 10 CONTINUE
619. 11 CONTINUE
620. GO TO 21
621. 19 ALPHA = 0.025
622. GO TO 21
623. 20 ALPHA = 0.
624. 21 CONTINUE
625. IF(ALPHA .GE. 1.0) ALPHA=1.0
626. RETURN
627. END

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628. SUBROUTINE VAPOR(TV,RHOV,CPV,MYUV,KV)
629. DIMENSION TVD(14),RHOVD(14),CPVD(14),MYUVD(14),KVD(14)
630. REAL MYUV,MYUVD,KV,KVD
631. CHANGE TV TO DEGREE F FROM DEGREE C
632. TV = TV*9./5.+32.
633. DATA TVD/212.,300.,400.,500.,600.,700.,800.,900.,1000.,
634. 1 1200.,1400.,1600.,1800.,2000./
635. DATA RHOVD/0.0372,0.0328,0.0288,0.0258,0.0233,0.0213,
636. 1 0.0196,0.0181,0.0169,0.0149,0.0133,0.0120,0.0109,0.01/
637. DATA CPVD/0.451,0.456,0.462,0.470,0.477,0.485,0.494,
638. 1 0.50,0.51,0.53,0.55,0.56,0.58,0.6/
639. DATA MYUVD/0.87,1.00,1.13,1.26,1.42,1.55,1.70,
640. 1 1.81,1.92,2.14,2.36,2.58,2.81,3.03/
641. DATA KVD/0.0145,0.0171,0.02,0.0228,0.0257,0.0288,
642. 1 0.0321,0.0355,0.0388,0.0457,0.053,0.061,0.068,0.076/
643. DO 10 I=1,14
644. IF(TV.GT. TVD(I)) GO TO 10
645. IF(TV.LT. TVD(1)) GO TO 1000
646. IF(TV.EQ. TVD(1)) GO TO 100
647. TVXI = (TV-TVD(I-1))/(TVD(I)-TVD(I-1))
648. RHOV = RHOVD(I-1)+(RHOVD(I)-RHOVD(I-1))*TVXI
649. CPV = CPVD(I-1)+(CPVD(I)-CPVD(I-1))*TVXI
650. MYUV = (MYUVD(I-1)+(MYUVD(I)-MYUVD(I-1))*TVXI)*1.E-5
651. KV = KVD(I-1)+(KVD(I)-KVD(I-1))*TVXI
652. GO TO 11
653. 10 CONTINUE
654. 100 RHOV = RHOVD(1)
655. CPV = CPVD(1)
656. MYUV = MYUVD(1)*1.E-5
657. KV = KVD(1)
658. 11 CONTINUE
659. C CHANGE THE UNITS FROM ENGLISH UNITS INTO C.G.S. UNITS
660. RHOV = RHOV / 62.43
661. MYUV = MYUV/0.0672
662. KV = KV / 241.9
663. C CHANGE TEMPERATURE BACK TO DEGREE C
664. TV = (TV-32.)*5./9.
665. 1000 RETURN
666. END

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667. SUBROUTINE SATPRE(TSW,PSW)
668. DIMENSION TSWD(17),PD(17)
669. TSW IS TEMPERATURE AND PSW IS CORRESPONDING SATURATION
670. PRESSURE
671. CHANGE TSW TO DEGREE F FROM DEGREE C
672. TSW = TSW*9./5.+32.
673. DATA TSWD/212.,300.,350.,400.,450.,500.,520.,540.,560.,
674. 1 580.,600.,620.,640.,660.,680.,700.,705.4/
675. DATA PD/14.696,67.013,134.63,247.31,422.6,680.8,
676. 1 812.4,962.5,1133.1,1325.8,1542.9,1786.6,2059.7,2365.4,
677. 2 2708.1,3093.7,3206.2/
678. DO 10 I=1,17
679. IF (TSW.GT. TSWD(I)) GO TO 5
680. IF (TSW.LT. TSWD(1)) GO TO 1000
681. IF (TSW.EQ. TSWD(1)) GO TO 100
682. PSW = PD(I-1)+(PD(I)-PD(I-1))/(TSWD(I)-TSWD(I-1))*(TSW-TSWD(I-1))
683. GO TO 11
684. 5 IF (I.EQ. 17) GO TO 13
685. 10 CONTINUE
686. 13 PSW = 3206.2
687. GO TO 11
688. 100 PSW = 14.696
689. 11 CONTINUE
690. C CHANGE THE UNITS FROM ENGLISH UNITS INTO C.G.S. UNITS
691. PSW = PSW * 144. / 0.00209
692. C CHANGE TEMPERATURE BACK TO DEGREE C
693. TSW = (TSW-32.)*5./9.
694. 1000 RETURN
695. END

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696. SUBROUTINE FF(XTTF,F)
697. XTTF IS INVERSE OF MARTINELLI NUMBER
698. F IS REYNOLDS NUMBER FACTOR
699. IF(XTTF.LT.3.) GO TO 10
700. F = 10.*(0.7254257*ALOG10(XTTF)+0.3942467)
701. GO TO 40
702. 10 IF(XTTF.LT.0.5) GO TO 20
703. F = 10.*(0.623388*ALOG10(XTTF)+0.442931)
704. GO TO 40
705. 20 IF(XTTF.LT.0.1) GO TO 30
706. F = 10.*(0.2204226*(ALOG10(XTTF)**2)+0.6164743*ALOG10(XTTF))+
707. 1 0.3960517)
708. GO TO 40
709. 30 F=1.
710. 40 CONTINUE
711. RETURN
712. END

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713. SUBROUTINE SF(RE,S)
714. RE IS TOTAL REYNOLDS NUMBER (RE = REL*F**1.25)
715. S IS SUPPRESSION FACTOR
716. IF(RE.GT.2.E4) GO TO 10
717. S = -0.365412*ALOG10(RE)+2.361648
718. GO TO 40
719. 10 IF(RE.GT.2.E5) GO TO 20
720. S = -0.58*ALOG10(RE)+3.284597
721. GO TO 40
722. 20 IF(RE.GT.4.E5) GO TO 30
723. S = 0.522883*(ALOG10(RE)**2.)-6.033233*ALOG10(RE)+17.498857
724. GO TO 40
725. 30 S = -0.050259*ALOG10(RE)+0.391553
726. 40 CONTINUE
727. IF(S.GE.1.0) S=1.0
728. IF(S.LE.0.1) S=0.1
729. RETURN
730. END

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SUBROUTINE EMISVT(TW,EX,TV,EW,EV)
  CHANGE TEMPERATURE FROM DEGREE C TO DEGREE F
  TW = TW*9./5.+32.
  TV = TV*9./5.+32.
  IF(EX.EQ.2.) GO TO 10
  EW = 0.44+(0.36-0.44)*(TW-420.)/(914.-420.)
  GO TO 20
10 EW = 0.62+(0.73-0.62)*(TW-450.)/(1650.-450.)
20 CONTINUE
  EV = 0.02355*(0.0071/0.015)**((TV-200.)/800.)
  CHANGE TEMPERATURE BACK TO DEGREE C
  TW = (TW-32.)*5./9.
  TV = (TV-32.)*5./9.
  RETURN
END
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TABLE 1
Physical Characteristics of Test Sections

Test Section Number	1	2	3	4
Material	s.s.304	s.s.304	Inconel X-705	s.s.304
Tube O.D mm	19.1	19.1	15.9	15.9
Tube wall Thickness mm	1.65	0.89	1.02	1.02
Tube length approx. m	4	3.5	3.5	3.5

TABLE 2
The Ranges of Experimental Parameters

Coolant flow rate, G , $g/cm^2 s$	10 - 40
Initial wall temperature, $T_w, ^\circ C$	270 - 800
Inlet coolant subcooling, $T_{sub}, ^\circ C$	10 - 80
Pressure, P , bar	1

TABLE 3

Quality x	Void Fraction α
0.00015,	0.025
0.0003	0.09
0.0005	0.15
0.001	0.3
0.002	0.41
0.003	0.48
0.004	0.53
0.006	0.59
0.008	0.63
0.01	0.66
0.02	0.75
0.03	0.79
0.04	0.82
0.05	0.84
0.06	0.85
0.1	0.89
0.2	0.93
0.3	0.955
0.4	0.97
0.5	0.98
0.7	0.99
1.0	1.0

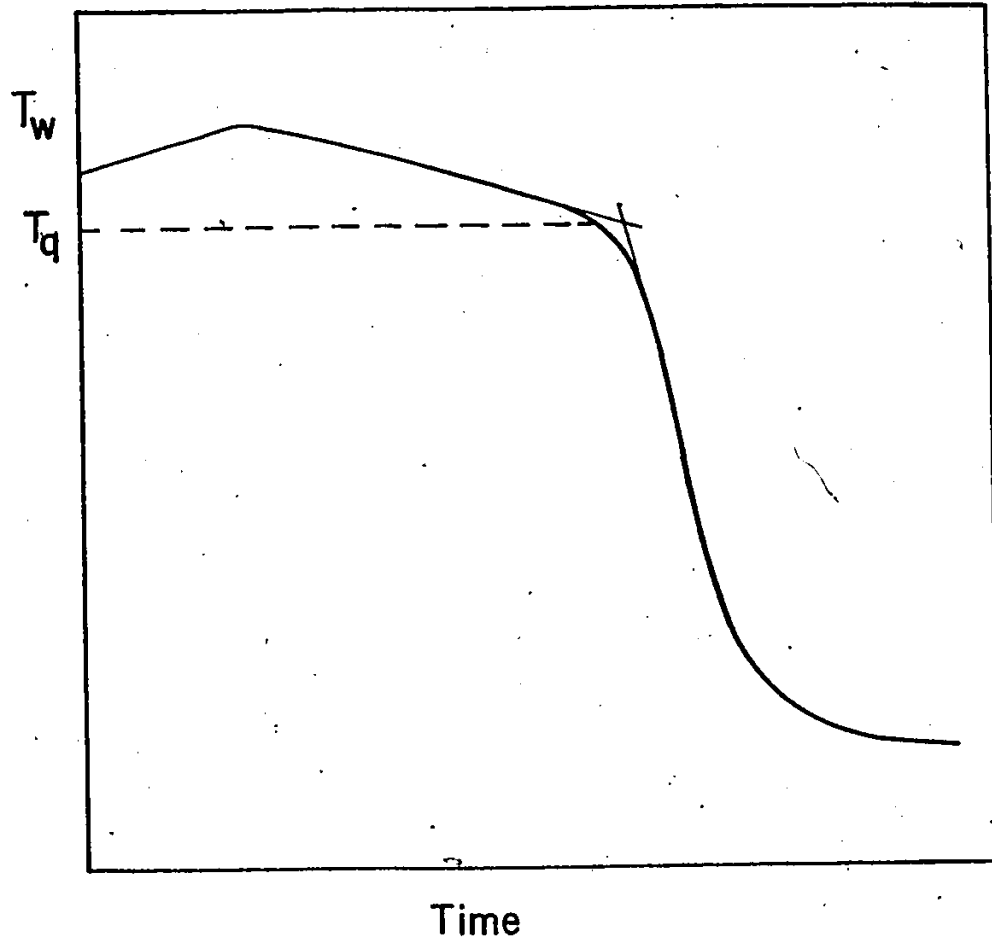


Fig. 2.1 Rewetting Temperature

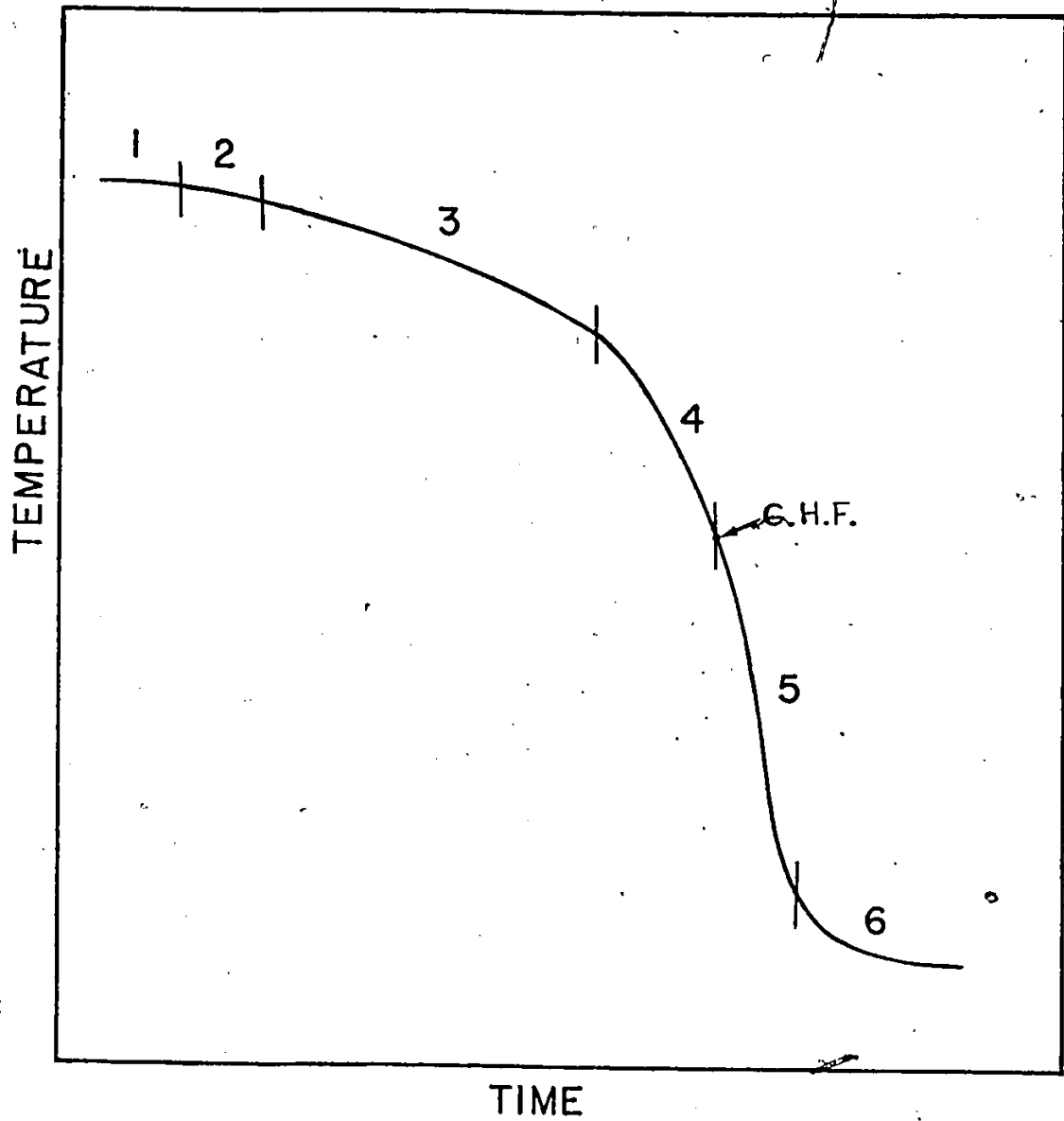


Fig. 2.2 Flow Regimes

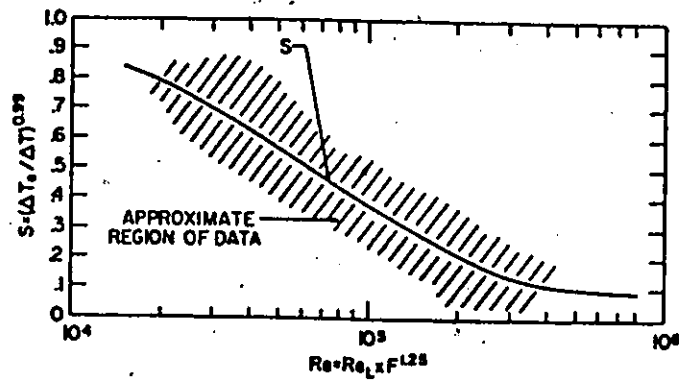


Fig. 2.3 Suppression Factor, S_p

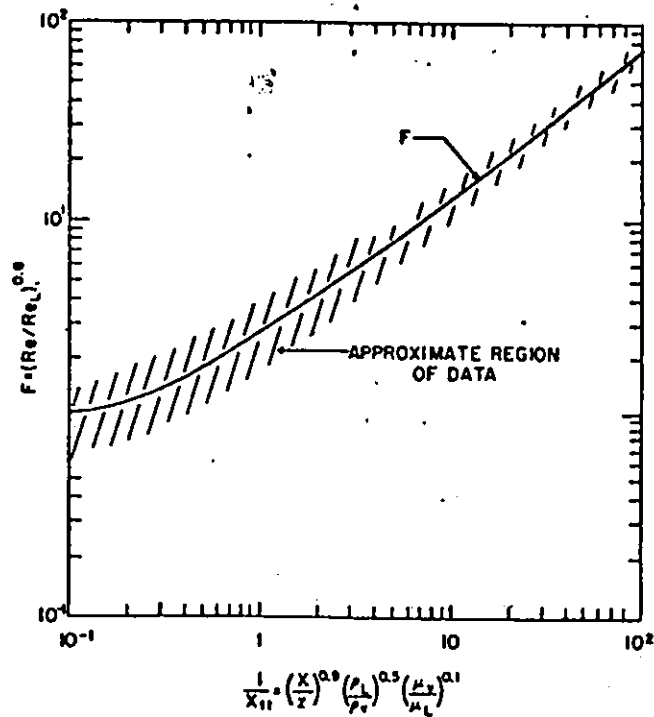


Fig. 2.4 Reynolds Number Factor, F

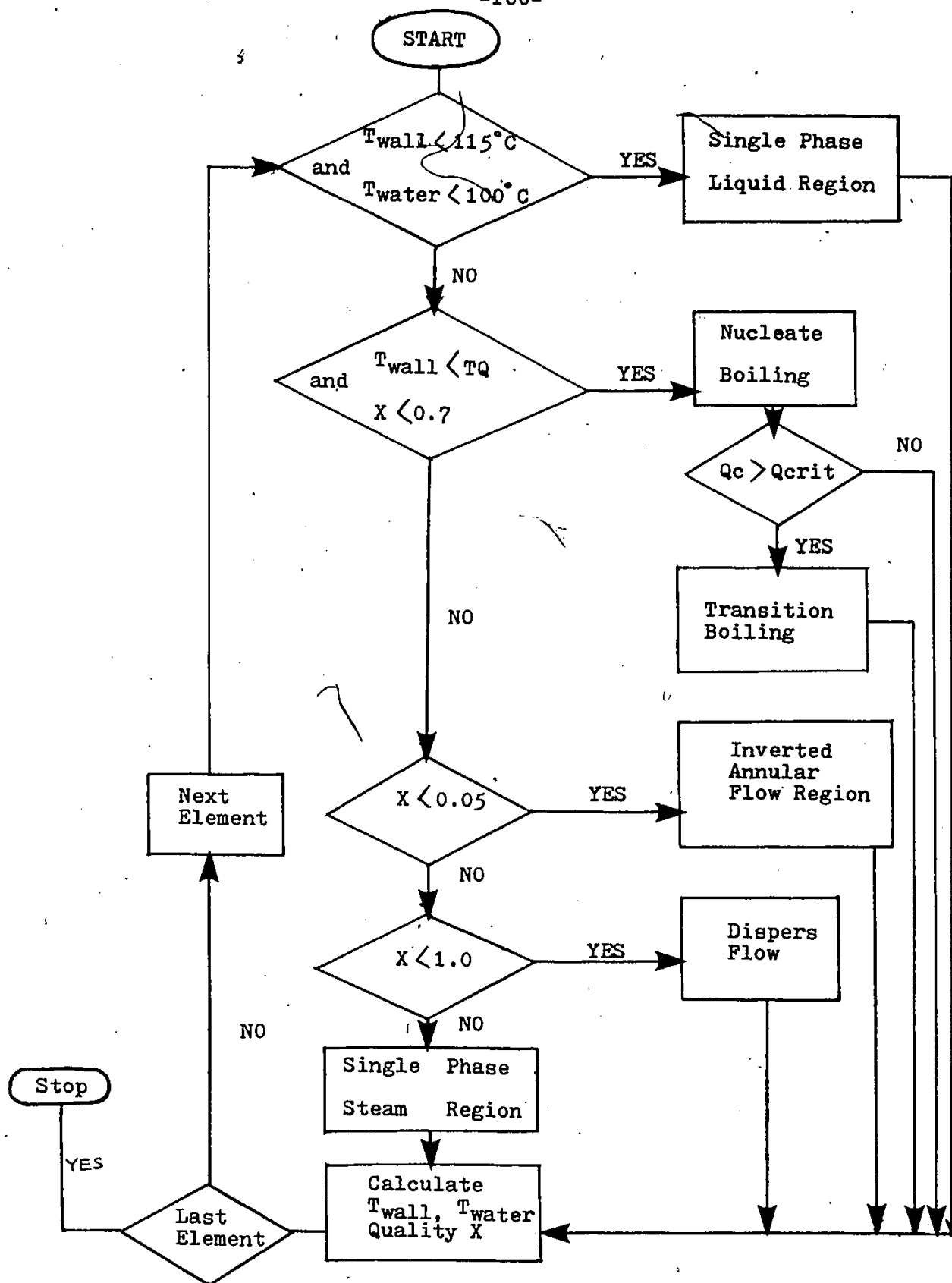
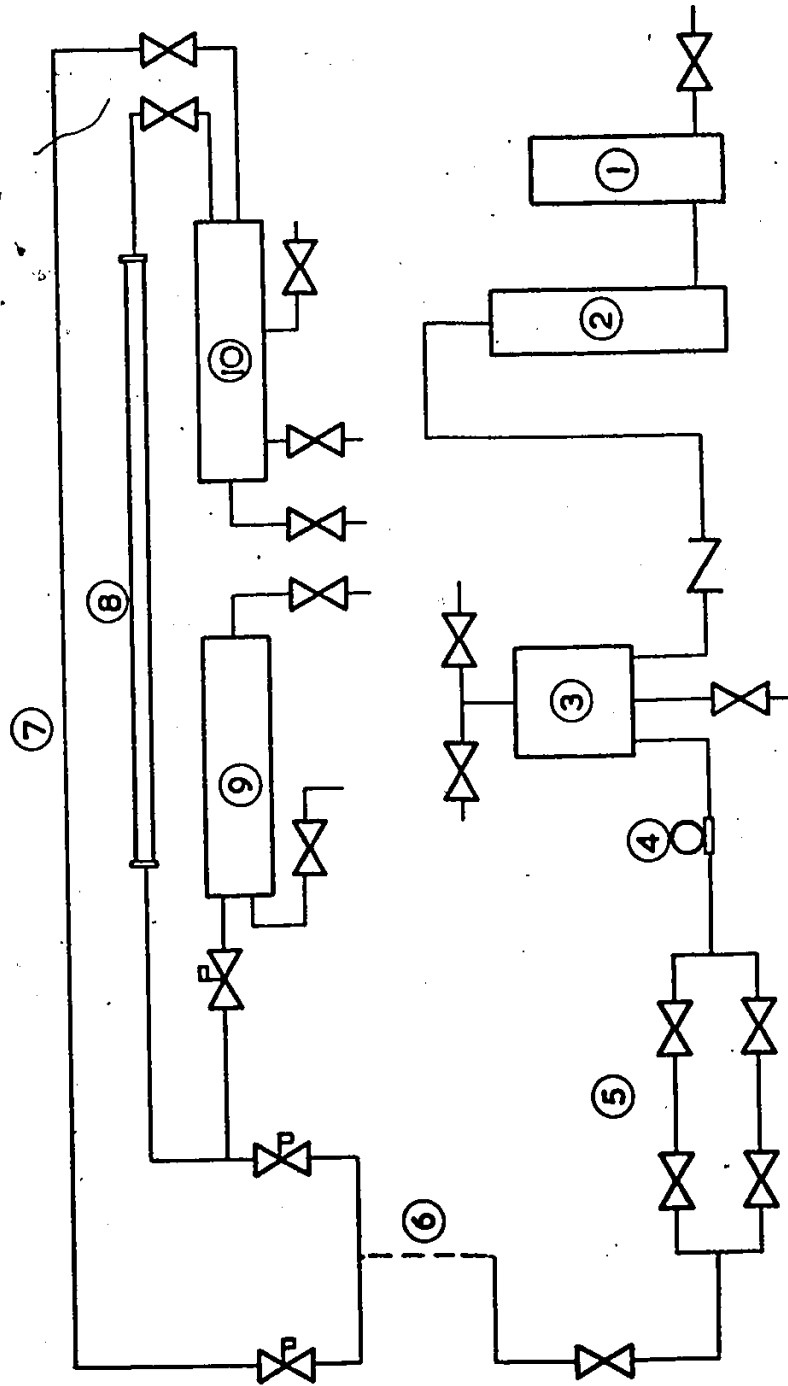


