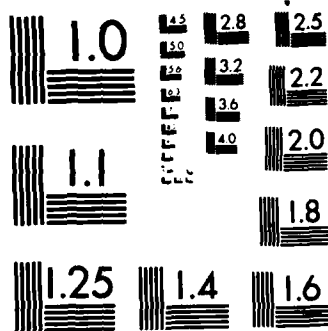


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THESIS

THE EFFECT OF CONDENSATE INUNDATION ON
STEAM CONDENSATION HEAT TRANSFER
TO WIRE-WRAPPED TUBING

by

Georgios Dimitriou Kanakis

June 1983

Thesis Advisor:

P.J. Marto

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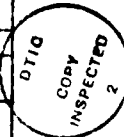
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REPORT DOCUMENTATION PAGE		READ INSTRUCTIONS BEFORE COMPLETING FORM
1. REPORT NUMBER	2. GOVT ACCESSION NO. AD-A132175	3. RECIPIENT'S CATALOG NUMBER
4. TITLE (and Subtitle) The Effect of Condensate Inundation on Steam Condensation Heat Transfer to Wire-Wrapped Tubing		5. TYPE OF REPORT & PERIOD COVERED Master's Thesis June 1983
7. AUTHOR(s) Georgios Dimitriou Kanakis		6. PERFORMING ORG. REPORT NUMBER
9. PERFORMING ORGANIZATION NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		8. CONTRACT OR GRANT NUMBER(s)
11. CONTROLLING OFFICE NAME AND ADDRESS Naval Postgraduate School Monterey, California 93940		10. PROGRAM ELEMENT, PROJECT, TASK AREA & WORK UNIT NUMBERS
14. MONITORING AGENCY NAME & ADDRESS (if different from Controlling Office)		12. REPORT DATE June 1983
		13. NUMBER OF PAGES 160
		15. SECURITY CLASS. (of this report)
		15a. DECLASSIFICATION/DOWNGRADING SCHEDULE
16. DISTRIBUTION STATEMENT (of this Report) Approved for public release; distribution unlimited.		
17. DISTRIBUTION STATEMENT (of the abstract entered in Block 20, if different from Report)		
18. SUPPLEMENTARY NOTES		
19. KEY WORDS (Continue on reverse side if necessary and identify by block number) Smooth, roped, wire wrapped tubes, inundation.		
20. ABSTRACT (Continue on reverse side if necessary and identify by block number) 4 Steam condensation heat transfer measurements were made in a 5-tube test condenser having an additional perforated tube to simulate up to 30 active tubes. Results were obtained for smooth tubes and roped tubes wrapped with wire. A Sieder-Tate equation was used to correlate the inside heat-transfer coefficient. For smooth tubes, a leading coefficient of 0.029 was found, while it was 0.061 for the roped tubes. The average		

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The Effect of Condensate Inundation on Steam
Condensation Heat Transfer to Wire-Wrapped Tubing

by

Georgios Dimitriou Kanakis
Lieutenant, Hellenic Navy
B.S., Naval Postgraduate School, 1982

Submitted in partial fulfillment of the
requirements for the degree of

MASTER OF SCIENCE IN MECHANICAL ENGINEERING

from the

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ABSTRACT

Steam condensation heat transfer measurements were made in a 5-tube test condenser having an additional perforated tube to simulate up to 30 active tubes. Results were obtained for smooth tubes and roped tubes wrapped with wire. A Sieder-Tate equation was used to correlate the inside heat-transfer coefficient. For smooth tubes, a leading coefficient of 0.029 was found, while it was 0.061 for the roped tubes. The average condensing coefficient measured for 30 smooth tubes was 0.59 times the Nusselt coefficient calculated for the first tube. When the smooth tubes were wrapped with wire, this ratio increased up to 0.86. Further, roped tubes without wire experienced a ratio of 0.63, while roped tubes wrapped with wire resulted in a ratio of 0.86. These preliminary data show that wire-wrapped tubes may lead to a significant reduction in condenser surface area.

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NOMENCLATURE

A_o	Outside, heat-transfer area of one tube (m^2)
A_i	Inside, heat-transfer area of one tube (m^2)
C_i	Sieder-Tate coefficient
C_{pw}	Specific heat of water evaluated at T_b (KJ/kg·K)
C_f	Correction factor $(\mu/\mu_w)^{0.14}$
D_i	Inner diameter of the tube (m)
D_o	Outer diameter of the tube (m)
g	Acceleration of gravity (9.81 m/s^2)
h_i	Experimentally-determined value for the inside, heat-transfer coefficient ($W/m^2 \cdot K$)
h_{fg}	Latent heat of vaporization (KJ/kg)
h_N	Local, outside, heat-transfer coefficient for the Nth tube ($W/m^2 K$)
h_{Nu}	Heat-transfer coefficient calculated from the Nusselt equation ($W/m^2 \cdot K$)
h_1	Outside heat-transfer coefficient for the first tube ($W/m^2 \cdot K$)
k_f	Thermal conductivity of the condensate film ($W/m \cdot K$)
k	Thermal conductivity of the cooling water evaluated at T_b ($W/m \cdot K$)
k_m	Thermal conductivity of titanium ($W/m \cdot K$)
L	Condensing length (m)
LMTD	Logarithmic Mean Temperature Difference ($^{\circ}C$)
m	Slope of the least-squares-fit, straight line
$\dot{m}$	Mass flow rate of cooling water (kg/min)

N	The number of tubes in a column or the tube number of a given tube
Nu	Water-side Nusselt number
Pr	Prandtl number evaluated at T_b
Q	Heat transfer rate (W)
q''	Heat flux based on outside area (W/m^2)
Re	Water-side Reynolds number
R_f	Fouling thermal resistance (m^2K/W)
R_w	Wall thermal resistance based on the outside area (m^2K/W)
R_1	Outside, local, heat-transfer coefficient ratio (h_N/h_1)
R_2	Outside, average, heat-transfer coefficient ratio ($\bar{h}_N/h_1$)
S/D	Spacing-to-diameter ratio of tubes
T_b	Average cooling water bulk temperature ($^{\circ}C$)
T_{ci}	Cooling water inlet temperature ($^{\circ}C$)
T_{co}	Cooling water outlet temperature ($^{\circ}C$)
T_w	Wall temperature ($^{\circ}C$)
T_f	Average condensate film temperature ($^{\circ}C$)
T_{sat}	Saturation temperature of steam ($^{\circ}C$)
T_v	Vapor (steam) temperature ($^{\circ}C$)
U_n	Overall heat-transfer coefficient (m^2K/W)
U_o	Outside heat-transfer coefficient (m^2K/W)
V_w	Cooling water velocity (m/s)
X	Sieder-Tate parameter ($X = Re^{+0.8} Pr^{+1/3} (\frac{\mu}{\mu_w})^{0.14}$)
ΔT	Temperature difference ($T_w - T_b$) ($^{\circ}C$)

Greek Symbols

ζ	Heat capacity parameter ($C_p \Delta T / h_{fg}$)
μ	Dynamic viscosity of water evaluated at T_b ($N \cdot s / m^2$)
μ_f	Dynamic viscosity of condensate evaluated at T_f ($N \cdot s / m^2$)
μ_w	Dynamic viscosity of water evaluated at T_w ($N \cdot s / m^2$)
ξ	Acceleration parameter ($K \Delta T / \mu h_{fg}$)
ρ	Density of cooling water (kg / m^3)
ρ_v	Vapor density (kg / m^3)

ACKNOWLEDGEMENT

Any work of this sort necessarily represents the influence of many people. I am especially indebted to Professor Paul J. Marto, my Thesis Advisor, for his support and patient guidance throughout the project.

My sincere appreciation to Dr. A.S. Wanniarachchi, my Second Reader, for his great help in overcoming even the smallest obstacles.

Special thanks to Mr. Thomas Christian, Mr. John Moulton, Mr. Willard Dames, and Mr. Ron Lonqueira for their technical assistance.

I wish to thank my wife Vassia and my family for their moral support, understanding and encouragement. Without it, this project certainly would not be complete today.

I. HISTORICAL BACKGROUND

In recent years, there has been a continued interest in the reduction of the size and weight of propulsion systems aboard both surface vessels and submarines. Especially, actual dimensions of naval condensers have a critical bearing on cost and performance of the ship. Often, compactness is more important than thermal effectiveness when the overall performance of the ship is considered; in a submarine, the diameter of the pressure hull can depend on the dimensions of the condenser.

The importance of compactness justifies measures to raise the overall heat-transfer coefficient of condenser tubes despite the penalties which may occur, i.e., the increased pumping power and tube cost.

Naval condenser design is based upon the Heat Exchange Institute (HEI) specifications for steam condensers [Ref. 1] and also the standards of the Tubular Exchange Manufacturers Association (TEMA) [Ref. 2]. Search [Ref. 3] investigated the present condenser design processes, including the feasibility of enhanced heat transfer in naval condensers. He concluded that the current design is very conservative, and he predicted that a forty-percent reduction in condenser weight and volume could be achieved depending on the heat-transfer enhancement method used.

In recent years, many research efforts have been directed to the study of heat-transfer enhancement techniques and their application to heat-exchanger design. Webb [Ref. 4] has summarized extensive works of augmentation techniques. At the Naval Postgraduate School, Beck [Ref. 5], Pence [Ref. 6], Reilly [Ref. 7], Fenner [Ref. 8] and Ciftci [Ref. 9] conducted experimental research into various kinds of enhancement schemes employing a single-tube test condenser.

The above-mentioned investigations concluded that, for the same diameter tube, the overall heat-transfer coefficient of enhanced tubes can exceed those for smooth tubes by almost 100 percent. Reilly and Fenner [Ref. 10] revealed that most of the above mentioned augmentation occurred on the cooling-water side due to a combination of increased surface area, and increased turbulence and swirl in the cooling water flow. Little or no improvement occurred on the steam side. Eissenberg [Ref. 11] performed an extensive study on condenser-tube, heat-transfer coefficients using a multi-tube bundle.

In order to investigate the outside heat-transfer performance of various enhanced tubes in tube bundles, research was conducted at the Naval Postgraduate School. Noftz [Ref. 12] modified a test apparatus initially designed by Morrison [Ref. 13] to simulate a tube bundle using five

active tubes arranged in a vertical plane. A perforated tube was located at the top of the bundle, through which water was flooded to simulate bundles having up to 30 tubes in a vertical row. His investigations determined that the heat-transfer coefficients for a given tube of the tube bundle increased as the mean vapor velocity increased, but decreased as the amount of condensate inundation increased. The experimentally found values for the heat-transfer coefficients were comparative with the Nusselt theory.

Based upon research done currently, it is evident that present day smooth-tube steam condensers, operating under typical conditions, have limitations in their thermal efficiency, due to a large thermal resistance which occurs on the tube side of the condenser. This resistance is generally larger than any of the thermal resistances that occur on the steam side, in the tube wall, those due to fouling, or due to noncondensable gases. However, employing enhanced tubes, the inside heat-transfer coefficient can be increased by 100 to 200 percent over the smooth-tube case. The outside heat-transfer coefficient, on the other hand, is increased by only 10 to 50 percent. In this situation, the thermal resistances on the inside and outside of the tube can be approximately equal.

Webb [Ref. 4] reported that the dominant thermal resistance in film condensation is that of conduction across the condensate film and, therefore, a surface geometry that promotes reduced film thickness will provide enhancement.

Thomas, et al, [Ref. 14] tested ammonia condensation on a smooth tube with a wire wrapped in a helical manner. The measured condensing coefficient was approximately three times that predicted by the Nusselt equation for a smooth tube. Surface-tension forces draw the condensate to the base of the wires, which act as condensate run off channels.

Webb [Ref. 4] stated that, when noncondensables are present, an additional thermal resistance is introduced in the gas at the vapor-liquid interface. Mixing in the gas film will substantially reduce this thermal resistance. Therefore, the maintenance of high vapor velocities, or special surface geometries that promote a higher heat-transfer coefficient in the gas film will substantially alleviate the performance deterioration due to noncondensables.

Cunningham [Ref. 15] presented in his paper that, for the roped tubes on the vapor (shell) side, the enhancement is achieved by improved condensate drainage, while on the coolant (tube) side the helical ridges increased turbulence and, as a result, the inside convective coefficient.

Improvements on the condensing side up to 100 percent have been reported for single-tube tests [Refs. 16;17]. Although titanium has a low thermal conductivity, it provides a high resistance to erosion and water-side fouling. Titanium tubes with enhancement both inside and outside are commercially available through the Wolverine Tube Division of Universal Oil Products, Inc. The applications of these tubes having all the inherited properties of titanium are promising for naval condensers.

The goals of this thesis were therefore to:

1. Obtain baseline heat-transfer performance data for the test condenser utilizing 16 mm O.D. smooth titanium tubes.
2. Conduct steam condensation tests with the following enhanced tube geometries to determine steam-side heat-transfer coefficients in relation to smooth-tube performance:
 - a. Wolverine "roped" tubes.
 - b. Wolverine "roped" wrapped with titanium wire.
 - c. Smooth tubes wrapped with titanium wire.

II. THEORETICAL BACKGROUND

The combined effect of vapor shear and inundation on the condensate film heat-transfer coefficient for cylindrical, horizontal tubes within tube bundles is a very complex and still insufficiently-understood subject, which is of importance to the efficient design of steam condensers. Although many researchers have studied this subject both theoretically and experimentally, there is no accurate methodology available for predicting the condensate-film, heat-transfer coefficient within tube bundles.

In 1916, Nusselt conducted his pioneering analysis for the simple case of condensation occurring on the outside of a single, isolated, horizontal tube. He idealized the problem by making the following assumptions for single tubes as stated by Nobbs [Ref. 18].

1. The wall temperature is constant.
2. The flow is laminar in the condensate film.
3. Heat transfer in the condensate is by conduction, and subcooling may be neglected.
4. The fluid properties are constant within the condensate film.
5. The forces due to hydrostatic pressure, surface tension, inertia, and vapor-liquid interfacial shear are negligible when compared to the viscous and gravitational forces.
6. The surrounding steam and vapor/liquid interface are at saturation temperature.

7. The film thickness is small when compared with normal tube diameters and the effects of curvature are small.

Based on the above assumptions, Nusselt predicted the famous relationship for the heat-transfer coefficient:

$$h_{Nu} = 0.725 \left[\frac{k^3 \rho (\rho - \rho_v) h_{fg} \cdot g}{\mu D (T_{sat} - T_w)} \right]^{1/4} \quad (2.1)$$

In order to simulate and analyze a tube bundle, Eissenberg [Ref. 11] stated the following additional assumptions:

8. Condensate drains as a laminar sheet from a tube on to the tube directly underneath in such a way that velocity and temperature gradients are not lost in the fall between tubes.
9. The saturation temperature and the tube-wall temperature are constant for all tubes in the bank.

Jakob [Ref. 19] extended the Nusselt analysis for filmwise condensation heat transfer on a vertical in-line row of horizontal tubes as shown in Figure 1a.

The above-mentioned assumptions were combined with the assumption of constant temperature drop across the condensate film for all the tubes, and the average coefficient for a vertical row of N tubes was predicted to be:

$$\bar{h}_N = 0.725 \left[\frac{k^3 \rho (\rho - \rho_v) h_{fg} \cdot g}{\mu N D (T_{sat} - T_w)} \right]^{1/4} \quad (2.2)$$

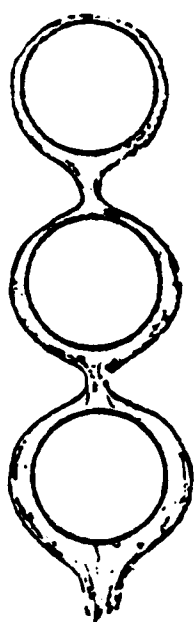


Figure 1a.
Idealized Condensation
on Banks of Tubes

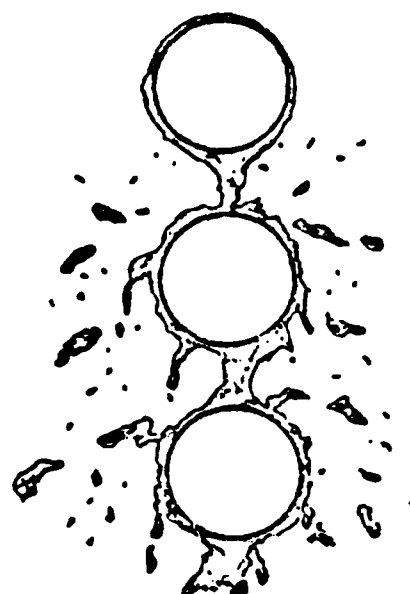


Figure 1b.
More Realistic Picture of
Condensation on Banks of Tubes

Upon, dividing equation (2.2) by equation (2.1), the Nusselt theory can be expressed as:

$$\frac{\bar{h}_N}{h_{Nu}} = N^{-1/4} \quad (2.3)$$

Equation (2.3) can be also expressed in terms of the local coefficient for the N-th tube:

$$\frac{h_N}{h_{Nu}} = N^{3/4} - (N-1)^{3/4} \quad (2.4)$$

In reality, condensate does not drop off in a continuous laminar sheet, but drops off instead by discrete droplets of liquid, as shown in Figure 1b, depending upon the surface tension of the condensate. These droplets create ripples in the condensate film, and thereby decrease the performance degradation due to inundation. Based on his research, Kern [Ref. 20] proposed a less conservative relationship:

$$\frac{\bar{h}_N}{h_1} = N^{-1/6} \quad (2.5)$$

or, in terms of the local coefficient for the N-th tube,

$$\frac{h_N}{h_1} = N^{5/6} - (N-1)^{5/6} \quad (2.6)$$

Chen [Ref. 21] considered the following conditions:

1. the momentum gain of the falling condensate between tubes, and
2. the condensation of vapor on the condensate between tubes,

and concluded that:

$$\frac{\bar{h}_N}{h_{Nu}} = N^{-1/4} \left[(1 + 0.2\zeta(N-1)) \left(\frac{1-0.68\zeta + 0.02\zeta\xi}{1+0.95\xi - 0.15\zeta\xi} \right) \right]^{1/4} \quad (2.7)$$

where;

$$\xi = \frac{k\Delta T}{\mu h_{fg}} \quad , \quad \zeta = \frac{C_p \Delta T}{h_{fg}}$$

and

$$\xi = \frac{\zeta}{P_r}$$

The above approximate expression, due to Chen, is valid for most ordinary applications.

Experimental work done by Eissenberg [Ref. 11], in order to investigate the effects of steam velocity, condensate inundation, and noncondensable gases on the heat-transfer coefficient, revealed that condensate does not always drain onto tubes aligned vertically, but can be diverted sideways, caused by local, vapor-flow conditions. The condensate thus follows a staggered path as shown in Figure 2.

Eissenberg, making the assumption that the flow is dominated by gravity, stated in this side-drainage model, that condensate strikes the lower tubes on their sides

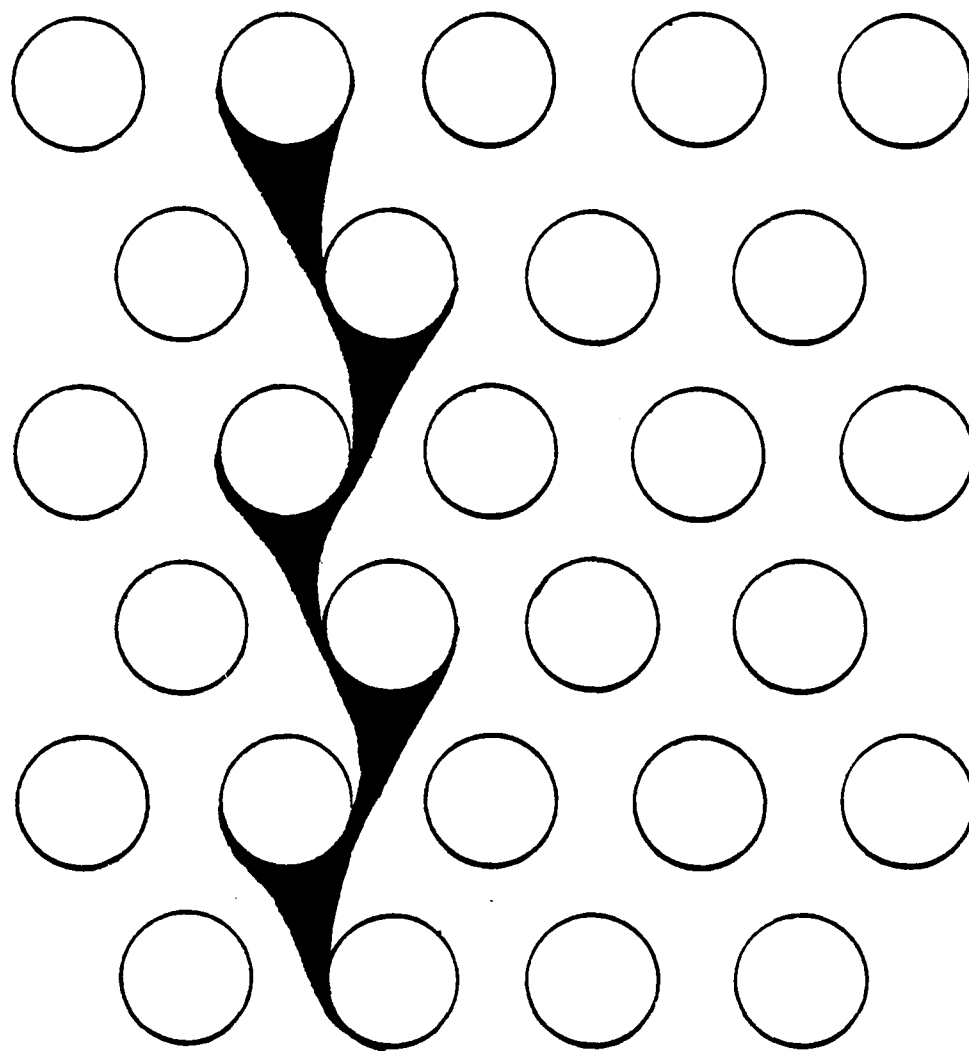


Figure 2. Droplet Path Through a Tube Bundle with Side Drainage

rather than their tops. Therefore, the inundation effects influence the condensate flow only on the lower half of the tubes, which transfer less heat than the upper half. Based on the above-mentioned model, Eissenberg obtained the following formula:

$$\frac{\bar{h}_N}{h_{Nu}} = 0.60 + 0.42 N^{-1/4} \quad (2.8)$$

Much experimental research has been conducted for studying the effect of condensate inundation. In general, the obtained data are highly scattered as shown in Figure 3. Berman [Ref. 22] made a comprehensive comparison of filmwise condensation data on bundles of horizontal tubes, and he concluded that the wide variation in experimental data for tube bundle inundation is caused by the following variables:

1. bundle geometry (in-line or staggered),
2. tube spacing,
3. type of condensing fluid,
4. operating pressure,
5. heat flux, and
6. local vapor velocity.

In addition, noncondensable gases, and insufficient steam for lower tubes can cause such scattering of data.

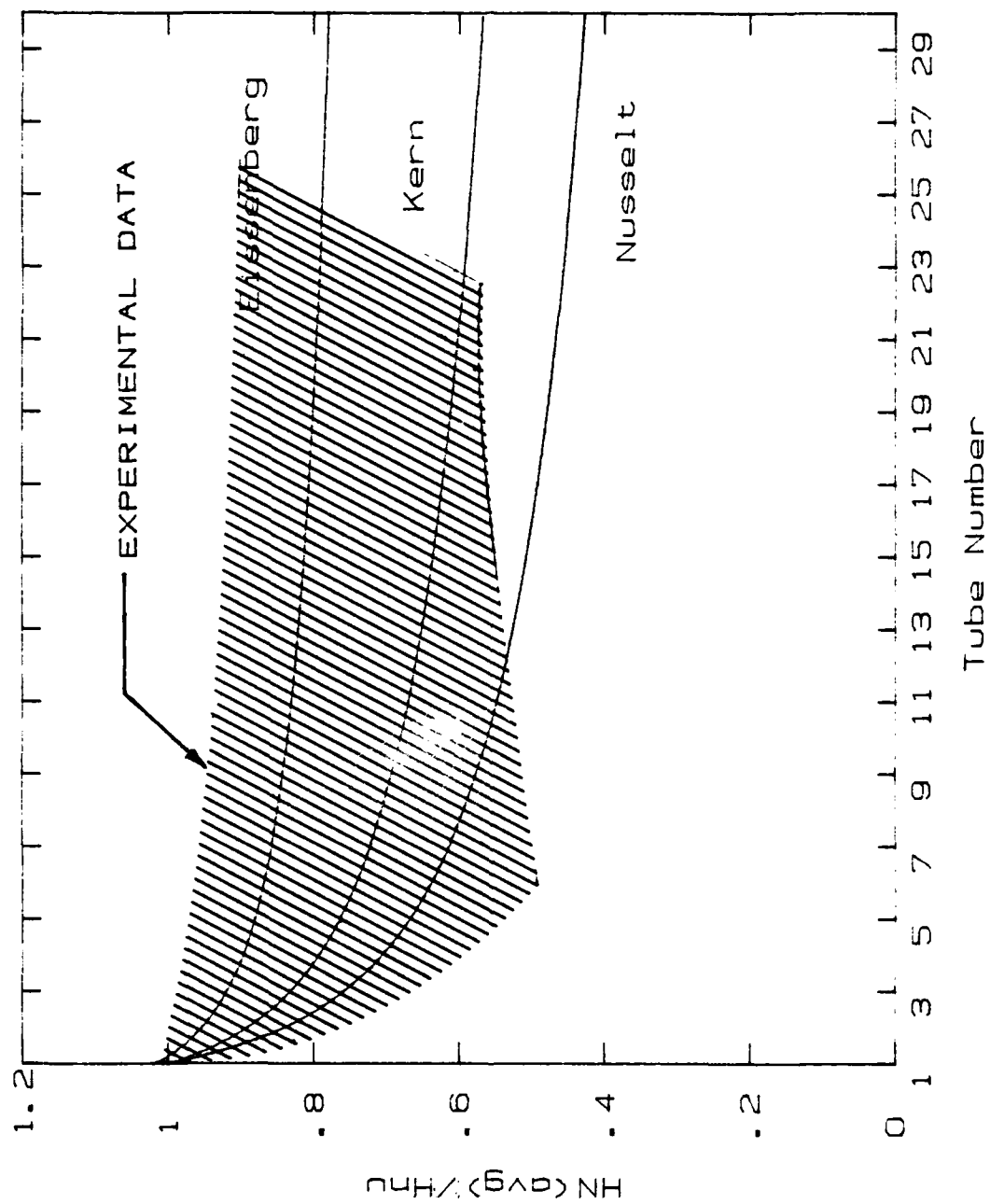


Figure 3. Schematic Comparison of Various Theories with Experimental Data for Condensation Inundation Studies

Nobbs [Ref. 18] used one active tube in a dummy tube bundle. He simulated additional condensate by using three porous tubes. Based on his results, he concluded the following:

1. Vapor velocity increases the condensate heat-transfer coefficient on both inundated and uninundated tubes in a given tube bundle.
2. The effect of inundation is to reduce the heat-transfer coefficient.
3. The condensate drainage path is often not vertically downwards but in a diagonal direction. This can result in tubes receiving different amounts of inundation.

Marto and Nunn [Ref. 23] made a comprehensive survey of the effects of noncondensable gases. They stated that these can be classified into one of two categories:

1. the introduction of an additional local thermal resistance due to the propagation of noncondensable gases towards a condensing tube surface under the influence of a gradient in partial pressure, and
2. the cumulative effect of gas blanketing where uneven rates of condensation in a condenser bundle eventually lead to regions where tubes are inoperative in a condensing role.

Experiments done recently by Cunningham [Ref. 15] revealed that noncondensable gases have a less effect on the performance of finned tubes compared to smooth tubes. Based on the above discussion, it is clear that the performance degradation due to noncondensables must be taken into consideration in realistic designs of tube bundles.

Another very important factor in the performance of a condenser is the effect of vapor velocity. As noted earlier, the vapor shear plays a beneficial role. Berman and Tumanov [Ref. 24] conducted experiments on a single horizontal tube placed in a bank of uncooled neighboring tubes. For vapor in vertical downflow, they found a relation between the vapor Reynolds number, and the heat flux as:

$$\frac{h}{h_N} = 1 + 9.5 \times 10^{-3} (\text{Rev})^{11.8/\sqrt{\text{Nu}}}$$

with the restriction, that $(\text{Rev})^{11.8/\sqrt{\text{Nu}}} < 50$

Eissenberg [Ref. 25] has stated that in designing experimental bundles, the combined effects of inundation and vapor shear are very important. In fact, the use of narrow condenser bundles to experimentally study vapor shear and inundation effects is preferred to simulate large condensers. However, small narrow tube bundles can create errors, due to the following causes:

1. Wall flow: condensate drainage may reach side walls;
2. Dummy tubes: condensate inundation may disperse;
3. Vapor lanes: vapor may bypass along the side walls; and
4. Noncondensable gases: they can affect condensation even at low concentration in the bulk stream, particularly if steam is recycled.

III. EXPERIMENTAL FACILITY

A. TEST FACILITY

The test facility shown in Figures 4 and 5, was designed and built by Morrison [Ref. 13] and modified by Noftz [Ref. 12] to simulate an active tube column having up to 30 tubes in increments of five tubes deep (i.e., five, ten, fifteen, etc.). Some elements of the original test facility were modified to allow for more efficient operation of the facility.

A detailed description of the components used in the test facility is given in Reference 12. Only a short description of these components will be found in this report. Particular attention, however, will be focused on the experimental tubes. Calibration procedures for components requiring calibration are outlined by Reilly [Ref. 7].

B. STEAM SYSTEM

The steam system shown in Figure 6 was modified from Noftz's initial design. From the house supply line, steam flows via a 19 mm O.D. stainless steel line through a steam supply valve (MS-3) to a cast iron steam separator. The steam continues through the system past two Nupro bellows valves, which were used in conjunction with the supply valve

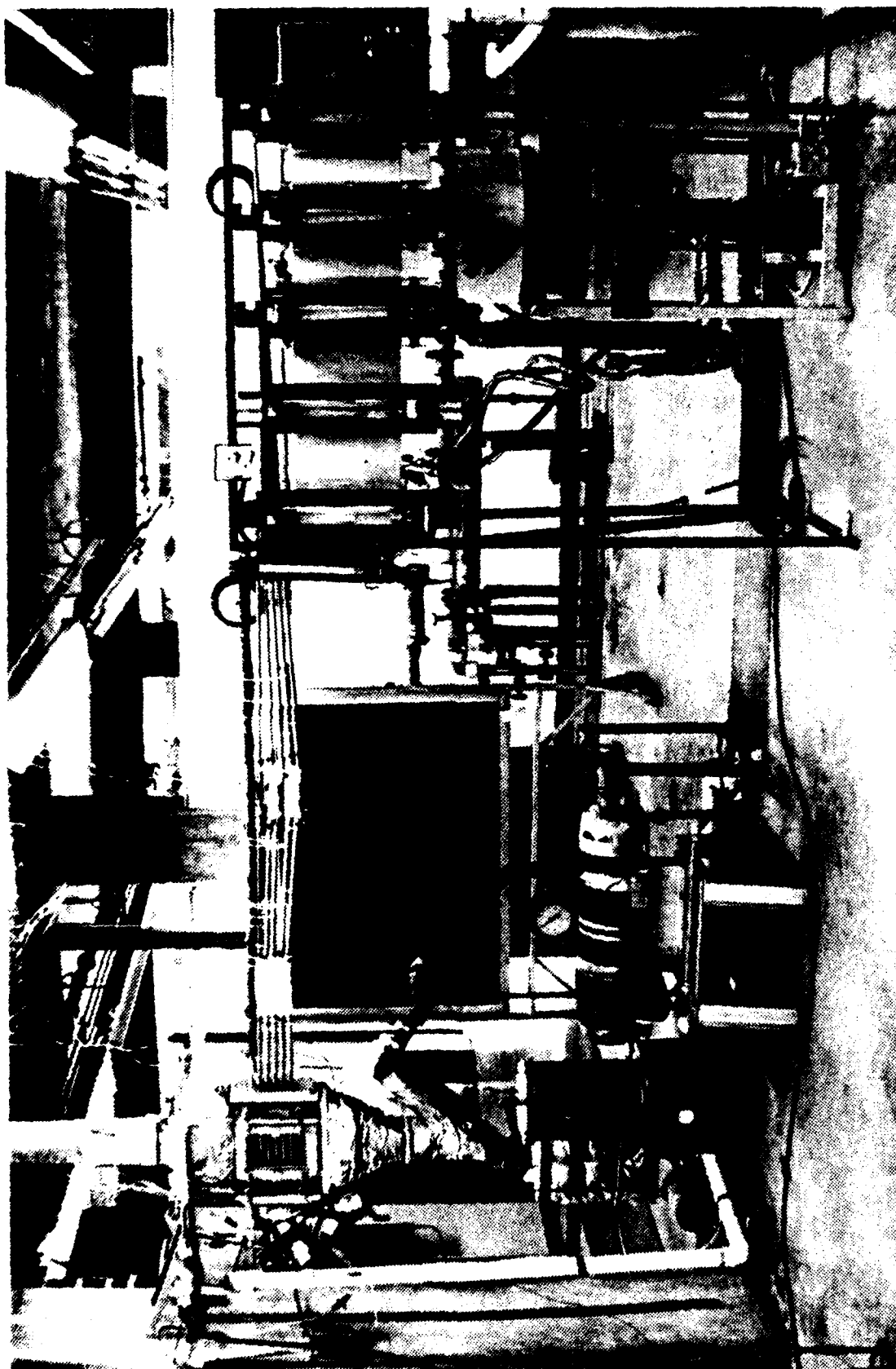


Figure 4. Front View of Test Facility

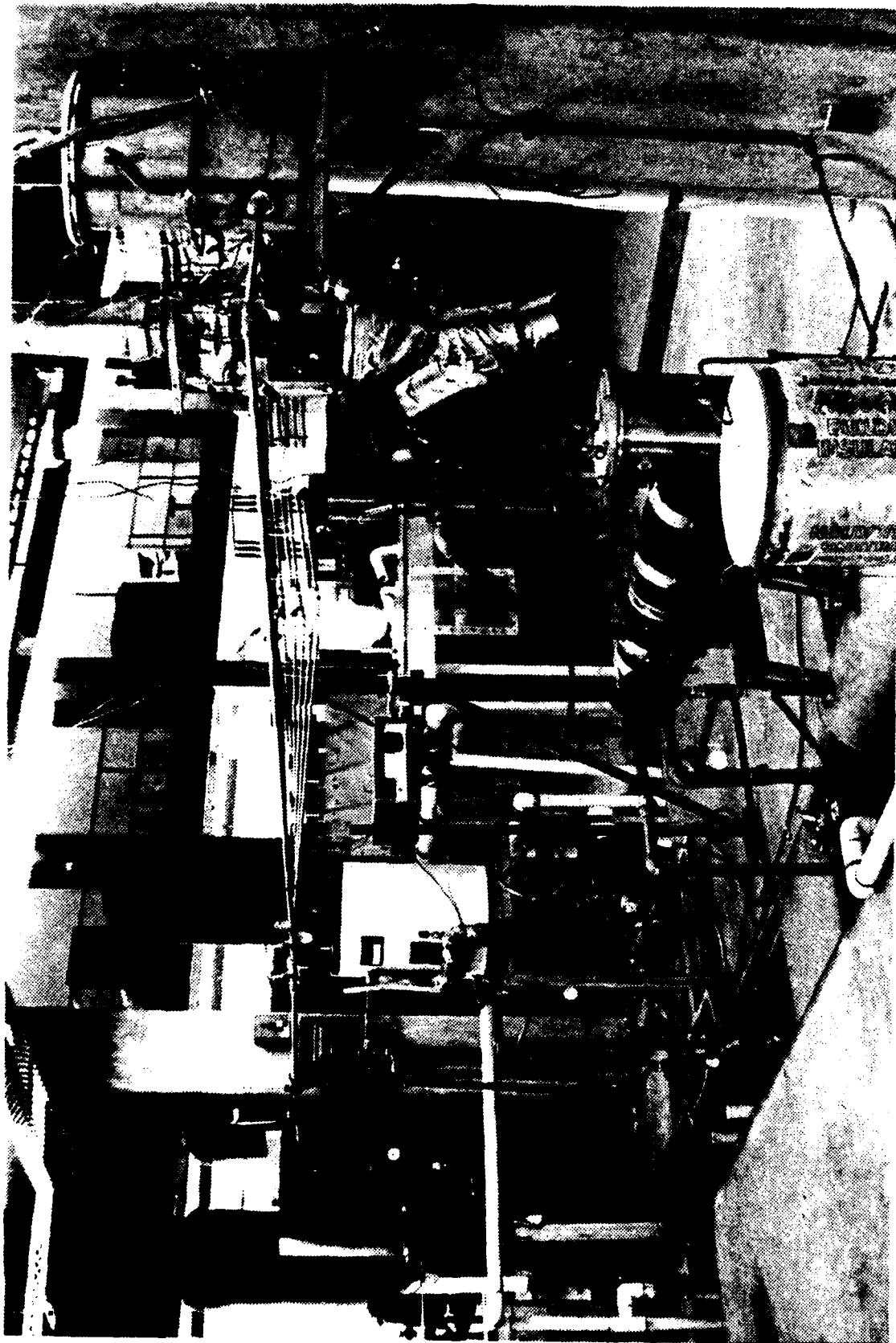


Figure 5. Rear View of Test Facility

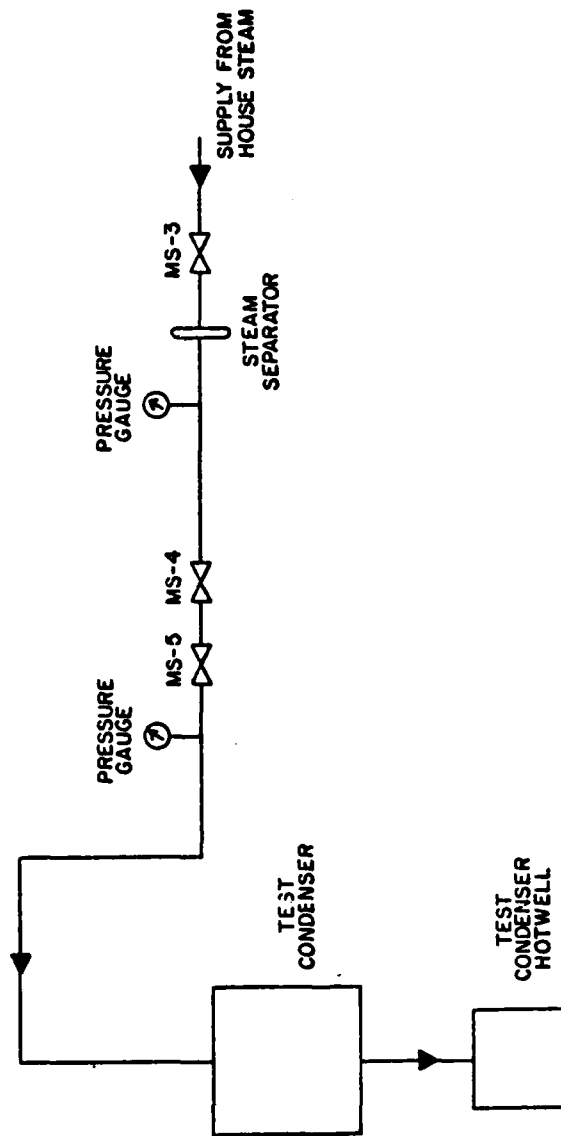


Figure 6. Schematic of Steam System

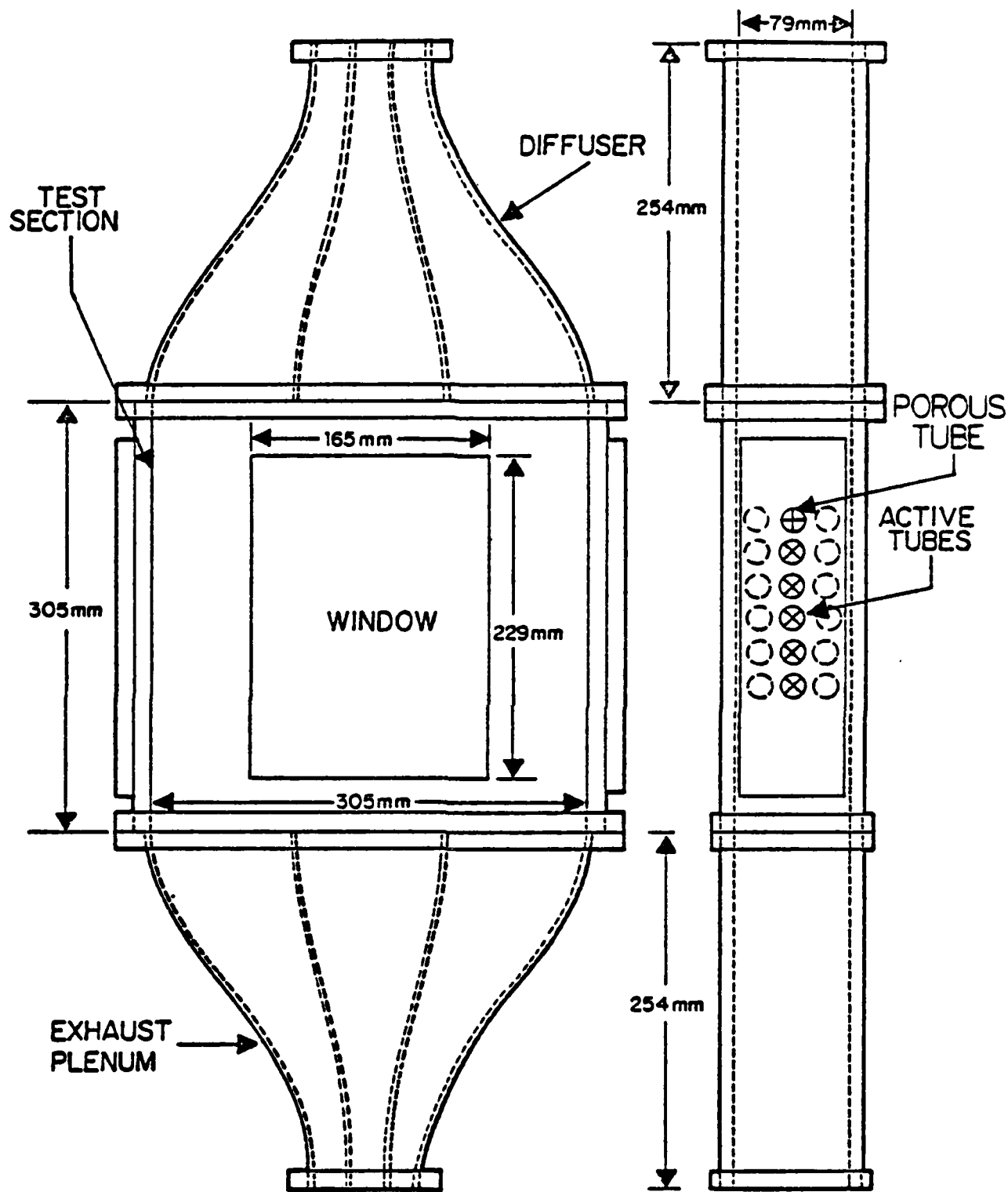
to regulate the steam supply pressure. From these valves, the steam flows into the test condenser diffuser.

The steam supply pressure was monitored by a pressure gage just downstream of the steam supply valve; also a compound gage just after the Nupro valves was used to monitor the pressure in this line.

The operator had no control over the state point, quality or noncondensable gas content since house steam was used. However, especially at nights and during weekends, the state point of the steam at the inlet to the test condenser was found to be nearly constant.

C. TEST CONDENSER

The test condenser shown in Figure 7 was unchanged from Noftz's initial design. Steam enters via the top, passes through the transition piece, the vortex annihilator, and finally through the diffuser. The dimensions of the test condenser were 305 mm X 305 mm X 79 mm and it was made of stainless steel. These dimensions allowed for a maximum of twenty-seven 16 mm O.D. tubes arranged in an in-line configuration of three columns of nine tubes each. For this experiment, the in-line configuration was used with a middle column of five active tubes flanked on either side by a column of five dummy tubes. Just on top of the upper active tube, a perforated, distilled water supply tube was positioned, and flanked on both sides by dummy tubes. In order



NOTE: ALL COMPONENTS DRAWN TO SCALE

Figure 7. Sketch of Test Condenser

to conduct the experiments, a square, in-line arrangement of tubes, with a pitch-to-diameter ratio of 1.5 was used.

A vertical slot (Figure 7) along the centerline of each condenser end plate was used for active and perforated tube installation. The tubes were positioned using nylon tube sheets that were attached to the exterior of the condenser end plates.

To minimize heat losses, and also to prevent leaks from the tube sheets, each test condenser side was provided with a nylon tube sheet one-inch thick with six holes ($S/D = 1.5$), having about 0.5 mm tube clearance, which allowed the tubes to be easily slid into the test condenser through the tube sheets. The exterior side of each tube sheet had grooves to support O-rings. The aluminum tube sheets had six holes, having a tube clearance of about 0.5 mm and were used as sealing plates. The diffuser, exhaust plenum, transition piece, vortex annihilator, and the exhaust piping which are shown in Figures 8 and 9, were insulated with rubber insulation.

A viewing window allowed viewing of the condensation process. A double-walled glass window was used, and heated air was fed through the clearance between the two glasses to eliminate fogging on the inner glass.

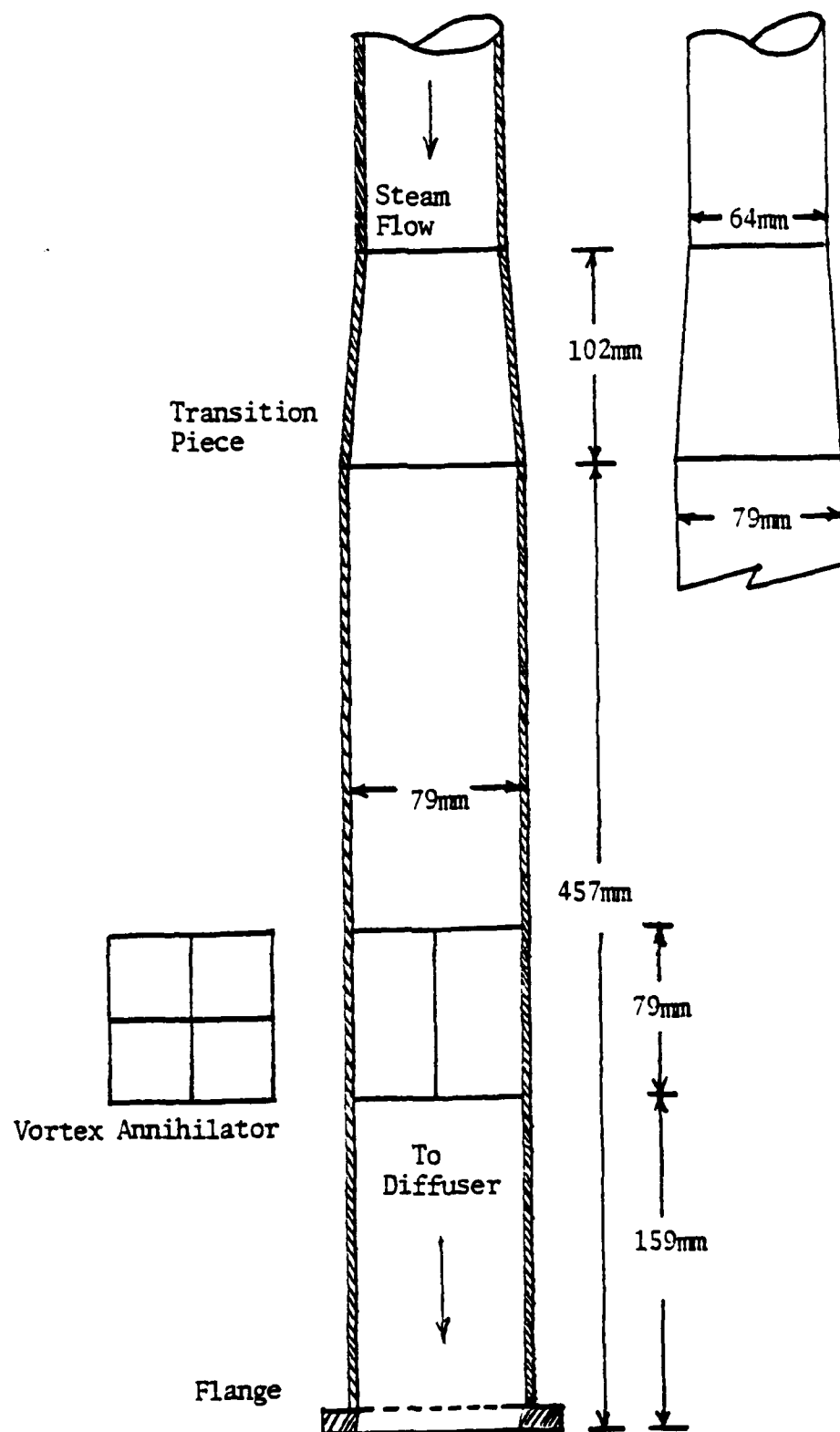


Figure 8. Details of Transition Piece and Vortex Annihilator

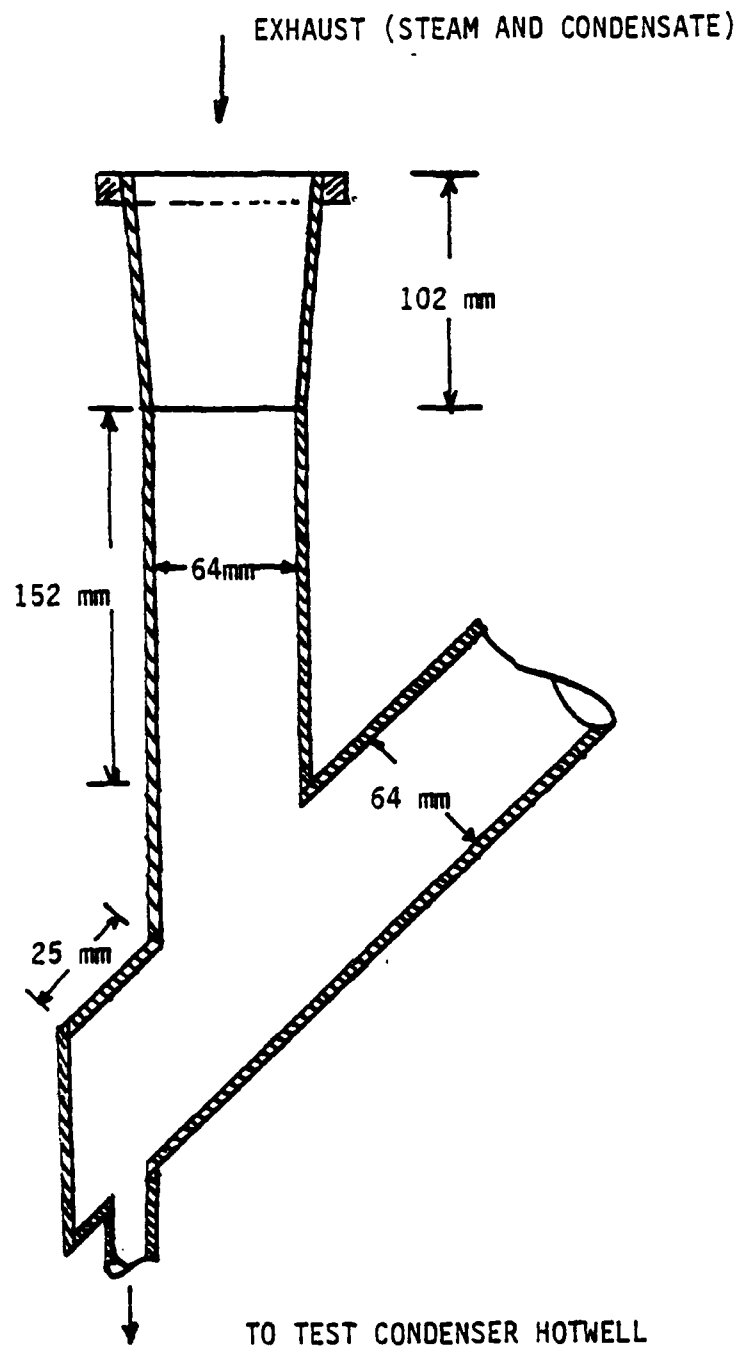


Figure 9. Details of Exhaust and Condensate Piping from Exhaust Plenum

D. TEST CONDENSER TUBES

Four kinds of active tubes were tested in this experiment. The tubes were manufactured by the Wolverine Division of Universal Oil Products, and all were made of titanium.

The first kind of tubes, shown in Figure 10 were smooth titanium tubes of 16 mm O.D. with a 1.65 mm wall thickness. The second kind of tubes, shown in Figure 11, were single-start, helically-corrugated tubes, designated as Low Pressure Drop (LPD), with 16 mm O.D. and a 1.65 mm wall thickness. The third kind of tubes, shown in Figure 12, were also single-start, helically-corrugated LPD tubes, wrapped with titanium wire of 1.58 mm O.D. The fourth kind of tubes, shown in Figure 13, were smooth tubes, wrapped with 1.58 mm O.D. titanium wire on the same pitch as the roped Wolverine tubes. In order to wrap the titanium wire, the tubes were attached into a lathe, and the wire was welded at the start of the helical groove. Then, under a constant tension of 4 kg, the wire was wrapped manually by turning the chuck of the lathe. Finally, the free end of the titanium wire was welded at the end of the helical groove.

The 660-mm-long active tubes were connected to separate cooling water supply and discharge lines. The dummy tubes flanking the active tubes were made of 16 mm O.D. stainless steel. These tubes served to direct the steam flow so as to

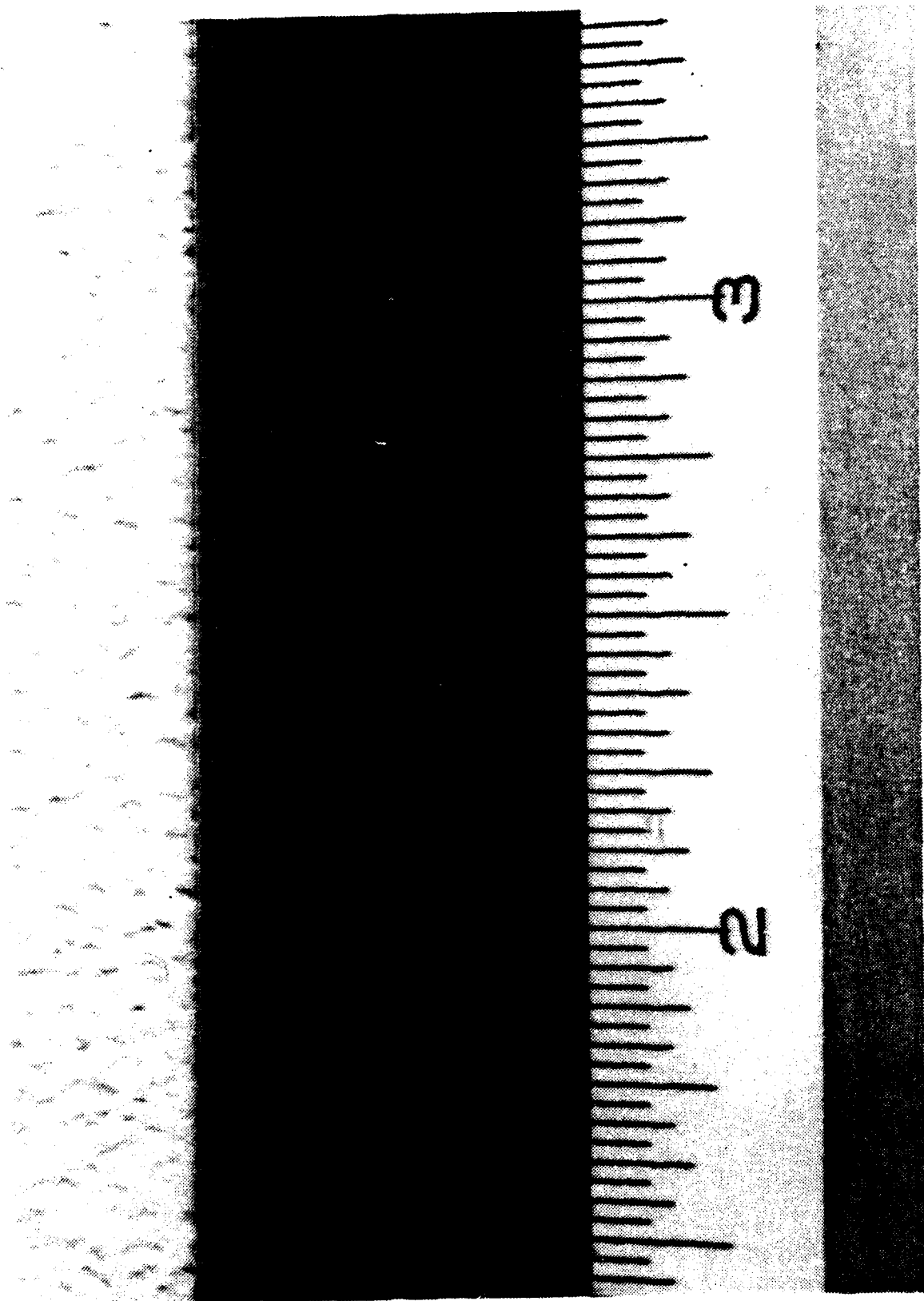


Figure 10. Photograph of Smooth Titanium Tube

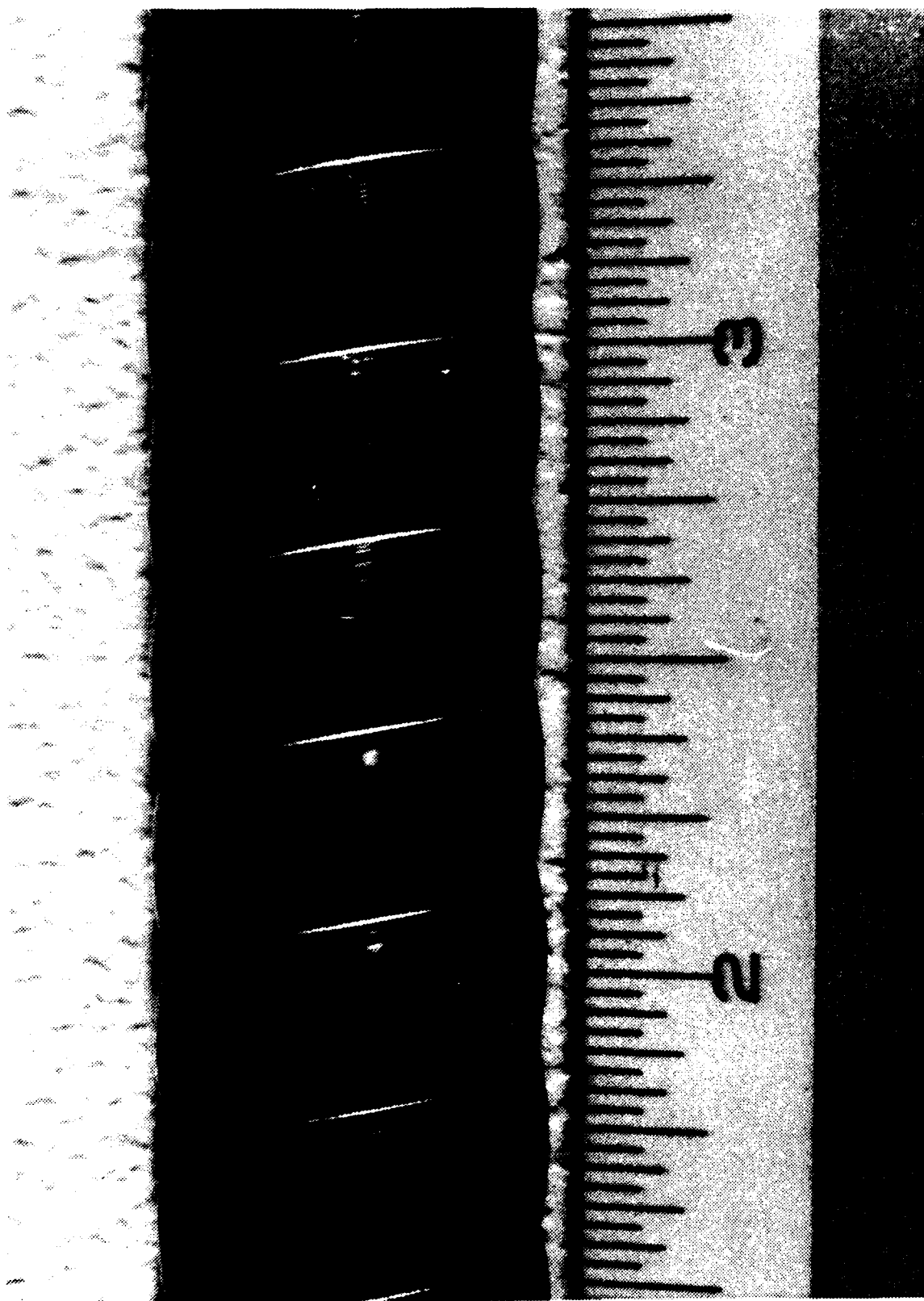


Figure 11. Photograph of Roped Titanium Tube



Figure 12. Photograph of Wire Wrapped Roped Titanium Tube

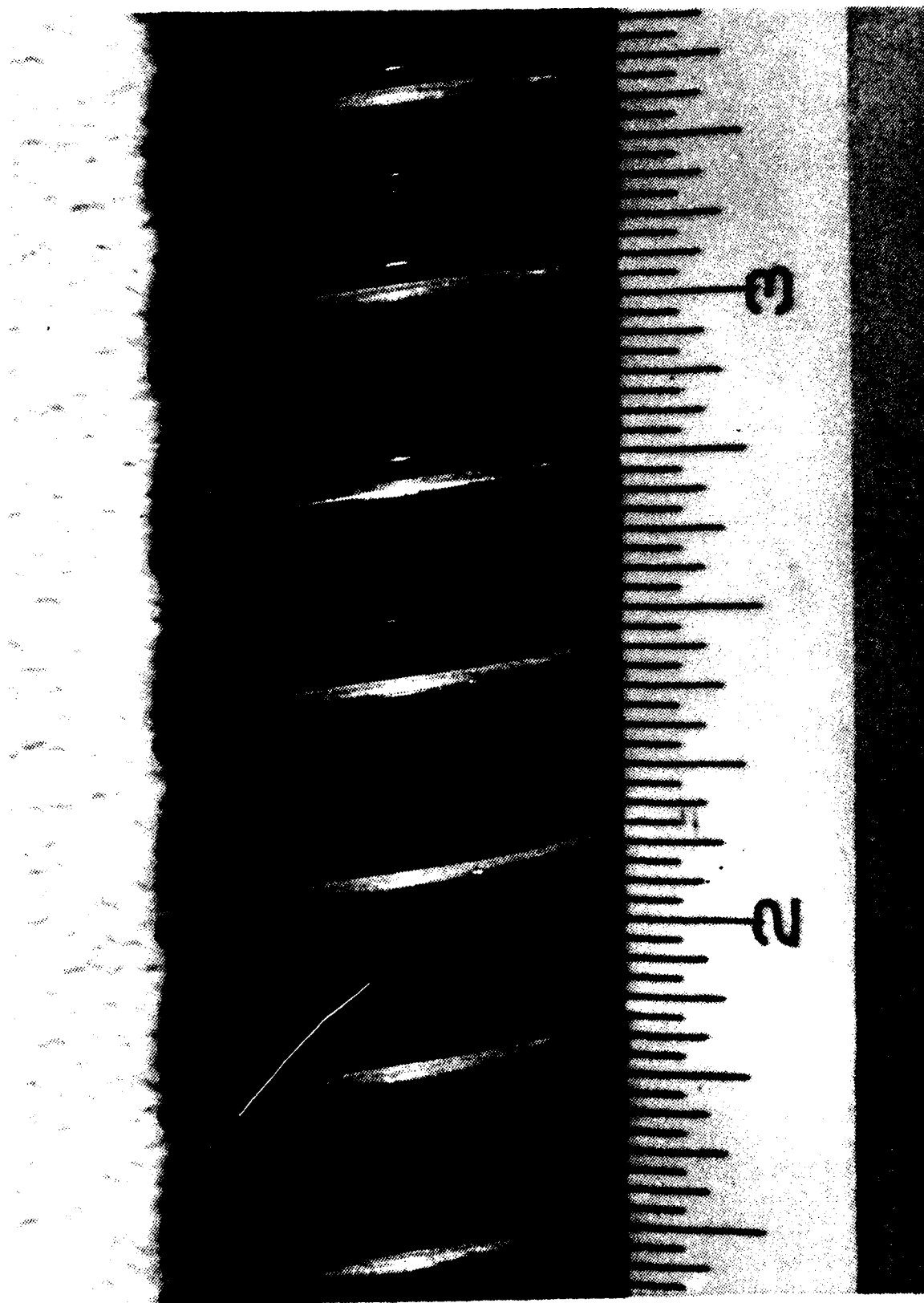


Figure 13. Photograph of Wire Wrapped Smooth Titanium Tube

simulate actual conditions in a condenser. Cooling water was not supplied to these tubes and they did not penetrate the test condenser end plates. Special characteristics of the Wolverine tubes are listed in Table I.