Fig. 3.1 Flow Chart



- | | | |
|------------------------|------------------------|---------------------|
| 1. Water Demineralizer | 5. Flow Measurement | 9. Reverse Tank |
| 2. Preheater | 6. Flexible Connection | 10. Condensing Tank |
| 3. Main Supply Boiler | 7. By-pass | |
| 4. Flow Circulator | 8. Test-Section | |

Fig. 4.1 Test Apparatus

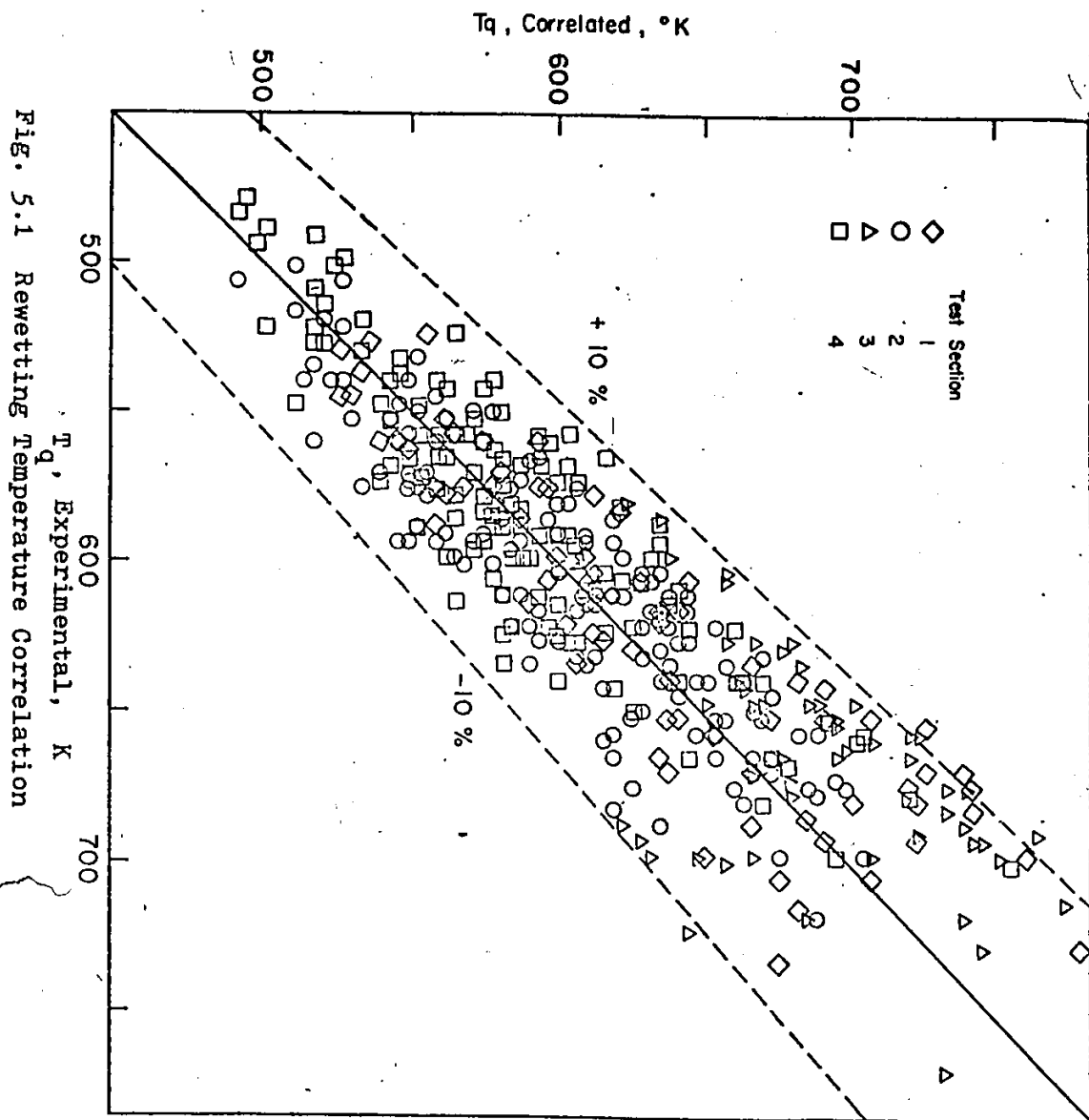


Fig. 5.1 Rewetting Temperature Correlation

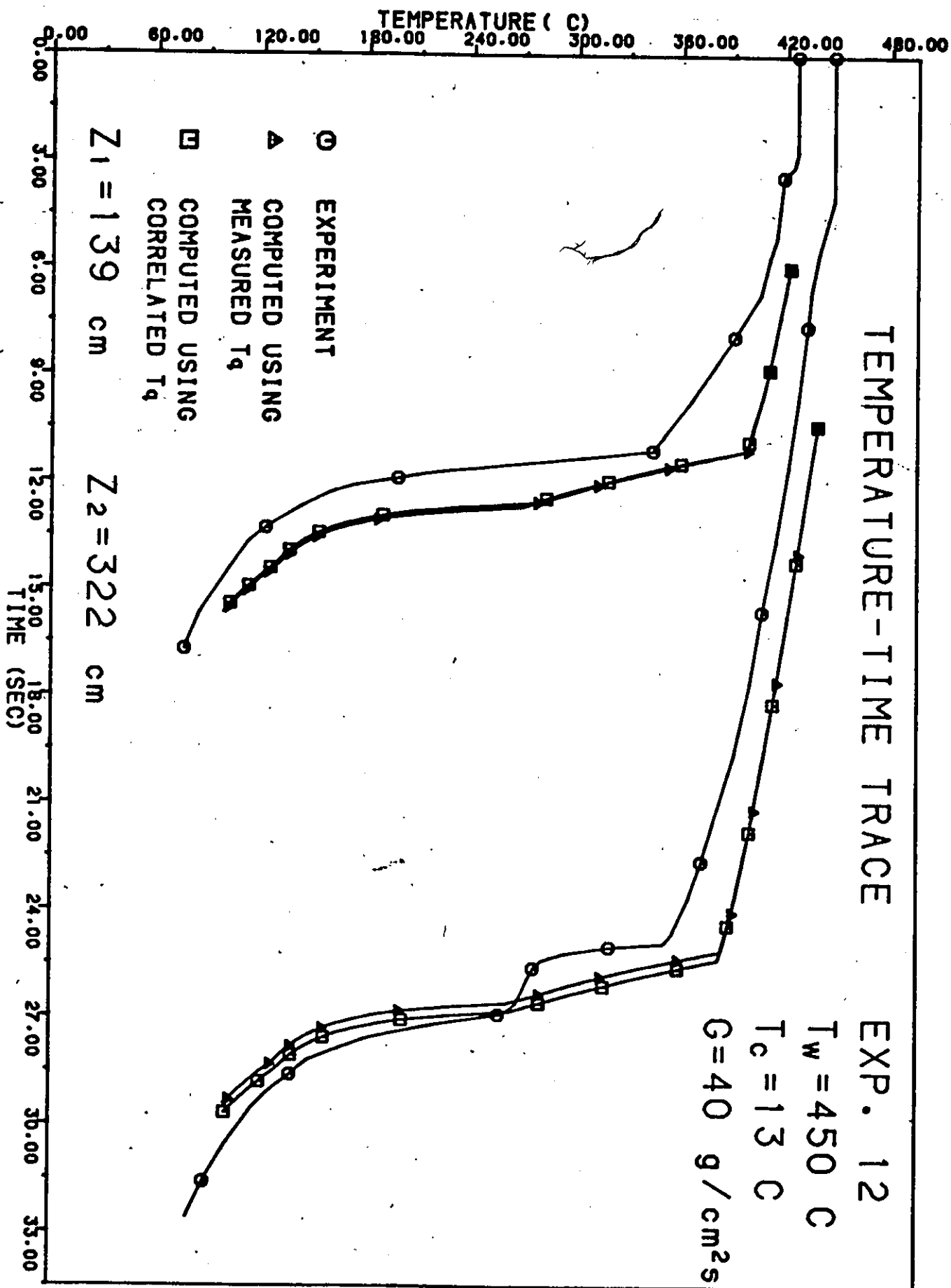


Fig. 5.2 T-T Plots with DX of 2 cm

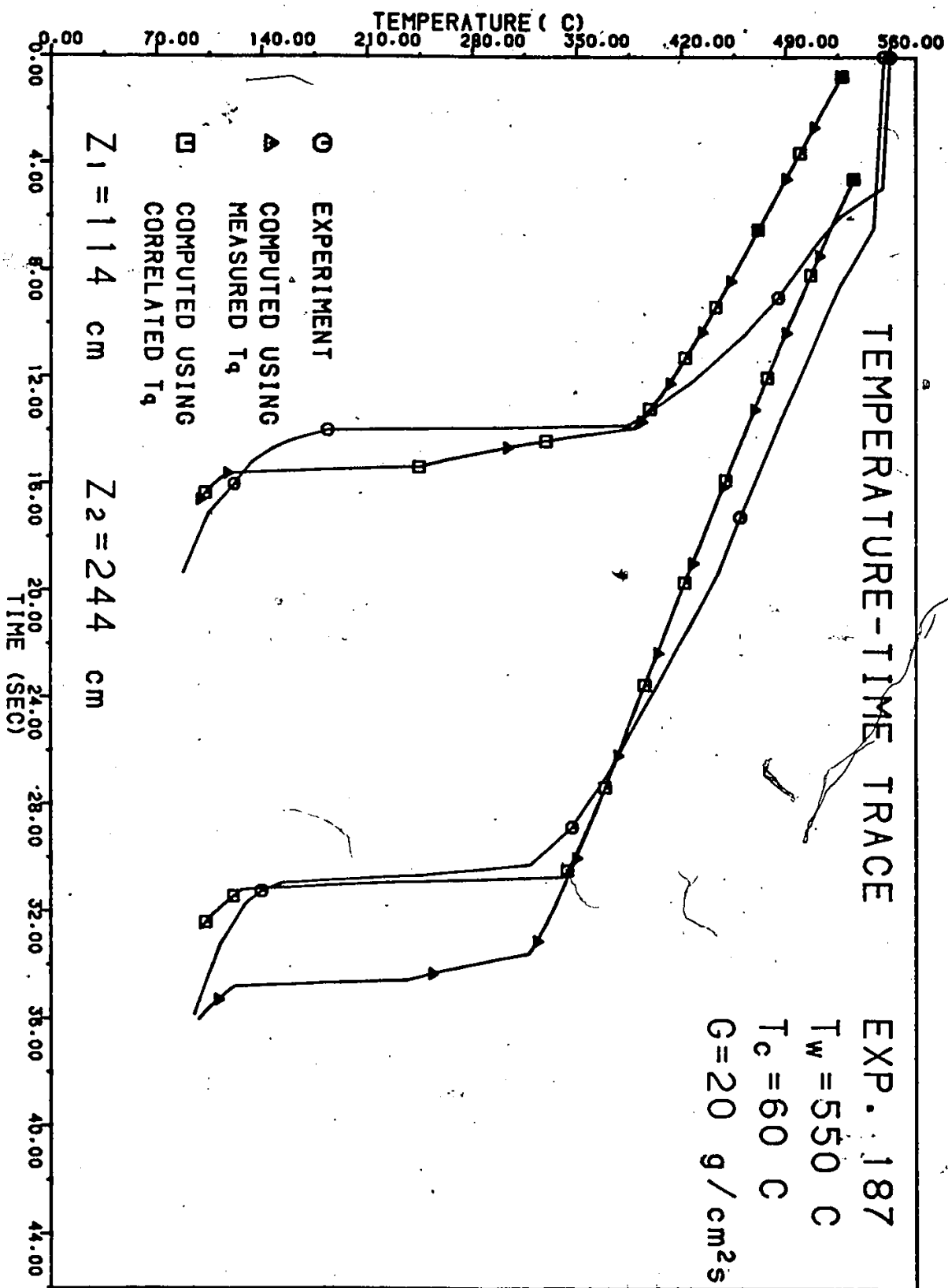


Fig. 5.3 T-T Plots with DX of 2 cm

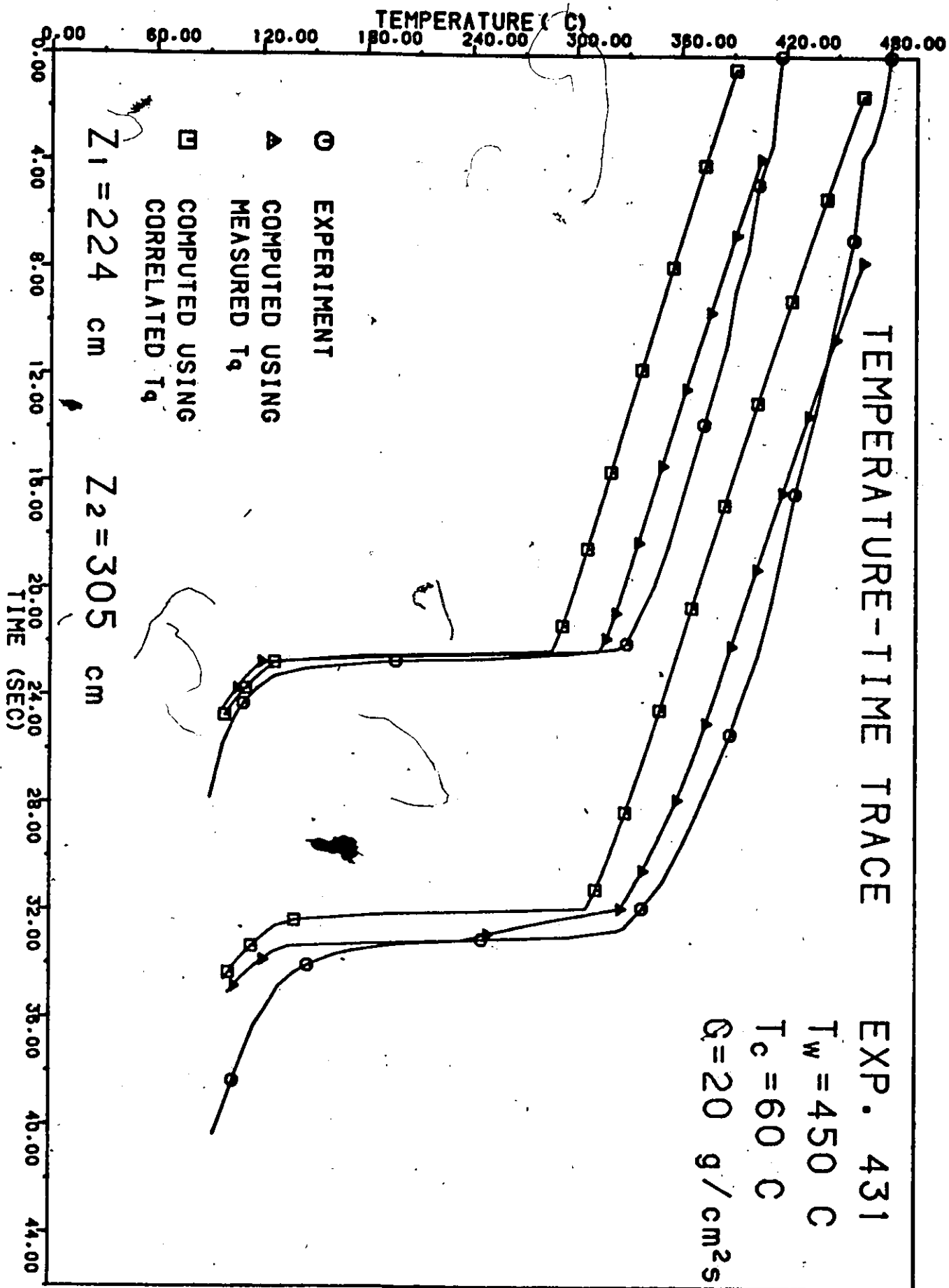