E. PERFORATED TUBE

The perforated tube water supply system is shown in Figure 14. This system consisted of a perforated tube (located above the uppermost active tube), a water heater which served as a supply tank, a rotameter in order to control the amount of water supplied to the perforated tube, a pump driven by a 1/2-HP electric motor, a condensate pump and associated piping system and valves. The active length of the copper-nickel, perforated tube was 305 mm, which was identical to the length of the test condenser. Supply water entered one end of this tube; the other end was sealed off.

The supply tank was unchanged from Noftz's initial design, except for the addition of a thermocouple in order to have a direct indication of the exact temperature of the water supplied to the perforated tube. The water, heated to the temperature of the condensate leaving the bottom tube, was fed to the perforated tube by a 1/2-HP, electric-motor-driven pump, via a rotameter and a recirculation valve. The flow rate to the perforated tube was controlled by using the rotameter and the valve provided in the heater-water recirculation line. Also, another modification was made to

TABLE I
DESIGN CHARACTERISTICS OF WOLVERINE
CORODENSE TUBES (TYPE LPD)

Outside diameter (in)	: 5/8
Wall thickness (in)	: 0.035
Outside diameter (ft)	: 0.0520
Heat Transfer Surface and Ratio	
a. Outside A_o^* (ft ² /ft)	: 0.163
b. Outside to inside A_o^*/A_i^*	: 1.141
Inside diameter D_i (ft)	: 0.0435
Cross section for flow A_{cs} (ft ²)	: 0.00163
Number of groove starts	: one ± 0
Pitch (in)	: 0.300
Groove radius (in)	: 1/32 nominal
Depth-transition length (next to plain section) (in)	: 1-3/4 max

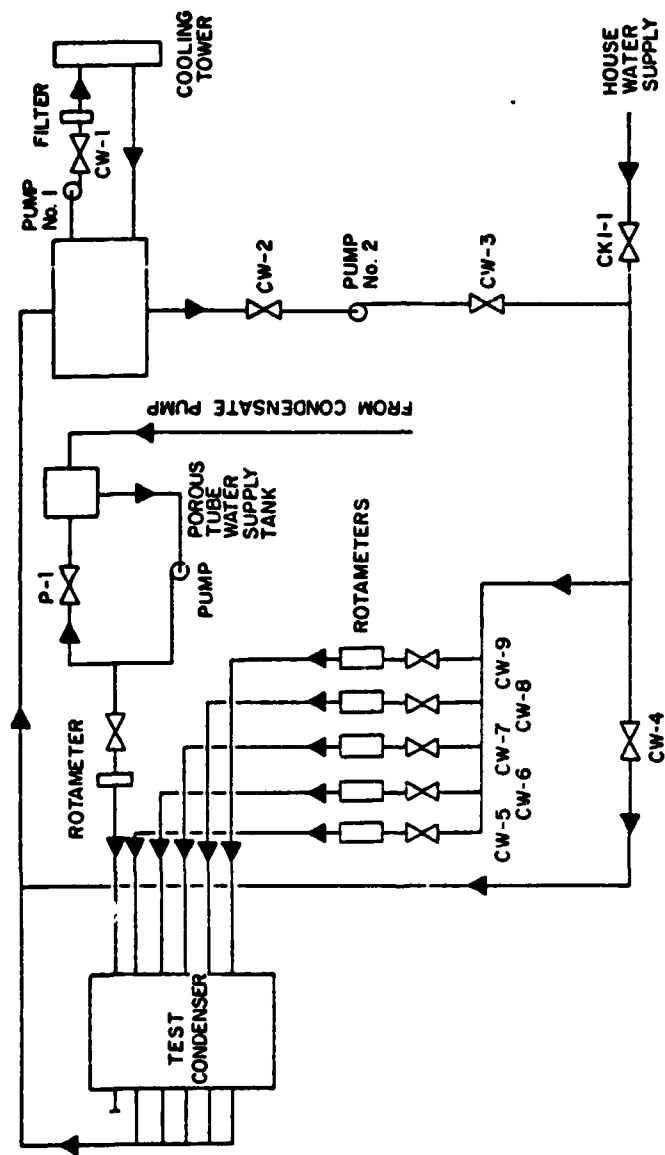


Figure 14. Schematic of Perforated Tube Water Supply System

feed the condensate collected in the hotwell, via the condensate pump, to the supply tank (heater) in order to facilitate the inundation of up to 30 tubes under atmospheric conditions.

The main feature of the perforated tube was to inundate with condensate from above, in order to simulate a tube column of more than five active tubes. When the wrapped Wolverine tubes and the smooth wrapped tubes were tested, the perforated tube was also wrapped with titanium wire, by using the above-mentioned technique.

F. CONDENSATE SYSTEM

The condensate system shown in Figure 15 was modified from Noftz's initial design, and consisted of the test condenser and hotwell, the condensate pump, piping and valves.

The test condenser hotwell collected the condensate produced by the test tubes. With valve C-1 closed, the condensate mass flow rate from the test condenser could be measured using a stop watch. Opening the valve, the condensate was fed, via the condensate pump, to the supply tank for the perforated tube, or dumped into the building's drainage system. The condensate line connecting the test condenser and the test condenser hotwell were insulated using rubber insulation.

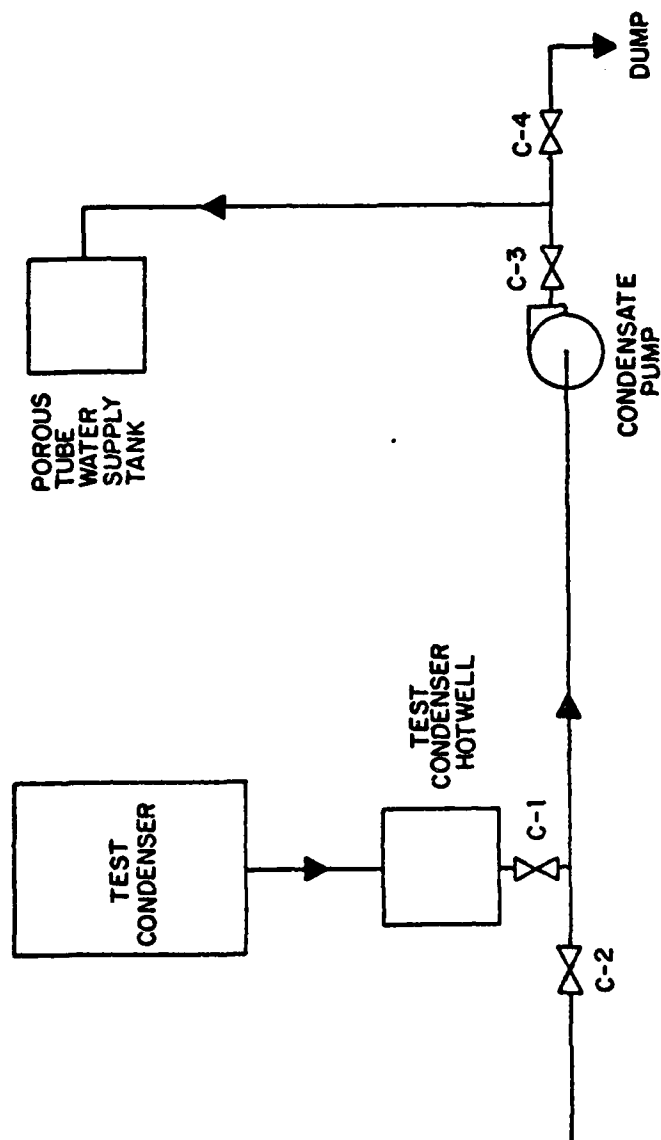


Figure 15. Schematic of Condensate System

G. COOLING-WATER SYSTEM

The cooling-water system was a partially-closed system as shown in Figure 14, and it was unchanged from Noftz's initial design. House water was used for the test facility. Cooling water was stored in a 1.2 meter, cubical, plexiglass supply tank, and was pumped by a 5-HP, electrically-driven pump via 51-mm-O.D. plastic piping to a manifold. Five rotameters were attached to the manifold to measure the flow rate through each active tube. Five regulating valves were used to obtain any water velocity between 0 and 5 m/s within the tubes. The cooling water passing through the rotameters was fed via 16 mm O.D. stainless steel tubing, and tygon tubing to the active tubes. The total tube lengths were long enough (over 2 meters) to ensure hydraulically fully-developed flow, and no swirling into the tubes. After flowing through the tubes, the cooling water was passed through mixing chambers, where the temperature profile was destroyed, to facilitate the steady measurement of the outlet cooling water temperatures. After the mixing chambers, the cooling water was collected in the supply tank. A 7.5-HP, electrically-driven pump was used to pump water from the supply tank, through a filter, to a cooling tower in order to minimize temperature rise at the water inlet.

The cooling tower was located outside the building and was composed of four truck radiators across which air was blown by means of a fan. The entire system, consisting of the heat exchanger and the fan was enclosed in a wooden structure with louvered openings to provide enough ventilation.

The tygon tubing was secured to the inlet and outlet sections of the active tubes by means of hose clamps. The outlet sections, including the mixing chambers, were insulated with rubber insulation.

H. INSTRUMENTATION

1. Flow Rates

a. Foulton rotameters were used to measure the flow rate of cooling water for each active tube. Starting from the top active tube to the bottom one, the rotameters were calibrated giving 100% maximum flow rates of 66.9 ± 1 , 72.6 ± 1 , 71.5 ± 1 , 72.8 ± 1 , and 73.3 ± 1 kg/min.

b. The perforated tube water rotameter was calibrated giving a 100 percent volumetric flow rate of 969 ± 30 ml/min.

c. All rotameters were calibrated using the procedures noted in Appendix A of Reference 7.

2. Temperature

Stainless-steel sheathed, copper-constantan thermocouples were used as the primary temperature monitoring devices. Three thermocouples were used to measure the inlet cooling water temperature; and twelve thermocouples were utilized to measure the outlet cooling water temperature.

Additionally, two thermocouples were used to measure the steam saturation temperature; one thermocouple was used to monitor the condensate temperature, and one was used to measure the vapor temperature. Table II lists the locations monitored.

A Gulton Industries, West 20, 0-500° F temperature controller was used to regulate the temperature of the perforated tube supply water. The controller had a manufacturer-stated accuracy of $\pm 0.5\%$ of the span (about $\pm 1.25^\circ$ F). It was impossible to obtain the precise temperature for the perforated tube water, because of the low response of the heating system.

During inundation runs, the condensate temperature rose with the increasing number of tubes. For example, the condensate temperature after the 5th tube was 94.6° C and it increased to 97.65° C after the 30th tube.

3. Pressure

Several different types of pressure measurement devices were used in this facility. They were: a Bourdon tube pressure gage which was used to measure the steam supply pressure, located downstream of the steam supply valve; an absolute pressure transducer which was used to measure the test condenser pressure; and, a compound pressure gage was also used to measure the pressure drop downstream of the Nupro valves. The pressure transducer was calibrated against a mercury manometer.

4. Data Collection and Display

A Hewlett-Packard HP-3054A Automatic Data Acquisition System, with a HP-9826 computer and a HP-2671G printer were used to record and display the thermocouple and pressure transducer readings. The temperatures were recorded in degrees Celsius, while the pressure was recorded in volts which was then converted, using the calibration curve, into mm Hg absolute. The pressure transducer was assigned to channel 019 of the HP Automatic Data Acquisition System, while the thermocouples were assigned channels as indicated in Table II.

TABLE II
CHANNEL NUMBERS FOR COPPER-CONSTANTAN THERMOCOUPLES

<u>Location</u>	<u>Channel</u>
T _{ci} #1	000
T _{ci} #3	001
T _{ci} #5	002
T _{co} #1	003
T _{co} #1	004
T _{co} #1	005
T _{co} #2	006
T _{co} #2	007
T _{co} #3	008
T _{co} #3	009
T _{co} #4	010
T _{co} #4	011
T _{co} #4	012
T _{co} #5	013
T _{co} #5	014
T _{sat}	015
T _{sat}	016
T _{cond}	017
T _{vap}	018

IV. PROCEDURES

A. INSTALLATION AND OPERATING PROCEDURES

1. Preparation of Condenser Tubes

Prior to installation, the titanium tubes were cleaned using a chemical cleaning method [Ref. 26]. The steps in this cleaning process are as follows:

- a. Swab the tube surface with acetone to remove grease.
- b. Using a test tube brush, brush the inside surface of the tube with a 50% sulfuric acid solution in order to remove any oxides. Also, apply this solution to the outside surface of the tube.
- c. Rinse the inside and outside of the tube with tap water.
- d. Apply, using a brush, a 50% solution of sodium hydroxide mixed with an equal amount of ethyl alcohol, at the boiling temperature (about 85° C) to the outside surface of the tube.
- e. Rinse the tube with tap water.
- f. Rinse thoroughly with distilled water.

Prior to any run, the condenser tubes had to be prepared to ensure filmwise condensation. Exterior and interior surfaces were cleaned to ensure proper wetting characteristics. Also, the tubes were cleaned by running steam at atmospheric pressure through the test condenser for about twenty minutes without the cooling water running through the tubes.

It was also found that, when drop-wise condensation occurred, rinsing the tubes using the perforated-tube water was sufficient to restore film-wise condensation.

2. System Operation and Steady-State Conditions

Complete operating instructions are listed in Appendix A. The steady-state condition was reached about three hours after initial system light-off, and about fifteen minutes after changes to the cooling water or perforated tube supply water flow.

When a steady-state condition was reached, the runs were made. The duration of each run was approximately one minute. For each condition, for example, for tubes 11 through 15, five consecutive runs were made and average values were computed.

The following data were taken automatically by the data-acquisition system:

- a. the thermocouple readings, and
- b. the pressure transducer reading.

Also, the following data were read into the computer, through the keyboard, for each run:

- a. the setting of each rotameter, and
- b. the test condenser hotwell levels.

B. DATA REDUCTION PROCEDURES

1. Overall Heat-Transfer Coefficient

The heat-transfer rate to the cooling water is given by:

$$Q = \dot{m} C_{pw} (T_{co} - T_{ci}) \quad (4.1)$$

The heat-transfer rate can also be found from the overall heat-transfer coefficient by:

$$Q = U_o A_o \cdot LMTD \quad (4.2)$$

where

$$LMTD = \frac{(T_{sat} - T_{ci}) - (T_s - T_{co})}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right)} \quad (4.3)$$

After combining equations (4.1), (4.2) and (4.3), it is found that

$$U_o = \frac{C_{pw}}{A_o} \ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right) \quad (4.4)$$

2. Inside Heat-Transfer Coefficient

The heat-transfer coefficient on the inside is calculated from the Sieder-Tate relationship as described in Holman [Ref. 27]:

$$N_u = \frac{h_i D_i}{k} = C_i R_e^{0.8} P_r^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (4.5)$$

In the above equation, C_i is referred to as the Sieder-Tate coefficient. The remainder of the right-hand side of the above equation,

$$(R_e^{0.8} \cdot P_r^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14})$$

is referred to as the Sieder-Tate parameter, X .

For thermally and hydrodynamically developed flow in tubes, the Sieder-Tate coefficient equals 0.027. However, when short tubes are considered, as in the case of this experiment, fully-developed conditions are not attained. Therefore, the value of C_i must be found experimentally.

The Wilson plot was used to arrive at the value of the Sieder-Tate coefficient. The Wilson plot was developed in 1915 [Ref. 28]. It is merely a plot of $1/U_o$ versus the inverse of the Sieder-Tate parameter (which is proportional to the inverse of cooling water velocity raised to the 0.8 power). The reasoning behind the Wilson plot is shown in the following development.

Consider the equation for the overall heat-transfer coefficient:

$$\frac{1}{U_o} = \frac{A_o}{A_i h_i} + R_w + \frac{1}{h_o} \quad (4.6)$$

or

$$\frac{1}{U_o} = \frac{A_o}{A_i} \frac{D_i}{C_i k X} + R_w + \frac{1}{h_o} \quad (4.7)$$

where

$$R_w = \frac{D_o \ln\left(\frac{D_o}{D_i}\right)}{2k_m}$$

For the Wilson-plot method to be successful, h_o must be kept constant. When steam condenses on the shell side, the above condition can be achieved only if the heat flux (q'') is kept constant.

As required for the Wilson plot, when the cooling water velocity increases, h_i increases, U_o increases, and finally q'' increases.

Consider

$$q'' = U_o \text{LMTD} \quad (4.8)$$

In order to keep q'' constant, one must therefore decrease LMTD by lowering the steam saturation temperature. This requires trial-and-error setting of condenser pressure which is a very difficult task.

To avoid this difficulty and yet arrive at a Sieder-Tate constant not affected by the varying q'' , a modified Wilson-plot method was used. This method was developed by Wanniarachchi [Ref. 29], and the required steps are listed below:

1. Assume $C_i = 0.027$

2. Calculate

$$LMTD = \frac{T_{co} - T_{ci}}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right)}$$

3. Calculate

$$Q = \dot{m} C_p (T_{co} - T_{ci})$$

4. Calculate

$$U_o = \frac{Q}{A_o \cdot LMTD}$$

5A. For the first data point only, do the following:

a. Assume

$$C_f = \left(\frac{\mu}{\mu_u} \right)^{0.14} = 1$$

b. Calculate the Sieder-Tate parameter

$$X = R_e^{0.8} P_r^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

c. Calculate

$$h_i = \frac{k}{D_i} C_i R_e^{0.8} P_r^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

d. Calculate

$$T_w = T_c + \frac{q''}{h_i}$$

e. Calculate

$$C_f = \left(\frac{\mu}{\mu_w}\right)^{0.14}$$

f. Repeat steps a through e until C_f assumed in step a approximately equals C_f calculated in step e.

g. Calculate

$$\frac{1}{h_o} = \frac{1}{U_o} - \frac{1}{h_i} \frac{A_o}{A_i} - R_w \quad (4.9)$$

h. Assign $Q_1 = Q$

NOTE: The second subscript of h on the left-hand side of equation (4.9) refers to the first data point.

5B. For all data points ~~except~~ for first data point, do the following:

a. Calculate

$$\frac{1}{h_i} = \left[\frac{1}{U_o} - R_w - \frac{1}{h_{o,N}} \right] \frac{A_i}{A_o}$$

where

$$\frac{1}{h_{o,N}} = \frac{1}{h_{o,1}} - \left(\frac{Q}{Q_1}\right)^{1/3}$$

b. Calculate

$$T_w = T_c + \frac{q''}{h_i}$$

c. Calculate the Sieder-Tate parameter

$$X = R_e^{0.8} P_r^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

6. Repeat steps 2 through 5 for all data points.

7. Plot $\frac{1}{h_i}$ versus $\frac{1}{X}$

NOTE: It is more reasonable to plot h_i versus X . However, $1/h_i$ versus $1/X$ was plotted in order to be consistent with the original Wilson-plot method

$$\left(\frac{1}{U_o} \text{ versus } \frac{1}{X} \right) .$$

8. Obtain the slope (m) . of the least-squares-fit, straight line.

9. Calculate the Sieder-Tate constant.

$$C_i = \frac{D_i}{k_w \cdot m}$$

10. Repeat steps 2 through 9 until the assumed and calculated C_i values are approximately equal.

A listing of the computer program used in this method can be found in Appendix D.

3. Outside Heat-Transfer Coefficient

The heat-transfer coefficient on the outside is calculated using the following steps:

1. Calculate the average bulk water temperature.

$$T_b = (T_{co} + T_{ci})/2$$

2. Evaluate the thermophysical properties based on the average bulk water temperature.
3. Calculate the cooling water velocity.

$$V_w = \frac{\dot{m}}{\rho A_i}$$

4. Calculate the water-side Reynolds number.

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

5. Calculate the heat transferred to the cooling water.

$$Q = \dot{m} (T_{co} - T_{ci}) C_{pw}$$

6. Calculate the heat flux.

$$q'' = \frac{Q}{\pi \cdot D_o \cdot L}$$

7. Calculate the Nusselt coefficient using the formula:

$$h_{Nu} = 0.651 \left[\frac{k_f^3 \cdot \rho_f^2 \cdot h_{fg} \cdot g}{\mu_f \cdot D_o \cdot q''} \right]^{1/3} \quad (4.10)$$

NOTE: The above formula was employed since no direct measurement of tube wall temperature was made.

To compare the condensate film temperature, as required for equation (4.9), an iterative scheme was used as outlined below:

- a. Assume $T_f = T_{sat}$.
- b. Evaluate the relevant thermophysical properties, which are included in equation (4.10).
- c. Calculate the Nusselt coefficient using equation (4.10).
- d. Evaluate $T_{f,c}$ using the formula:

$$T_{f,c} = T_{sat} - \frac{q''}{h_{Nu}} + 0.5$$

- e. Repeat steps a through d until T_f assumed in step a approximately equals $T_{f,c}$ calculated in step d.
8. Calculate the inside heat-transfer coefficient using the formula:

$$h_i = \frac{k_w}{D_i} C_i Re^{0.8} Pr^{1/3} C_f \quad (4.11)$$

NOTE: In order to determine the C_f , the average inner wall temperature must be known. As noted earlier, this temperature is not known directly and it must be found iteratively, as described below:

- a. Assume a correction factor C_f (say 1).
- b. Calculate the h_i using equation (4.11).
- c. Calculate the tube wall temperature using the formula:

$$T_w = T_b + \frac{q''}{h_i} \left(\frac{D_i}{D_o} \right)$$

- d. Calculate a new correction factor, at the evaluated tube wall temperature

$$C_f = \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

- e. Repeat steps a through d until C_f assumed in step a approximately equals C_f calculated in step d.
9. Calculate the log-mean-temperature difference, (LMTD).

$$LMTD = \frac{T_{co} - T_{ci}}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{c.}} \right)} \quad (4.12)$$

10. Calculate the overall heat-transfer coefficient using the formula:

$$U_o = \frac{q''}{LMTD} \quad (4.13)$$

11. Calculate the outside heat-transfer coefficient using the equation:

$$h = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w} \quad (4.14)$$

NOTE: The validity of equation (4.13) is based on the assumption of negligible water-side fouling resistance and the resistance due to noncondensables.

12. Calculate the normalized, local, outside heat-transfer coefficient, h_N/h_1 .
13. Calculate the normalized average, local outside heat-transfer coefficient (h_N/h_1).

C. DATA-REDUCTION PROGRAM

A computer program was utilized to analyze the raw data. The program was in BASIC language and was run on an HP-9826 computer system. A peripheral plotter was used to plot the results. The program is presented in Appendix D.

V. RESULTS AND DISCUSSION

A. SIEDER-TATE COEFFICIENTS FOR SMOOTH AND ROPED TUBES

The modified Wilson Plots for smooth and roped tubes are shown in Figures 16 and 17. Of particular interest is the very small intercept, which represents the estimated fouling factor. For example, the estimated fouling factor for data run STSD-11 (for smooth tubes) is $1.37 \times 10^{-6} \text{ m}^2 \cdot \text{K/W}$ based on the inside area. This value is only three percent of the typical value ($4.4 \times 10^{-5} \text{ m}^2 \cdot \text{K/W}$) used for design purposes. This very small value supports the initial assumption of negligible fouling factor for the success of the modified Wilson-Plot method.

Table III is a summary of the Sieder-Tate coefficients calculated for smooth and roped tubes. The average Sieder-Tate coefficient was calculated to be 0.029 ± 0.001 for smooth tubes and 0.061 ± 0.002 for roped tubes. Thus, the roped tubes have a Sieder-Tate coefficient 2.1 times greater than that for the smooth tubes. This increase is mainly due to the increased surface area, turbulence and swirl effects.

The Sieder-Tate coefficient derived for the smooth tubes is 7.4 percent greater than the value (0.027) published in the original, generalized correlation. This increase can be easily explained by the short condensing tubes used in this

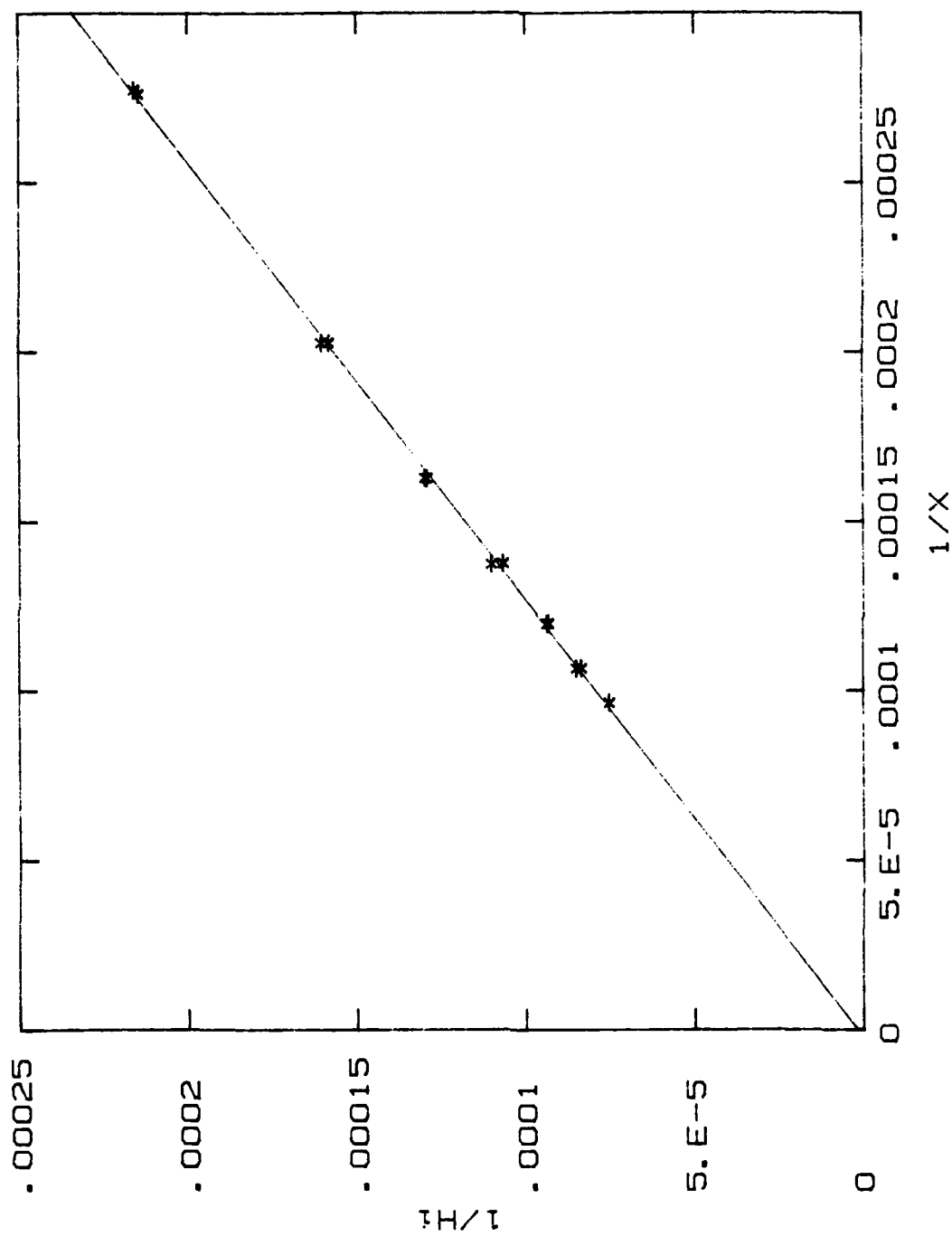


Figure 16. Modified Wilson-Plot for Smooth Tubes (Run STSD-11)

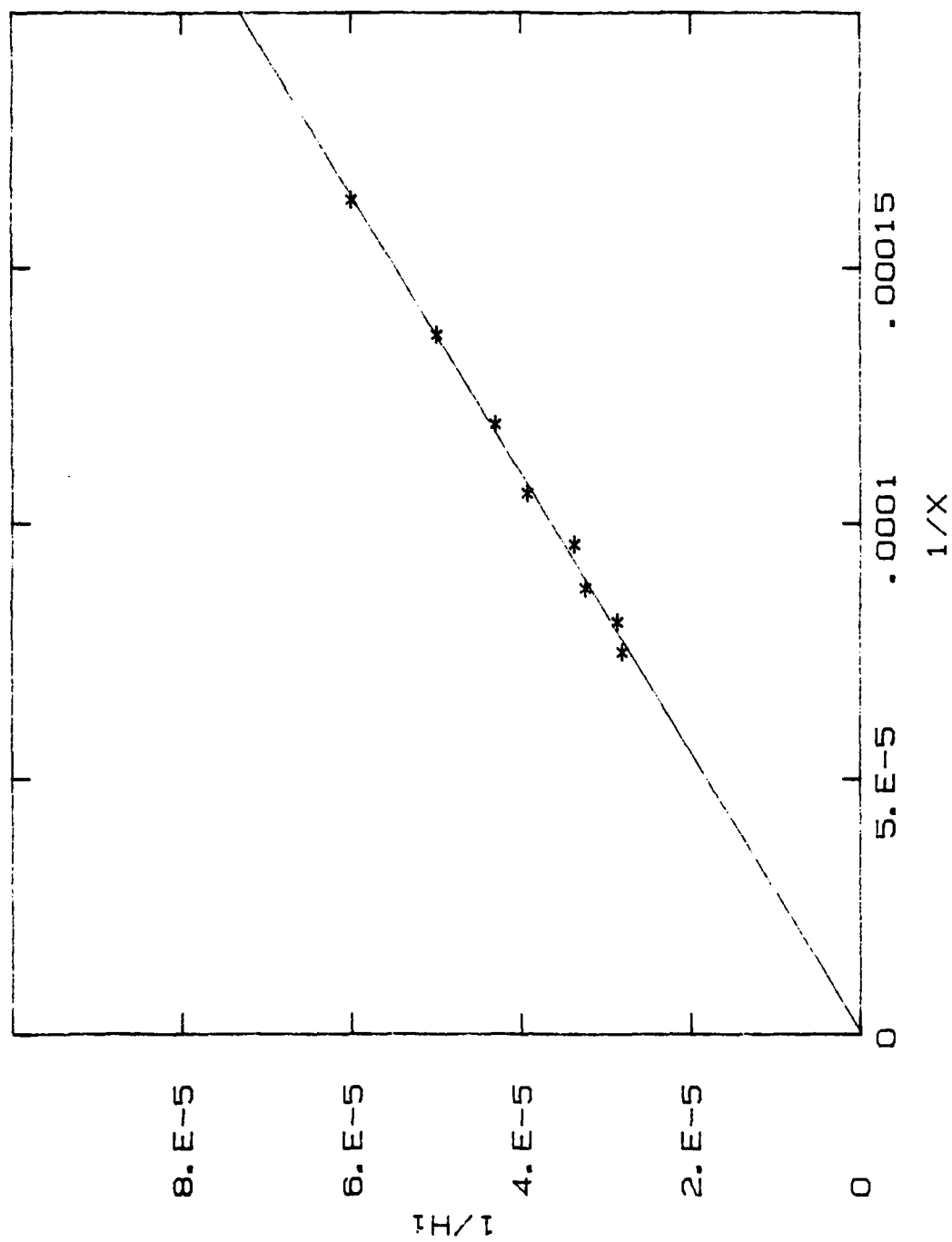


Figure 17. Modified Wilson-Plot for Roped Tubes (Run RTSD-11)

TABLE III

SUMMARY OF SIEDER-TATE COEFFICIENTS
FOR SMOOTH AND ROPED TUBES

<u>File Name</u>	<u>Tube Type</u>	<u>Sieder-Tate Coefficient</u>
STSD-2	Smooth	0.0287
STSD-10	Smooth	0.0291
STSD-11	Smooth	0.0296
RTSD-3	Roped	0.0591
RTSD-4	Roped	0.0589
RTSD-6	Roped	0.0621
RTSD-7	Roped	0.0627

experiment. Even though the flow condition was hydrodynamically fully-developed, it was thermally developing throughout the condensing length, resulting in a greater Sieder-Tate coefficient.