Fig. 5.4 T-T Plots with DX of 2 cm

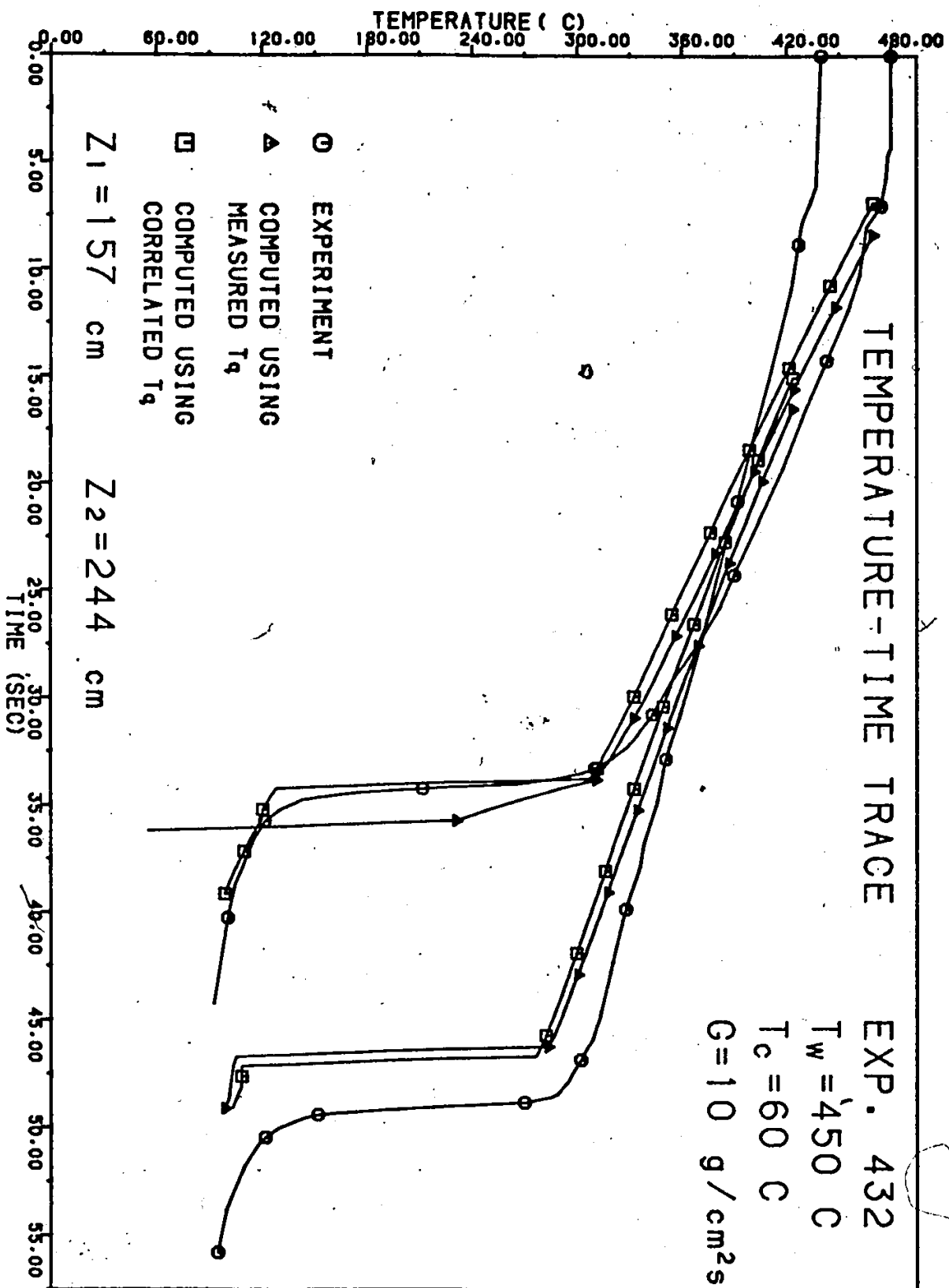


Fig. 5.5 T-T Plots with DX of 2 cm

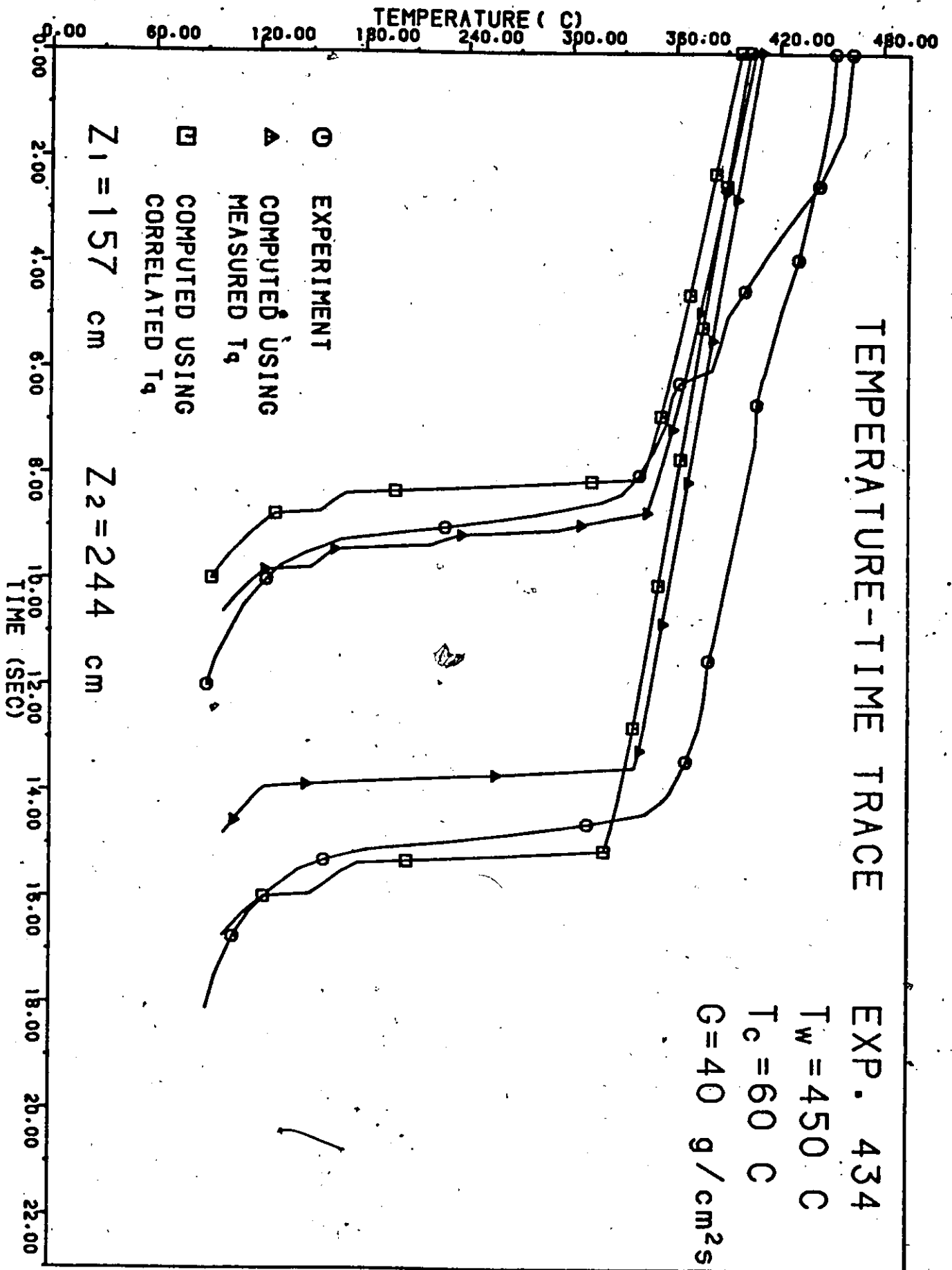


Fig. 5.6 T-T Plots with DX of 2 cm

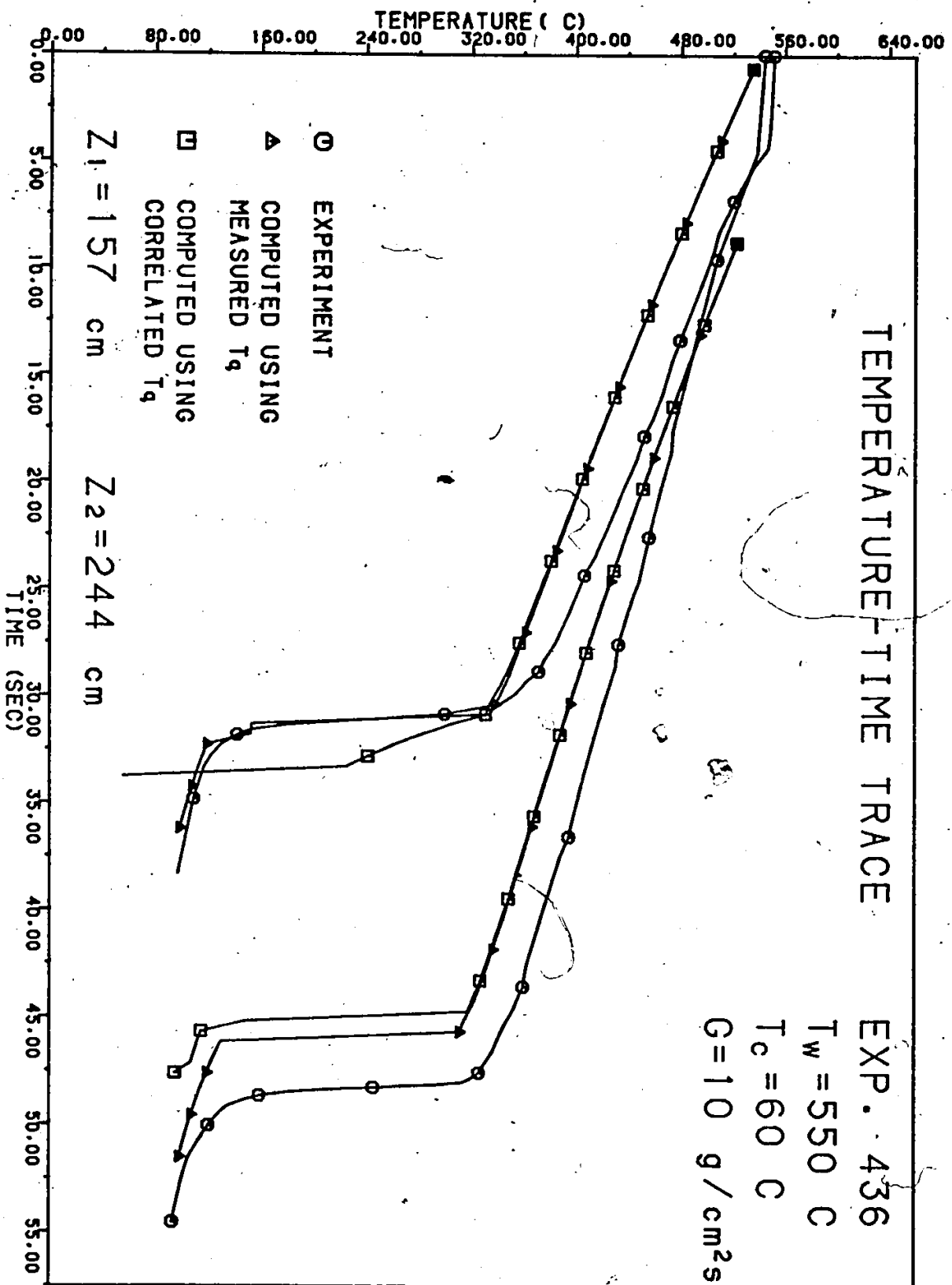


Fig. 5.7 T-T Plots with DX of 2 cm

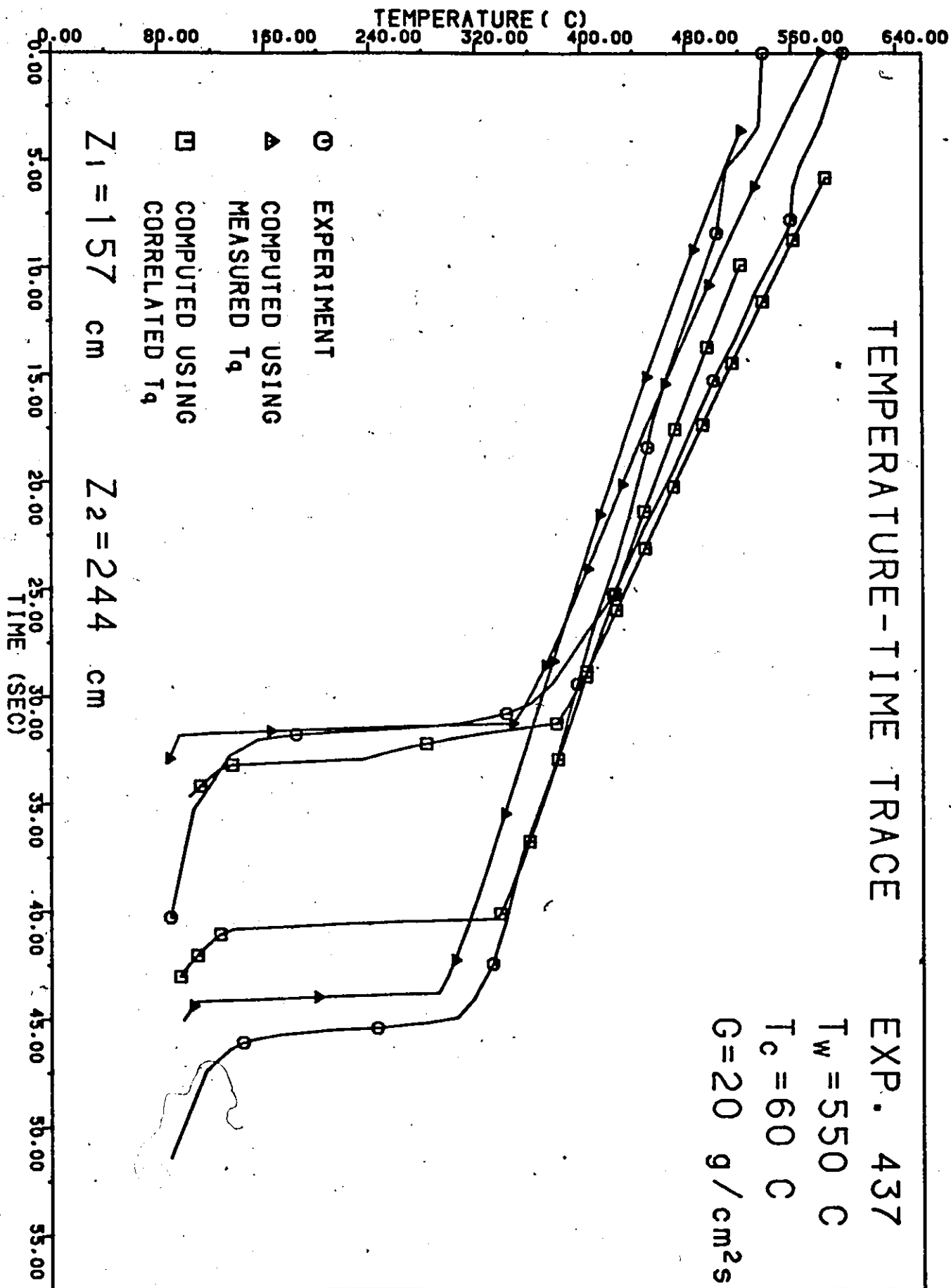


Fig. 5.8 T-T Plots with DX of 2 cm

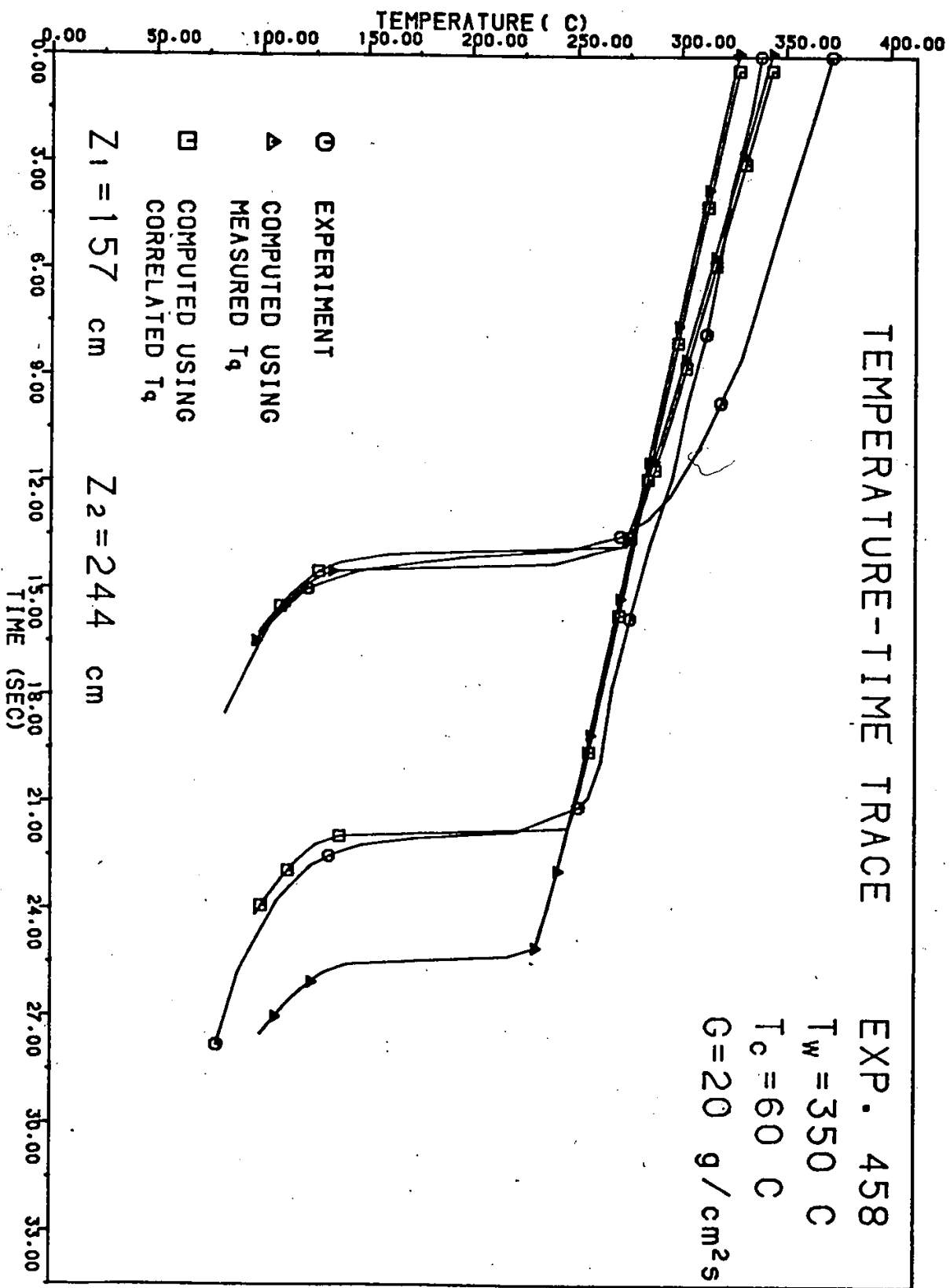


Fig. 5.9 T-T Plots with DX of 2 cm

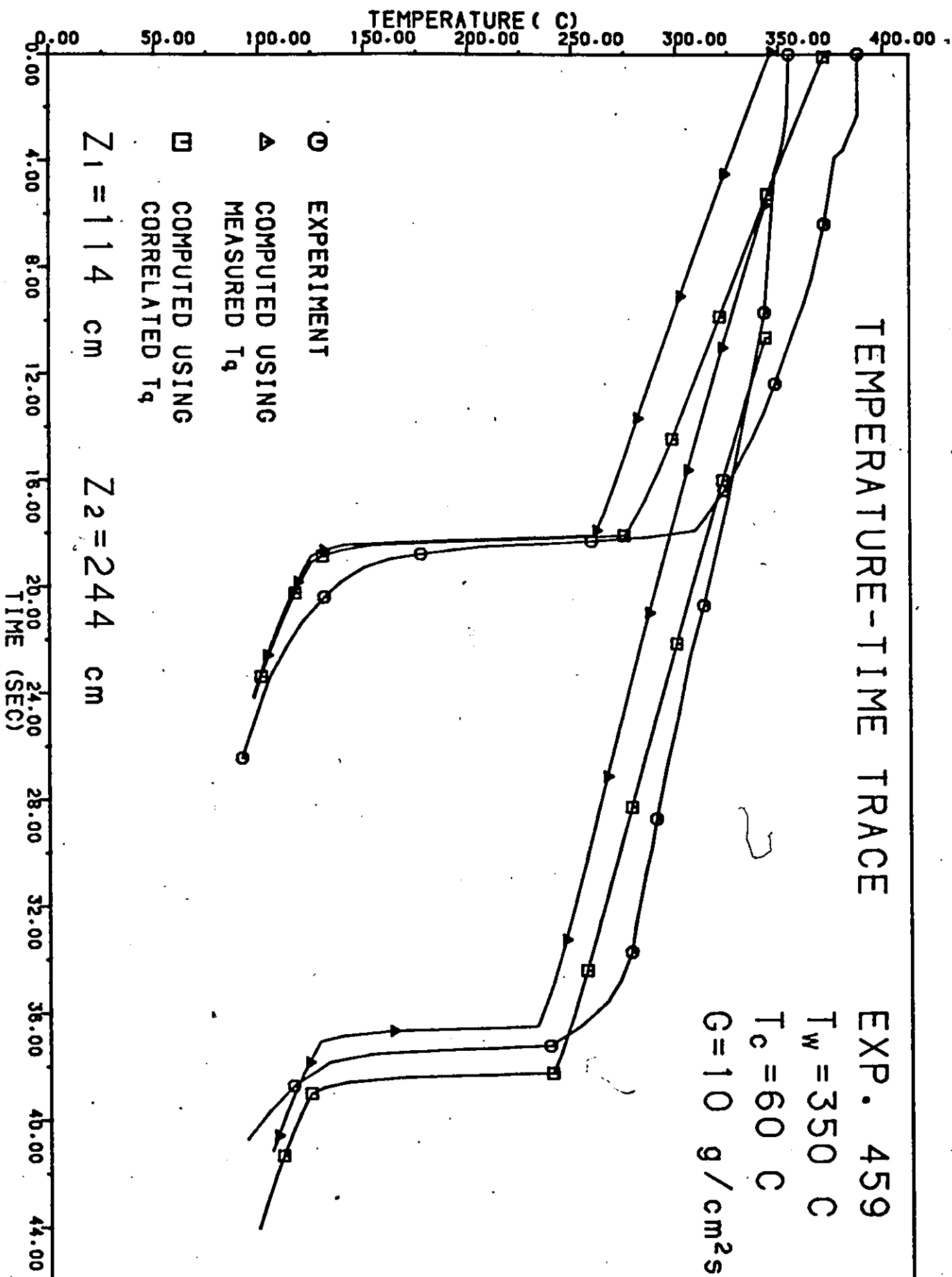


Fig. 5.10 T-T Plots with DX of 1 cm

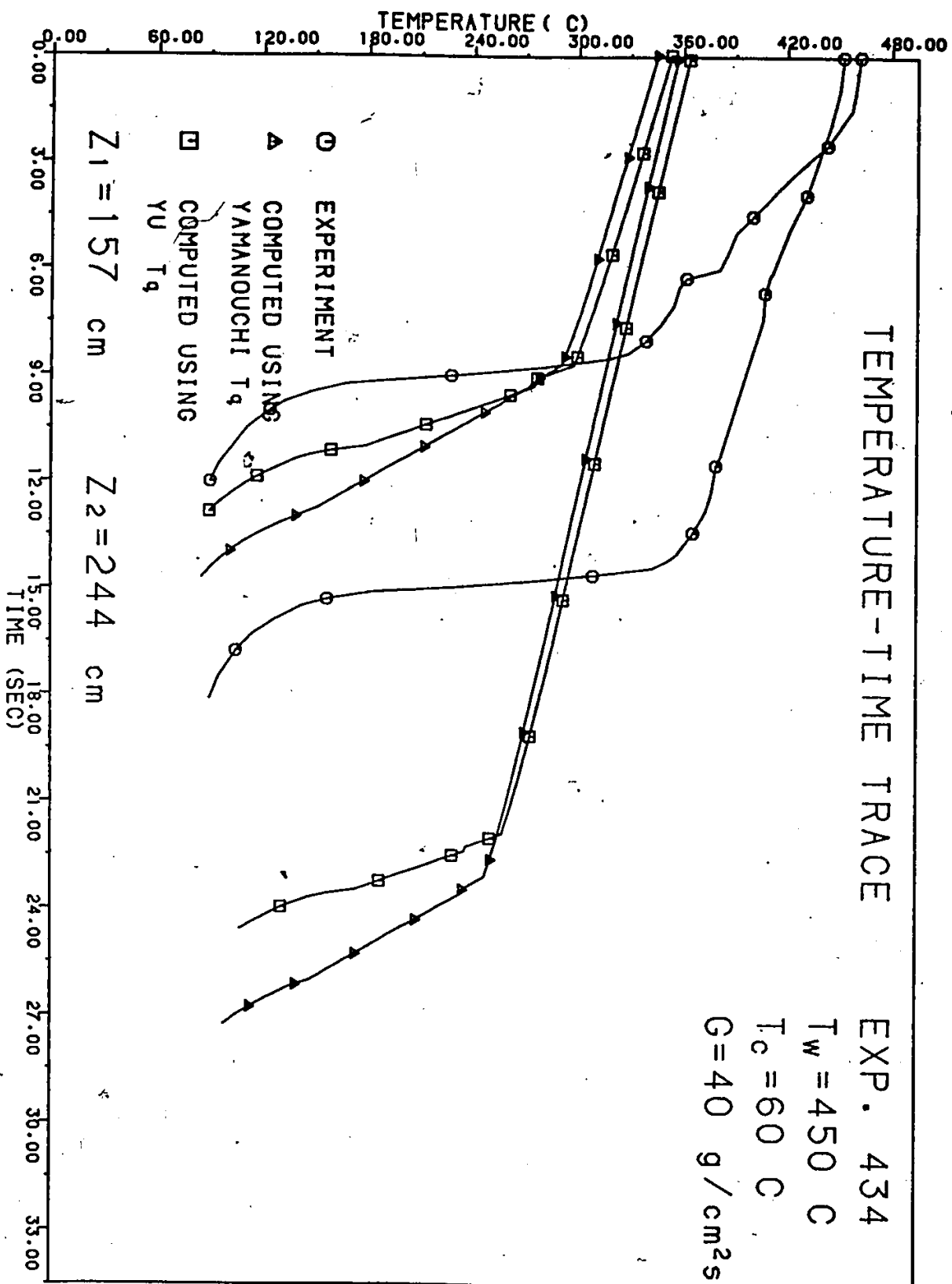


Fig. 5.11 T-T Plots with Yamanouchi and Yu T_q

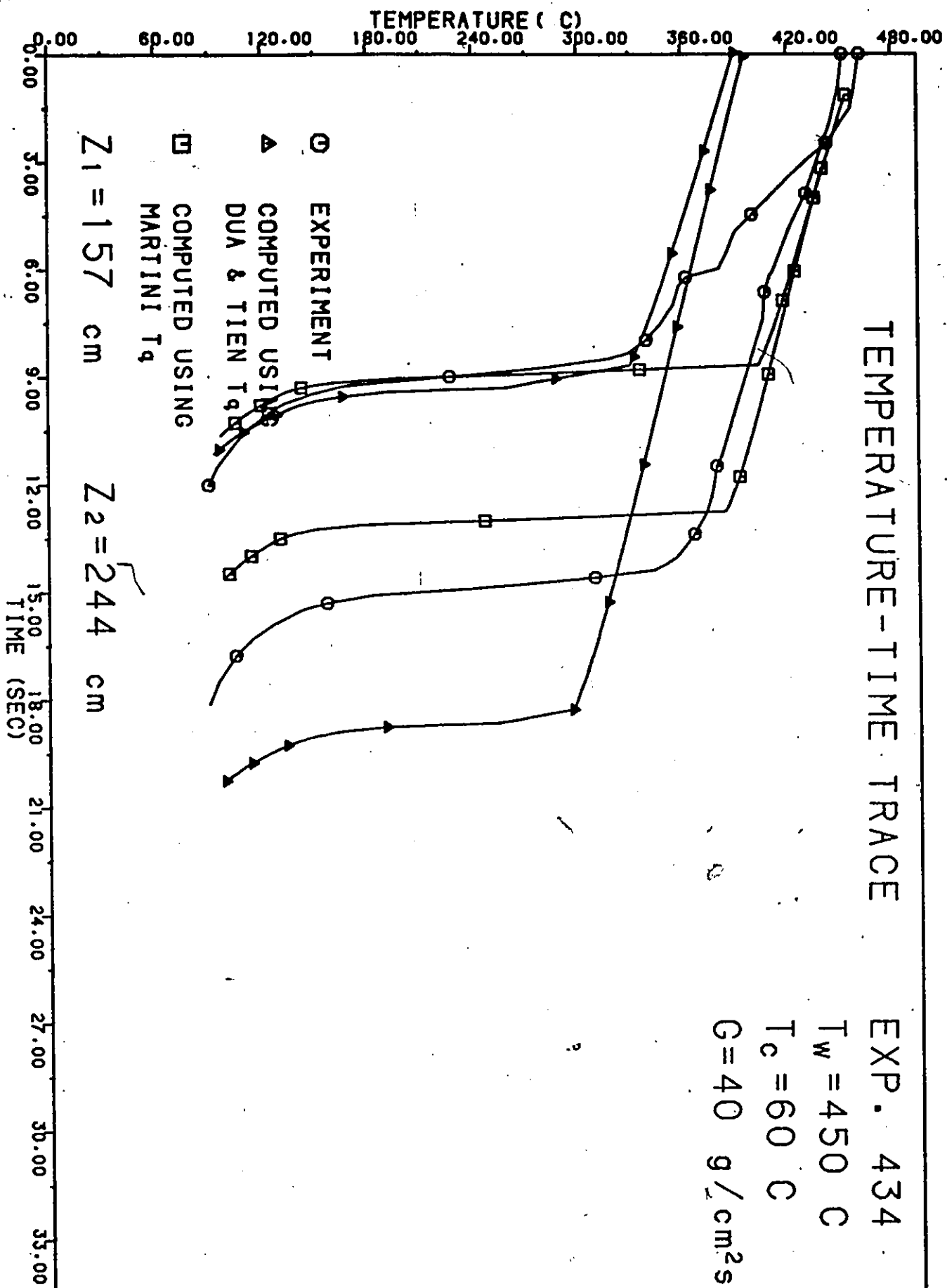