It is worth noting that the Sieder-Tate coefficient derived by using the original Wilson-Plot method (i.e., plotting $1/U_o$ versus $1/X$), consistently gave values about 10% greater than the values obtained using the modified Wilson-Plot method.

B. SMOOTH TUBES

Figure 18 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie about 40 percent above the curve predicted by the Nusselt theory. This close agreement was considered to be an indication of the proper operation of the test apparatus and the data reduction procedures.

As discussed earlier in Chapter II, the Nusselt theory for a tube bundle is based on a number of basic assumptions. For example, in reality, the condensate does not fall as a continuous sheet as assumed for the Nusselt theory, but it forms drops before leaving the tube. This phenomenon has a tendency to result in a larger condensing coefficient than that predicted by the Nusselt theory.

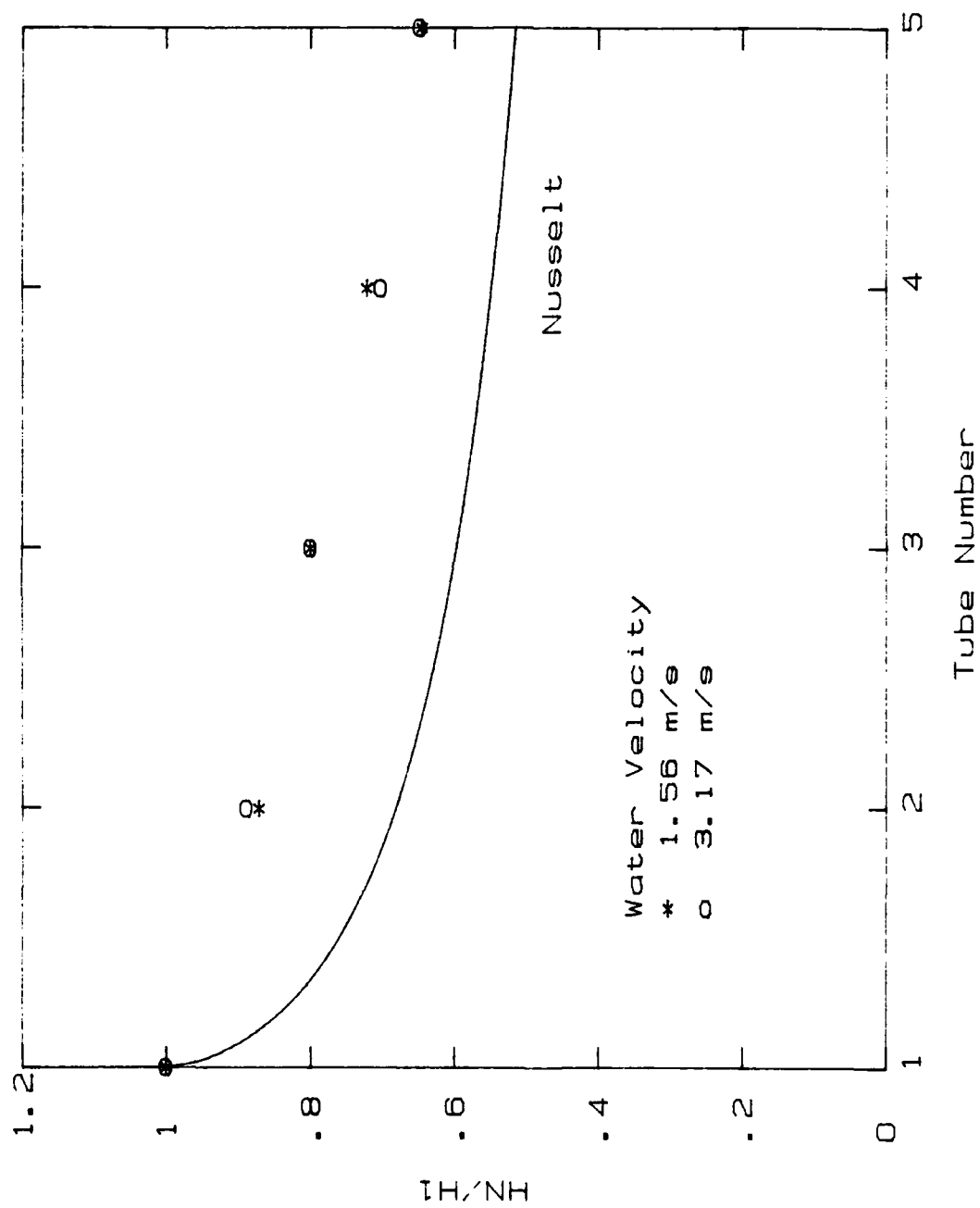


Figure 18. Variation of Local Condensing Coefficient with Tube Number (Run STNWN1-1)

Figure 19 shows the normalized, average condensing coefficient for five tubes. These data points show a much smoother trend than that for local values. Further, these data points are closer to the Eissenberg correlation than the Nusselt prediction.

Figure 20 shows the variation of the normalized, local, condensing coefficient for 30 tubes. The data points lie up to 35 percent above the curve predicted by the Nusselt theory. The outside heat-transfer coefficient for the first tube was ten percent smaller than the value predicted by the Nusselt theory.

Figure 21 displays the normalized, average condensing coefficient for 30 tubes. Again, these data points show a smoother trend than that for local values. These points lie between the Eissenberg and Nusselt predictions.

Figure 22 shows the least-squares-fit curve for the data points for smooth tubes. The exponent derived is -0.154 , and it is in close agreement with the value of -0.14 derived by Noftz [Ref. 12].

C. ROPED TUBES

Figure 23 displays the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 15 percent above the curve representing the Nusselt theory.

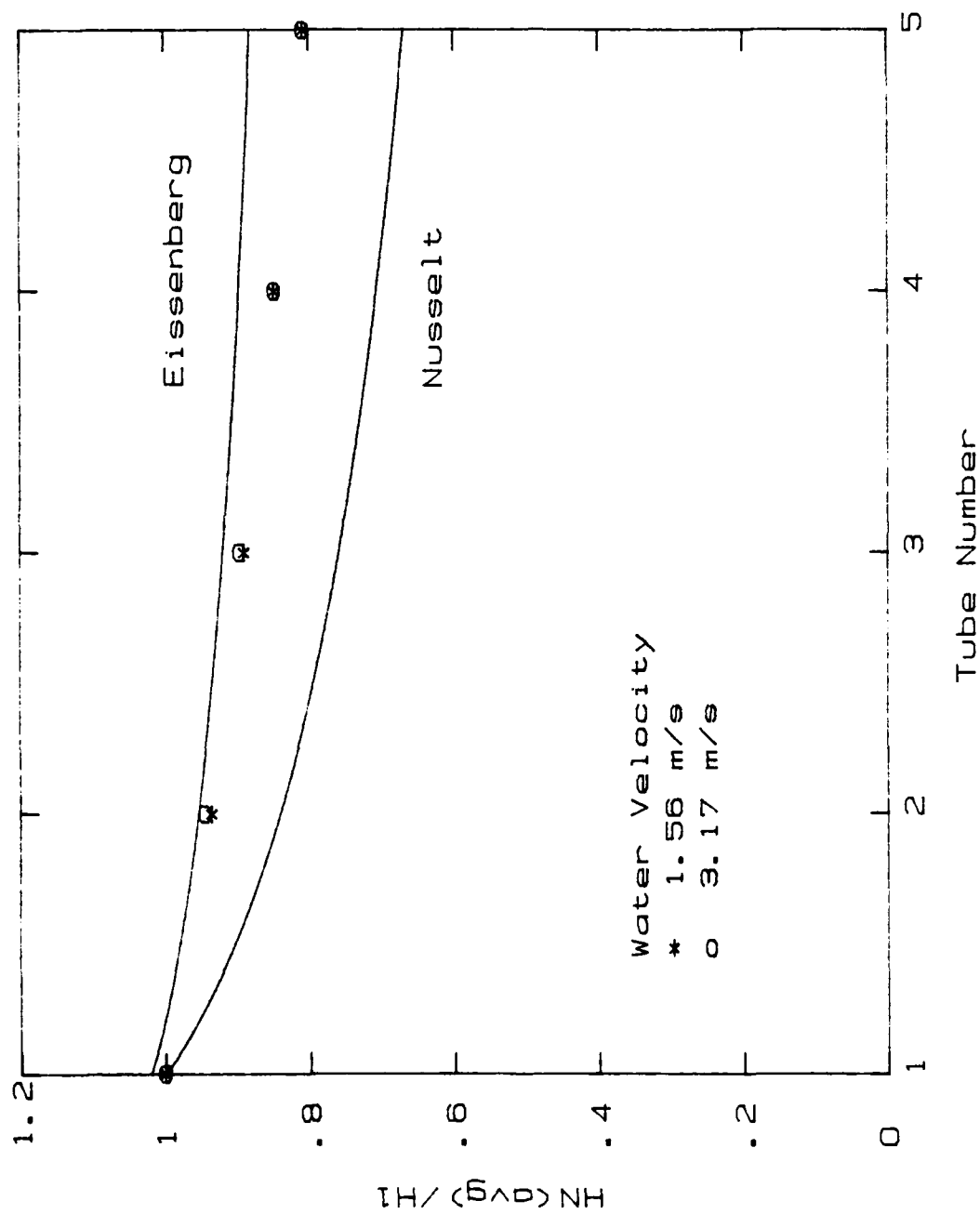


Figure 19. Variation of Average Condensing Coefficient with Tube Number (Run STNWN1-1)

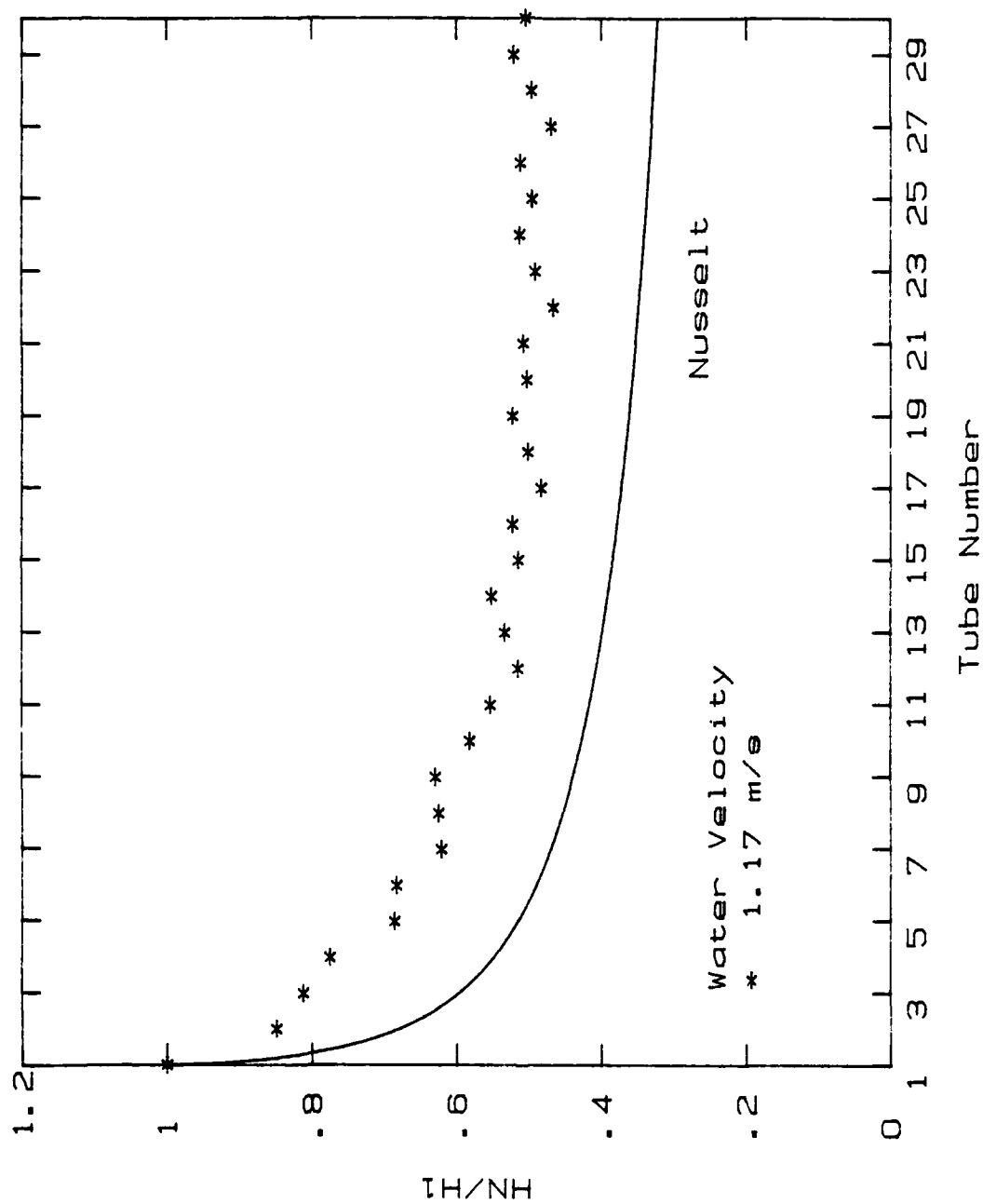


Figure 20. Variation of Local Condensing Coefficient with Tube Number (Run STNWI-1)

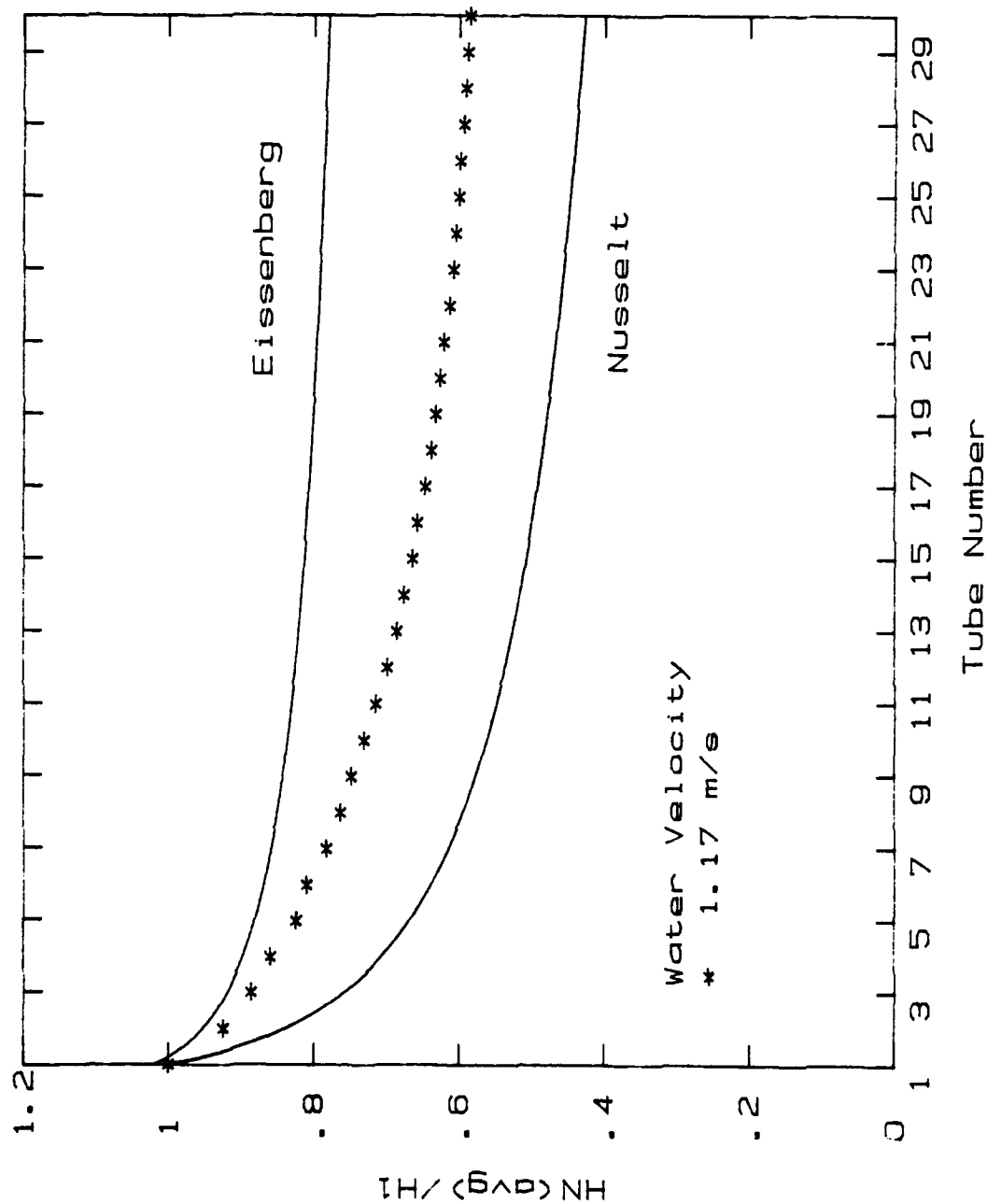


Figure 21. Variation of Average Condensing Coefficient with Tube Number (Run STNWI-1)

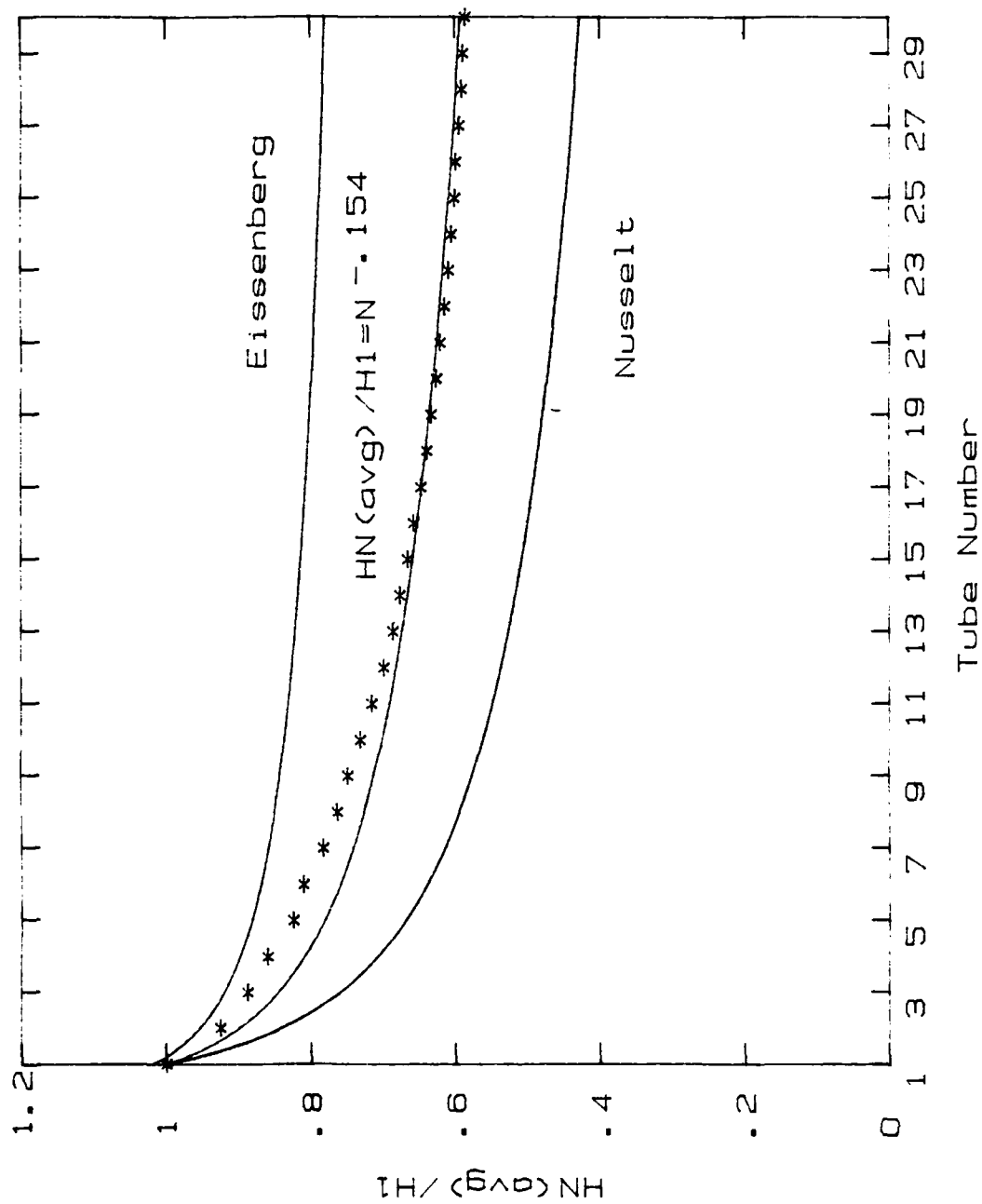


Figure 22. Least-Squares-Curve Fit for Data (Run STNW-1)

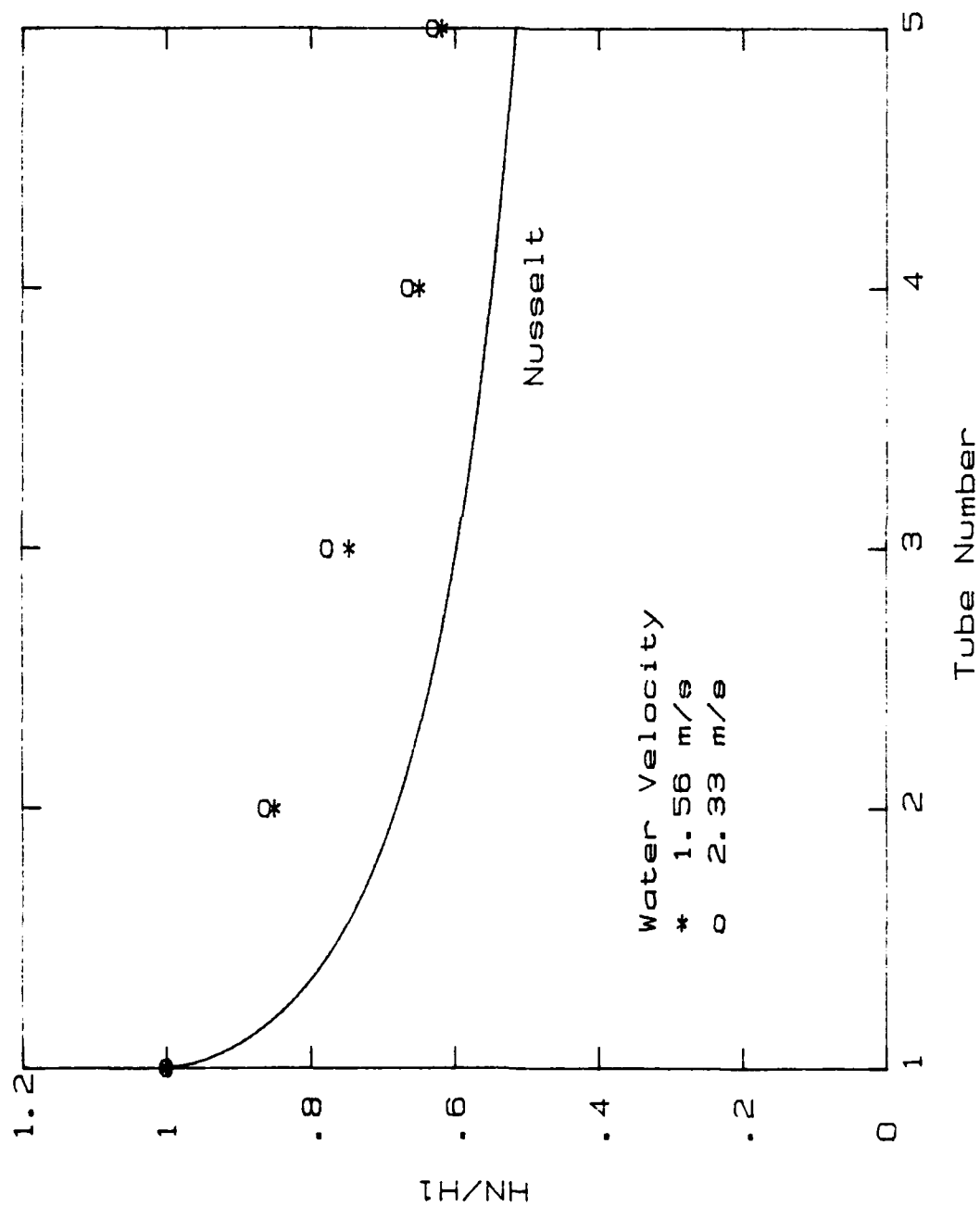


Figure 23. Variation of Local Condensing Coefficient with Tube Number
(Run RTNWN1-3)

Figure 24 shows the normalized, average, condensing coefficient for five tubes. These points show a trend very close to the Eissenberg relationship.

Figure 25 represents the plot of the normalized, local, condensing coefficient under inundation conditions up to 30 tubes.

Figure 26 shows the normalized, average, condensing coefficient for 30 tubes. These data points lie about midway between the Eissenberg and Nusselt predictions.

Figure 27 shows the least-squares-fit curve for the data points for roped tubes. The exponent calculated is -0.183 . Since the interest is in the large tube bundles, the curve fit was generated only for tubes 11 through 30. This is evident from the figure as the curve fit shows poor agreement when compared to the data points for the first ten tubes.

D. ROPED TUBES WRAPPED WITH WIRE

The outside, heat-transfer coefficient for the first tube was 11.3 percent greater than the value predicted by the Nusselt theory. This increase is in agreement with the manufacturer's claim that the special geometry of the roped tubes has a thinning effect on the condensate film, resulting in a larger condensing coefficient.