Fig. 5.12 T-T Plots with Dua and Martini T_q

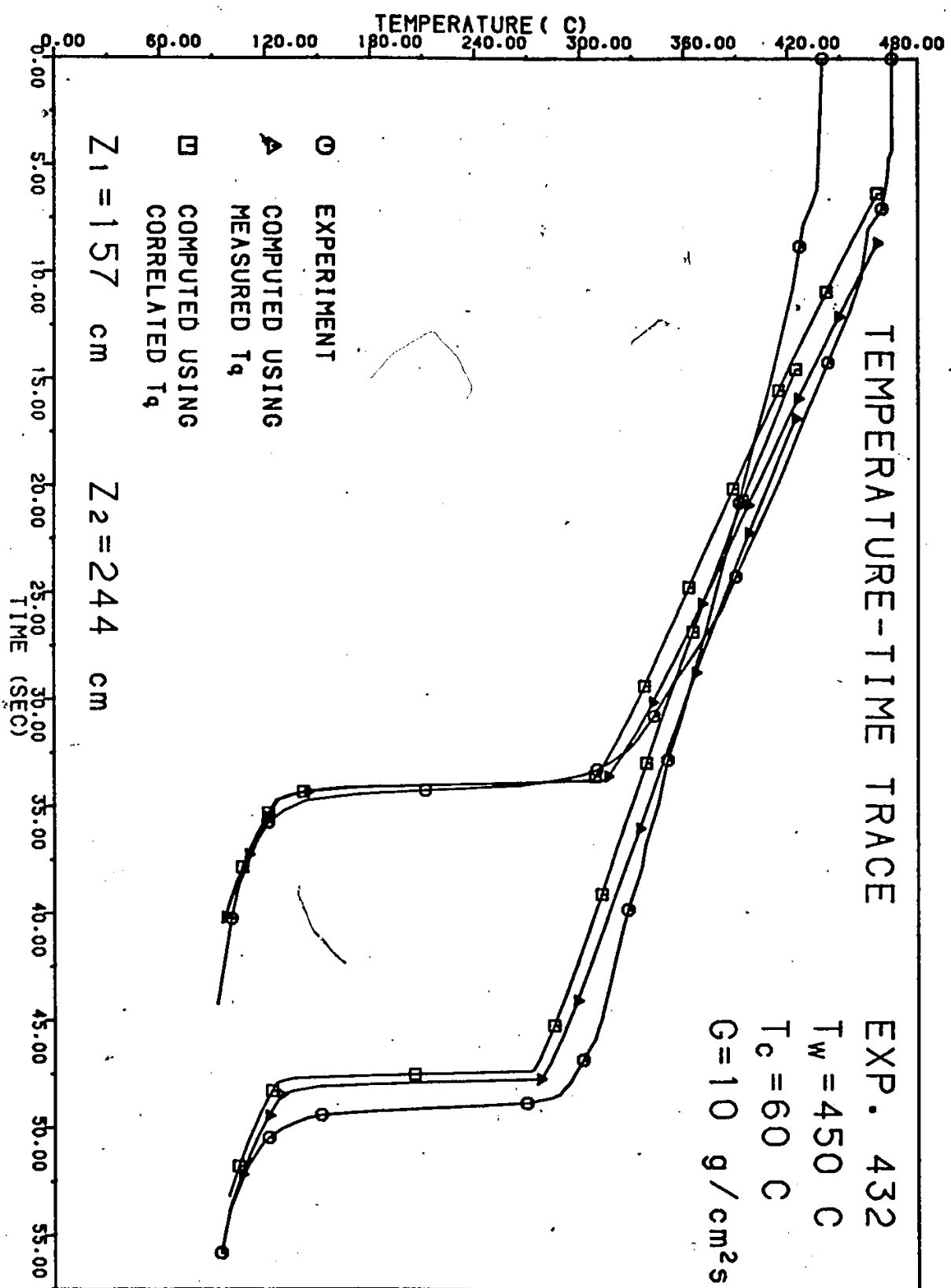


Fig. 5.13 T-T Plots with DX of 1 cm

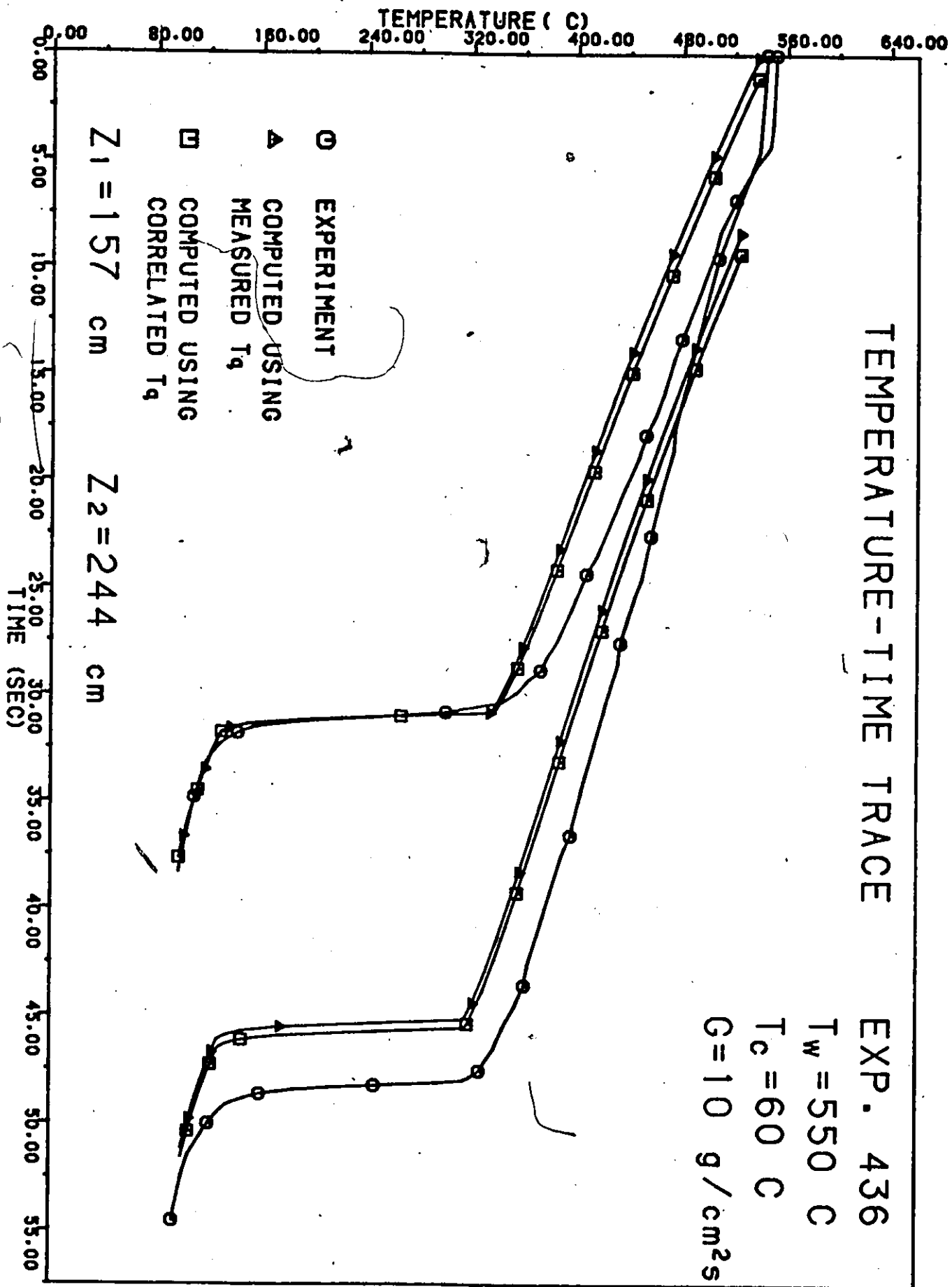


Fig. 5.14 T-T Plots with DX of 1 cm

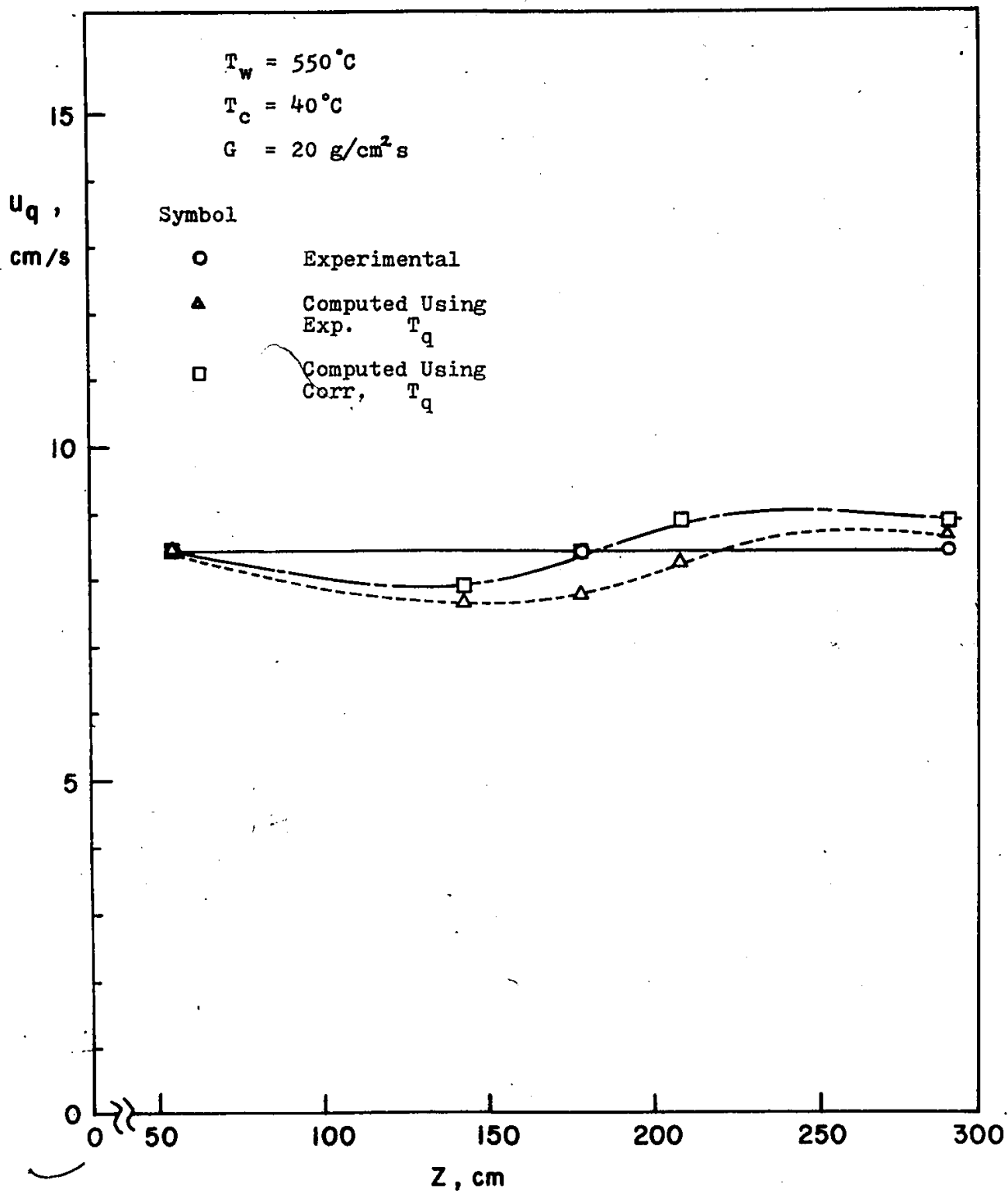


Fig. 5.15 Rewetting Velocity Distribution over Axial Location

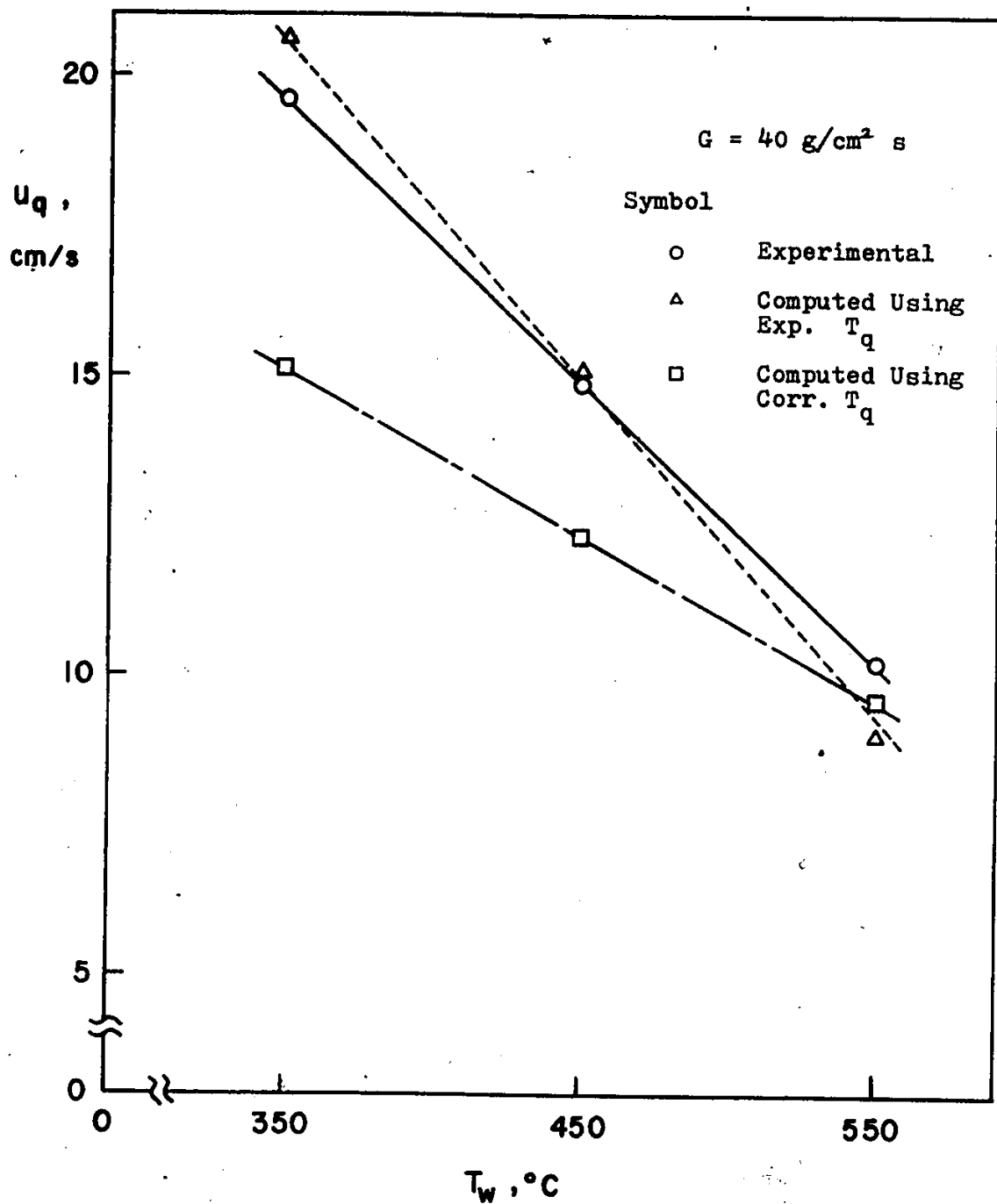


Fig. 5.16 Rewetting Velocity vs. Initial Wall Temperature

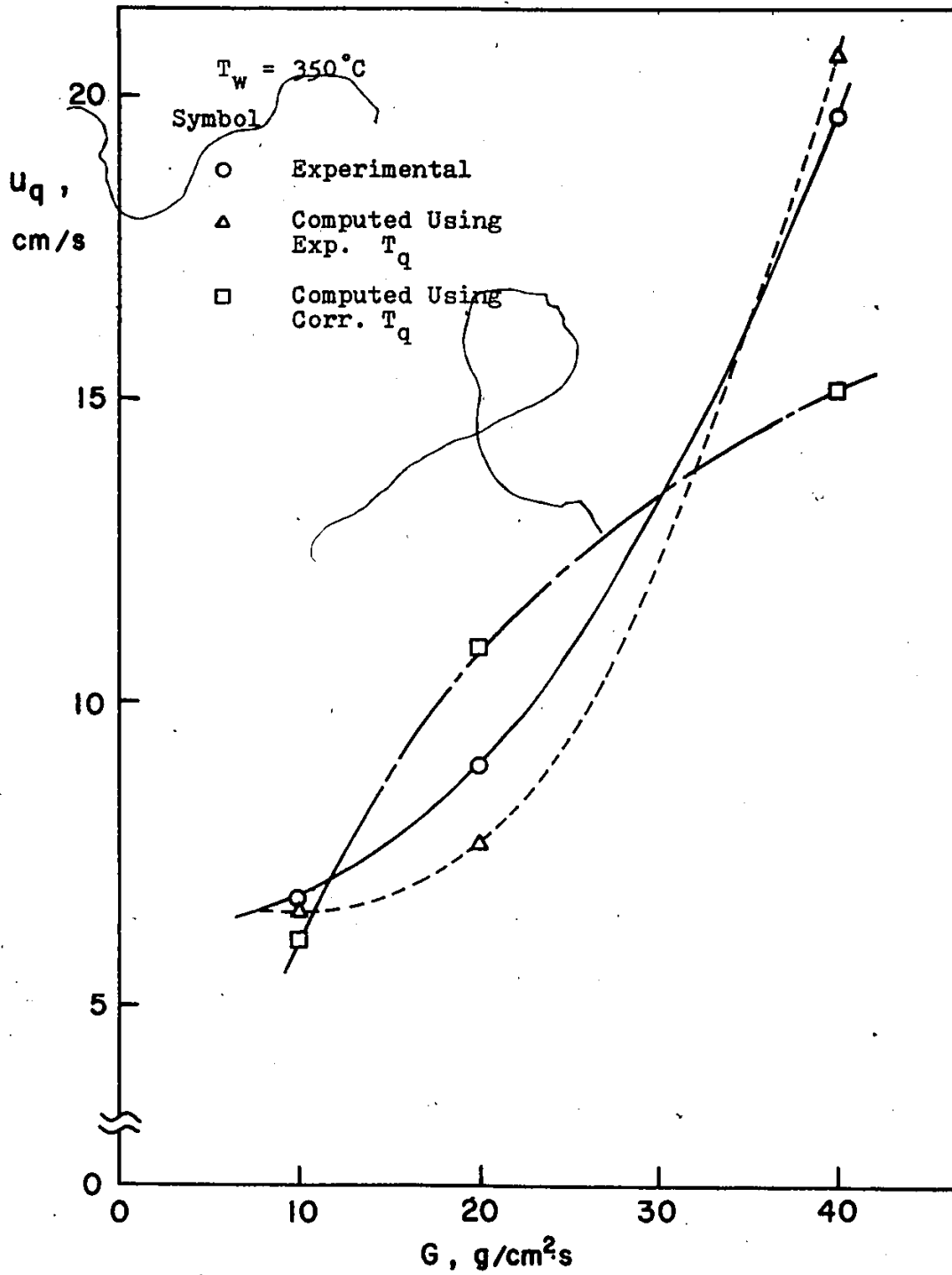


Fig. 5.17 Rewetting Velocity vs. Mass Flow Rate ($T_w = 350^\circ\text{C}$)

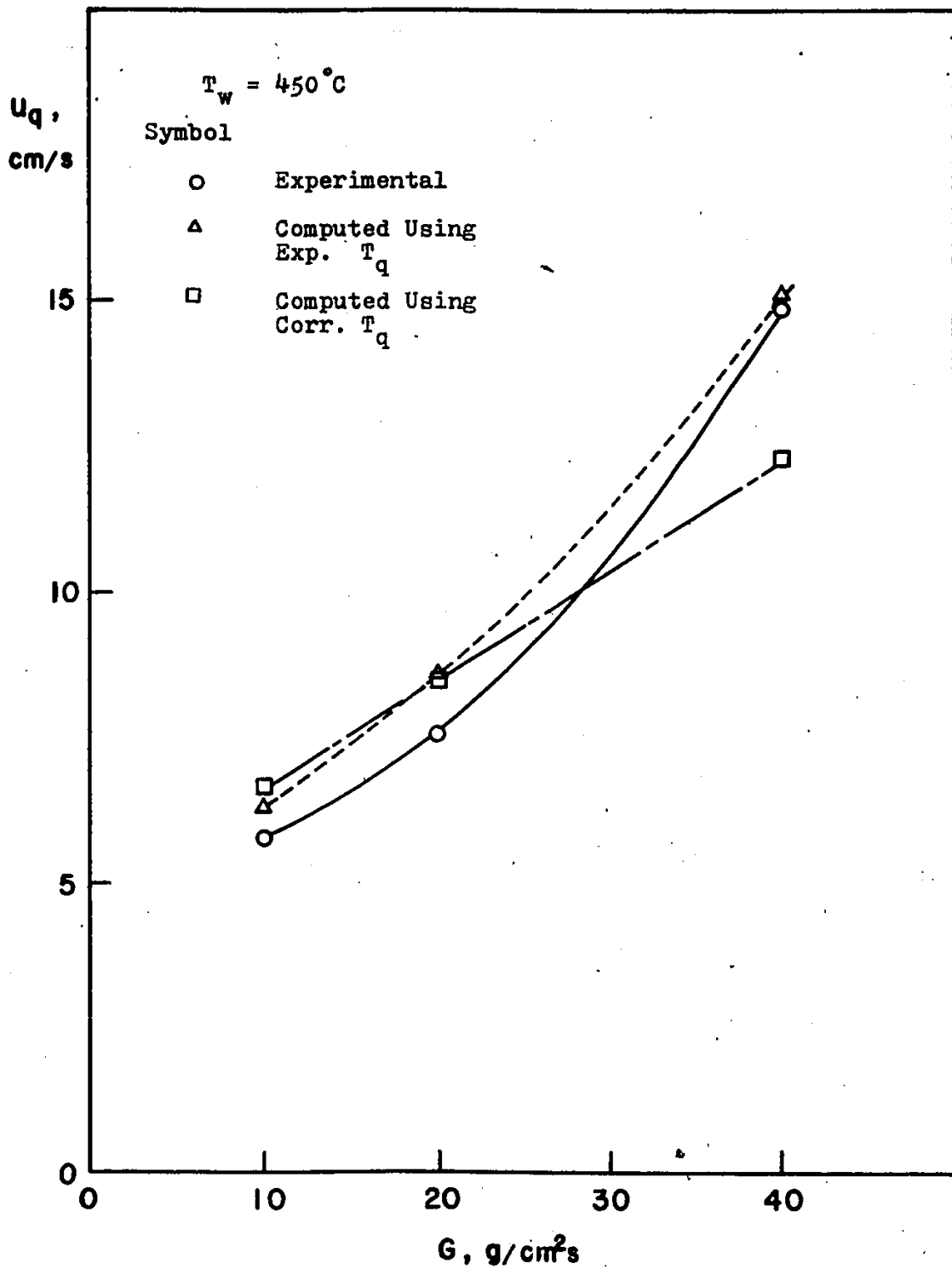


Fig. 5.18 Rewetting Velocity vs. Mass Flow Rate ($T_w = 450^\circ\text{C}$)

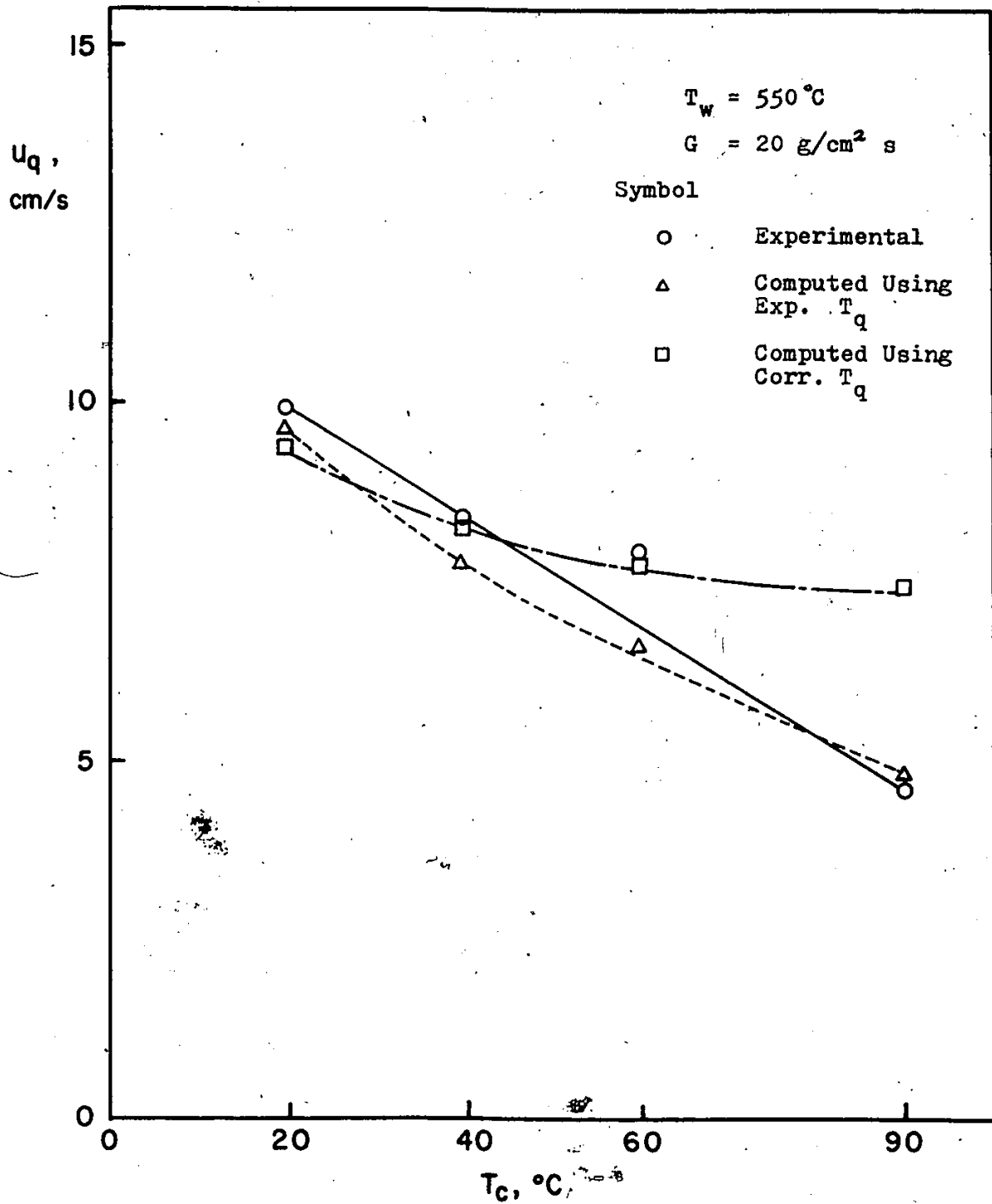


Fig. 5.19 Rewetting Velocity vs. Coolant Temperature

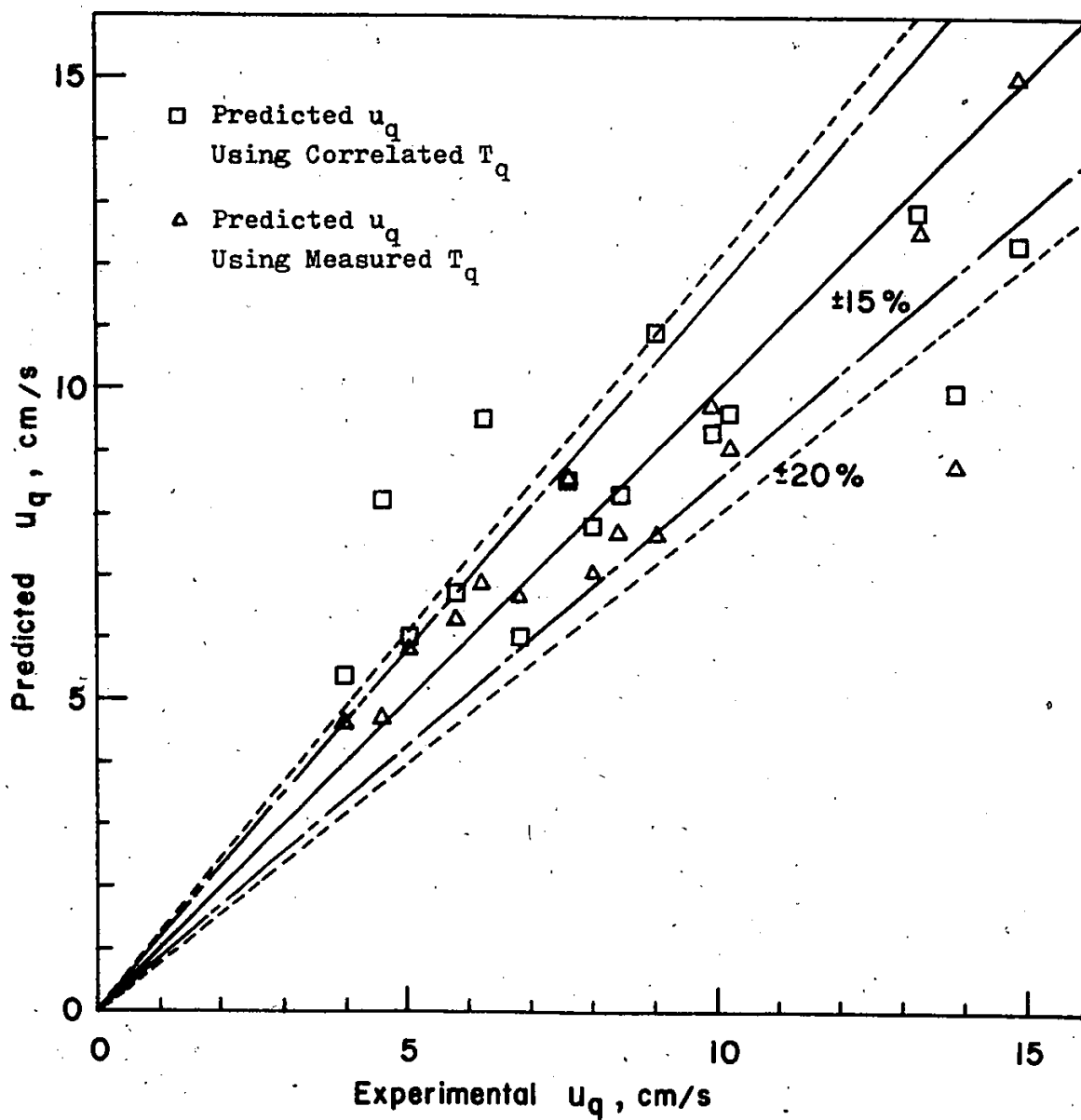


Fig. 5.20 Computed Rewetting Velocity vs. Experimental Rewetting Velocity

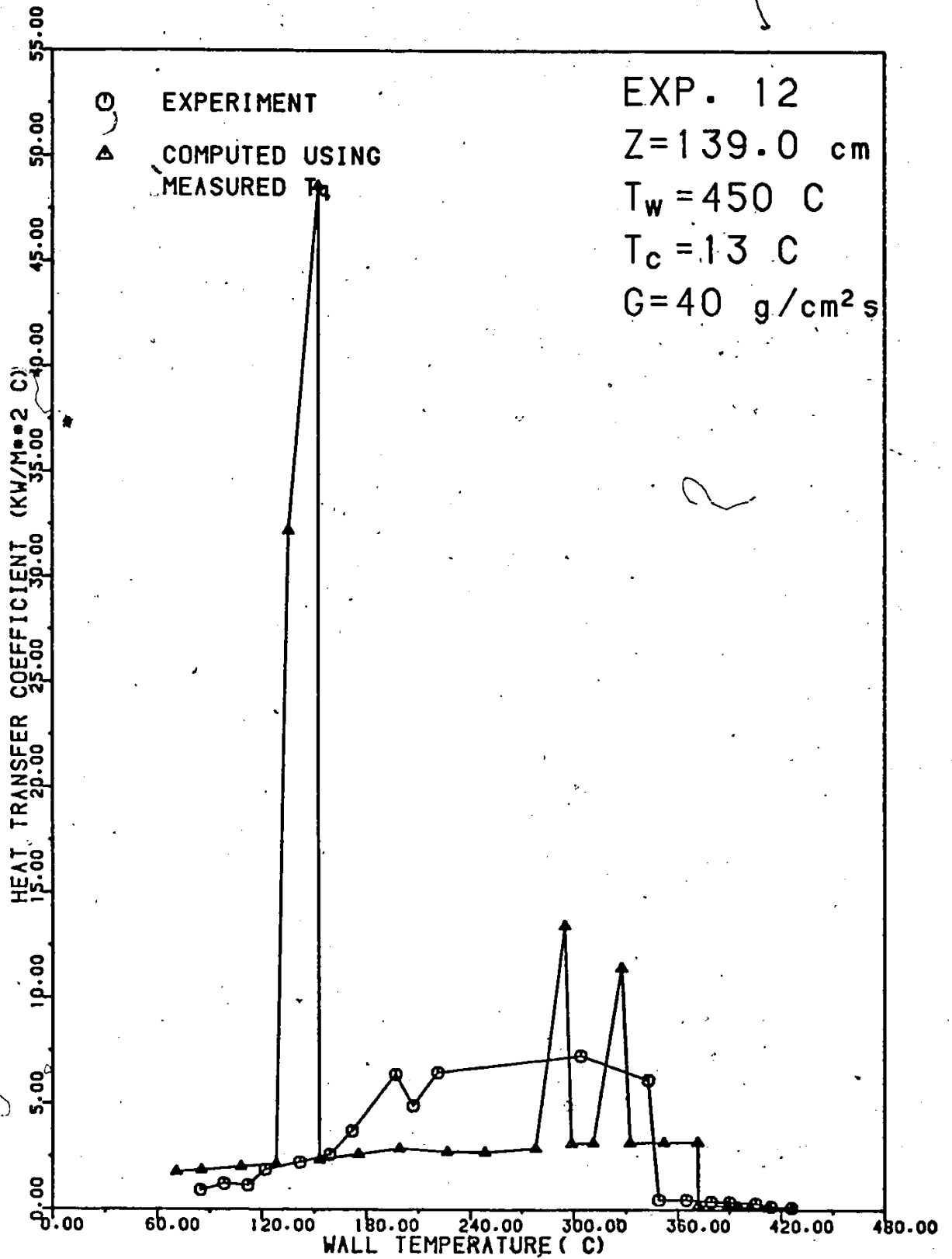


Fig. 5.21 Heat Transfer Coefficient vs. Wall Temperature

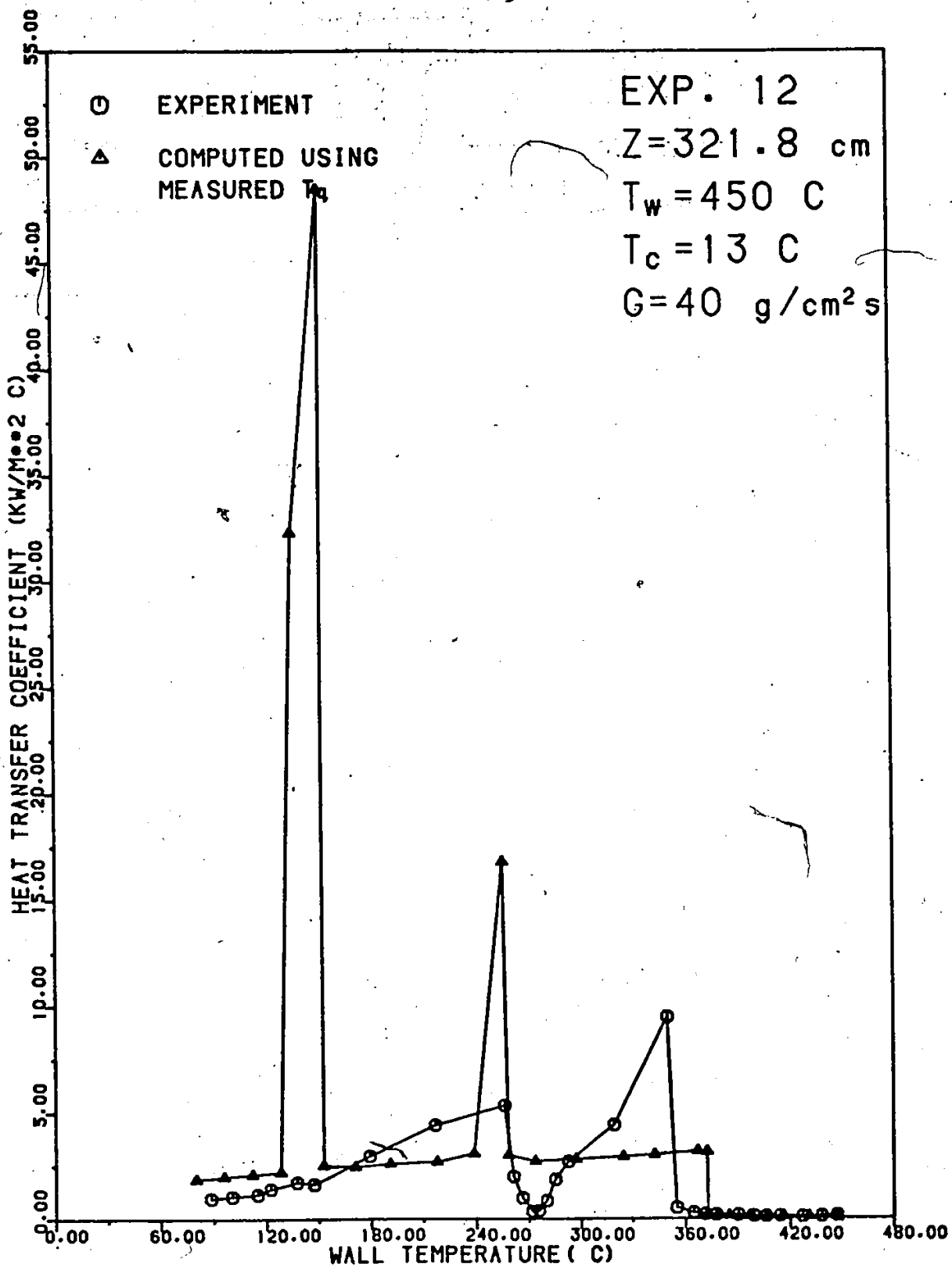


Fig. 5.22 Heat Transfer Coefficient vs. Wall Temperature

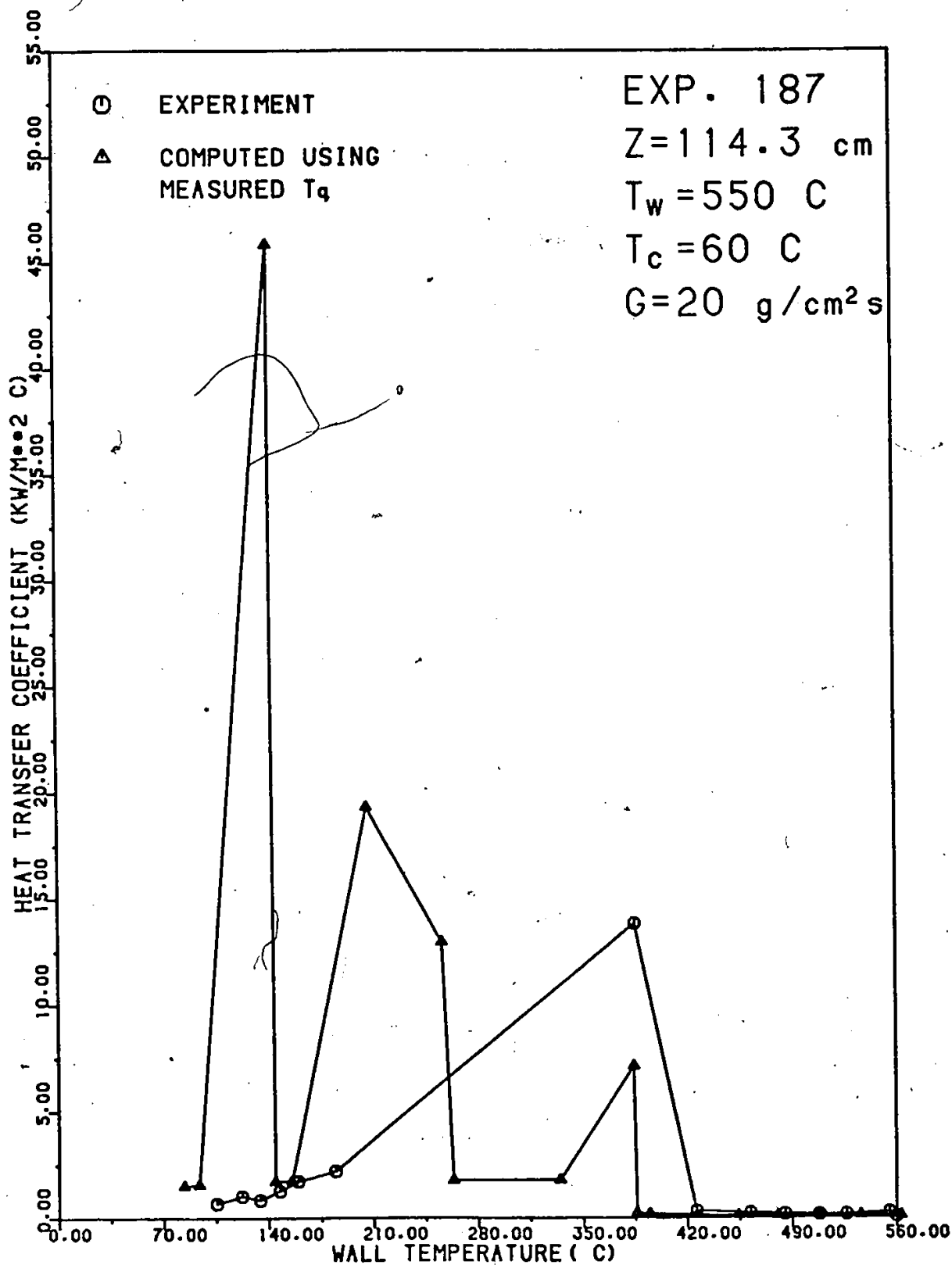


Fig. 5.23 Heat Transfer Coefficient vs. Wall Temperature

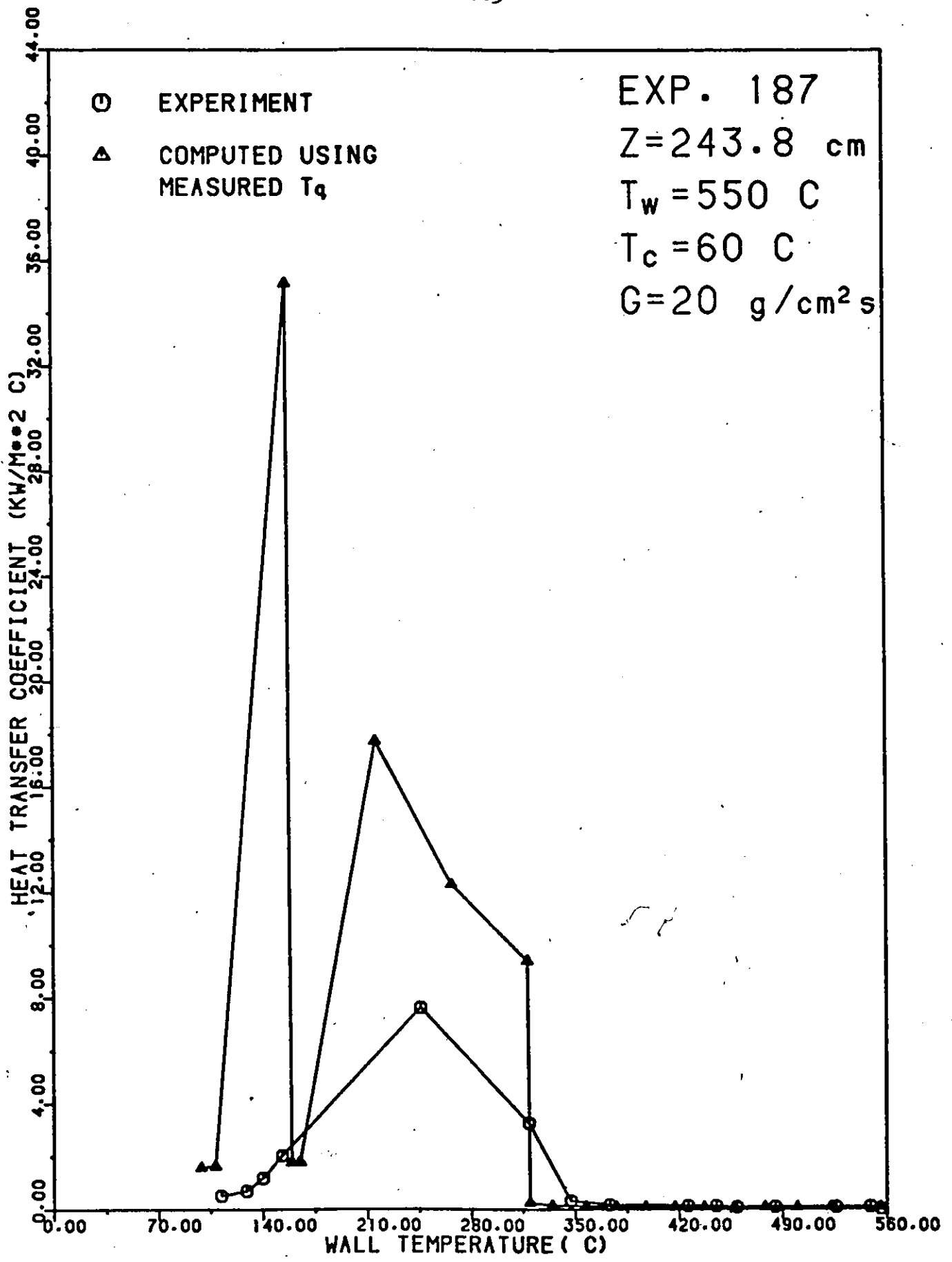


Fig. 5.24 Heat Transfer Coefficient vs. Wall Temperature

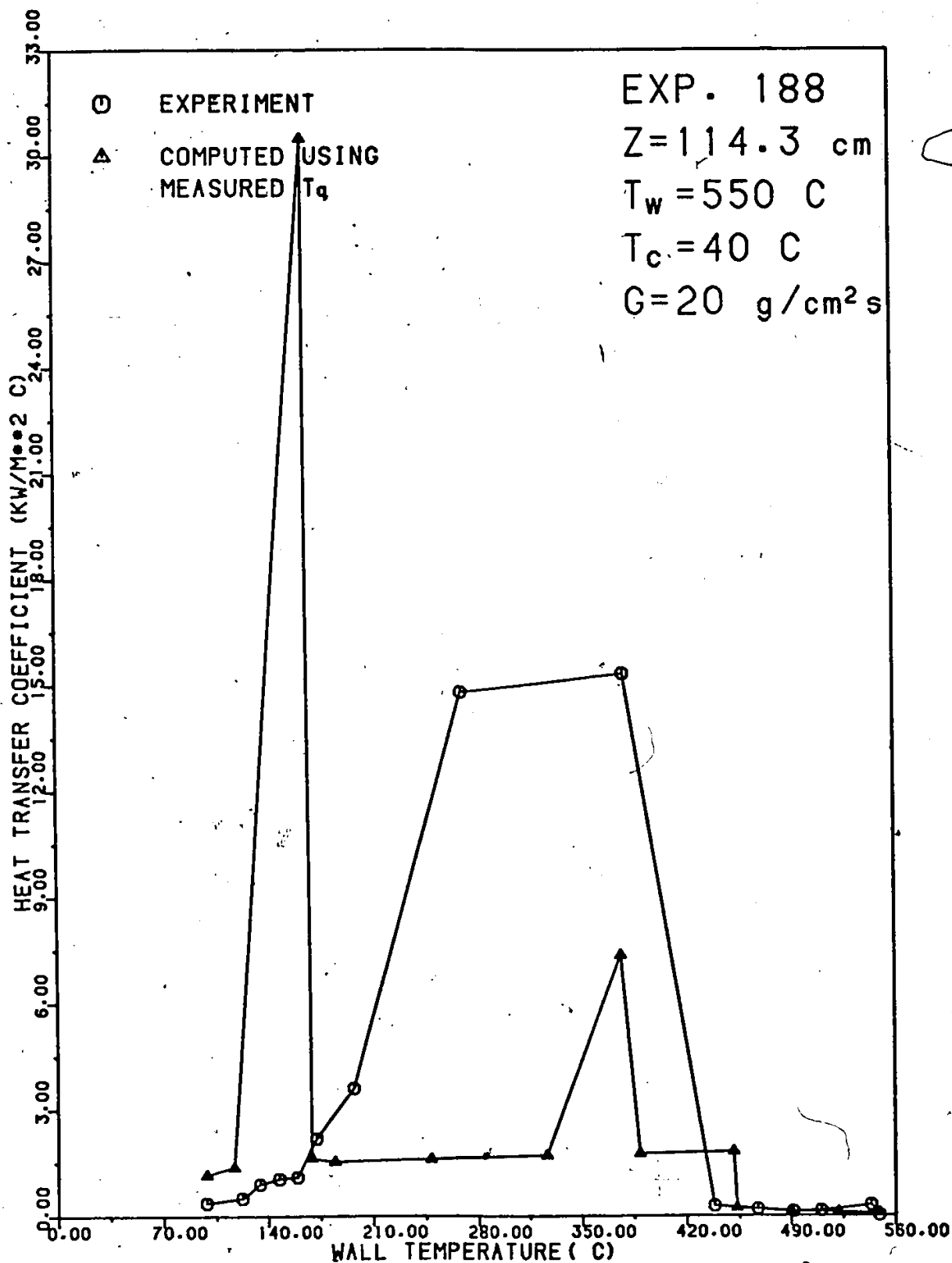


Fig. 5.25 Heat Transfer Coefficient vs. Wall Temperature