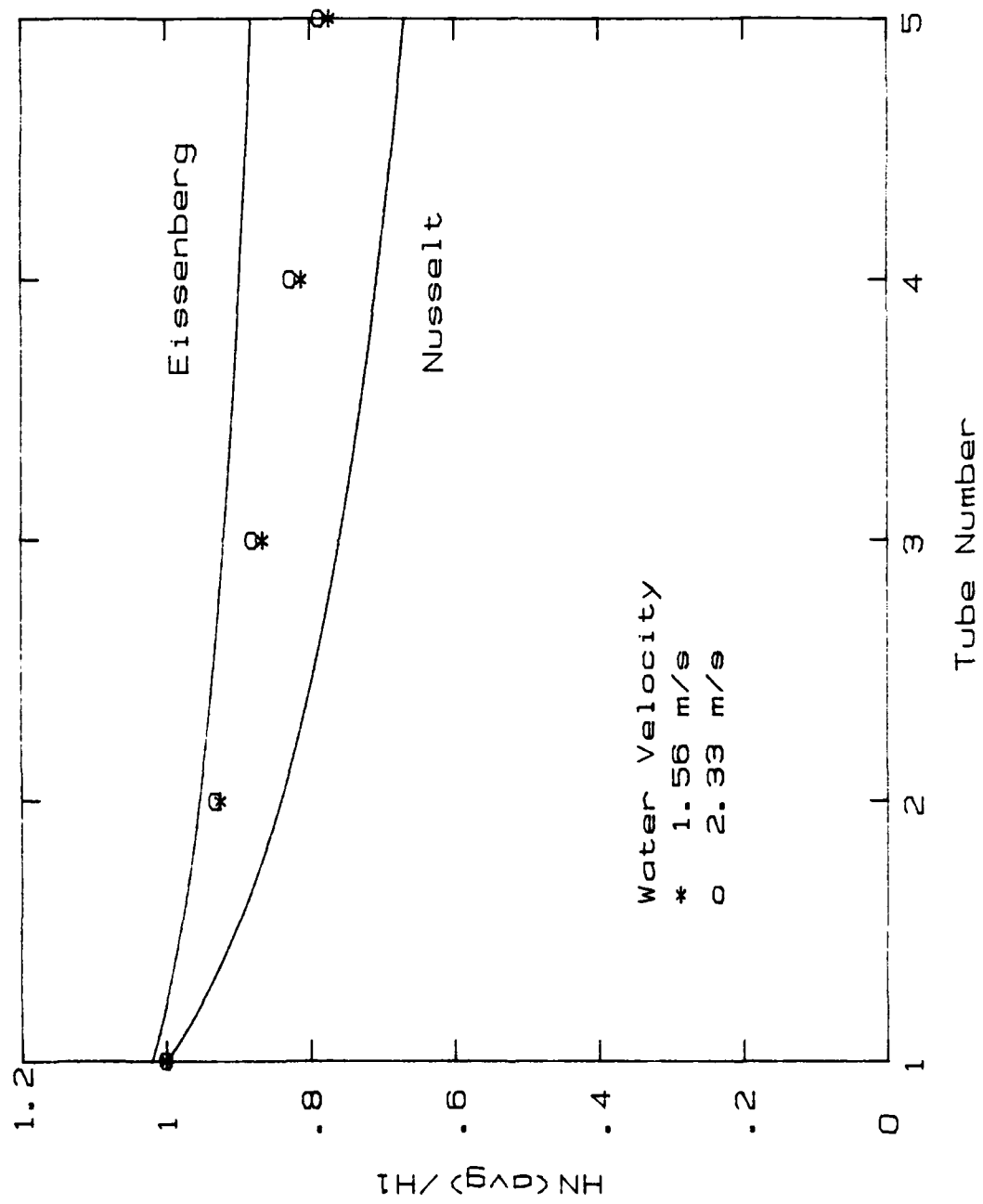


Figure 24. Variation of Average Condensing Coefficient with Tube Number (Run RTNWNII-3)

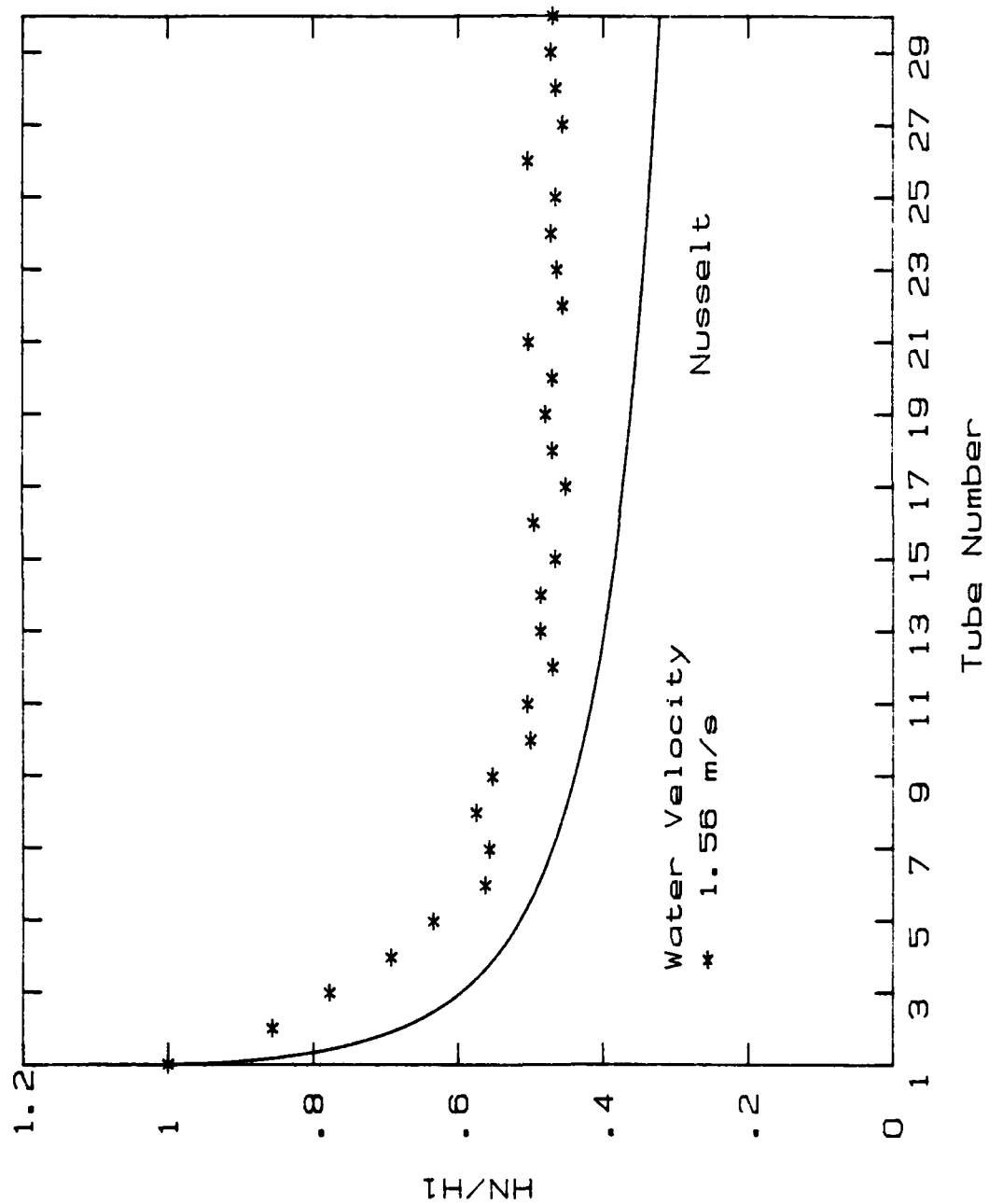


Figure 25. Variation of Local Condensing Coefficient with Tube Number (Run RTNWN1-3)

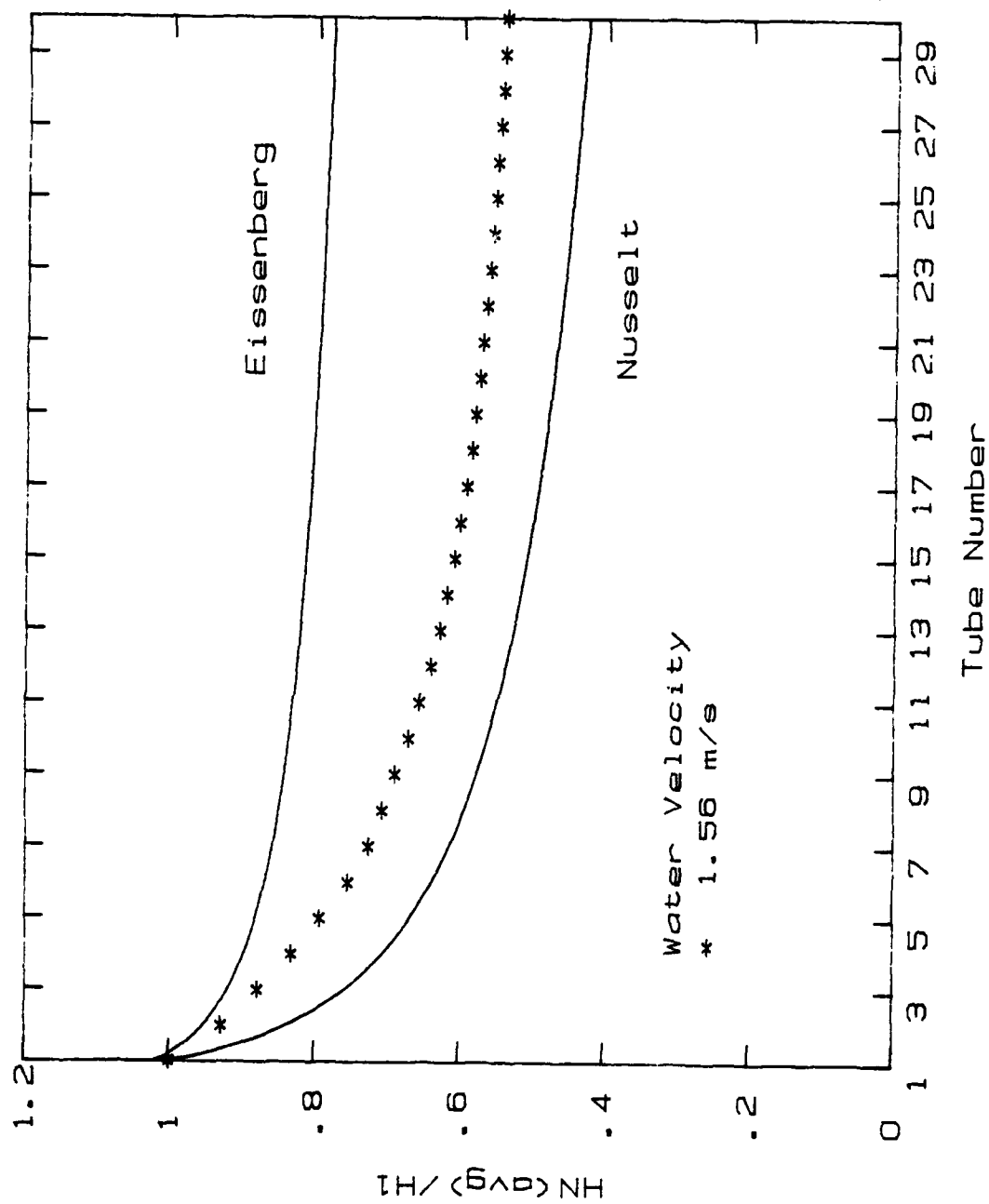


Figure 26. Variation of Average Condensing Coefficient with Tube Number (Run RTNWI-1)

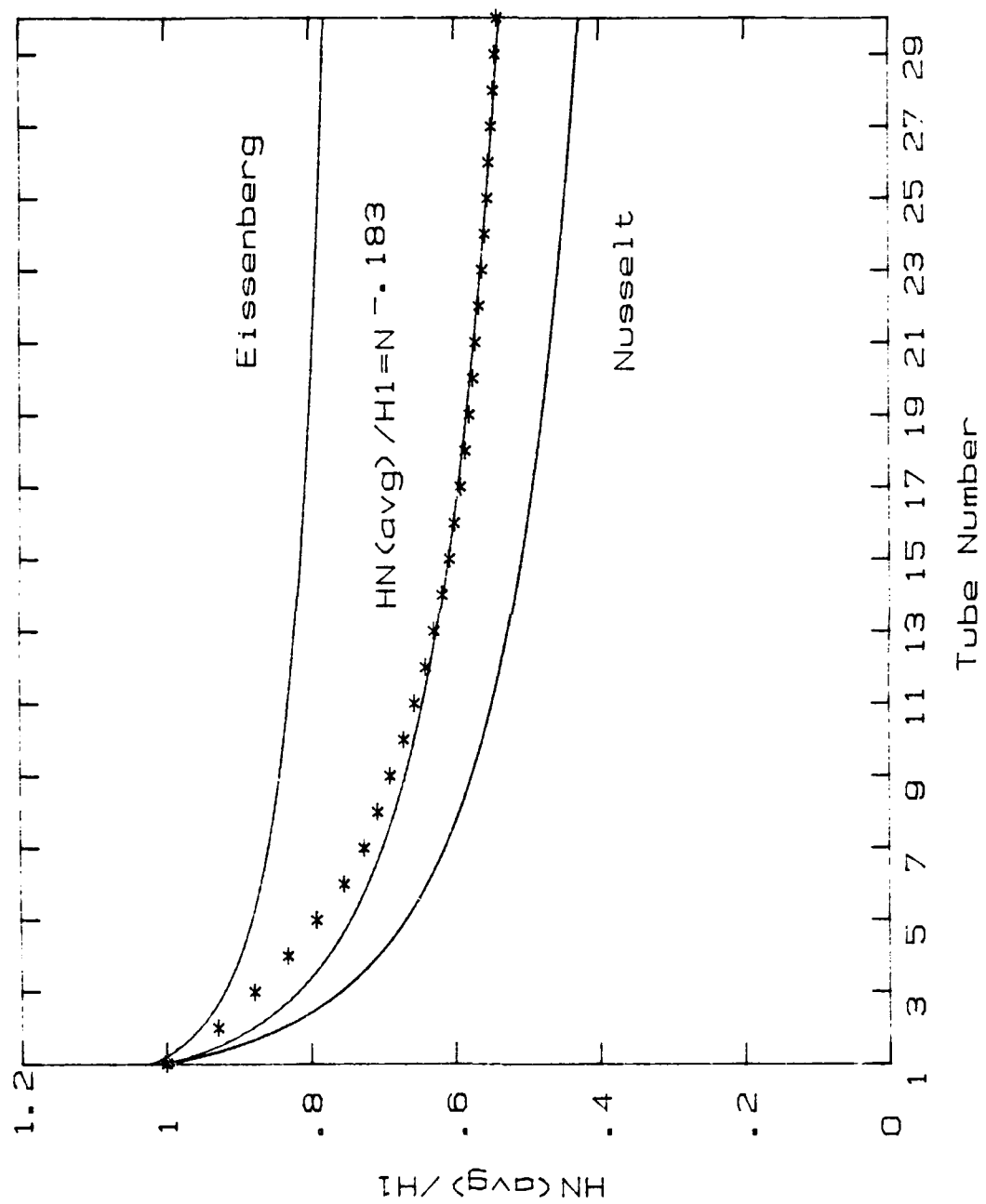


Figure 27. Least-Squares-Curve Fit for Data (Run RTNWI-1)

Figure 28 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 63 percent above the curve predicted by the Nusselt theory.

Figure 29 displays the normalized, average, condensing coefficient. The data points are well above the Eissenberg correlation.

Figure 30 shows the normalized, local, condensing coefficient for a bundle of 30 tubes. The data points are scattered within the limits of uncertainty, as predicted by the error analysis (see Appendix C).

Figure 31 shows the variation of the data points representing the normalized, average, condensing coefficient under inundation up to 30 tubes. The data points lie up to 100 percent above the curve predicted by the Nusselt theory.

Figure 32 represents the least-squares-fit curve, which has a derived exponent of -0.039 . The curve fit is in good agreement with all the data points.

E. SMOOTH TUBES WRAPPED WITH WIRE

The outside heat-transfer coefficient for the first tube was up to six percent greater than the value predicted by the Nusselt theory. This increase is caused by the thinning effect on the condensate film, resulting from the surface-tension forces acting toward the wrapped wire.

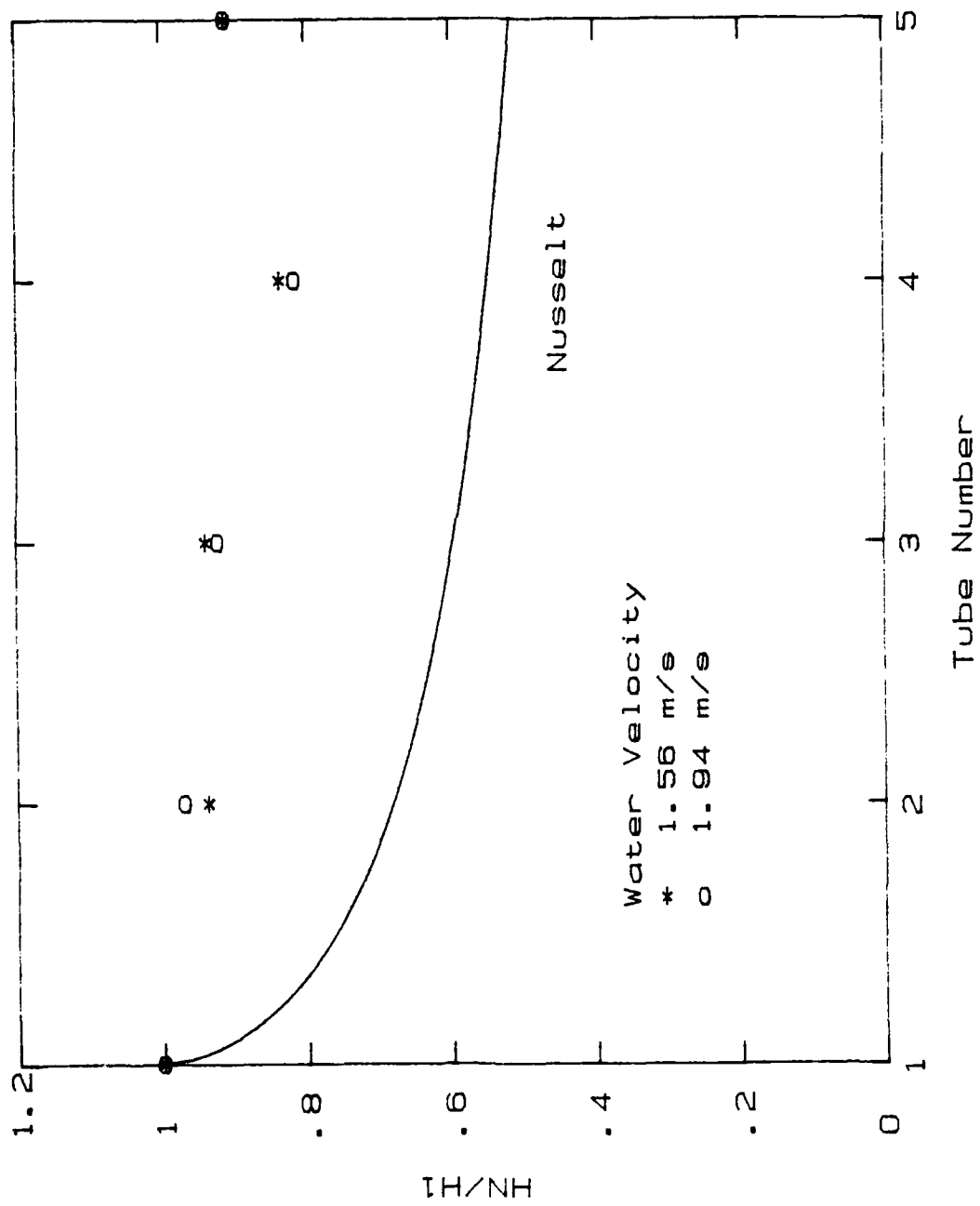


Figure 28. Variation of Local Condensing Coefficient with Tube Number
(Run RTWNI-3)

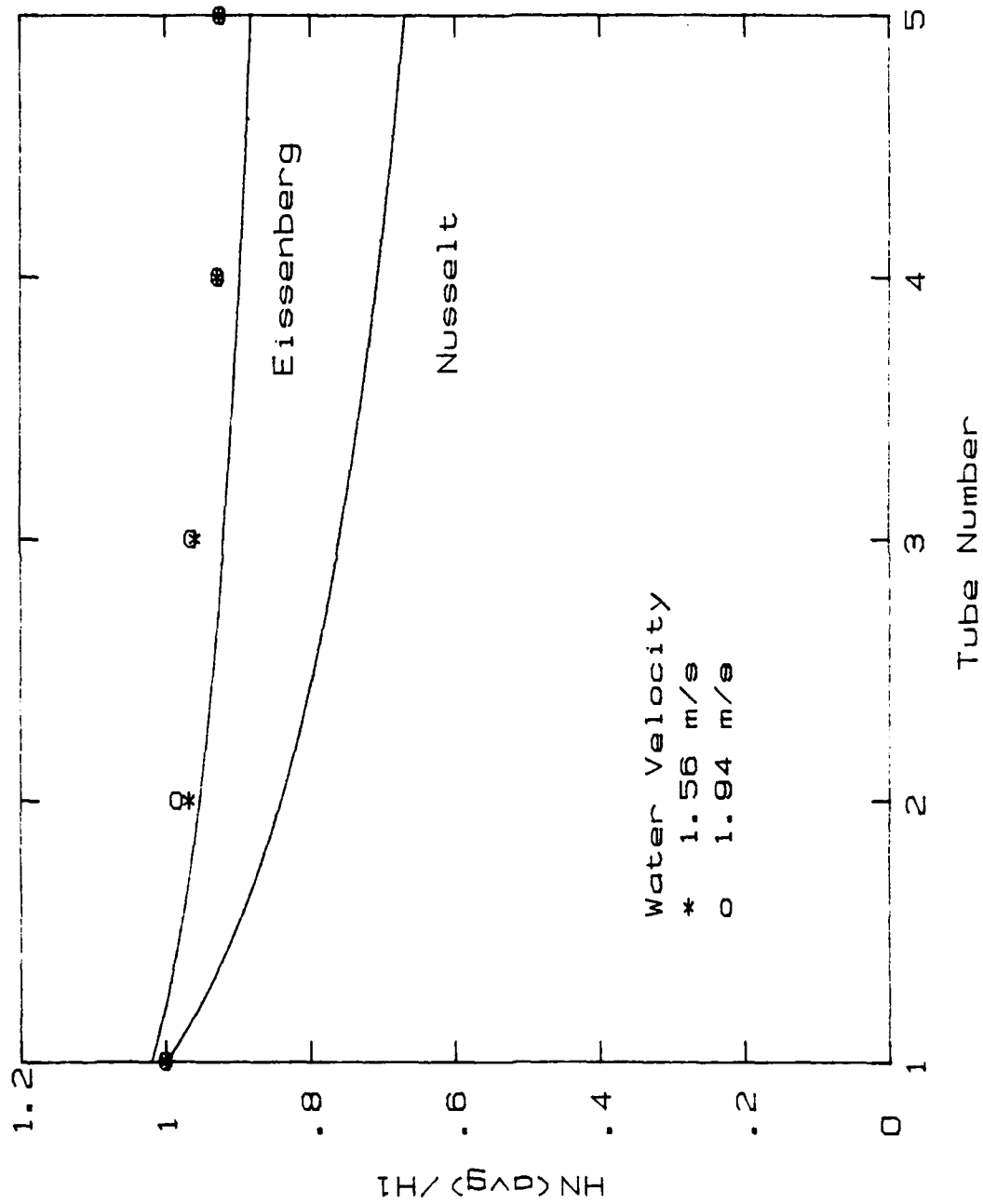


Figure 29. Variation of Average Condensing Coefficient with Tube Number (Run RTWNI-3)

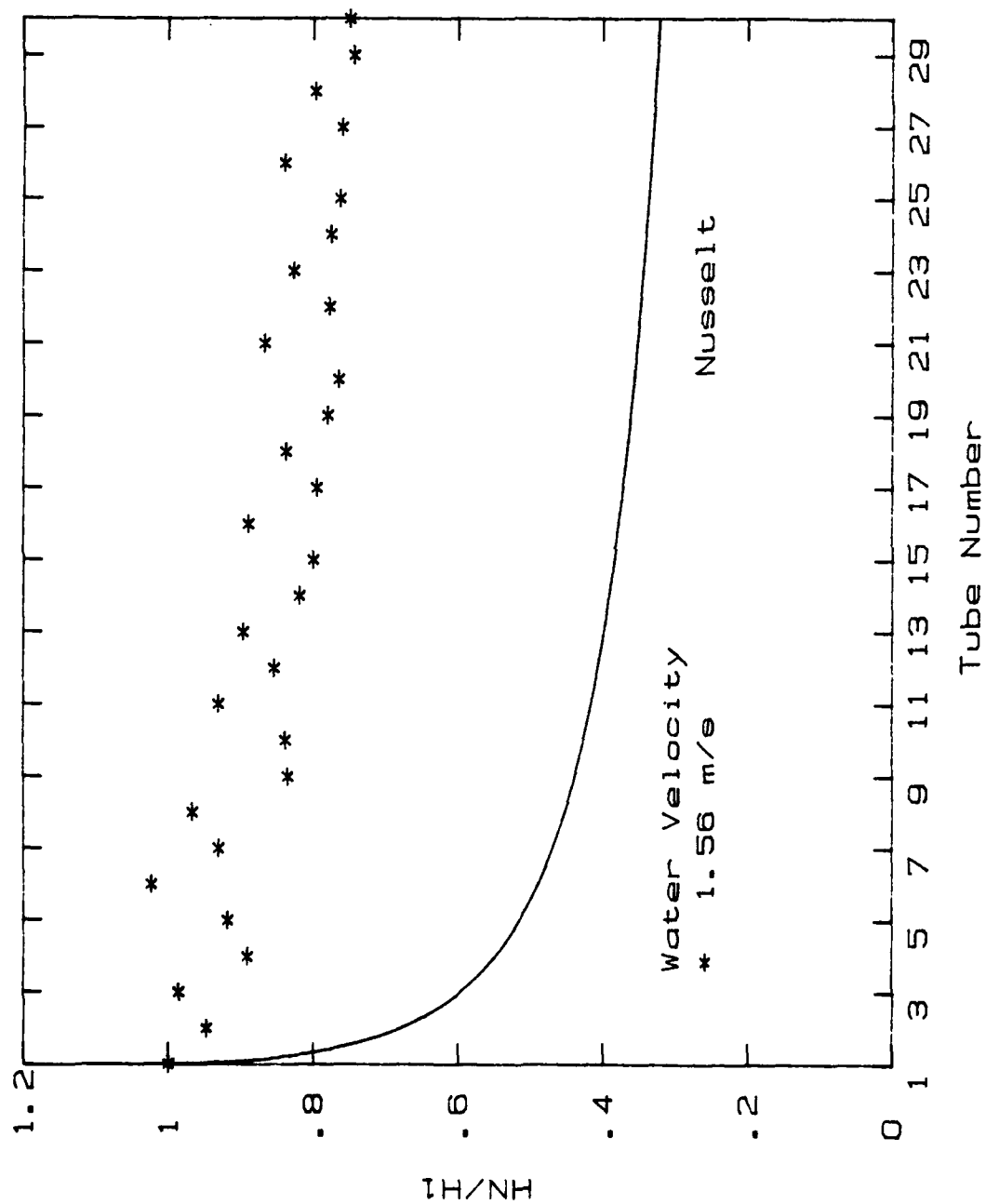


Figure 30. Variation of Local Condensing Coefficient with Tube Number
(Run RTWI-3)

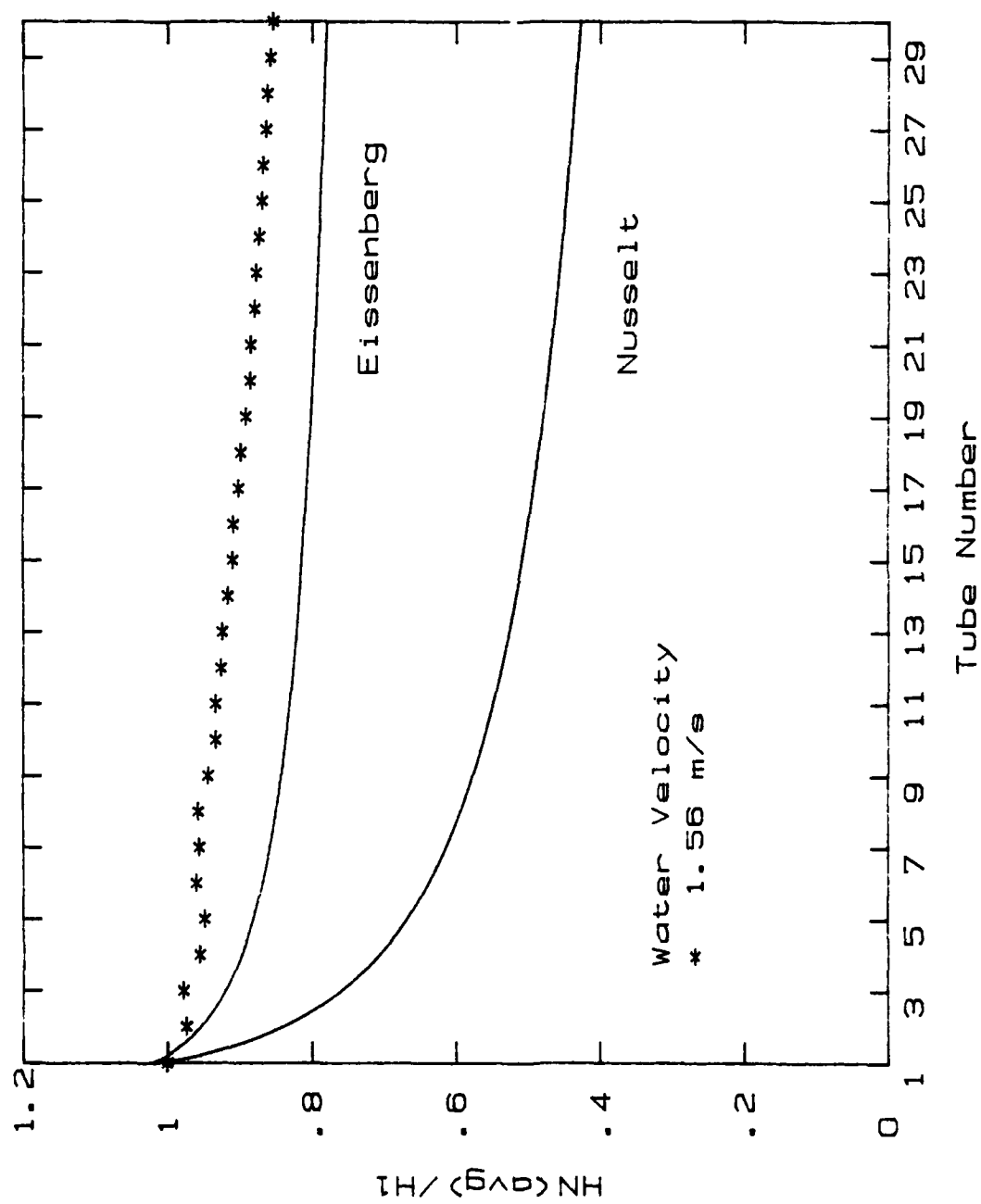


Figure 31. Variation of Average Condensing Coefficient with Tube Number (Run RTWI-1)

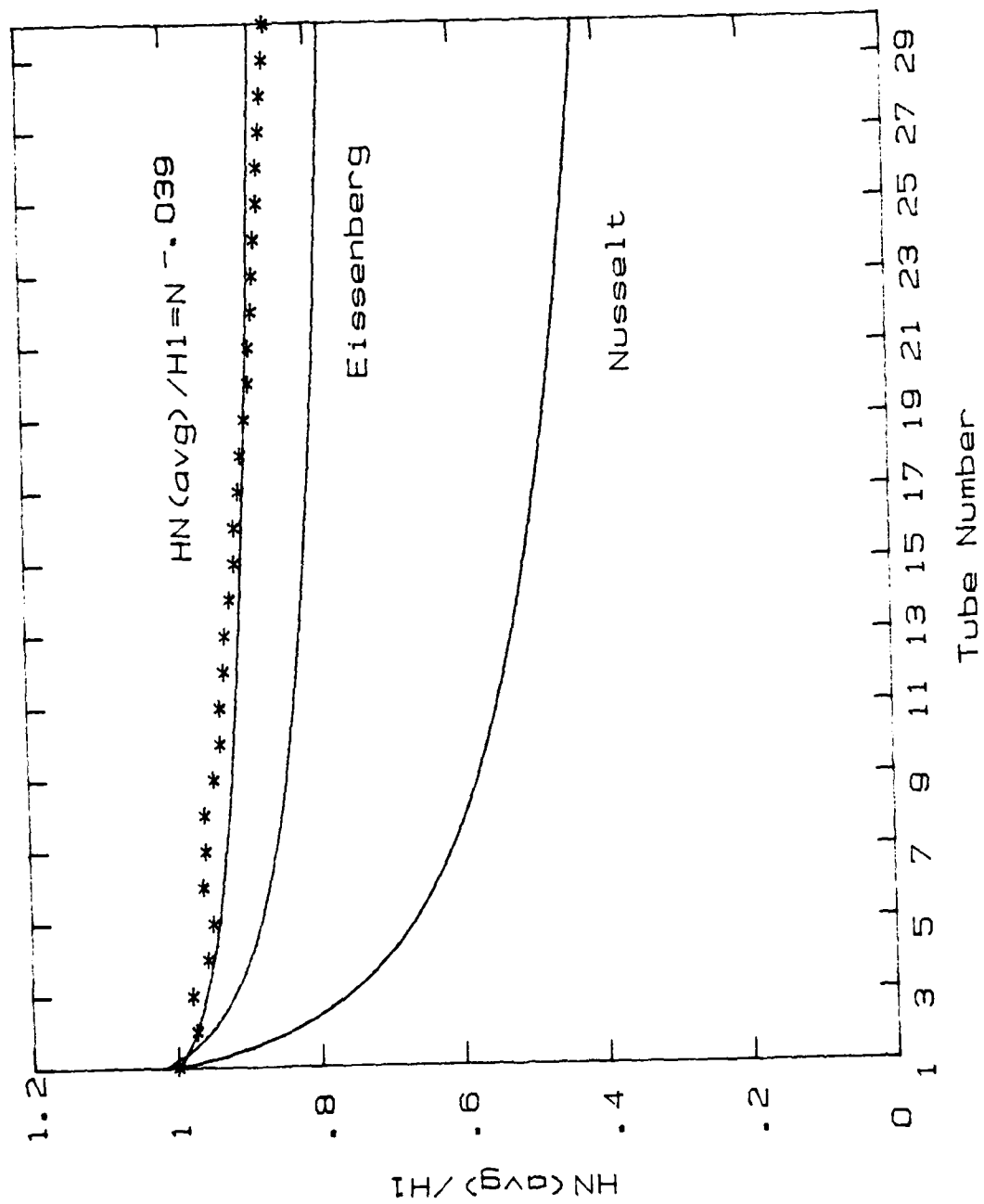


Figure 32. Least-Squares-Curve Fit for Data (Run RTWI-1)

Figure 33 shows the variation of the normalized, local, condensing coefficient for five tubes. The data points lie up to 76 percent above the curve predicted by the Nusselt theory.

Figure 34 shows the normalized, average condensing coefficient for five tubes. These data points are well above the Eissenberg correlation.

Figure 35 shows the variation of the normalized, local, condensing coefficient for 30 tubes. The data points lie up to 107 percent above the curve predicted by the Nusselt theory.

Figure 36 displays the normalized, average, condensing coefficient for 30 tubes. The data points lie above the curve representing the Eissenberg correlation.

Figure 37 shows that the curve fit is in good agreement with all the data points. The derived exponent is -0.037 .

F. OBSERVATIONS

1. Smooth Tubes

During all runs, complete film-wise condensation was observed without any visible evidence of drop-wise condensation. This was mainly achieved by the tube cleaning procedures used.

A slow rate of condensate droplet migration, from cooling water outlet end to the inlet end, was observed at the bottom of each active tube. This problem was minimized

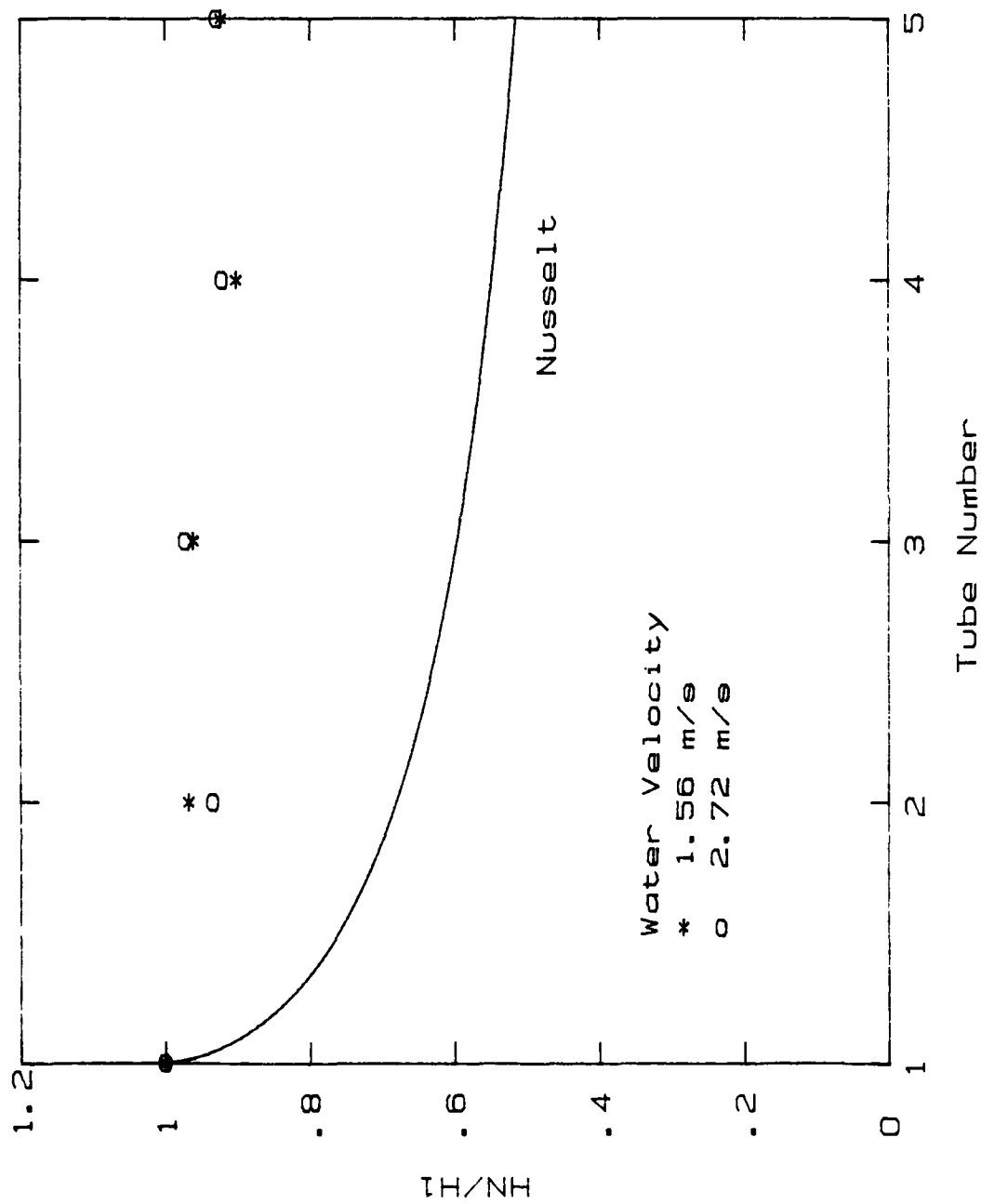


Figure 33. Variation of Local Condensing Coefficient with Tube Number
(Run STWNI-2)

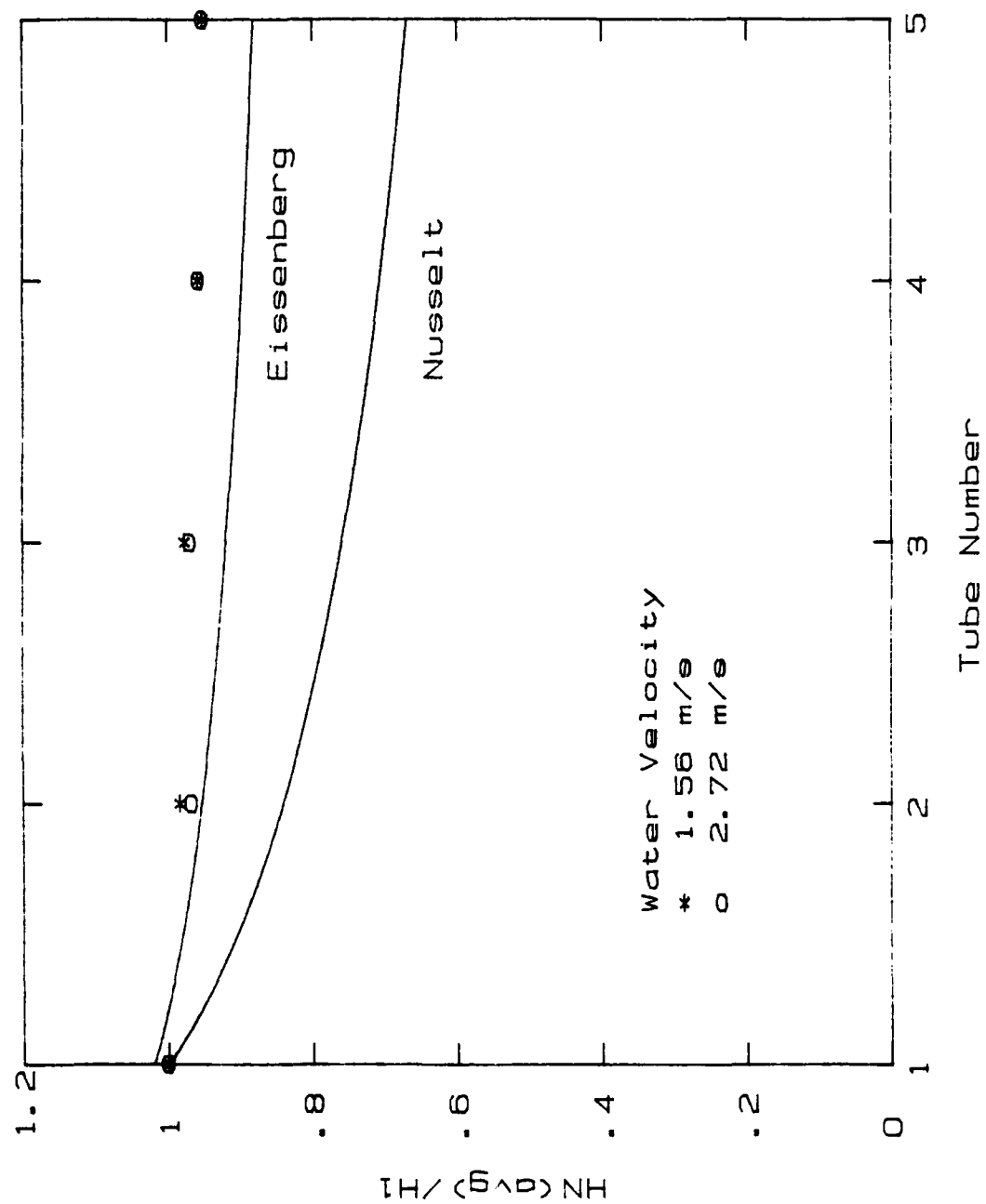


Figure 34. Variation of Average Condensing Coefficient with Tube Number (Run STWNI-2)

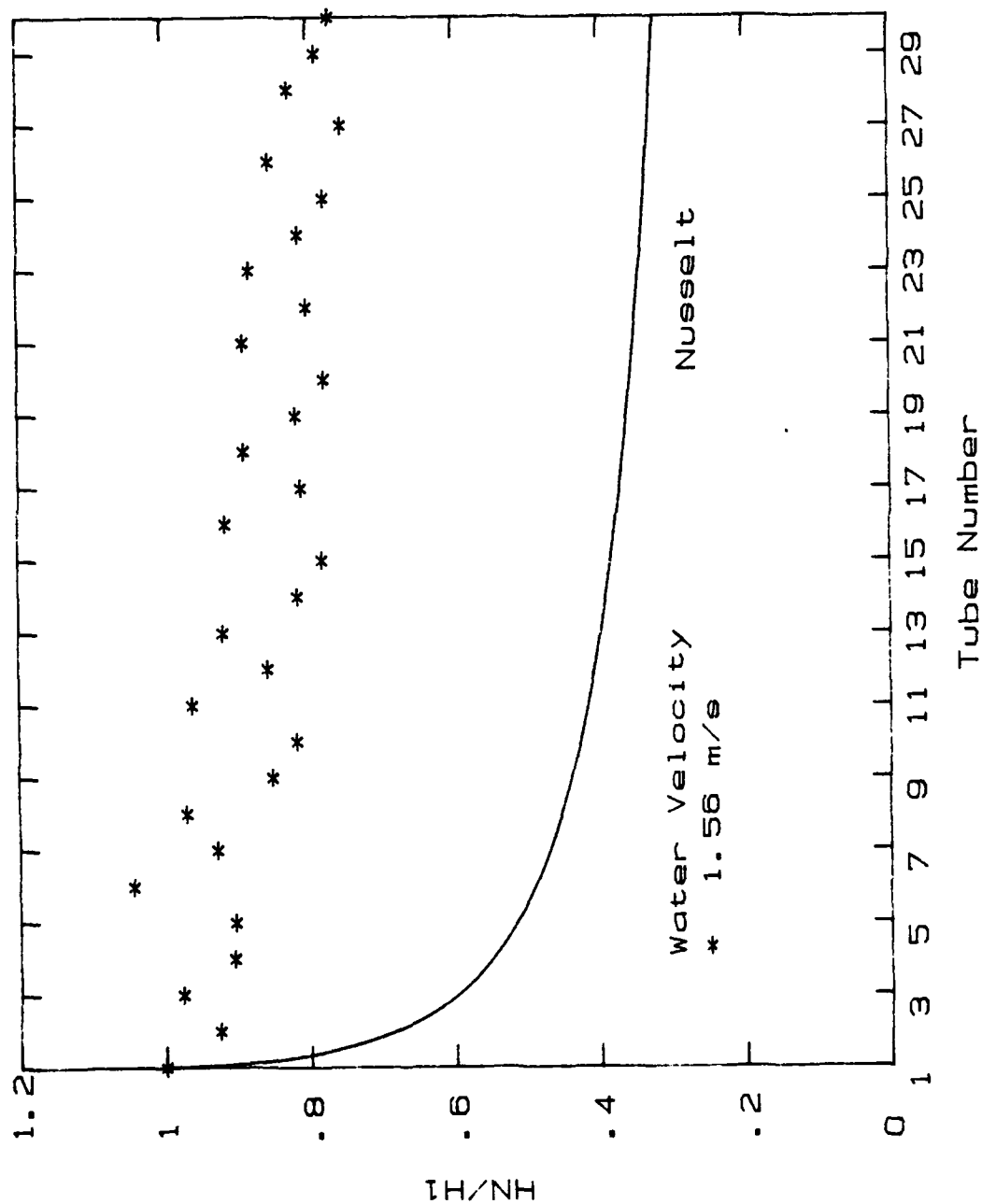


Figure 35. Variation of Local Condensing Coefficient with Tube Number
(Run STWI-1)

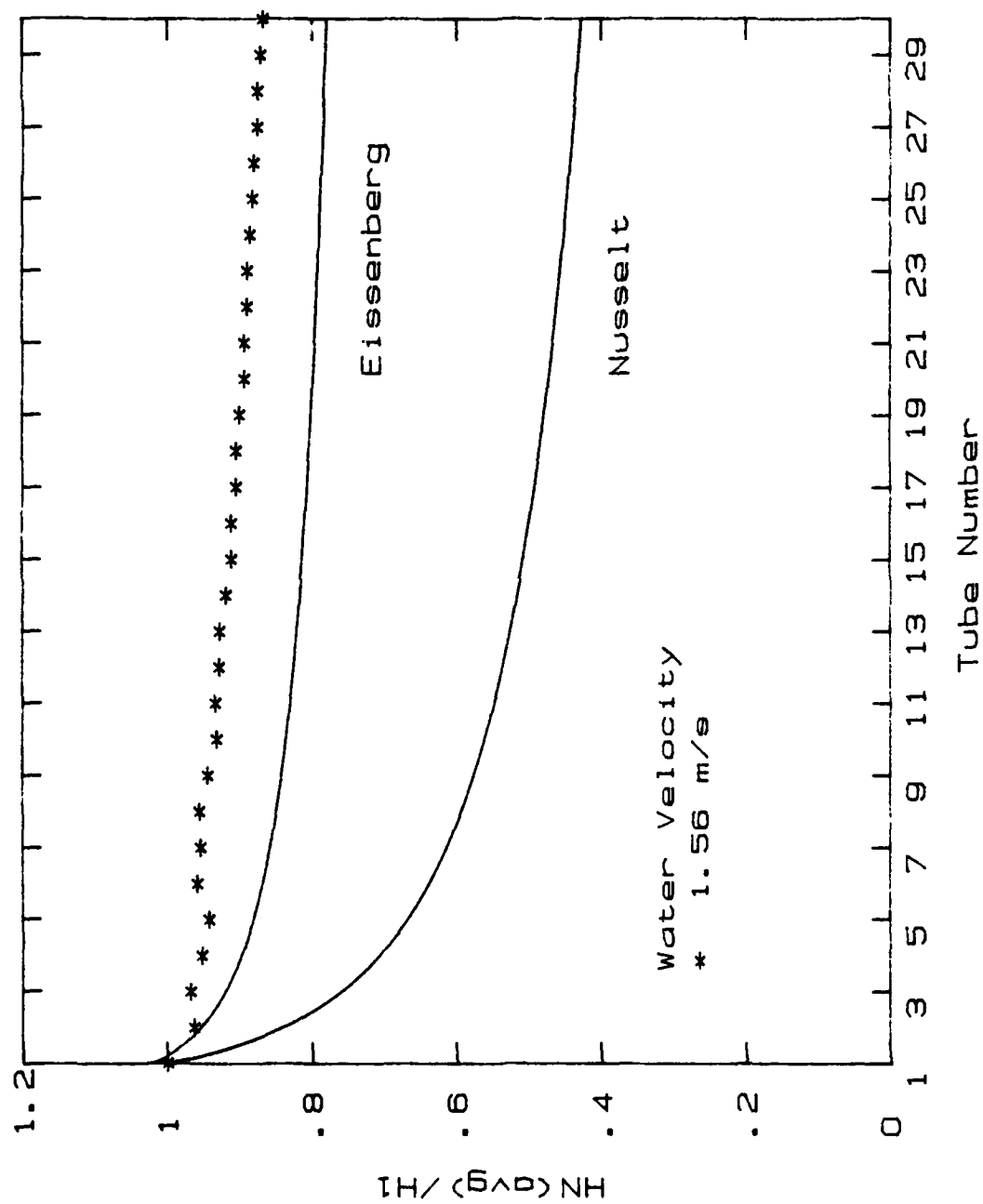


Figure 36. Variation of Average Condensing Coefficient with Tube Number (Run STWI-1)

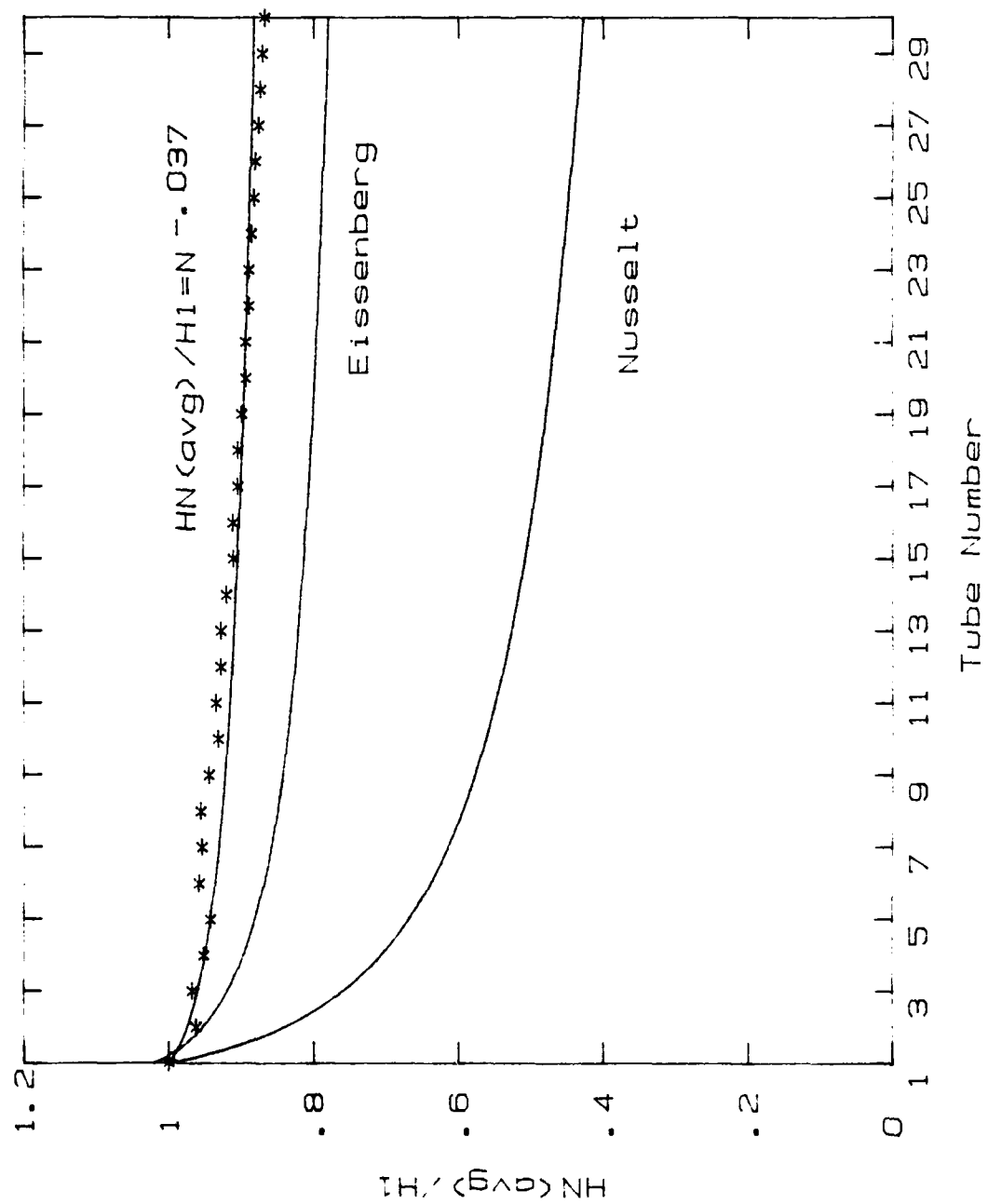


Figure 37. Least-Squares-Curve Fit for Data (Run STWI-1)

TABLE IV
SUMMARY OF RUNS WITH INUNDATION

<u>File Name</u>	<u>Tube Type</u>	<u>External Wire Wrapped or Not</u>
STNWI-1	Smooth	No
RTNWI-1	Roped	No
RTNWI-2	Roped	No
RTNWI-3	Roped	No
RTNWI-4	Roped	No
RTNWI-5	Roped	No
RTWI-1	Roped	Yes
RTWI-2	Roped	Yes
STWI-1	Smooth	Yes
STWI-2	Smooth	Yes

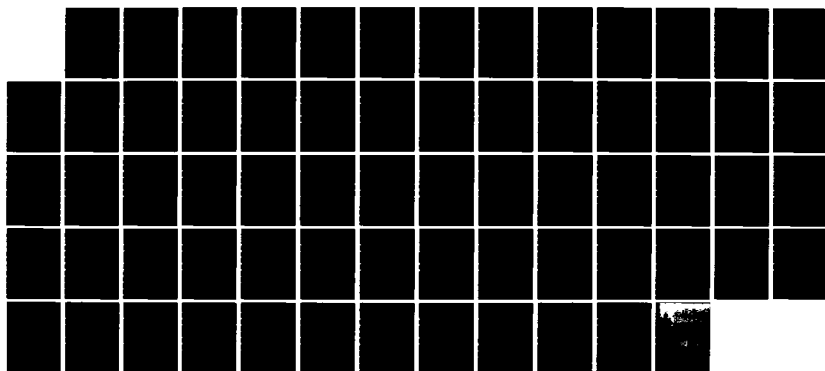
AD-A132 175

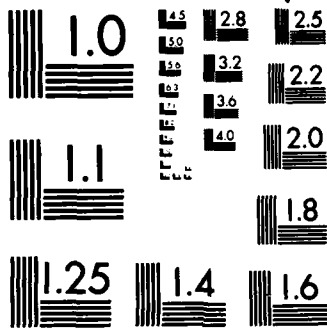
THE EFFECT OF CONDENSATE INUNDATION ON STEAM
CONDENSATION HEAT TRANSFER TO WIRE-WRAPPED TUBING(U)
NAVAL POSTGRADUATE SCHOOL MONTEREY CA G D KANAKIS
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MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

TABLE V
SUMMARY OF RUNS WITHOUT INUNDATION

<u>File Name</u>	<u>Tube Type</u>	<u>External Wire Wrapped or Not</u>
STNWEV-2	Smooth	No
STNWN1-1	Smooth	No
STNWN1-2	Smooth	No
STNWN1-3	Smooth	No
RTNWN1-3	Roped	No
RTWNI-1	Roped	Yes
RTWNI-2	Roped	Yes
RTWNI-3	Roped	Yes
STWNI-2	Smooth	Yes
STWNI-4	Smooth	Yes
STWNI-5	Smooth	Yes

TABLE VI

RESULTS FOR RUNS WITHOUT INUNDATION

File Name	Tube Type	External Wire Wrap or not	h_N/h_1	$\bar{h}_N/h_1$	Cooling Water Velocity (m/s)
STNWN1-1	Smooth	No	0.7476	0.8662	1.56
-1	Smooth	No	0.7586	0.8722	3.17
STNWN1-2	Smooth	No	0.6522	0.8031	1.56
-2	Smooth	No	0.6503	0.8102	3.17
STNWN1-3	Smooth	No	0.6473	0.8031	1.56
-3	Smooth	No	0.6393	0.8002	3.17
RTNWN1-3	Roped	No	0.6181	0.7728	1.56
-3	Roped	No	0.6298	0.7866	2.33
RTWN1-1	Roped	Yes	0.9465	0.9466	1.56
RTWN1-2	Roped	Yes	0.9479	0.9579	1.56
-2	Roped	Yes	0.9250	0.9303	2.33
RTWN1-3	Roped	Yes	0.9098	0.9244	1.56
-3	Roped	Yes	0.9094	0.9228	1.94
STWN1-2	Smooth	Yes	0.9278	0.9508	1.56
-2	Smooth	Yes	0.9224	0.9509	2.72

TABLE VI (continued)

STWNI-4	Smooth	Yes	0.8404	0.9051	1.56
-4	Smooth	Yes	0.8496	0.8713	2.72
STWNI-5	Smooth	Yes	0.9117	0.9545	1.56
-5	Smooth	Yes	0.9322	0.9506	2.72

TABLE VII

RESULTS FOR RUNS WITH INUNDATION UP TO 30 TUBES

File Name	Tube Type	External Wire Wrap or not	h_N/h_1	$\bar{h}_N/h_1$	Cooling Water Velocity (m/s)
STNWI-1	Smooth	No	0.504	0.5857	1.17
RTNWI-1	Roped	No	0.4694	0.5404	1.56
RTNWI-2	Roped	No	0.4751	0.5306	2.72
RTNWI-3	Roped	No	0.5039	0.5554	1.56
RTNWI-4	Roped	No	0.4885	0.5420	1.56
RTNWI-5	Roped	No	0.5029	0.5543	2.72
STWI-1	Smooth	Yes	0.7693	0.8674	1.56
STWI-2	Smooth	Yes	0.7227	0.8309	1.56
RTWI-1	Smooth	Yes	0.75	0.8536	1.56
RTWI-2	Smooth	Yes	0.7876	0.8623	1.56

TABLE VIII

COMPARISON OF HEAT-TRANSFER COEFFICIENTS FOR TUBE #1 IN THE BUNDLE
(WITHOUT INUNDATION) AT 1.56 m/s COOLING WATER VELOCITY

File Name	Tube Type	External Wire Wrap or not	$h_1 (W/m^2K)$	$h_{Nu} (W/m^2K)$
STNWN1-1	Smooth	No	11136.4	10801.9
			10257.7	11023.9
			10448.9	10991.6
			10328.9	11018.6
			9960.9	11102.2
STNWN1-2	Smooth	No	10521.7	10915.7
			10624.5	10902.4
			10690.9	10893.9
			10799.4	10879.2
			10597.9	10909.1
STNWN1-3	Smooth	No	10453.1	11050.4
			10620.3	11028.9
			10643.8	11027.8
			10613.8	11032.3
			10628.7	11027.9
RTNWN1-3	Roped	No	11329.	9758.6
			11423.3	9747.2
			11328.7	9757.5
			11359.2	9755.1
			11470.4	9744.6
RTWN1-1	Roped	Yes	9946.4	9989.1
			9975.8	9908.5
			9889.7	9908.5
			10052.3	9873.
			10019.1	9884.7

TABLE VIII (continued)

RTWNI-2	Roped	Yes	9951.3 9933.6 9954.1 9832.2 9931.1	9946.5 9960.4 9953.5 9973.3 9960.6
RTWNI-3	Roped	Yes	10594.4 10188.2 10040.8 10104. 10328.9	9511.4 9612.6 9648.7 9645.9 9607.
STWNI-2	Smooth	Yes	10624.7 10681.8 10746.1 10772.1 10789.3	11056.5 11051. 11047.5 11044.9 11044.7
STWNI-4	Smooth	Yes	11846.2 11275. 11259.3 11299.5 11285.8	10809. 10878.9 10882.1 10878.6 10887.
STWNI-5	Smooth	Yes	10607.6 10542.9 10854.4 10681.3 10698.3	10994.6 11010.1 10962.7 10991.2 10984.9

TABLE IX

COMPARISON OF HEAT-TRANSFER COEFFICIENT FOR TUBE WITH INUNDATION

File Name	Tube Type	External Wire Wrap or not	$h_i ({}^w/m^2k)$	$h_{Nu} ({}^w/m^2k)$
STNWI-1	Smooth	No	11005.6	12102.7
			11132.5	12089.5
			11219.7	12079.5
			11126.1	12086.2
			11028.4	12100.1
RTNWI-1	Roped	No	11359.8	9785.8
			11394.3	9783.7
			11461.4	9777.1
			11329.1	9806.1
			11351.3	9796.1
RTNWI-2	Roped	No	11089.7	9225.9
			11103.1	9221.
			11096.7	9225.6
			11058.4	9237.6
			10865.6	9281.3
RTNWI-3	Roped	No	10790.5	9793.2
			11056.1	9741.9
			10932.4	9764.3
			11004.2	9743.3
			11030.8	9734.1
RTNWI-4	Roped	No	11999.5	9502.2
			11322.1	9624.6
			11329.9	9622.3
			11239.5	9631.4
			11389.2	9616.5

TABLE IX (continued)

RTNWI-5	Roped	No	10493.5 10657.8 10894. 10663.6 10635.5	9312.8 9285.9 9241.1 9297.8 9313.8
RTWI-5	Roped	Yes	9909.75 9881.61 9956.31 9862.91 9923.13	9972.6 9980.3 9964.8 9981.2 9970.5
RTWI-2	Roped	Yes	9953.69 10026.1 10003.7 10085.7 10095.9	9837.7 9830.7 9844.9 9826.0 9826.1
STWI-1	Roped	Yes	10777.9 10785.3 10800.3 10767.1 10754.7	11048.4 11046.4 11048.4 11050.2 11051.
STWI-2	Roped	Yes	11461. 11731.7 11533.1 11440.1 11561.0	10909.8 10869.0 10897.8 10908.0 10891.1

TABLE X
COMPARISON OF $\bar{h}_N/h_{Nu}$ FOR UNINUNDATED TUBE RUNS

Tube Type	External Wire Wrap or not	$\bar{h}_N/h_{Nu}$
Smooth	No	0.7986
Roped	No	0.9019
Roped	Yes	0.9651
Smooth	Yes	0.9339

TABLE XI
COMPARISON OF h_N/h_{Nu} FOR INUNDATION TUBE RUNS

Tube Type	External Wire Wrap or not	$\bar{h}_N/h_{Nu}$
Smooth	No	0.59
Roped	No	0.63
Roped	Yes	0.86
Smooth	Yes	0.86

TABLE XII
EXPONENTS OF THE LEAST-SQUARES-FIT

File Name	Tube Type	External Wire Wrap or not	Exponent
STNWI-1	Smooth	No	0.154
RTNWI-1	Roped	No	0.183
-2	Roped	No	0.191
-3	Roped	No	0.179
-4	Roped	No	0.185
RTWI-1	Roped	Yes	0.039
-2	Roped	Yes	0.039
STWI-1	Smooth	Yes	0.037
-2	Smooth	Yes	0.056

after leveling the tubes, by adjusting the leveling nuts on the test condenser support bracket. When the condensation or the inundation rate increased, it was observed that the drops were formed at more sites along the tubes, but the droplet size for each tube was nearly the same.

2. Roped Tubes

Again, during the experiments, there was no evidence of drop-wise condensation. It was observed that the falling drops were coalesced on the lower surface of the tube, especially in the space between the two successive grooves. and then rivulets were formed and fell on the tube below. Under condensate inundation, the phenomenon of the rivulet formation was more intense. It is also worth noting that the droplet-formation frequency was higher near the cooling water inlet end. This can be easily explained by the larger, local, Sieder-Tate coefficient and by the larger temperature difference, $T_{\text{sat}} - T_{\text{ci}}$, at the inlet compared to $T_{\text{sat}} - T_{\text{co}}$, at the outlet end.

3. Roped Tubes Wrapped with Wire

The condensate was formed in the space between the successive grooves wrapped with wire. Then due to the surface tension forces, the condensate was drawn to the base of the wire, and drops were formed at the bottom surface of the tubes.