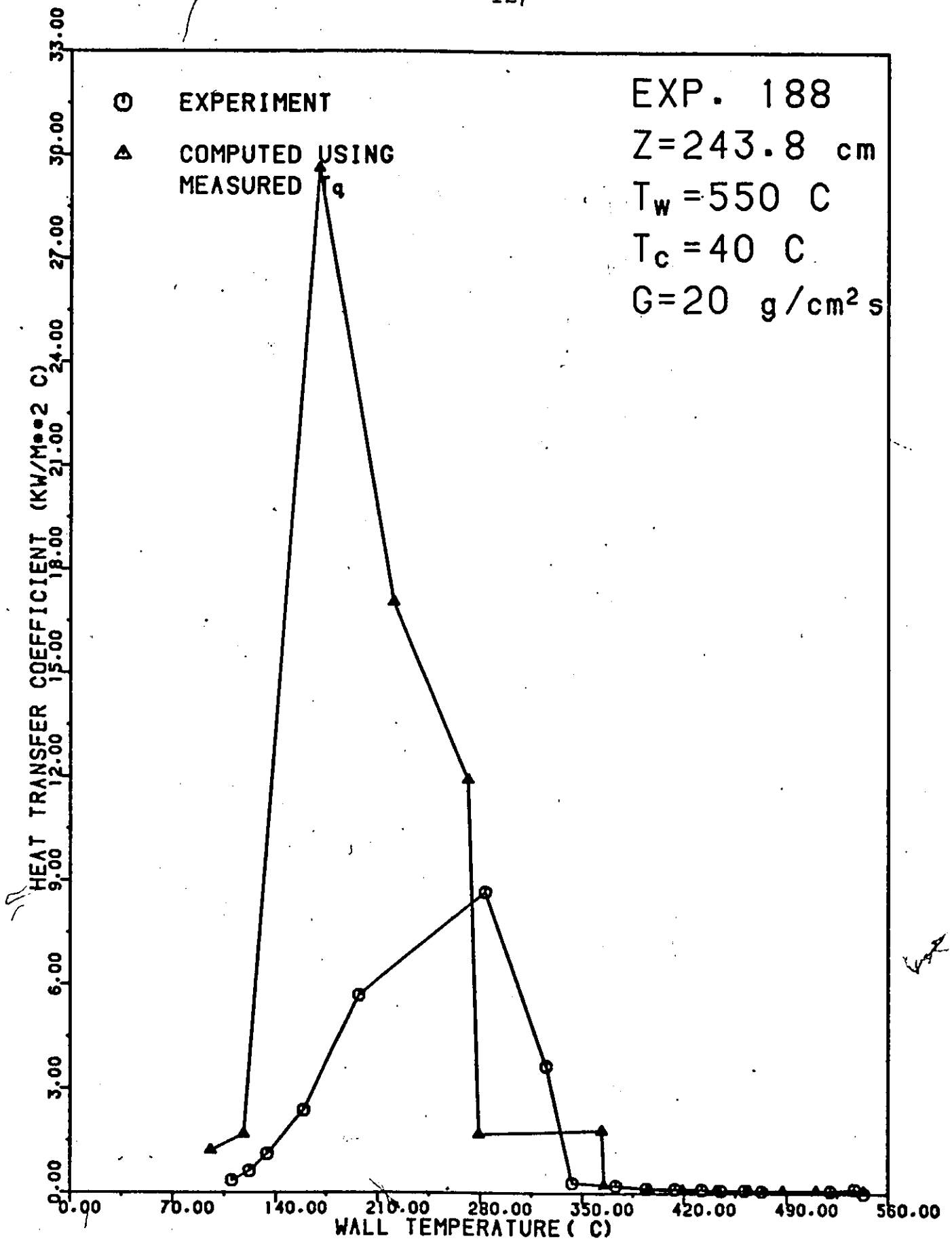


Fig. 5.26 Heat Transfer Coefficient vs. Wall Temperature

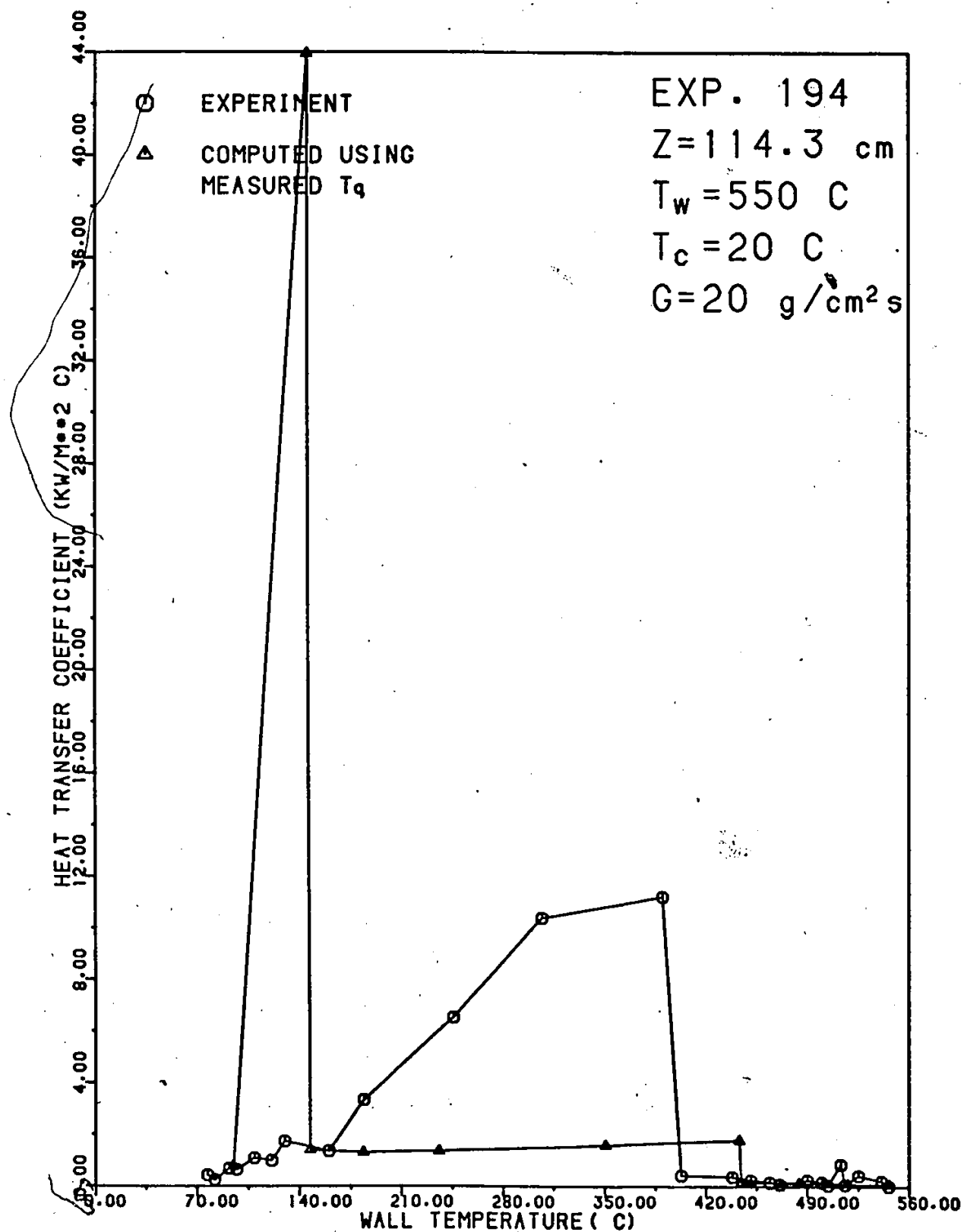


Fig. 5.27 Heat Transfer Coefficient vs. Wall Temperature

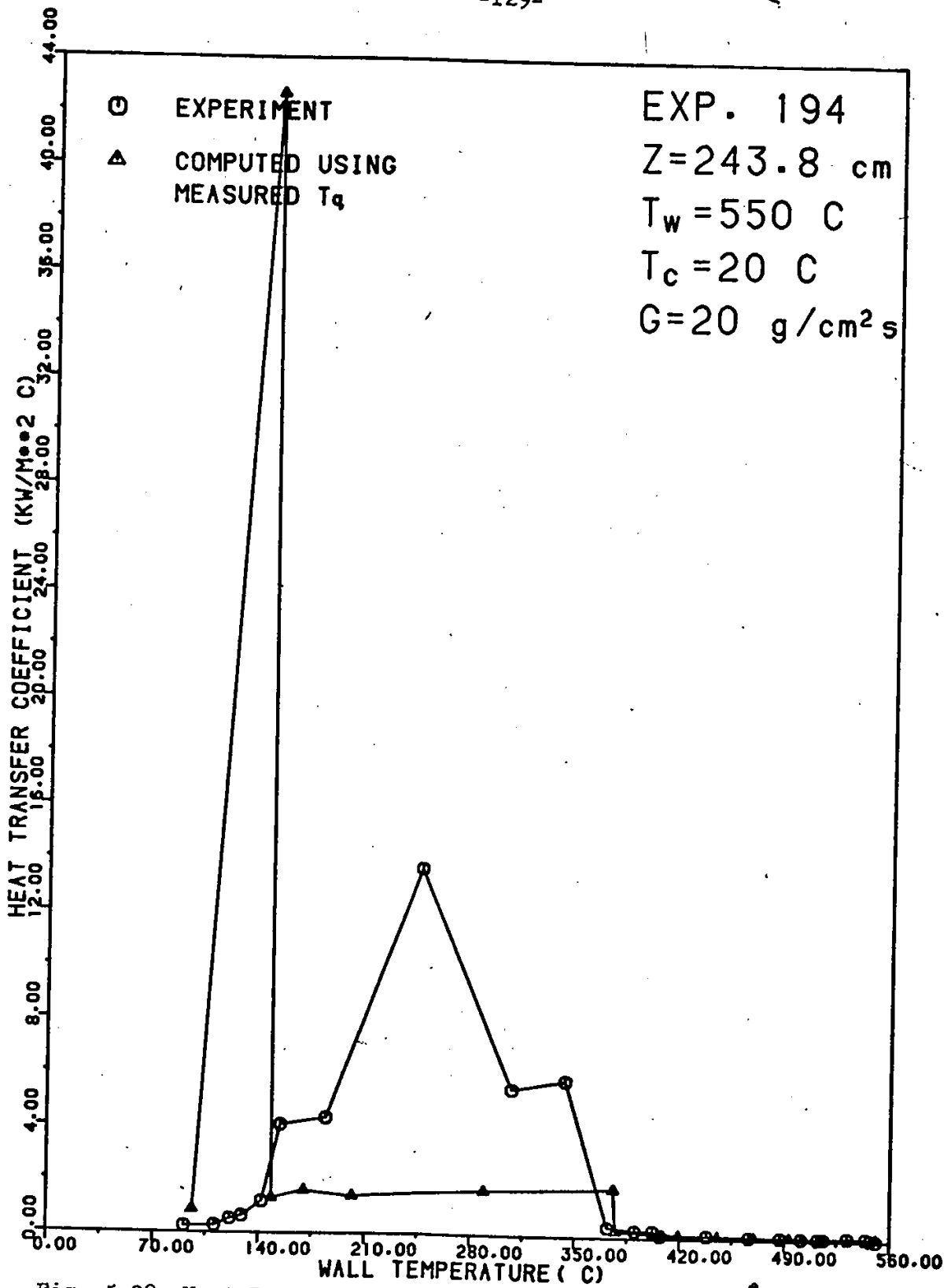


Fig. 5.28 Heat Transfer Coefficient vs. Wall Temperature

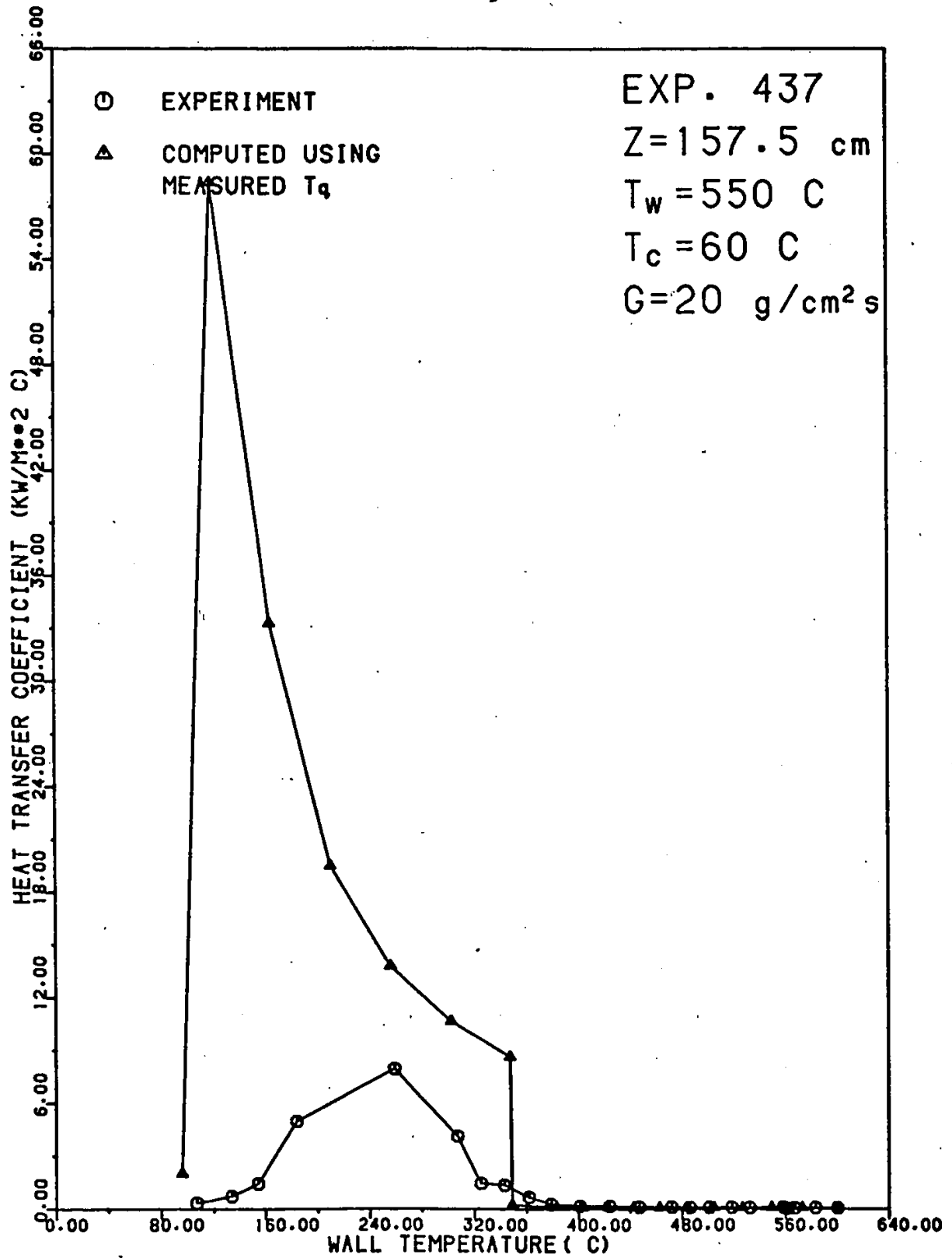


Fig. 5.29 Heat Transfer Coefficient vs. Wall Temperature

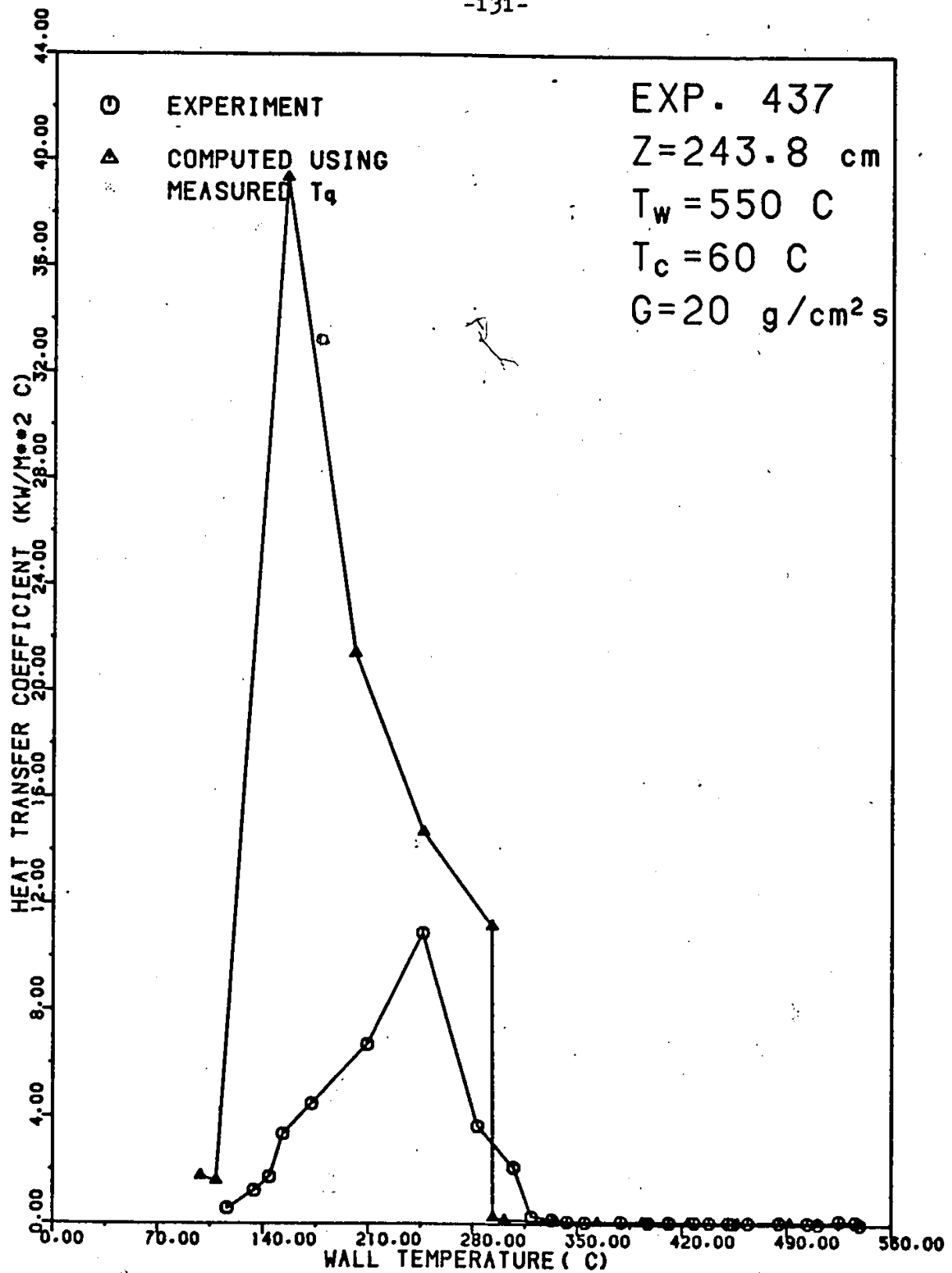


Fig. 5.30 Heat Transfer Coefficient vs. Wall Temperature

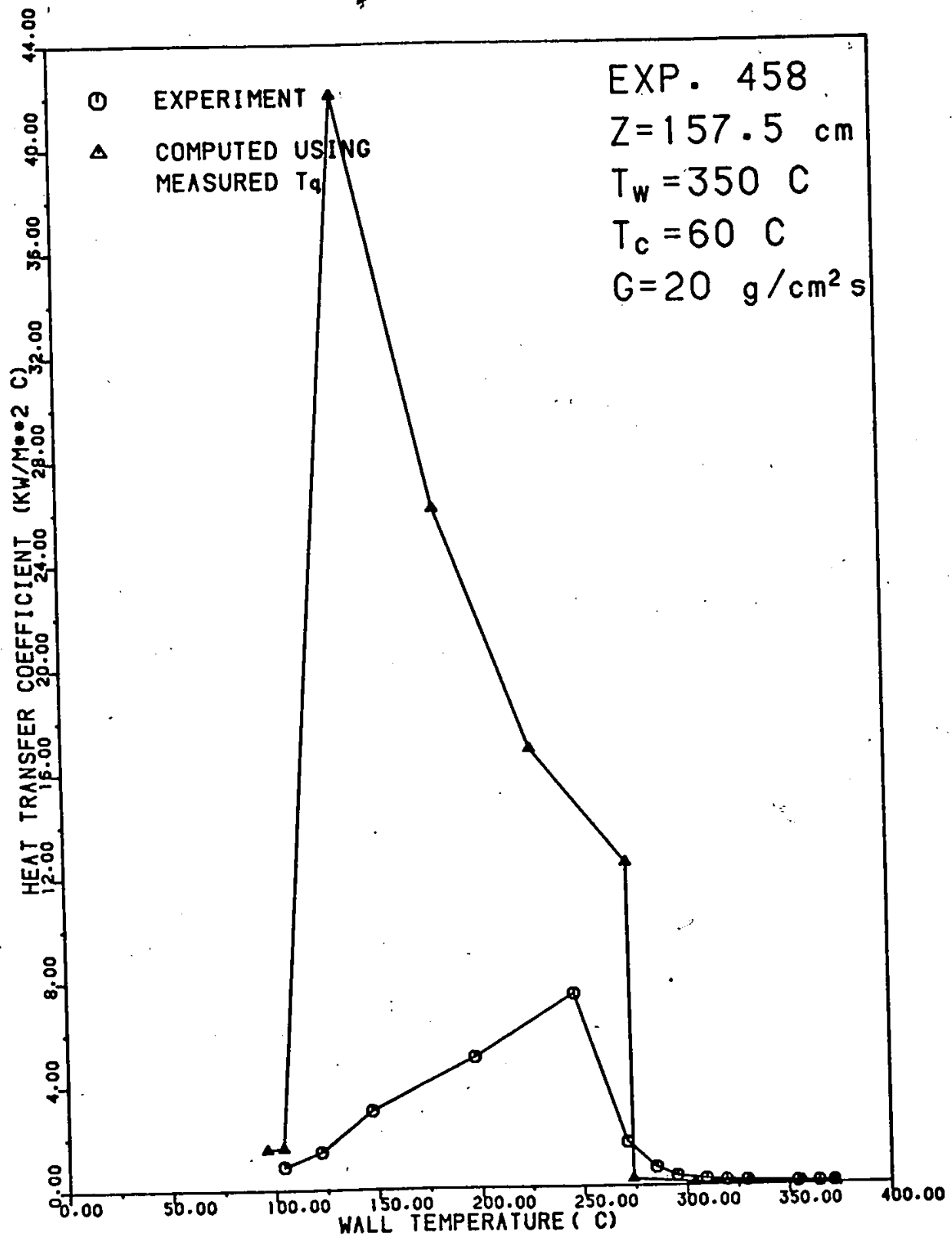


Fig. 5.31 Heat Transfer Coefficient vs. Wall Temperature

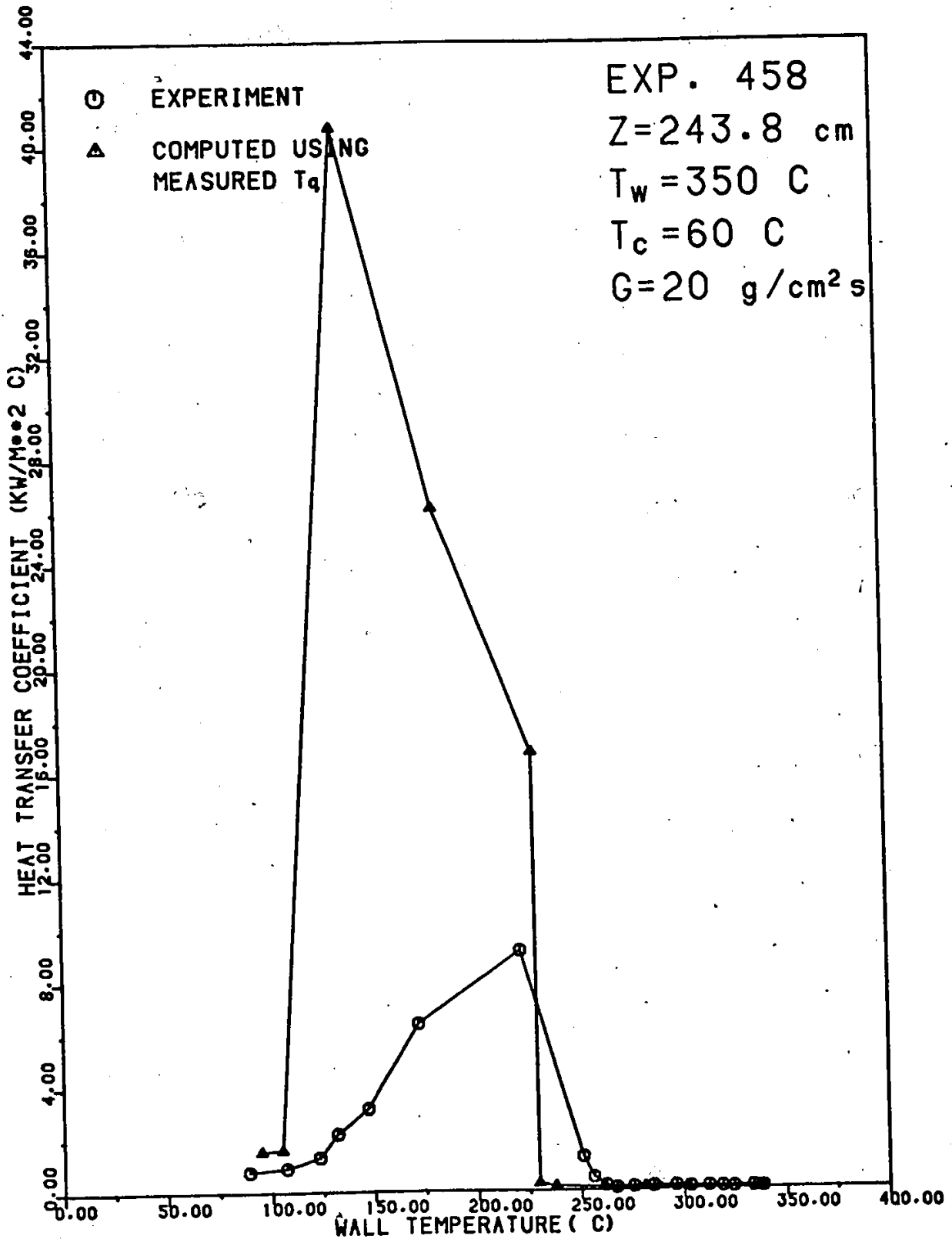