Under inundation conditions, the above-mentioned drop formation and movement were more intense. It is worth noting that no splashing was observed during the experiment.

4. Smooth Tubes Wrapped with Wire

It was observed that the condensate forms between two successive, helically-wrapped wires, and was then drawn immediately towards the space between the wire and the tube surface. Rivulets were drawn from the base of the wire to the next tube below. It was observed that there was no drop migration along the tube.

Under condensate inundation, the above mentioned drop formation was more intensive, but again no splashing was observed.

VI. CONCLUSIONS

1. The average, outside, heat-transfer coefficient for 30 smooth tubes was 59 percent of the Nusselt coefficient calculated for the first tube in the bank.

2. The average, outside, heat-transfer coefficient for 30 smooth tubes wrapped with wire was 86 percent of the Nusselt coefficient calculated for the first tube in the bank.

3. The average, outside, heat-transfer coefficient for 30 roped tubes was 63 percent of the Nusselt coefficient calculated for the first tube in the bank.

4. The average, outside, heat-transfer coefficient for 30 roped tubes, wrapped with wire was 86 percent of the Nusselt coefficient calculated for the first tube in the bank.

5. The Sieder-Tate coefficient for roped tubes was 2.1 times greater than that for smooth tubes.

6. Wire wrapping considerably improves the average condensing coefficient for both smooth and roped tubes in tube bundles.

7. Of all cases investigated in this study, roped tubes with wire wrap would be the best candidate for designing compact condensers. However, the water side pumping power, which was not investigated in this work, could be considerably higher than that for the smooth-tube case.

VII. RECOMMENDATIONS

A. TEST APPARATUS MODIFICATIONS

The following test apparatus modifications are considered advisable:

1. Redesign the test condenser flanges to allow a large sealing area, and also to allow for the possibility of further experiments with different S/D ratios.
2. Redesign the existing test condenser hotwell to allow more reliable and convenient measurement of condensate flow rate.
3. Install a larger heater for the perforated-tube water supply tank to facilitate the simulation of larger tube bundles. Also, modify the temperature control system for the perforated-tube water supply tank, to allow for more rapid heating and cooling to reduce long delays between runs.
4. Install a glass window on the rear side of the test condenser, in order for the operator to have a view on both sides of the test tubes.
5. Make the system vacuum tight so that data can be taken at vacuum conditions.

B. ADDITIONAL TESTS

The following additional tests would be important in this continued investigation.

1. Conduct tests with enhanced tubes manufactured by Yorkshire Imperial Metals, Ltd.
2. Take movies of the condensation process so that further conclusions could be drawn with regard to condensate drop phenomena and their relationship to condenser performance.
3. Conduct tests varying the pitch of the wrapped wire on smooth titanium tubes and determine the optimum wire pitch.

4. Withdraw gas samples from the test condenser and analyze on the gas chromatograph to determine the effect of noncondensable gases on the condenser performance.
5. Conduct tests to investigate the effect of vapor velocity on the heat-transfer coefficient.

APPENDIX A
OPERATING PROCEDURES

A. INITIAL PROCEDURES

1. Energize the main circuit breaker located in Power Panel P-2 on the wall to the right of the test apparatus.
2. Energize the circuit breaker on the left side of the old control board by pressing the ON button.
3. Energize the following switches in the control panel:
 - a. #1 - Perforated tube water supply tank (feed pump).
 - b. #2 - Outlets.
 - c. #3 - Hot water heater.
 - d. #4 - Condensate pump.
 - e. #6 - Cooling tower.
 - f. #7 - Cooling water pump.
4. Ensure all test apparatus valves are closed.
5. Fill the perforated tube condensate supply tank with distilled water. Set the temperature controller for the perforated tube water supply tank at about 95° C, fully open the recirculation valve, P-1, and start the pump to begin heating the water. The controller will have to be reset to the proper supply temperature once steady-state conditions are obtained.
6. Fill the cooling water supply tank. This can be done by backfilling with valves CK-1 and CW-4 open.
7. Energize the data acquisition system.

B. OPERATION

1. House Steam

- a. Open the main supply valve.
- b. Open valve MS-3 until the pressure gage indicates the desired steam supply pressure.
- c. Fully open MSD-1 to drain any condensate.
- d. Open valves MS-4 and MS-5 until the desired steam supply pressure is obtained and re-adjust MS-3 as necessary.

2. Condensate System. To collect the condensate in the test condenser hotwell, operate the system with valve C-1 closed. After a test run is completed, open valve C-1 to drain the condensate by opening valve C-2 to the bilges. or operating the condensate pump in order to fill the perforated tube supply tank.

3. Cooling-Water System

- a. Open valves CW-1, CW-2 and CW-3.
- b. Ensure valves CK1-1 and CW-4 are closed.
- c. Energize the two cooling water pumps.
- d. Open valves CW-5, CW-6, CW-7, CW-8 and CW-9 to obtain the desired cooling rates.

4. Perforated Tube Water Supply System

- a. Once steady-state conditions are achieved, reset the temperature controller to the proper inundation temperature.
- b. Adjust the rotameter to the required flow rate for each run. The supply tank recirculation valve may have to be adjusted to achieve the desired flow rate, but should never be fully closed to avoid damaging the pump and to ensure uniform water temperature in the tank.
- c. Refill the supply tank as required, by using the existing piping filling system, by operating the condensate pump; or filling the supply tank with distilled water filling line.

5. Miscellaneous.

To maintain a clear test condenser window, open valve A-2 and then energize and adjust the air heater power supply. When securing, always turn off the power supply first and allow the air heater to cool before securing valve A-2.

C. SECURING THE TEST APPARATUS

1. Secure the steam valves MS-5, MS-4, MS-3 and the main supply valve.
2. Secure the air compressor.
3. Secure the perforated-tube water supply system by securing the pump, temperature controller, and valves P-1 and P-4. Drain the supply tank by opening valve P-3 (if desired).
4. Secure the test condenser viewing window air heater as prescribed above.
5. Secure the data acquisition system.
6. Allow the test condenser to cool down for about 30 minutes. then secure the cooling water pumps and close valves CS-1, CW-2, CW-3, CW-5, CW-6, CW-7, CW-8, CW-9 and C-4.
7. Drain the test condenser hotwell.
8. Secure all circuit breakers.
9. Drain the cooling water system piping and rotameters by opening valve CW-10 and leaving open the cooling water rotameter supply valves.
10. Drain the cooling water supply tank by opening the drain valve via the remote operating rod.
11. Ensure all valves are secured.

APPENDIX B SAMPLE CALCULATIONS

A. RUN STNWN1-1

A sample calculation is performed in this section to illustrate the solution procedure used in the data reduction program [Ref 29].

The STNWN1-1 run was selected to perform this analysis:

INPUT PARAMETERS

File	STNWN1-1				
Pressure condition	Atmospheric				
Inundation condition	5 tubes				
Month, date and time	05:11:10:08:50				
Run number	1				
Tube number	1	2	3	4	5
Inlet temp (°C)	28.53	28.54	28.55	28.54	28.53
Outlet temp (°C)	32.55	32.20	32.17	32.05	31.83
Saturation temperature	100.24 (°C)				
Degree of superheat	1.88 (°C)				
Condensate temperature	92.41 (°C)				
Static pressure	767.33 (mm Hg)				

The following calculations are further limited only to the first tube.

1. Determination of Average Bulk Temperature

$$T_b(1) = T_{ci}(1) + T_{co}(1) \times 0.5$$

$$T_b(1) = (28.53 + 32.55) \times 0.5$$

$$T_b(1) = 30.54^{\circ}\text{C}$$

2. Thermophysical Properties

$$P_r = 5.329$$

$$\rho = 995.2 \text{ kg/m}^3$$

$$\mu = 798 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$C_{pw} = 4.178 \text{ Kj/kg}\cdot\text{K}$$

$$k_w = 619 \times 10^{-3} \text{ W/m}\cdot\text{K}$$

NOTE: All properties are calculated at the average bulk water temperature from Table A.6 p. 782 [Ref 30]

3. Cooling Water Mass Flow Rate

$$\dot{m} = 14.22 \text{ Kg/min} = 0.237 \text{ Kg/s}$$

4. Determination of Cooling Water Velocity

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(14.22) \frac{1}{60}}{(995.2) (1.56 \times 10^{-4})}$$

$$V_w = 1.53 \text{ m/s}$$

5. Determination of Reynolds Number

$$R_e = \frac{\rho V_w D_i}{\mu_w}$$

$$R_e = \frac{(995.2) (1.53) (0.141)}{798 \times 10^{-6}}$$

$$R_e = 26,904$$

6. Determination of Heat Transfer

$$Q = \dot{m} \cdot (T_{co} - T_{ci}) \cdot C_{pw}$$

$$Q = (14.22) \left(\frac{1}{60}\right) (32.55 - 28.53) (4,178)$$

$$Q = 3,980.5 \text{ W}$$

7. Determination of Heat Flux

$$q'' = \frac{Q}{\pi \cdot D_o \cdot L}$$

$$q'' = \frac{3,980.5}{\pi \cdot (0.015875) (0.305)}$$

$$q'' = 261,682.2 \frac{\text{W}}{\text{m}^2}$$

8. Determination of Nusselt Coefficient

$$h_{Nu} = 0.651 \left[\frac{k_f^3 \cdot \rho_f^2 \cdot h_{fg} \cdot (9.81)}{\mu_f \cdot D_o \cdot q''} \right]^{1/3}$$

$$\text{Assume } T_f = T_{sat}$$

$$T_f = 100.24^\circ\text{C}$$

$$\rho_f = 957.8 \text{ kg/m}^3$$

$$h_{fg} = 2.2564 \times 10^3 \text{ J/kg}$$

$$k_f = 682 \times 10^{-3} \text{ W/m}\cdot\text{K}$$

$$u_f = 281 \times 10^{-6} \frac{\text{N} \cdot \text{s}}{\text{m}^2}$$

NOTE: All properties are calculated at the Saturation Temperature. From Table A.6 p. 782 [Ref 30]

$$h_{\text{Nu}} = 0.651 \left[\frac{(682 \times 10^{-3})^3 \cdot (957.8)^2 (2256.4 \times 10^3) (9.81)}{(281 \times 10^{-6}) (0.015975) (261682.2)} \right]^{1/3}$$

$$h_{\text{Nu}} = 11492.6 \text{ W/m}^2\text{K}$$

9. Determination of $T_{f,c}$

$$T_{f,c} = T_{\text{sat}} - \frac{q''}{h_{\text{Nu}}} 0.5$$

$$T_{f,c} = 100.24 - \frac{261685.3}{11492.6} 0.5$$

$$T_{f,c} = 88.85^\circ\text{C}$$

10. Thermophysical Properties

$$k = 673 \times 10^{-3} \text{ W/mK}$$

$$\rho = 963 \text{ Kg/m}^3$$

$$\mu = 326 \times 10^{-6} \text{ N} \cdot \text{s/m}^2$$

$$h_{fg} = 2.289.5 \times 10^3 \text{ J/kg}$$

NOTE: All properties are calculated at the film temperature. From Table A.6 p. 782 [Ref 30]

11. Determination of Nusselt Coefficient

$$h_{Nu} = 0.651 \left[\frac{k_f^3 \cdot \rho_f^2 h_{fg} \cdot 9.81}{\mu_f \cdot D_o \cdot q''} \right]^{1/3}$$

$$h_{Nu} = 0.651 \left[\frac{(673 \times 10^{-3})^3 (963)^2 (2289.5 \times 10^3) (9.81)}{(326 \times 10^{-6}) (0.015875) (261685.4)} \right]^{1/3}$$

$$h_{Nu} = 10,885.3 \text{ W/m}^2 \cdot \text{K}$$

12. Determination of $T_{f,c}$

$$T_{f,c} = T_{sat} - \frac{q''}{h_{Nu}} 0.5$$

$$T_{f,c} = 100.24 - \frac{261685.4}{10885.3} 0.5$$

$$T_{f,c} = 88.22^\circ\text{C}$$

13. Determination of Logarithmic Mean Temperature Difference (LMTD)

$$\text{LMTD} = \frac{T_{co} - T_{ci}}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right)}$$

$$\text{LMTD} = \frac{32.55 - 28.53}{\ln \left(\frac{100.24 - 28.53}{100.24 - 32.55} \right)}$$

$$\text{LMTD} = 69.68^\circ\text{C}$$

14. Determination of Overall Heat - Transfer Coefficient

$$U_o = \frac{q''}{\text{LMTD}}$$

$$U_o = \frac{261685.4}{69.68}$$

$$U_o = 3755.5 \frac{W}{m^2 K}$$

15. Determination of Inside Heat-Transfer Coefficient

$$\text{Assume } C_f = 1.1$$

$$C_i = 0.029$$

$$h_i = \frac{K_w}{D_i} \cdot C_i \cdot Re^{0.8} \cdot Pr^{0.333} \cdot C_f$$

$$h_i = \frac{0.673}{0.0141} (0.029) (26904^{0.8}) (5.829^{0.3333}) \cdot 1.1$$

$$h_i = 9303.2 \text{ W/m}^2 K$$

16. Determination of Inner Wall Temperature

$$T_w = T_b + \frac{q''}{h_i} \frac{D_o}{D_i}$$

$$T_w = 30.54 + \frac{(261685.4)}{(9303.2)} \frac{(0.015875)}{(0.0141)}$$

$$T_w = 62.2^\circ C$$

17. Determination of μ_w at the Average Wall Temperature

$$\mu_w (62.2^\circ C) = 453 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

18. Determination of Correction Factor

$$C_{fc} = \left[\frac{\mu_w}{\mu_w (65.59^\circ)} \right]^{0.14}$$

$$C_{fc} = \left[\frac{798 \times 10^{-6}}{453 \times 10^{-6}} \right]^{0.14}$$

$$C_{fc} = 1.08$$

19. Determination of Outside Heat-Transfer Coefficient

$$h = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w}$$

$$h = \frac{1}{\frac{1}{3755.5} - \frac{0.015875}{(0.1014)(9303.2)} - 0.000042925}$$

$$h = 9772.3 \text{ W/m}^2\text{K}$$

B. RUN STSD-11

A sample calculation is performed in this section to illustrate the solution procedure used in the modified Wilson plot program [Ref 29].

The DP-11 run was selected to perform this analysis

INPUT PARAMETERS

File	DP-11
Month, date and time	04:01:13:57:20
Data point	#1
Steam Saturation Temperature	100.01°C
Inlet Temperature	24.42°C
Outlet Temperature	30.75°C
Flowmeter Reading	10%

1. Determination of Average Bulk Water Temperature

$$T_b = (T_{ci} + T_{co}) 0.5$$

$$T_b = (24.42 + 30.75) 0.5 = 27.59^\circ\text{C}$$

2. Thermophysical Properties

$$P_r = 5.83$$

$$\rho = 997 \frac{\text{kg}}{\text{m}^3}$$

$$C_{pw} = 4.180 \text{ Kj/Kg} \cdot \text{K}$$

$$\mu = 857 \times 10^{-6} \text{ N} \cdot \text{s/m}^2$$

$$k_w = 0.613 \text{ W/m} \cdot \text{K}$$

NOTE: All properties are calculated at the average bulk water temperature from Table A.6 p. 782 [Ref. 30]

3. Cooling Water Mass Flow Rate

$$M_f = 6.36 \text{ kg/min}$$

4. Determination of Cooling Water Velocity

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(6.36) \frac{1}{60}}{(997) (1.56 \times 10^{-4})}$$

$$V_w = 0.68 \text{ m/s}$$

5. Determination of Reynolds Number

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

$$R_e = \frac{(997) (0.68) (0.0141)}{857 \times 10^{-6}}$$

$$R_e = 11,154$$

6. Determination of Heat Transfer

$$Q = M_f \cdot (T_{co} - T_{ci}) \cdot C_{pw}$$

$$Q = (6.36) \left(\frac{1}{G_o}\right) (30.75 - 24.42) (4180)$$

$$Q = 2,805 \text{ W}$$

7. Determination of Logarithmic Mean Temperature Difference (LMTD)

$$LMTD = \frac{T_{co} - T_{ci}}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right)}$$

$$LMTD = \frac{30.75 - 24.42}{\ln \left(\frac{100.01 - 24.42}{100.01 - 30.75} \right)}$$

$$LMTD = 72.38^{\circ}\text{C}$$

8. Determination of Overall Heat - Transfer Coefficient

$$U_o = \frac{Q}{\pi \cdot D_o \cdot L \cdot LMTD}$$

$$U_o = \frac{2805}{\pi (0.015875) (0.305) (72.38)}$$

$$U_o = 2,547.715 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$$

9. Determination of Sieder - Tate Parameter

$$X = R_e^{-0.8} \cdot P_r^{-0.3333}$$

$$X = (11,154)^{-0.8} (5.83)^{-0.3333}$$

$$X = 0.0003212$$

10. Determination of Inside Heat - Transfer Coefficient, based on assumed Sieder - Tate Coefficient

Assume $C_f = 1.1$ and

$$C_i = 0.03$$

$$h_i = \frac{K_w}{D_i} C_i \cdot R_e^{0.8} \cdot P_r^{0.333} C_f$$

$$h_i = \frac{613 \times 10^{-3}}{0.141} (0.03) (11,154^{0.8}) (5.83^{0.333}) (1.1)$$

$$h_i = 4,463.4 \frac{W}{m^2 \cdot K}$$

11. Determination of Average Wall Temperature

$$T_w - T_b = \frac{Q}{\pi \cdot D_i \cdot L \cdot h_i}$$

$$T_w - T_b = \frac{2805}{\pi (0.0141) (0.305) (4,453.4)}$$

$$T_w - T_b = 46.51^\circ C$$

$$T_w = 46.51 + T_b$$

$$T_w = 46.51 + 27.59 = 74^\circ C$$

12. Obtain μ_w at the Average Inner Wall Temperature

$$\mu_w (T_w) = 375 \times 10^{-6} \text{ N}\cdot\text{s}/\text{m}^2$$

13. Determination of C_{fc}

$$C_{fc} = \left(\frac{\mu_w}{\mu_w (T_w)} \right)^{0.14}$$

$$C_{fc} = \left(\frac{857 \times 10^{-6}}{375 \times 10^{-6}} \right)^{0.14}$$

$$C_{fc} = 1.12267$$

14. Iterate for h_i

Assume $C_f = 1.12$ and

$$C_i = 0.03$$

$$h_i = \frac{k_w}{D_i} C_i R_e^{0.8} P_r^{0.3333} \cdot C_f$$

$$h_i = \frac{613 \times 10^{-3}}{0.0141} \cdot (0.03) (11,154^{0.8}) (5.83^{0.3333}) (1.12)$$

$$h_i = 4,546.97 \text{ W/m}^2\text{K}$$

15. Determination of Wall Thermal Resistance, Based on the Outside Diameter

$$R_w = \frac{(D_o - D_i) D_o}{k_m \cdot (D_o + D_i)}$$

$$R_w = \frac{(0.015875 - 0.0141)(0.015875)}{(21.9)(0.015875 + 0.0141)}$$

$$R_w = 0.000042925 \frac{\text{m}^2 \cdot \text{K}}{\text{W}}$$

Assume $R_{f,} = 0$

16. Determination of Outside Heat - Transfer Coefficient

$$h = \frac{1}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}}$$

$$h = \frac{1}{\frac{1}{2,547.715} - 0.000042925 - \frac{0.015875}{(0.0141)(4,463.4)}}$$

$$h = 10273.8 \frac{\text{W}}{\text{m}^2 \cdot \text{K}}$$

Set $Q_o = Q = 2805 \text{ W}$

17. Determination of Actual Sieder-Tate Parameter

$$X = \frac{X}{C_{fc}}$$

$$X = \frac{0.0003212}{1.12267}$$

$$X = 0.0002861$$

INPUT PARAMETERS

File	STSD-11
Data point	#2
Steam saturation temperature	100.03°C
Inlet temperature	24.39°C
Outlet temperature	29.46°C
Rotameter setting	15%

18. Determination of Average Bulk Water Temperature

$$T_b = (T_{ci} + T_{co}) \times 0.5$$

$$T_b = (24.39 + 29.46) \times 0.5$$

$$T_b = 26.92^\circ\text{C}$$

19. Thermophysical Properties

$$P_r = 5,823$$

$$\mu = 996 \text{ Kg/m}^3$$

$$C_{pw} = 4.180 \text{ Kj/KgK}$$

$$\mu = 855 \times 10^{-6} \text{ N.s/m}^2$$

$$k_w = 613 \times 10^{-3} \text{ W/mK}$$

20. Cooling Water Mass Flow Rate

$$M_f = 9.72 \text{ Kg/min}$$

22. Determination of Cooling Water Velocity

$$V_w = \frac{M_f}{\rho A_i}$$

$$V_w = \frac{(9.72) \frac{1}{60}}{(996) (1.56 \times 10^{-4})}$$

$$V_w = 1.04 \text{ m/s}$$

22. Determination of R_e

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

$$R_e = \frac{(996) (1.04) (0.0141)}{855 \times 10^{-6}}$$

$$R_e = 17,082$$

23. Determination of Heat - Transfer

$$Q = M_f (T_{co} - T_{ci}) C_{pw}$$

$$Q = (9.72) \left(\frac{1}{60}\right) (29.46 - 23.39) (4180)$$

$$Q = 3,433.2 \text{ W}$$

24. Determination of Logarithmic Mean Difference Temperature (LMTD)

$$LMTD = \frac{T_{co} - T_{ci}}{\ln \left(\frac{T_{sat} - T_{ci}}{T_{sat} - T_{co}} \right)}$$

$$LMTD = \frac{29.46 - 24.39}{\ln \left(\frac{100.03 - 24.39}{100.03 - 29.46} \right)}$$

$$LMTD = 73.08^\circ\text{C}$$

25. Determination of Overall Heat-Transfer Coefficient

$$U_o = \frac{Q}{\pi \cdot D_o \cdot L \cdot LMTD}$$

$$U_o = \frac{3433.2}{(0.015875)(0.305)(73.08)}$$

$$U_o = 3,088.4 \text{ W/m}^2\text{K}$$

26. Determination of Sieder-Tate Parameter

$$X = R_e^{-0.8} \cdot P_r^{-0.3333}$$

$$X = (17,082^{-0.8}) (5.823^{-0.3333})$$

$$X = 0.000228$$

27. Determination of $\frac{1}{h_i}$

$$\frac{1}{h_i} = \frac{1}{U_o} - \frac{1}{h_o} \left(\frac{Q_o}{Q} \right)^{1/3} - R_w \left(\frac{D_i}{D_o} \right)$$

$$\frac{1}{h_i} = \left[\frac{1}{3088.4} - \frac{1}{10273} \left(\frac{2805}{3,433.2} \right)^{1/3} - 0.00004295 \right] \frac{0.015875}{0.0141}$$

$$\frac{1}{h_i} = 0.0002095 \frac{\text{m}^2\text{K}}{\text{W}}$$

28. Determination of Average Wall Temperature

$$T_w = T_b + \frac{Q}{\pi \cdot D_i \cdot L \cdot h_i}$$

$$T_w = 26.92 + \frac{3433.2}{\pi \cdot (0.015875) (0.305) (4,773.26)}$$

$$T_w = 74.20^\circ\text{C}$$

29. Determination of μ_w at the Average Wall Temperature

$$\mu_w(T_w) = 375.2 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

30. Determination of C_{fc}

$$C_{fc} = \frac{u_w}{u_w(T_w)}^{0.14}$$

$$C_{fc} = \frac{855 \times 10^{-6}}{375.2 \times 10^{-6}}^{0.14}$$

$$C_{fc} = 1.1222$$

31. Determination of Actual Sieder-Tate Parameter

$$X = \frac{X}{C_{fc}}$$

$$X = \frac{0.000228}{1.12459}$$

$$X = 0.0002027$$

APPENDIX C

UNCERTAINTY ANALYSIS

The general form of Kline and McClintock [Ref. 31] "second order" equation is used to compute the probable uncertainty in the results. For some resultant, R , which is a function of primary variables $X_1, X_2, \dots, X_n$, the probable uncertainty in R , δR is given by:

$$\delta R = \left[\left(\frac{\partial R}{\partial X_1} \delta X_1 \right)^2 + \left(\frac{\partial R}{\partial X_2} \delta X_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial X_n} \delta X_n \right)^2 \right]^{1/2} \quad (C.1)$$

where $\delta X_1, \delta X_2, \dots, \delta X_n$ is the possible uncertainty in each of the measured variables.

A. UNCERTAINTY IN THE COOLING WATER VELOCITY

$$V_w = \frac{\dot{m}}{\rho A_i}$$

Applying equation (C.1) the following equation results:

$$\frac{\delta V_w}{V_w} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \right)^2 + \left(\frac{\delta \rho}{\rho} \right)^2 + \left(\frac{\delta A_i}{A_i} \right)^2 \right]^{1/2} \quad (C.2)$$

The following uncertainties were assigned to the variables:

$$\delta \dot{m} = \pm 0.01 \text{ kg/s}$$

$$\delta \rho = \pm 3 \text{ kg/m}^3$$

$$\delta A_i = \pm 0.0001 \text{ m}^2$$

B. UNCERTAINTY IN THE REYNOLD'S NUMBER

$$R_e = \frac{\rho_w V_w D_i}{\mu_w}$$

The probable uncertainty is given by:

$$\frac{\delta R_e}{R_e} = \left[\left(\frac{\delta \rho}{\rho} \right)^2 + \left(\frac{\delta V_w}{V_w} \right)^2 + \left(\frac{\delta D_i}{D_i} \right)^2 + \left(\frac{\delta \mu}{\mu} \right)^2 \right]^{1/2} \quad (C.3)$$

The following uncertainties were assigned to the variables:

$$\delta \rho = \pm 3 \text{ kg/m}^3$$

$$\delta D_i = \pm 0.0001 \text{ m}$$

$$\delta \mu = \pm 8 \times 10^{-6} \text{ N}\cdot\text{s/m}^2$$

$$\delta V_w = \text{from equation (C.2)}$$

C. UNCERTAINTY IN HEAT TRANSFER

$$Q = \dot{m}(T_{co} - T_{ci})C_{pw}$$

The probable uncertainty is given by:

$$\frac{\delta Q}{Q} = \left[\left(\frac{\delta \dot{m}}{\dot{m}} \right)^2 + \left(\frac{\delta T_{co}}{T_{co} - T_{ci}} \right)^2 + \left(\frac{\delta T_{ci}}{T_{co} - T_{ci}} \right)^2 + \left(\frac{\delta C_{pw}}{C_{pw}} \right)^2 \right]^{1/2} \quad (C.4)$$

The following uncertainties were assigned to the variables:

$$\delta \dot{m} = \pm 0.01 \text{ kg/s}$$

$$\delta T_{co} = \pm 0.025 \text{ } ^\circ\text{C}$$

$$\delta T_{ci} = \pm 0.025 \text{ } ^\circ\text{C}$$

$$\delta C_{pw} = \pm 8 \text{ J/Kg}\cdot\text{C}$$

D. UNCERTAINTY OF THE HEAT-FLUX

$$q'' = \frac{Q}{\pi D_o L}$$

The probable uncertainty is given by:

$$\frac{\delta q''}{q''} = \frac{1}{\pi} \left[\left(\frac{\delta Q}{Q} \right)^2 + \left(\frac{\delta D_o}{D_o} \right)^2 + \left(\frac{\delta L}{L} \right)^2 \right]^{1/3} \quad (C.5)$$

The following uncertainties were assigned to the variables:

$$\delta D_o = \pm 0.0001 \text{ m}$$

$$\delta L = \pm 0.0001 \text{ m}$$

$$\delta Q = \text{as found from equation (C.4)}$$

E. UNCERTAINTY OF h_{Nu}

$$h_{Nu} = 0.651 \left[\frac{k_f^3 \rho^2 h_{fg} \cdot g}{\mu_f D_o q} \right]^{1/3}$$

The probable uncertainty is given by:

$$\begin{aligned} \frac{\delta h_{Nu}}{h_{Nu}} = 0.651 \left[\left(\frac{k_f}{k_f} \right)^2 + \left(\frac{2}{3} \frac{\delta \rho}{\rho} \right)^2 + \left(\frac{\delta h_{fg}}{3 h_{fg}} \right)^2 + \left(\frac{\delta q}{3 q} \right)^2 + \left(\frac{\delta \mu_f}{3 \mu_f} \right)^2 + \right. \\ \left. + \left(\frac{\delta D_o}{3 D_o} \right)^2 + \left(\frac{\delta q''}{3 q''} \right)^2 \right]^{1/2} \quad (C.6) \end{aligned}$$

The following uncertainties were assigned to the variables:

$$\delta k_f = \pm 0.0012 \text{ w/m} \cdot \text{k}$$

$$\delta \rho = \pm 3 \text{ kg/m}^3$$

$$\delta h_{fg} = \pm 0.48 \text{ J/kg}$$

$$\delta \mu_f = \pm 8 \times 10^{-6} \text{ N} \cdot \text{s/m}^2$$

$$\delta D_o = \pm 0.0001 \text{ m}$$

$$\delta g = \pm 0.001 \text{ m/s}^2$$

$$\delta q'' = \text{as found from equation (C.5)}$$

F. UNCERTAINTY OF T_{film_C}

$$T_{f_C} = T_{\text{sat}} - \frac{q''}{h_{\text{Nu}}} 0.5$$

The probable uncertainty is given by:

$$\frac{\delta T_{f_C}}{T_{f_C}} = \left[\left(\frac{\delta T_{\text{sat}}}{T_{\text{sat}}} \right)^2 + \left(\frac{\delta q''}{q''} \right)^2 + \left(\frac{\delta h_{\text{Nu}}}{h_{\text{Nu}}} \right)^2 \right]^{1/2} \quad (\text{C.7})$$

The following uncertainties were assigned to the variables:

$$\delta T_{\text{sat}} = \pm 0.025 \text{ } ^\circ\text{C}$$

$$\delta q'' = \text{as found from equation (C.5)}$$

$$\delta h_{\text{Nu}} = \text{as found from equation (C.6)}$$

G. UNCERTAINTY OF OVERALL HEAT-TRANSFER COEFFICIENT

$$U_O = \frac{q''}{\text{LMTD}}$$

The probable uncertainty is given by:

$$\frac{\delta U_O}{U_O} = \left[\left(\frac{\delta q''}{q''} \right)^2 + \left(\frac{\delta (\text{LMTD})}{\text{LMTD}} \right)^2 \right]^{1/2} \quad (\text{C.8})$$

The following uncertainties were assigned to the variables:

$$\delta q = \text{as found from equation (C.5)}$$

$$\delta (\text{LMTD}) = \text{as found from equation (C.9)}$$

H. UNCERTAINTY FOR LOGARITHMIC MEAN TEMPERATURE DIFFERENCE (LMTD)

$$\text{LMTD} = \frac{T_{\text{co}} - T_{\text{ci}}}{\ln\left(\frac{T_{\text{s}} - T_{\text{ci}}}{T_{\text{s}} - T_{\text{co}}}\right)}$$

The probable uncertainty is given by:

$$\begin{aligned} \frac{\delta \text{LMTD}}{\text{LMTD}} = & \left[\left(\frac{\delta T_{\text{s}} (T_{\text{ci}} - T_{\text{co}})}{(T_{\text{s}} - T_{\text{ci}})(T_{\text{s}} - T_{\text{co}}) \ln\left(\frac{T_{\text{s}} - T_{\text{ci}}}{T_{\text{s}} - T_{\text{co}}}\right)} \right)^2 + \right. \\ & + \left(\frac{\delta T_{\text{ci}}}{(T_{\text{s}} - T_{\text{ci}}) \ln\left(\frac{T_{\text{s}} - T_{\text{ci}}}{T_{\text{s}} - T_{\text{co}}}\right)} \right)^2 + \\ & \left. + \left(\frac{\delta T_{\text{co}}}{(T_{\text{s}} - T_{\text{co}}) \ln\left(\frac{T_{\text{s}} - T_{\text{ci}}}{T_{\text{s}} - T_{\text{co}}}\right)} \right)^2 \right]^{1/2} \quad (\text{C.9}) \end{aligned}$$