Fig. 5.32 Heat Transfer Coefficient vs. Wall Temperature

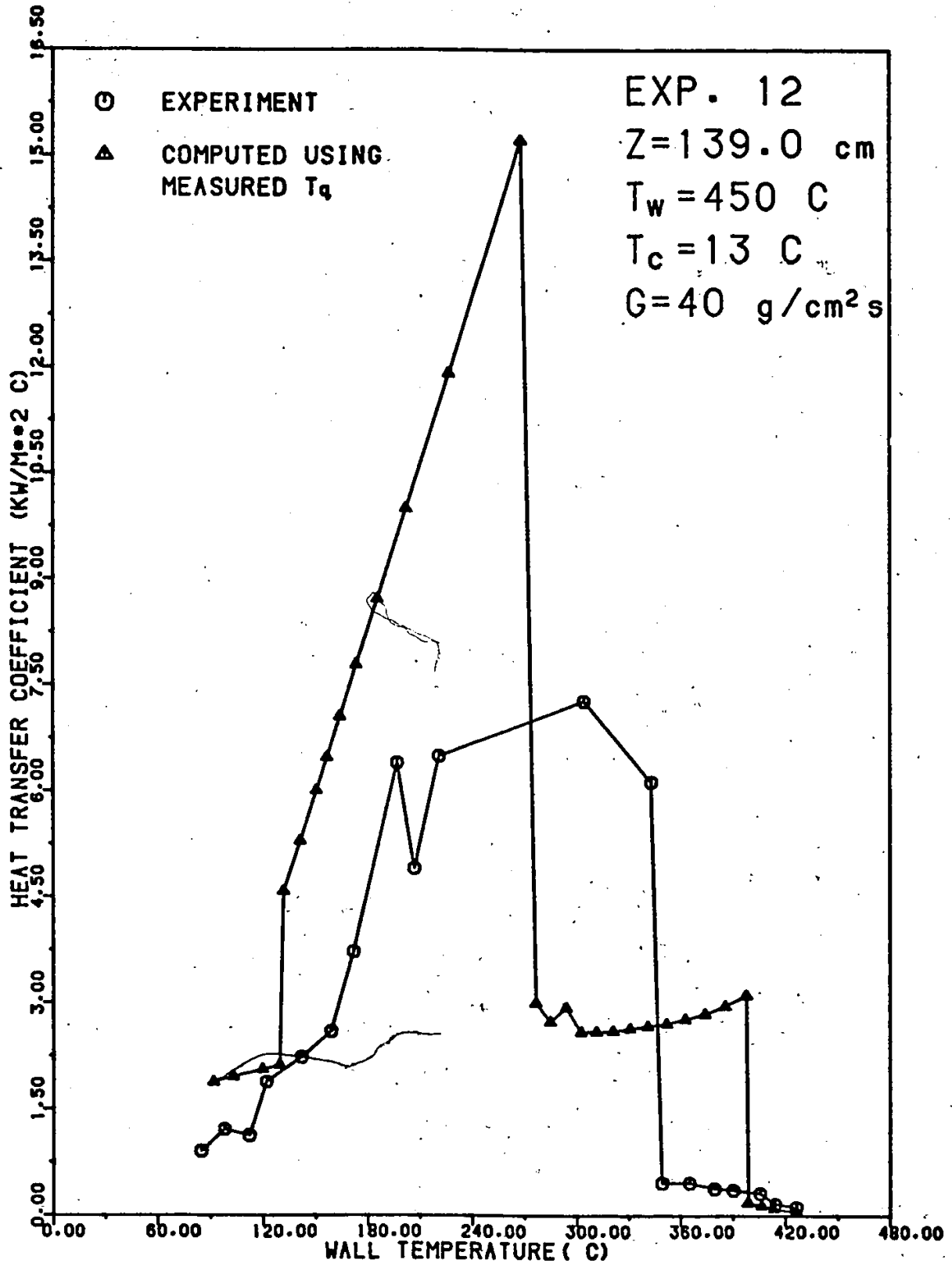


Fig. 5.33 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

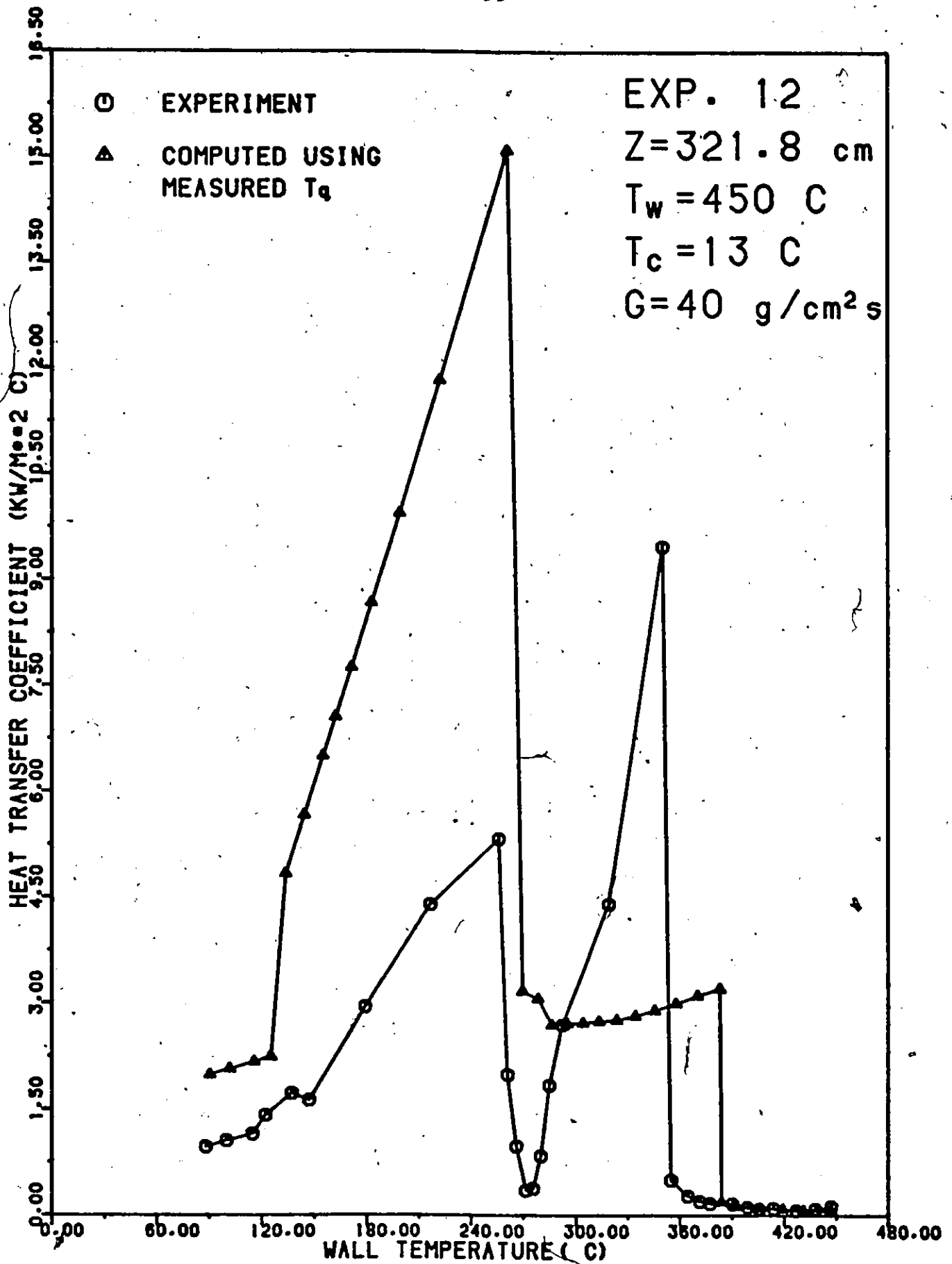


Fig. 5.34 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

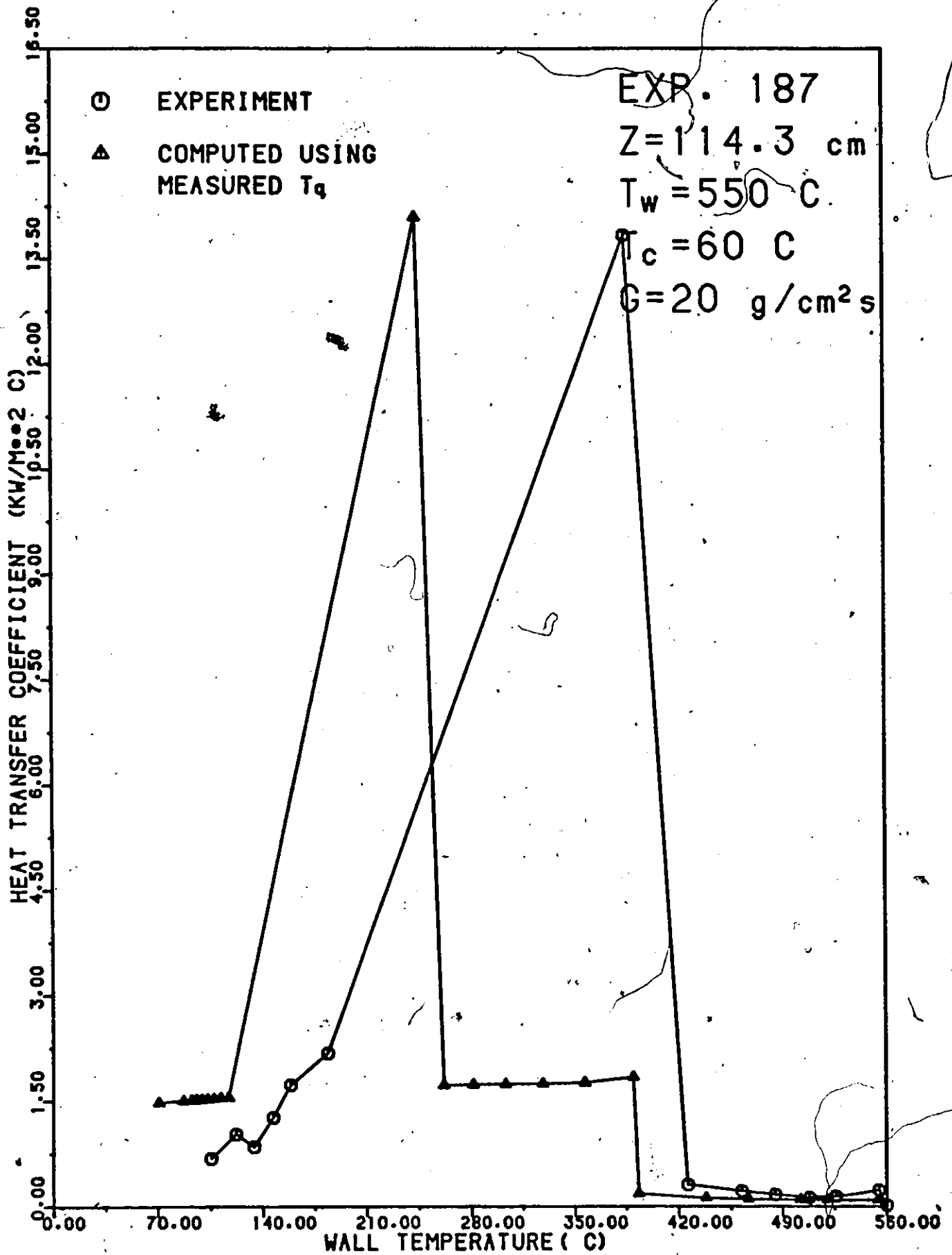


Fig. 5.35 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

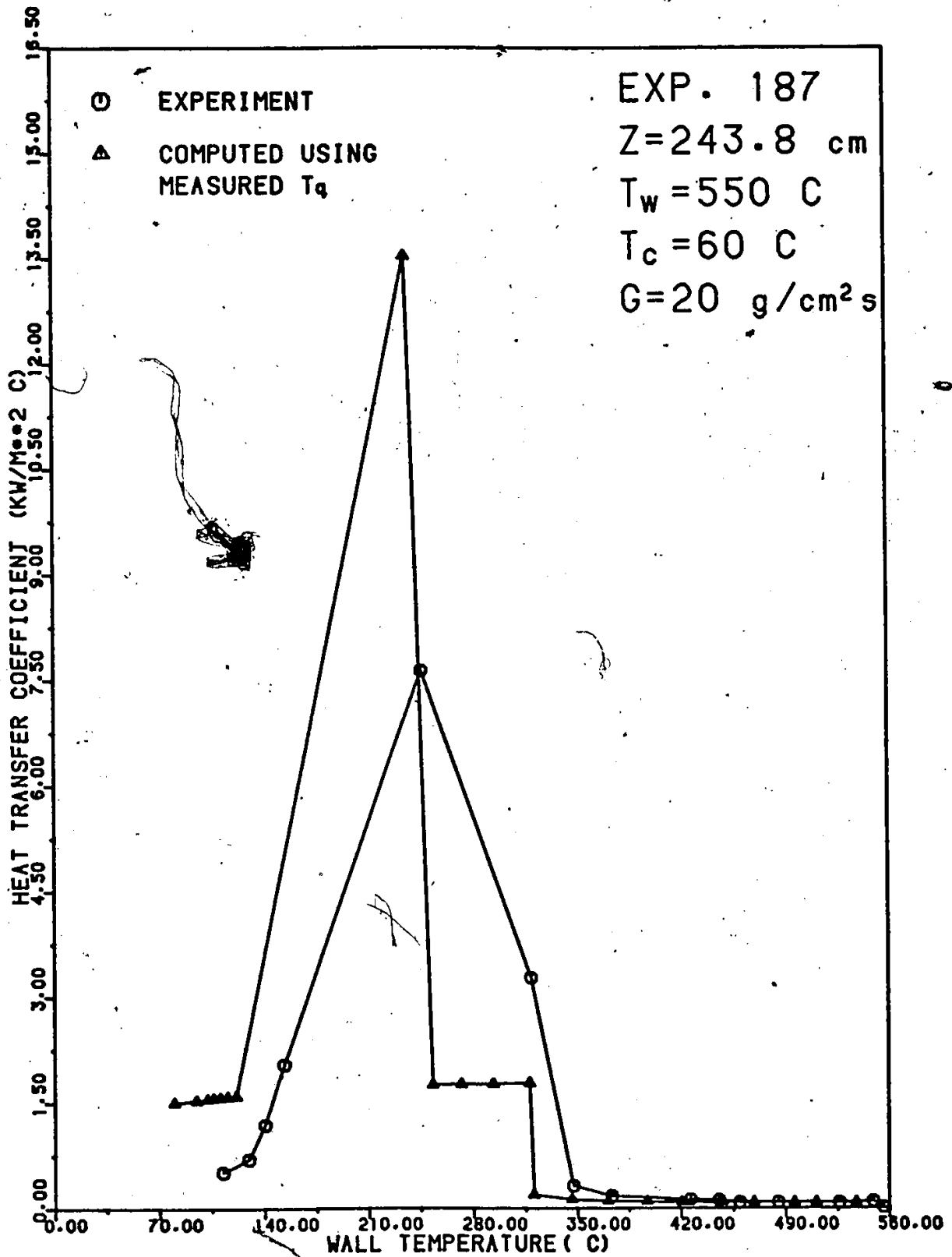


Fig. 5.36 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

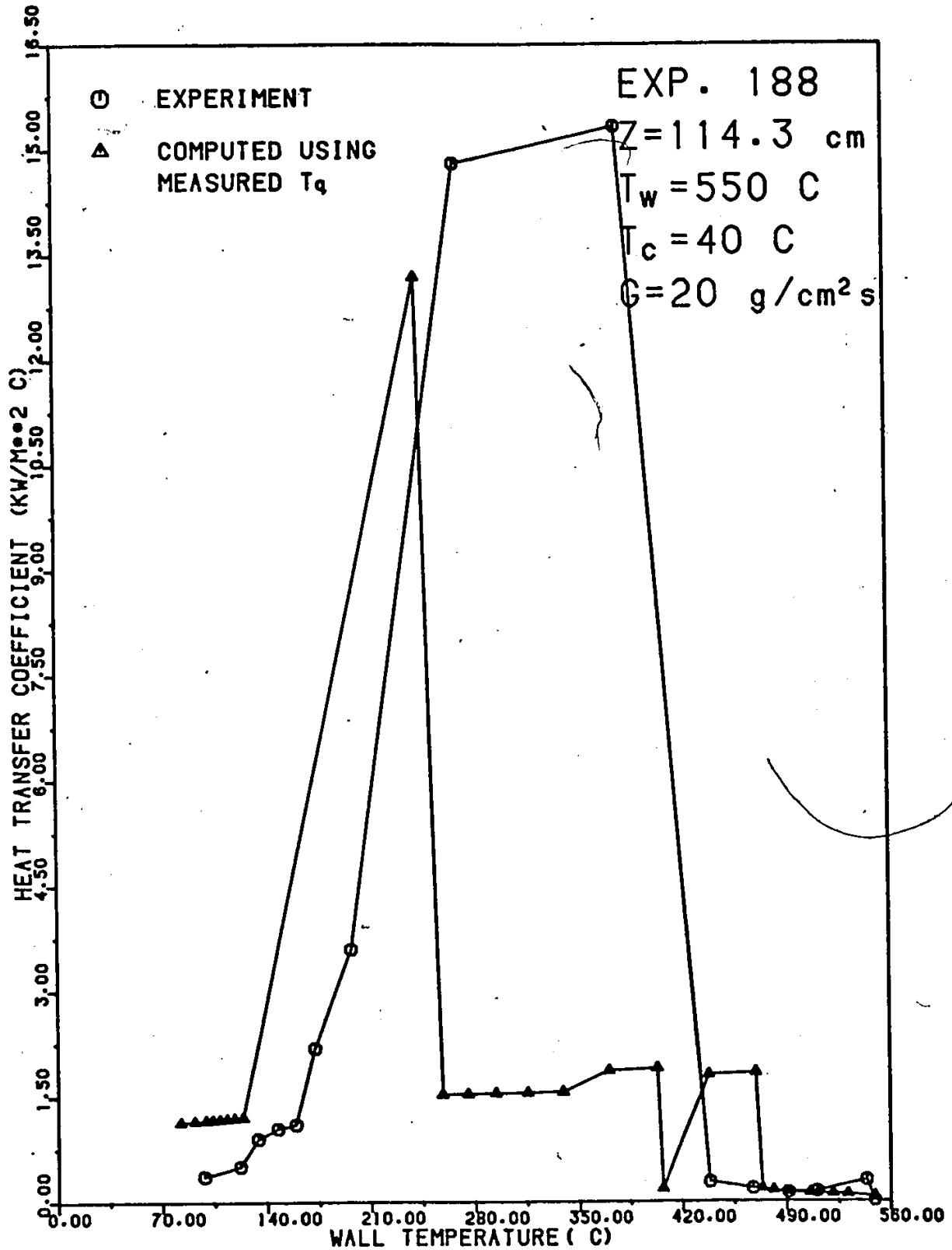


Fig. 5.37 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

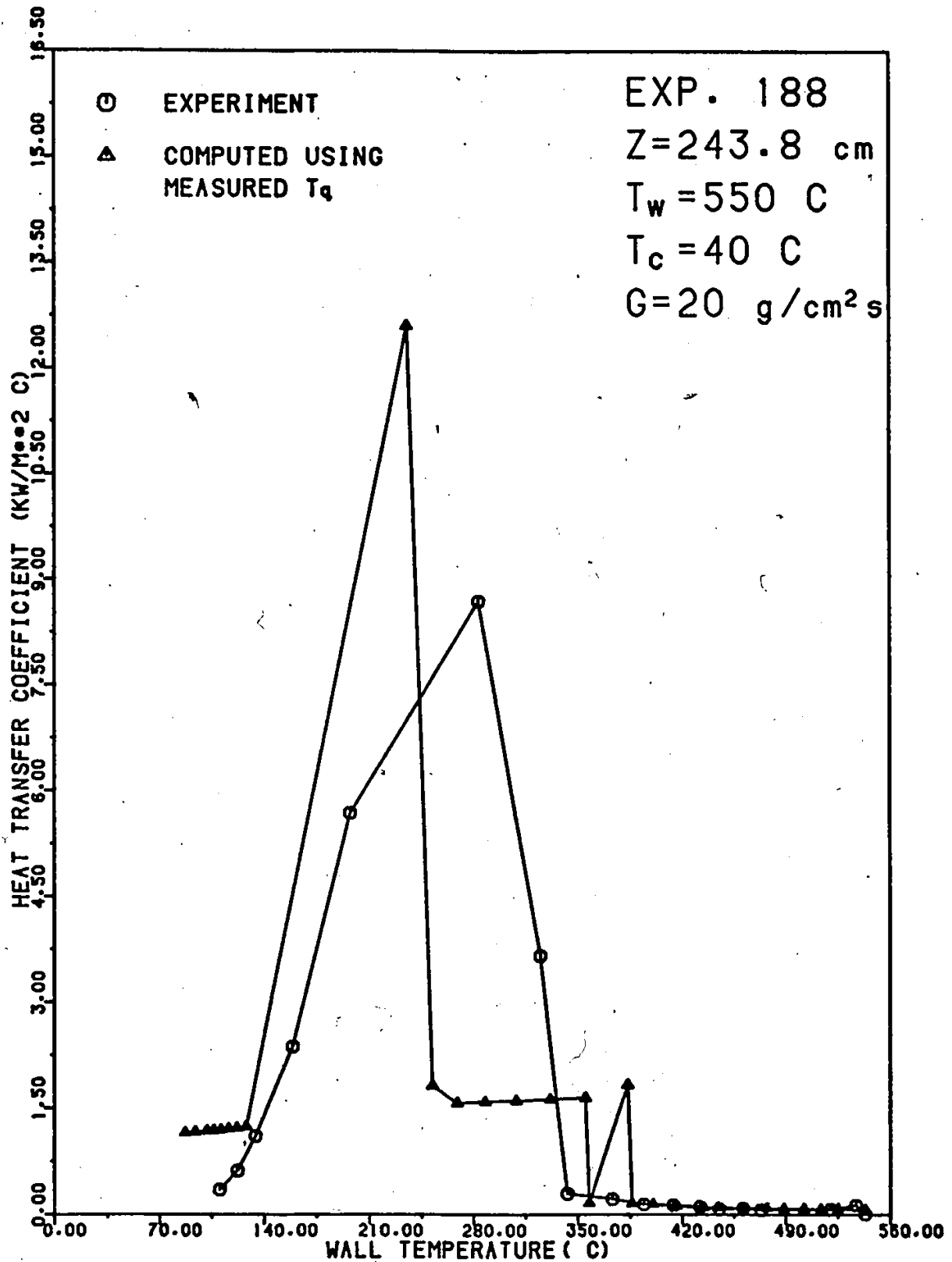


Fig. 5.38 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

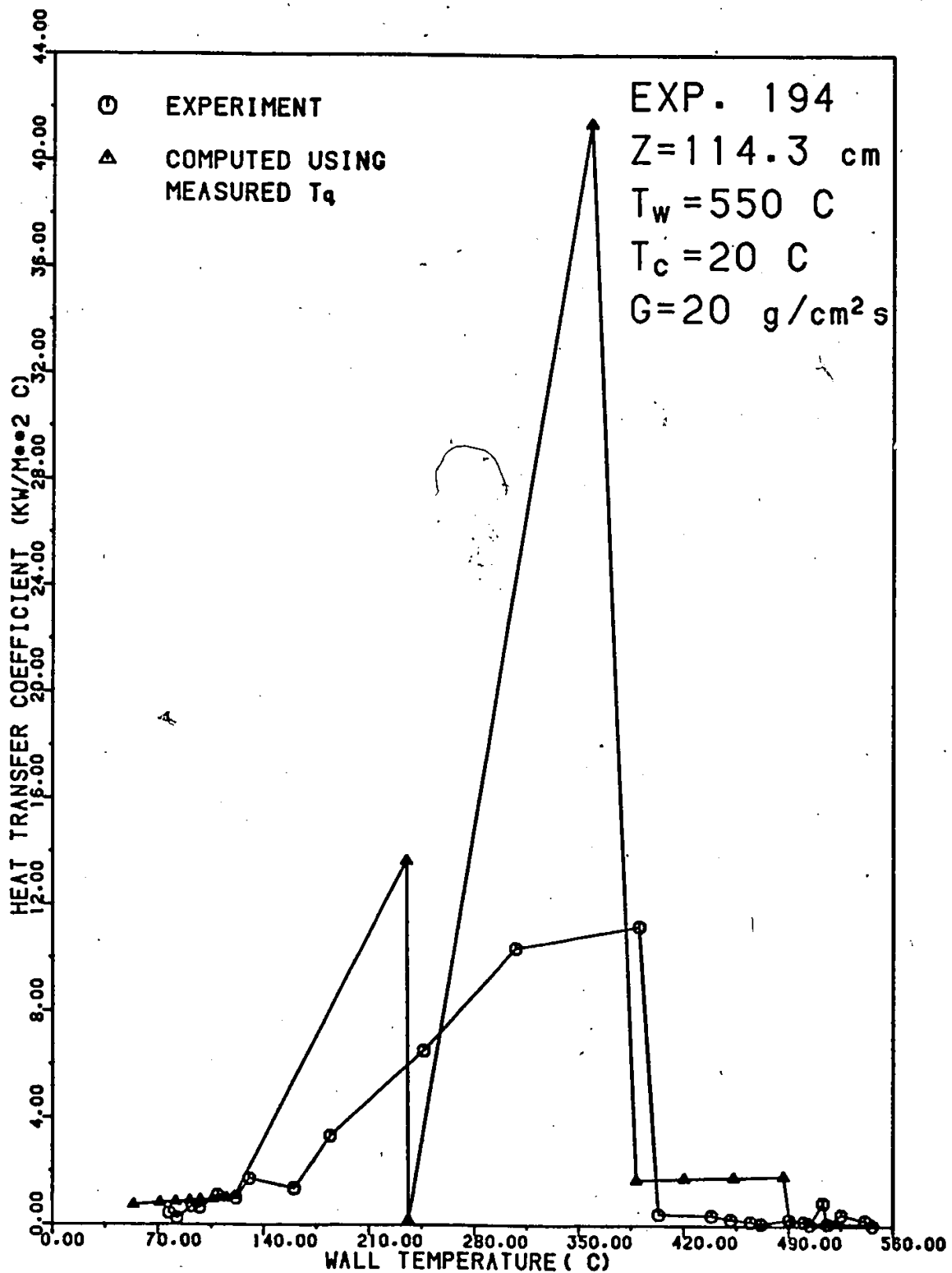


Fig. 5.39 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

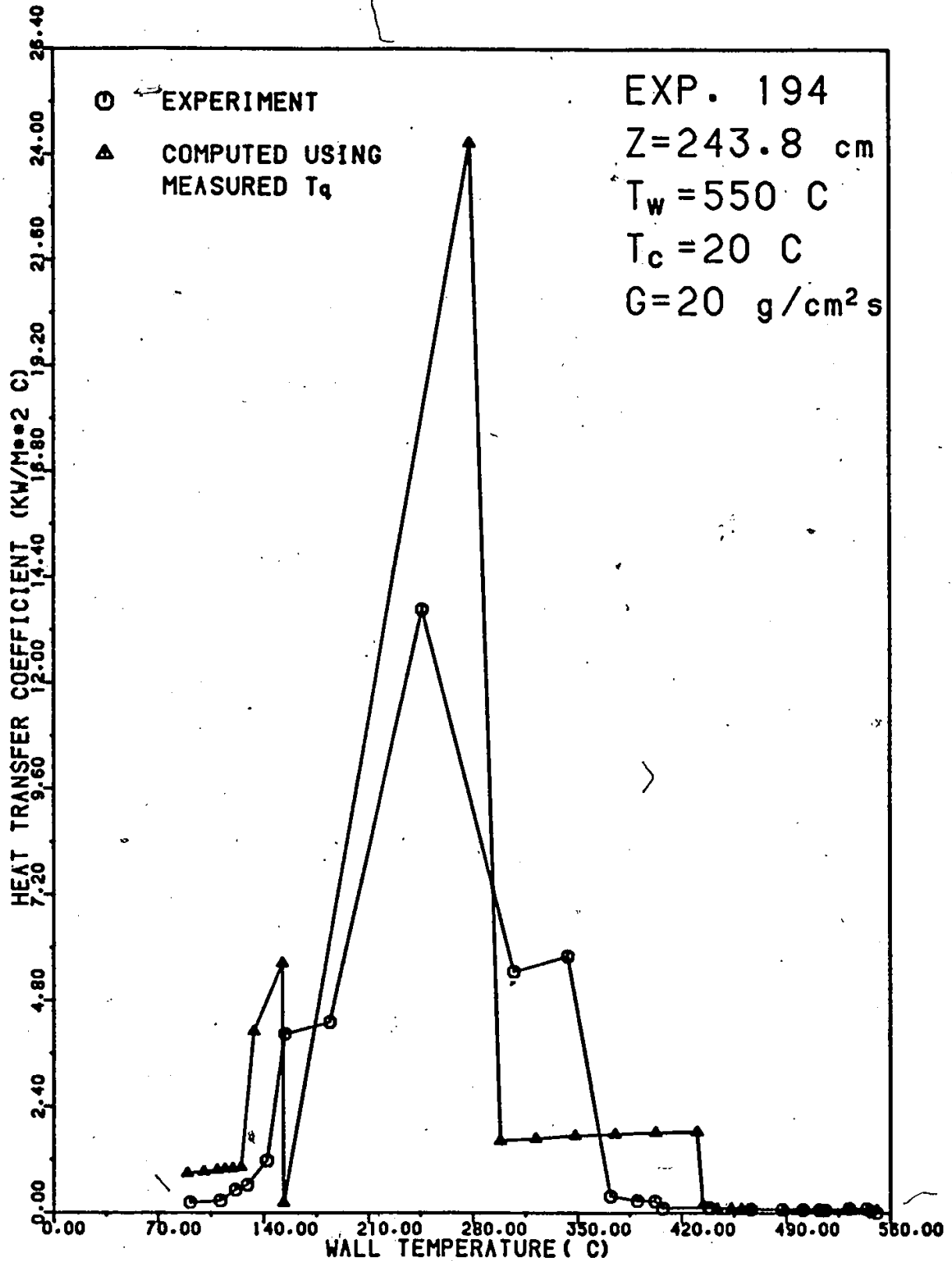


Fig. 5.40 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

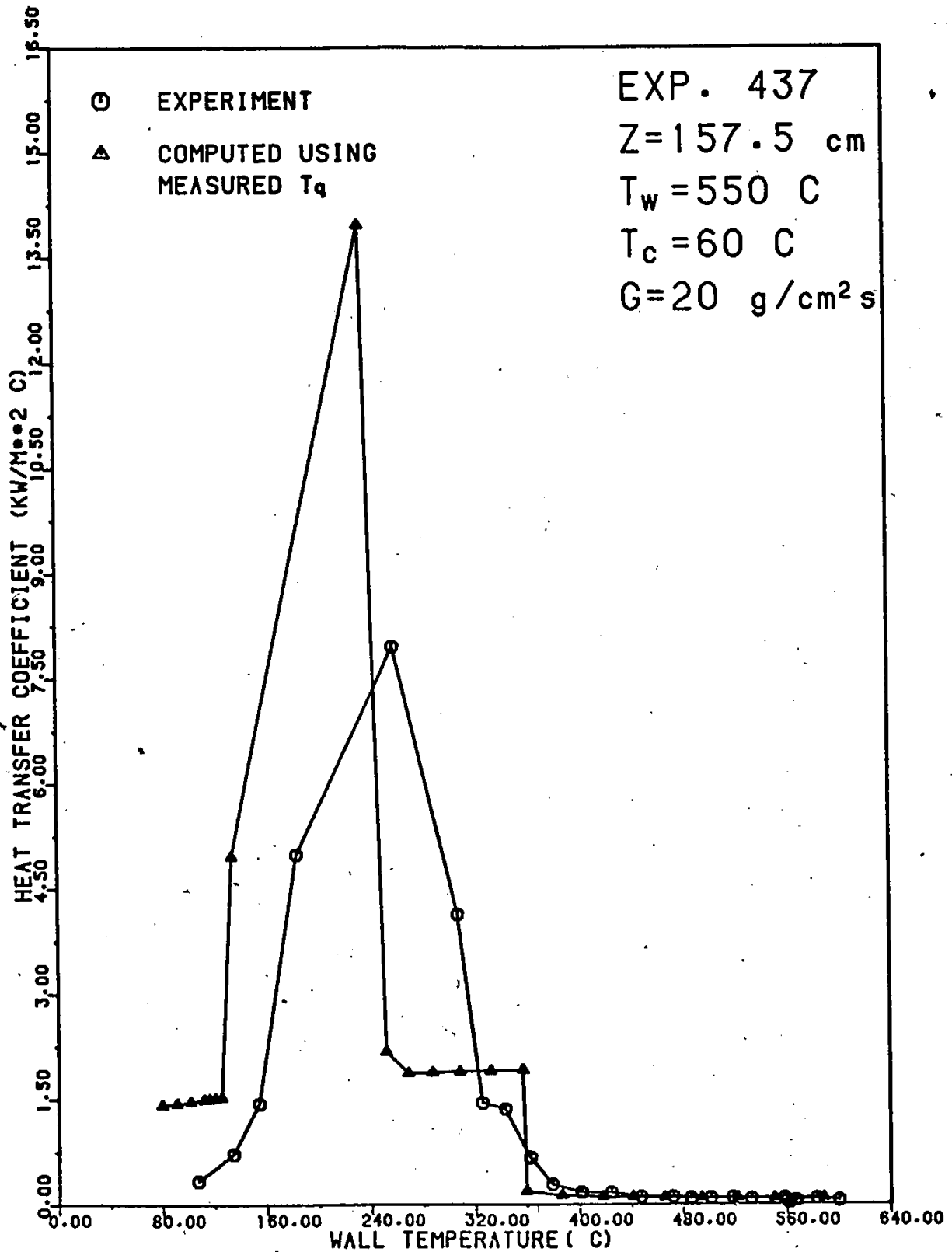


Fig. 5.41 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

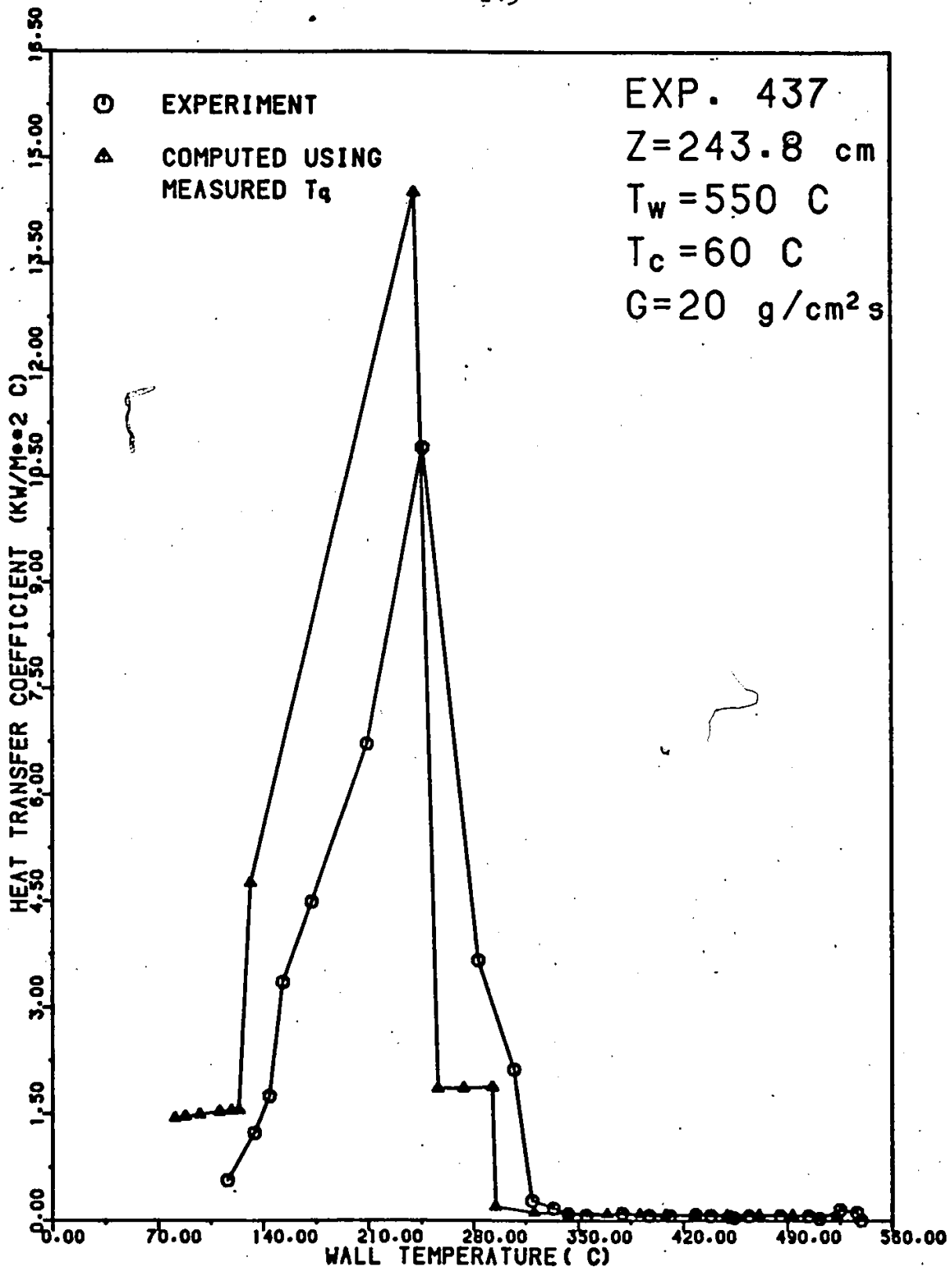


Fig. 5.42 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

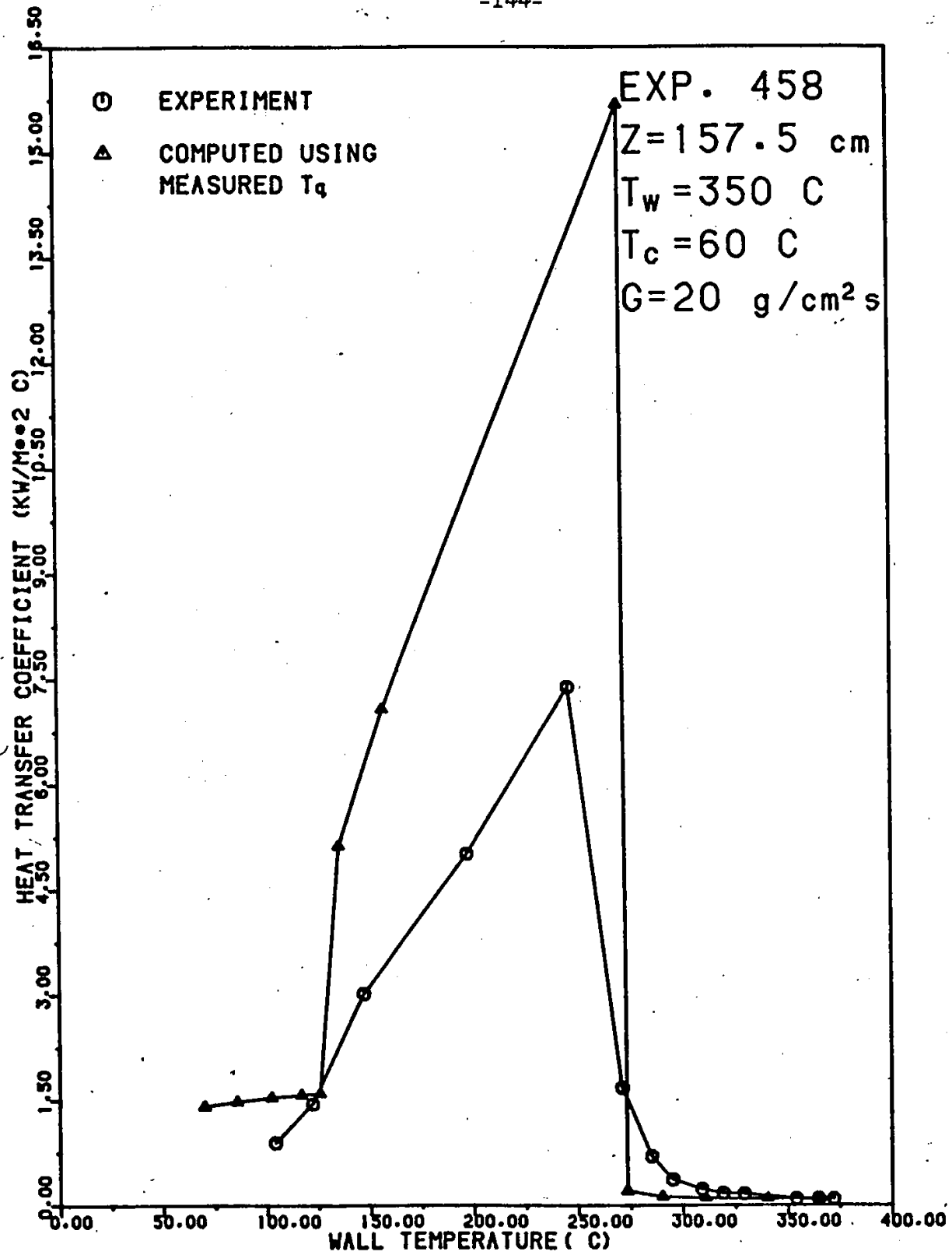


Fig. 5.43 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

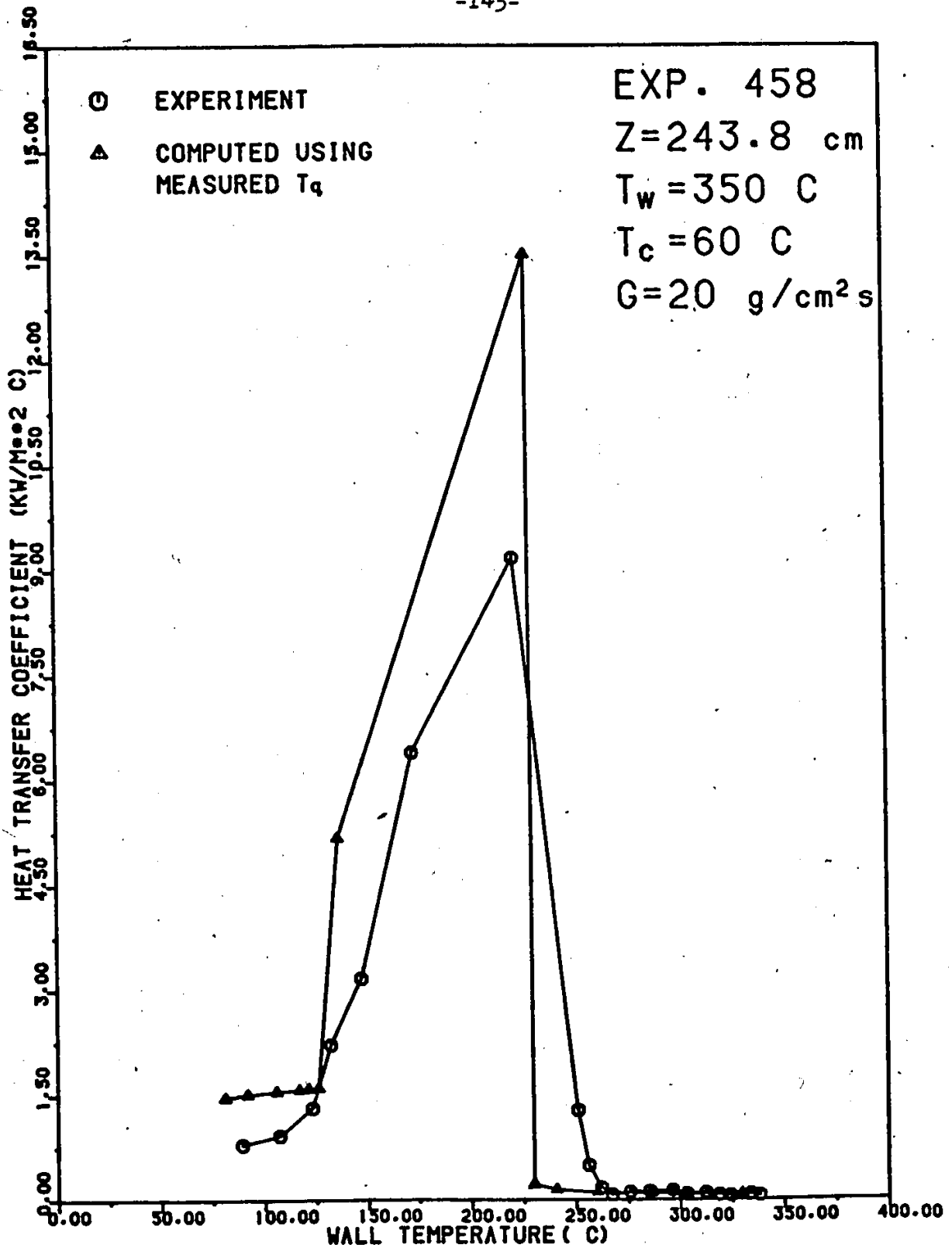


Fig. 5.44 h vs. T_w , Using Modified Chen's Nucleate Boiling Correlation

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