The following uncertainties were assigned to the variables:

$$\delta T_{\text{s}} = \pm 0.025 \text{ } ^\circ\text{C}$$

$$\delta T_{\text{ci}} = \pm 0.025 \text{ } ^\circ\text{C}$$

$$\delta T_{\text{co}} = \pm 0.025 \text{ } ^\circ\text{C}$$

I. UNCERTAINTY OF INSIDE HEAT-TRANSFER COEFFICIENT

$$h_{\text{i}} = \frac{k_{\text{w}}}{D_{\text{i}}} \cdot C_{\text{i}} R_{\text{e}}^{0.8} P_{\text{r}}^{0.333} \left(\frac{\mu}{\mu_{\text{w}}} \right)^{0.14}$$

The probable uncertainty is given by:

$$\frac{\delta h_i}{h_i} = \left[\left(\frac{\delta k_w}{k_w} \right)^2 + \left(\frac{\delta D_i}{D_i} \right)^2 + \left(\frac{0.8 \delta R_e}{R_e} \right)^2 + \left(\frac{0.333 \delta P_r}{P_r} \right)^2 + \left(\frac{\delta C_i}{C_i} \right)^2 + \left(\frac{0.14 \delta (\mu/\mu_m)}{\mu/\mu_m} \right)^2 \right]^{1/2} \quad (C.10)$$

The following uncertainties were applied to the variables:

$$\delta k_w = \pm 0.0012 \text{ w/m} \cdot \text{k}$$

$$\delta D_i = \pm 0.0001 \text{ m}$$

$$\delta C_i = \pm 0.0001$$

$$\delta R_e = \text{as found from equation (C.3)}$$

$$\delta P_r = \pm 0.17$$

$$\delta (\mu/\mu_w) = 8 \times 10^{-6} \text{ Ns/m}^2$$

J. UNCERTAINTY IN TEMPERATURE DIFFERENCE

$$\Delta T = \frac{q''}{h_i} \frac{D_o}{D_i}$$

$$\frac{\delta \Delta T}{\Delta T} = \left[\left(\frac{\delta q''}{q''} \right)^2 + \left(\frac{\delta h_i}{h_i} \right)^2 + \left(\frac{\delta D_o}{D_o} \right)^2 + \left(\frac{\delta D_i}{D_i} \right)^2 \right]^{1/2} \quad (C.11)$$

The following uncertainties were assigned to the variables:

$$\delta q'' = \text{as found from equation (C.5)}$$

$$\delta h_i = \text{as found from equation (C.10)}$$

$$\delta D_o = \pm 0.0001 \text{ m}$$

$$\delta D_i = \pm 0.0001 \text{ m}$$

K. UNCERTAINTY IN OUTSIDE HEAT-TRANSFER COEFFICIENT

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{D_o}{D_i h_i} - R_w}$$

$$\begin{aligned} \frac{\delta h_o}{h_o} = & \left[\left(\frac{\delta U_o}{U_o^2 \left(\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i} \right)} \right)^2 + \right. \\ & + \left(\frac{\delta R_w}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}} \right)^2 \\ & \left. + \left(\frac{\left(\frac{D_o}{D_i h_i} \right) \left(\frac{\delta h_i}{h_i} \right)}{\frac{1}{U_o} - R_w - \frac{D_o}{D_i h_i}} \right)^2 \right]^{1/2} \end{aligned} \quad (C.12)$$

The following uncertainties were assigned to the variables:

δU_o = as found from equation (C.8)

δh_i = as found from equation (C.10)

$\delta R_w = \pm 0.00001 \text{ m}^2 \cdot \text{k/w}$

L. UNCERTAINTIES FOR THE NORMALIZED LOCAL HEAT-TRANSFER COEFFICIENT h_N/h_i

This ratio is simply the heat transfer coefficient of a given tube, N, divided by that of the first tube, i.e., for the fifth tube, N=5 and:

$$\frac{h_N}{h_1} = \frac{h_5}{h_1}$$

An application of equation (C.1) results in the following equation:

$$\frac{\delta(h_N/h_i)}{(h_N/h_i)} = \left[\left(\frac{\delta h_i}{h_i}\right)^2 + \left(\frac{\delta h_N}{h_N}\right)^2 \right]^{1/2} \quad (C.13)$$

Note: Equation (C.13) is valid only for $N \geq 2$. For example:

$$\frac{\delta(h_2/h_1)}{h_2/h_1} = \left[\left(\frac{\delta h_1}{h_1}\right)^2 + \left(\frac{\delta h_2}{h_2}\right)^2 \right]^{1/2}$$

M. UNCERTAINTY IN THE NORMALIZED AVERAGE HEAT-TRANSFER COEFFICIENT, $\bar{h}_N/h_1$

The normalized average heat-transfer coefficient is obtained for the Nth tube by taking the average of the heat-transfer coefficients of the first N tubes and dividing this by the heat-transfer coefficient of the first tube:

$$\frac{\bar{h}_N}{h_1} = \frac{(h_1 + h_2 + \dots + h_N)N}{h_1}$$

Applying equation (C.1) to the above, the following equation results:

$$\frac{\delta(\bar{h}_N/h_1)}{(\bar{h}_N/h_1)} = \left[\sum_{i=1}^N \left(\frac{\delta h_i}{\sum_{i=1}^N h_i} \right)^2 + \left(\frac{\delta h_1}{h_1} \right)^2 \right]^{1/2} \quad (C.14)$$

where N = the tube number. For example:

$$\frac{\delta(\bar{h}_2/h_1)}{(\bar{h}_2/h_1)} = \left[\left(\frac{\delta h_1}{h_1+h_2} \right)^2 + \left(\frac{\delta h_2}{h_1+h_2} \right)^2 + \left(\frac{\delta h_1}{h_1} \right)^2 \right]^{1/2}$$

For Run STNWN1-1

$$V_w = 1.54 \pm 0.11 \text{ m/s}$$

$$Re = 27546 \pm 2121$$

$$Q = 3934 \pm 169 \text{ W}$$

$$q'' = 267830 \pm 11516 \text{ W/m}^2$$

$$h_{Nu} = 10802 \pm 164 \text{ W/m}^2\text{K}$$

$$LMTD = 69.7 \pm 0.6 \text{ }^\circ\text{C}$$

$$U_o = 3843.7 \pm 165 \text{ W/m}^2\text{K}$$

$$h_i = 9381 \pm 122 \text{ W/m}^2\text{K}$$

$$h_o = 11361.4 \pm 1249 \text{ W/m}^2\text{K}$$

$$h_2/h_1 = 0.8891 \pm 0.14$$

$$\bar{h}_2/h_1 = 0.9446 \pm 0.12$$

APPENDIX D

COMPUTER PROGRAMS

A. DATA REDUCTION PROGRAM

```

1000! FILE NAME: DRP
1010! REVISED: May 20, 1983
1020! COM /C1/ C(7)
1030! DIM Tc1(2),Tco(4,2),Ti(4),Mft(4),Vw(4),Ho(4)
1040! DIM To(4),Ts(1),Tb(4),R3(4),R4(4),S3(4),S4(4)
1050!
1060! ASSIGN COEFFICIENTS FOR THE 3-TH ORDER
1070! POLYNOMIAL FOR TYPE-T (COPPER-CONSTANTAN)
1080! THERMOCOUPLES
1090! DATA 0.10086091,25727.94369,-767345.9295,78025595.81
1100! DATA -9247486589.6,97688E+11,-2.66192E+13,3.94078E+14
1110! READ C(*)
1120!
1130! ASSIGN FULL-SCALE FLOW RATES THROUGH THE 5
1140! FLOW METERS (kg/min)
1150! DATA 66.86,73.35,72.44,72.52,72.24
1160! READ Mft(*)
1170!
1180! ASSIGN SIEDER-TATE COEFFICIENT AND EXPONENT
1190! FOR REYNOLDS NUMBER
1200! Ci=.029
1210! Ex=.8
1220!
1230! ASSIGN GEOMETRIC VARIABLES
1240! Di=.0141 ! Inner diameter (m)
1250! Do=.015875 ! Outer diameter (m)
1260! Ktm=21.9 ! Thermal conductivity of titanium (W/m-K)
1270! L=.305 ! Condensing length (m)
1280! Nc=3.3333 ! Number of unit cells across condenser width
1290! Pt=1.5 ! Transverse tube pitch-to-diameter ratio
1300!
1310! COMPUTE THE MINIMUM STEAM FLOW AREA IN THE TEST CONDENSER (m^2)
1320! Amf=Nc*Do*(Pt-PI/(4*Pt))*L
1330!
1340! COMPUTE INSIDE AREA AND WALL RESISTANCE
1350! Ai=PI*Di^2/4
1360! Rw=Do*LOG(Do/Di)/(2*Ktm)
1370!
1380! PRINTER IS 701
1390! CLEAR 709
1400! BEEP
1410! INPUT "ENTER MONTH, DATE, AND TIME (MM:DD:HH:MM:SS)".Times$
1420! OUTPUT 709:"TD":Times$
1430! BEEP
1440! INPUT "ENTER THE INPUT MODE (1=3054A,2=FILE)".Im
1450! IF Im=2 THEN
1460! BEEP
1470! INPUT "ENTER THE NAME OF THE EXISTING DATA FILE".Olddata$
1480! PRINT USING "10X.""This analysis was performed for data stored in file """,
1490! Olddata$
1490! ASSIGN @File2 TO Olddata$
1500! END IF
1510! IF Im=1 THEN
1520! BEEP
1530! INPUT "GIVE A NAME FOR THE DATA FILE TO BE CREATED".Newdata$
1540! CREATE 8DAT Newdata$.20
1550! ASSIGN @File1 TO Newdata$

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1560 END IF
1570 BEEP
1580 INPUT "GIVE A NAME FOR THE OUTPUT FILE",File_out$
1590 BEEP
1600 INPUT "ENTER THE PRESSURE CONDITION (1=ATM,2=VACUUM)",Mp
1610 IF Mp=1 THEN PRINT "          Pressure condition: ATMOSPHERIC"
1620 IF Mp=0 THEN PRINT "          Pressure condition: VACUUM"
1630 BEEP
1640 INPUT "ENTER THE INUNDATION CONDITION (1=5 TUBES, 2=30 TUBES)",Mi
1650 IF Mi=2 THEN PRINT "          Inundation condition: 30 TUBES"
1660 IF Mi=1 THEN PRINT "          Inundation condition: 5 TUBES"
1670 CREATE BDAT File_out$.6
1680 ASSIGN %File3 TO File_out$
1690 Ja=0
1700 Nrun=0
1710 FOR I=0 TO 4
1720 S3(I)=0.
1730 S4(I)=0.
1740 NEXT I
1750 Repeat: !
1760 Nrun=Nrun+1
1770 OUTPUT 709:"TD"
1780 ENTER 709:Time$
1790 PRINT " "
1800 PRINT USING "10X."Month, date, and time: "",15A":Time$
1810 IF Im=2 THEN Rdf
1820 BEEP
1830 INPUT "ENTER FLOW METER READINGS (AS PERCENTAGES)",Fm1,Fm2
1840 IF Nrun MOD 5=1 AND Mi=2 AND Nrun>5 THEN
1850 BEEP
1860 INPUT "ENTER FLOW RATE FOR POROUS TUBE (AS A PERCENT)",Fpt
1870 OUTPUT %File1:Fpt
1880 Mpt=-8.361613+10.076742*Fpt
1890 END IF
1900 DISP "START COLLECTING CONDENSATE"
1910 BEEP
1920 WAIT 20
1930 OUTPUT 709:"AR AFO AL19"
1940 OUTPUT 722:"F1 R1 T1 Z1 FL1"
1950!
1960! READ INLET WATER TEMPERATURES
1970!
1980 FOR I=0 TO 2
1990 OUTPUT 709:"AS SA"
2000 ENTER 722:Tci(I)
2010 CALL Tvsv(Tci(I))
2020 Tci(I)=FNTemp(Tci(I),I)
2030 NEXT I
2040!
2050! READ OUTLET WATER TEMPERATURES
2060!
2070 Ii=2
2080 FOR I=0 TO 4
2090 IF I=0 OR I=3 THEN
2100 Iu=2
2110 ELSE
2120 Iu=1
2130 END IF
2140 FOR J=0 TO Iu
2150 Ii=Ii+1
2160 OUTPUT 709:"AS SA"

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2170 ENTER 722:Tco(I,J)
2180 CALL Tvsv(Tco(I,J))
2190 Tco(I,J)=FNTemp(Tco(I,J),Ii)
2200 NEXT J
2210 NEXT I
2220!
2230! READ STEAM TEMPERATURES
2240!
2250 FOR I=15 TO 16
2260 OUTPUT 709:"AS SA"
2270 ENTER 722:Ts(I-15)
2280 CALL Tvsv(Ts(I-15))
2290 Ts(I-15)=FNTemp(Ts(I-15),I)
2300 NEXT I
2310!
2320! READ CONDENSATE TEMPERATURE
2330!
2340 OUTPUT 709:"AS SA"
2350 ENTER 722:Tcon
2360 CALL Tvsv(Tcon)
2370 Tcon=FNTemp(Tcon,17)
2380!
2390! READ VAPOR TEMPERATURE
2400!
2410 OUTPUT 709:"AS SA"
2420 ENTER 722:Tv
2430 CALL Tvsv(Tv)
2440 Tv=FNTemp(Tv,18)
2450!
2460! READ VAPOR PRESSURE
2470!
2480 OUTPUT 709:"AS SA"
2490 ENTER 722:P_volts
2500!
2510! COMPUTE AVERAGE WATER TEMPERATURES AT INLET
2520!
2530 Ti(0)=Tci(0)
2540 Ti(1)=(Tci(0)+Tci(1))*0.5
2550 Ti(2)=Tci(1)
2560 Ti(3)=(Tci(1)+Tci(2))*0.5
2570 Ti(4)=Tci(2)
2580!
2590! COMPUTE AVERAGE WATER TEMPERATURES AT OUTLET
2600!
2610 FOR I=0 TO 4
2620 IF I=0 OR I=3 THEN
2630 To(I)=(Tco(I,0)+Tco(I,1)+Tco(I,2))*0.3333
2640 ELSE
2650 To(I)=(Tco(I,0)+Tco(I,1))*0.5
2660 END IF
2670 NEXT I
2680 Tsa=(Ts(0)+Ts(1))*0.5
2690 Pvap=FNpvs(P_volts)
2700 Tsat=FNpvs(Pvap)
2710 Dsup=Tv-Tsat
2720!
2730! READ INFORMATION FOR CONDENSATE FLOW RATE
2740!
2750 BEEP
2760 INPUT "ENTER INITIAL AND FINAL LEVELS IN HOT WELL 1".H1,H2

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```

2770 Dh=H2-H1
2780 IF Nrun MOD 5=1 THEN Msum=0
2790 Mf1=540.4836*Dh
2800 Md1=Mf1*FNRhow(Tsat-10)*1.0E-6/60
2810 Msum=Msum+Mf1
2820 IF M1=2 AND Nrun<>30 AND Nrun MOD 5=0 THEN
2830 Mave=Msum/5
2840 Set=(Mave*FNRhow(Tsat-10)/10^6+.03238)/.042132
2850 END IF
2860!
2870 Rdf: !
2880!
2890 PRINT USING "10X,""Run number = "",DD":Nrun
2900 PRINT "      Tube #      :      1      2      3      4      5"
2910 IF Im=2 THEN
2920 IF Nrun MOD 5=1 AND M1=2 AND Nrun>5 THEN ENTER @File2:Fot
2930 ENTER @File2:Ti(*).To(*).Tsa.Tcon.Tv.Pvap.Tsat.Dsup.Fm1.Fm2
2940 ENTER @File2:H1.H2
2950 END IF
2960 IF Im=1 THEN OUTPUT @File1:Ti(*).To(*).Tsa.Tcon.Tv.Pvap.Tsat.Dsup.Fm1.Fm2
2970 PRINT USING "10X,""Inlet temp (Deg C) :","",5(DDD.DD.2X)":Ti(*)
2980 PRINT USING "10X,""Outlet temp (Deg C) :","",5(DDD.DD.2X)":To(*)
2990 PRINT USING "10X,""Saturation temperature =","",3D.DD."" (Deg C)""":Tsat
3000 PRINT USING "10X,""Degree of superheat =","",3D.DD."" (Deg C)""":Dsup
3010 PRINT USING "10X,""Condensate temperature =","",3D.DD."" (Deg C)""":Tcon
3020 PRINT USING "10X,""Static pressure =","",3D.DD."" (mm Hg)""":Pvap
3030 IF Im=1 THEN OUTPUT @File1:H1.H2
3040!
3050! CALCULATE AVERAGE BULK TEMPERATURES
3060!
3070 FOR I=0 TO 4
3080 Tb(I)=(Ti(I)+To(I))*0.5
3090 NEXT I
3100!
3110 IF M1=1 OR (M1=2 AND Nrun<6) THEN As1=0.
3120 IF M1=2 AND Nrun>5 THEN S1=As1
3130 S1=As1
3140 FOR J=0 TO 4
3150 IF J=0 THEN Cwf=Fm1
3160 IF J=1 THEN Cwf=Fm2
3170 Mf=Mft(J)*Cwf/(100*60)
3180 Tx=Tb(J)
3190 Vw(J)=Mf/(FNRhow(Tx)*A1)
3200!
3210! CALCULATE INSIDE AND OUTSIDE COEFFICIENTS
3220!
3230 Rew=FNRhow(Tx)*Vw(J)*Di/FNMuw(Tx)
3240 Cf=1.
3250 Q=Mf*FNCpw(Tx)*(To(J)-Ti(J))
3260 Qp=Q/(PI*Do*L)
3270 IF (M1=1 OR (M1=2 AND Nrun<6)) AND J=0 THEN
3280 Tfilm=Tsats
3290 Kf=FNKw(Tfilm)
3300 Rhof=FNRhow(Tfilm)
3310 Hfg=FNHfg(Tsat)*1000
3320 Muf=FNMuw(Tfilm)
3330 Hnu=.651*(Kf^3*Rhof^2*Hfg*9.81/(Muf*Do*Qp))^0.3333
3340 Tfilmc=Tsats-Qp/Hnu*.5
3350 IF ABS((Tfilmc-Tfilm)/Tfilmc)>.01 THEN
3360 Tfilm=Tfilmc

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```

3370 GOTO 3290
3380 END IF
3390 PRINT USING "10X." "Nusselt coefficient for first tube = ".5D.D." (W/m^2)
      Hnu
3400 END IF
3410 IF J=0 THEN
3420 IF M1=1 AND Nrun=1 THEN Ho1=0.
3430 PRINT "      Tube      Vw      Heat flux      Cond coef      R1      R2
      RR"
3440 PRINT "      #      (m/S)      (W/m^2)      (W/m^2-K)"
3450 END IF
3460 Muw=FNMuw(Tx)
3470 H1=FNKw(Tx)/D1*Ci*Rew*Ex*(FNPrw(Tx))^.3333*CF
3480 Dt=Qp/H1*Do/D1
3490 Cfc=(Muw/(FNMuw(Tx+Dt)))^.14
3500 IF ABS((Cf-Cfc)/Cfc)>.01 THEN
3510 Cf=(Cf+Cfc)*.5
3520 GOTO 3470
3530 END IF
3540 Lmtd=(To(J)-T1(J))/LOG((Tsat-T1(J))/(Tsat-To(J)))
3550 Uo=Qp/Lmtd
3560 Ho(J)=1./(1./Uo-Do/(D1*H1)-Rw)
3570 Rr=Uo/Ho(J)
3580 S1=S1+Ho(J)
3590 IF Nrun MOD 5=1 THEN
3600 IF M1=1 OR (M1=2 AND Nrun=31) THEN Ja=0
3610 IF M1=2 AND 5<Nrun AND Nrun<30 THEN Ja=Nrun-1
3620 IF M1=2 AND 35<Nrun THEN Ja=Nrun-1
3630 END IF
3640 IF M1=1 OR (30<Nrun AND Nrun<36 AND M1=1) OR Nrun<6 THEN
3650 R1=Ho(J)/Ho(0)
3660 R2=S1/((J+1+Ja)*Ho(0))
3670 ELSE
3680 R1=Ho(J)/Ho1
3690 R2=S1/((J+1+Ja)*Ho1)
3700 END IF
3710!
3720! PRINT RESULTS
3730!
3740! PRINT USING "11X.DD,4X.DD.DD,2X.2(D.5DE,2X),3(Z.4D,2X)":J+1+Ja,Vw(J),Qp,Ho
      (J),R1,R2,Rr
3750!
3760!
3770!
3780 IF M1=2 AND Nrun<6 AND J=0 THEN
3790 Ho1=Ho1+Ho(0)/5
3800 END IF
3810 FOR K=0 TO 4
3820 IF K=J THEN S3(K)=S3(K)+R1
3830 IF K=J THEN S4(K)=S4(K)+R2
3840 NEXT K
3850 NEXT J
3860 IF Nrun MOD 5=0 THEN
3870 FOR K=0 TO 4
3880 R3(K)=S3(K)/5
3890 R4(K)=S4(K)/5
3900 S3(K)=0.
3910 S4(K)=0.
3920 NEXT K
3930 IF M1=2 AND Nrun MOD 5=0 AND Nrun<>30 THEN As1=Nrun*R4(4)*Ho1
3940!

```

```

3950! PRINT AVERAGE KH1US
3960!
3970 PRINT " "
3980 PRINT "          Tube #      R3          R4"
3990 FOR J=1 TO 5
4000 PRINT USING "12X,DD.2(4X,Z.4D)":J+Ja,R3(J-1),R4(J-1)
4010 OUTPUT @File3:J+Ja,R3(J-1),R4(J-1)
4020 NEXT J
4030 PRINT " "
4040 END IF
4050 IF Nrun MOD 5=0 AND Mi=2 AND Nrun<>30 AND Im=1 THEN
4060 BEEP
4070 PRINT USING "10X,.""Set porous-tube flowmeter reading to ""',3D.D,."" PERCENT
      ""',Set
4080 END IF
4090!
4100!
4110!
4120 IF Im=1 THEN
4130 BEEP
4140 INPUT "DO YOU HAVE ANY MORE DATA (1=YES,0=NO)?".Go_on
4150 IF Go_on=1 THEN Repeat
4160 ELSE
4170 IF Mi=2 AND Nrun<30 THEN Repeat
4180 IF Mi=1 AND Nrun<10 THEN Repeat
4190 END IF
4200 IF Im=1 THEN PRINT USING "10X,DD,."" Data runs were stored in file ""',10A":
      Nrun,Newdata$
4210 ASSIGN @File1 TO *
4220 ASSIGN @File2 TO *
4230 ASSIGN @File3 TO *
4240 END
4250!
4260! THIS SUBROUTINE CONVERTS THERMOCOUPLE VOLTAGE INTO TEMPERATURE
4270!
4280 SUB TvsV(T)
4290 COM /C1/ C(7)
4300 Sum=0.
4310 FOR I=0 TO 7
4320 Sum=Sum+C(I)*T^I
4330 NEXT I
4340 T=Sum
4350 SUBEND
4360!
4370! THIS FUNCTION CALCULATES PRANDTL NUMBER OF WATER IN THE
4380! RANGE 15 TO 45 DEG C
4390!
4400 DEF FNPw(T)
4410 Y=10^(1.09976605-T*(1.3749326E-2-T*(3.968875E-5-3.45026E-7*T)))
4420 RETURN Y
4430 FNEND
4440!
4450! THIS FUNCTION CALCULATES THERMAL CONDUCTIVITY OF WATER
4460! IN THE RANGE OF 15 TO 105 DEG C
4470!
4480 DEF FNkw(T)
4490 Y=.5625894+T*(2.2964546E-3-T*(1.509766E-5-4.0581652E-8*T))
4500 RETURN Y
4510 FNEND
4520!

```

```

4530! THIS FUNCTION CALCULATES SPECIFIC HEAT OF WATER
4540! IN THE RANGE 15 TO 45 DEG C
4550!
4560 DEF FNCpw(T)
4570 Y=(4.21120858-T*(2.26826E-3-T*(4.42361E-5+2.71428E-7*T)))*1000
4580 RETURN Y
4590 FNEND
4600!
4610! THIS FUNCTION CALCULATES DENSITY OF WATER IN THE
4620! RANGE 15 TO 105 DEG C
4630!
4640 DEF FNRhow(T)
4650 Ro=999.52946+T*(.01269-T*(5.482513E-3-T*1.234147E-5))
4660 RETURN Ro
4670 FNEND
4680!
4690! THIS FUNCTION APPLIES CORRECTIONS TO THERMOCOUPLE READINGS
4700!
4710 DEF FNTemp(T,I)
4720 DIM A(14),B(14)
4730 DATA 0.640533,0.573054,0.593101,0.57298,0.56228,0.567384,0.569577
4740 DATA 0.553951,0.552008,0.566955,0.520998,0.522661,0.531008,0.560783,0.5524
4750 DATA 11.8744,8.63163,9.39412,8.570246,8.299436,8.36677,8.04507,7.459766
4760 DATA 7.498928,7.9408,5.87072,5.391556,6.13399,6.48586,6.326224
4770 READ A(*),B(*)
4780 IF I<15 THEN
4790 T=T-(A(I)-B(I)*.001*T)
4800 ELSE
4810 T=T-.5
4820 END IF
4830 RETURN T
4840 FNEND
4850!
4860! THIS FUNCTION COMPUTES THE SPECIFIC VOLUME OF STEAM
4870!
4880 DEF FNVvst(T)
4890 V=58.4525588-T*(1.51508776-T*(.01372746585-T*4.25366711E-5))
4900 RETURN V
4910 FNEND
4920!
4930! THIS FUNCTION CONVERTS THE VOLTAGE READING OF THE PRESSURE
4940! TRANSDUCER INTO PRESSURE IN MM HG
4950!
4960 DEF FNPvsv(V)
4970 Y=1.1103462+163.36413*V
4980 RETURN Y
4990 FNEND
5000!
5010! THIS FUNCTION CALCULATES THE SATURATION TEMPERATURE OF STEAM AS A FUNCTION
5020! OF PRESSURE
5030!
5040 DEF FNTvsp(P)
5050 IF P<600 THEN
5060 T=31.8776158+P*(.235854929-P*(3.6613664E-4-P*2.41652372E-7))
5070 ELSE
5080 T=59.36562+P*(.07379467-P*(3.15662E-5-P*6.27246E-9))
5090 END IF
5100 RETURN T
5110 FNEND

```

```

5120!
5130! THIS FUNCTION COMPUTES THE VISCOSITY OF WATER
5140!
5150 DEF FNMuw(T)
5160 Mu=1.57609473E-3-T*(3.51198576E-5-T*(3.5835816E-7-1.365586115E-9*T))
5170 RETURN Mu
5180 FNEND
5190!
5200! THIS FUNCTION COMPUTES THE LATENT HEAT OF VAPORIZATION
5210!
5220 DEF FNHfg(T)
5230 Hfg=2497.7389-T*(2.2074+T*(1.7079E-3-2.8593E-6*T))
5240 RETURN Hfg
5250 FNEND

```

B. MODIFIED WILSON-PLOT METHOD

```

1000! *****
1010! FILE NAME: WILSON *
1020! *
1030! THIS PROGRAM COMPUTES THE SIEDER-TATE *
1040! COEFFICIENT FOR FLOW IN TUBES *
1050! *****
1060 COM /Cc/ C(7)
1070 DIM Vw(18),Tcl(18),Tco(18),Ts(18),Md(18),Ta(18)
1080 DATA 0.10086091,25727.94369,-767345.8295,78025595.81
1090 DATA -9247486589.6,97688E+11,-2.66192E+13,3.94078E+14
1100 READ C(*)
1110 PRINTER IS 701
1120 BEEP
1130 CLEAR 709
1140 INPUT "ENTER MONTH, DATE AND TIME (MM:DD:HH:MM:SS)".Bs
1150 OUTPUT 709:"TD":Bs
1160 Di=.0141
1170 Ai=PI*Di^2/4
1180 Do=.015875
1190 L=.305
1200 Km=21.9
1210 Ru=(Do-Di)*Do/(Km*(Do+Di))
1220 Rfi=0.
1230 Series: !
1240 OUTPUT 709:"TD"
1250 ENTER 709:AS
1260 PRINT USING "10X, ""Month, date and time: "",15A":AS
1270 BEEP
1280 INPUT "ENTER INITIAL GUESS FOR SIEDER-TATE COEFFICIENT".Ci
1290 BEEP
1300 INPUT "ENTER EXPONENT FOR REYNOLDS NUMBER".Xn
1310 PRINT USING "10X, ""Initial guess for Sieder-Tate coefficient = "",Z.DD":Ci
1320 PRINT USING "10X, ""Exponent for the Reynolds number = "",Z.DD":Xn
1330 BEEP
1340 INPUT "ENTER THE INPUT MODE (1=3054A,2=FILE)".Im
1350 IF Im=1 THEN
1360 BEEP
1370 INPUT "GIVE A NAME FOR THE DATA FILE".D_files
1380 CREATE BDAT D_files.5
1390 ELSE
1400 BEEP
1410 INPUT "GIVE THE NAME OF THE DATA FILE".D_files
1420 PRINT USING "10X, ""Following analysis was performed for data stored in file "",10A":D_files
1430 BEEP
1440 INPUT "ENTER THE NUMBER OF DATA RUNS STORED".Nrun
1450 END IF
1460 ASSIGN @File TO D_files
1470 BEEP
1480 INPUT "GIVE A NAME FOR PLOTTING DATA FILE".Dplots
1490 CREATE BDAT Dplots.5
1500 ASSIGN @Filep TO Dplots
1510 BEEP
1520 INPUT "ENTER ANALYSIS TYPE (1=HI,2=UI)".It
1530 IF It=1 THEN PRINT USING "10X, ""Analysis type = H1-METHOD""
1540 IF It=2 THEN PRINT USING "10X, ""Analysis type = U1-METHOD""

```

```

1550 K=0
1560 PRINT " "
1570 PRINT "          Data   Vw   Tsat   Ti   To"
1580 PRINT "          #   (m/S)   (C)   (C)   (C)"
1590 Repeat:
1600!
1610! RECORDS THERMOCOUPLE AND PRESSURE TRANSDUCER
1620! READINGS AUTOMATICALLY THROUGH THE HP 3054A
1630! AUTOMATIC DATA ACQUISITION/CONTROL SYSTEM
1640!
1650 IF Im=1 THEN
1660 BEEP
1670 INPUT "ENTER FLOWMETER READING (AS A PERCENT)",Fm
1680 OUTPUT 709:"AR AF0 AL0"
1690 OUTPUT 722:"F1 R1 T1 Z1 FL1"
1700 OUTPUT 709:"AS SA"
1710! READS THERMOCOUPLE FOR WATER INLET
1720 ENTER 722:T(0)
1730 OUTPUT 709:"AR AF3 AL5"
1740 OUTPUT 722:"F1 R1 T1 Z1 FL1"
1750 FOR I=1 TO 3
1760 OUTPUT 709:"AS SA"
1770! READS THREE THERMOCOUPLES FOR WATER OUTLET
1780 ENTER 722:T(I)
1790 NEXT I
1800 OUTPUT 709:"AR AF19 AL19"
1810 OUTPUT 722:"F1 R1 T1 Z1 FL1"
1820 OUTPUT 709:"AS SA"
1830! READS PRESSURE TRANSDUCER
1840 ENTER 722:P_volts
1850 Pvap=FNPvsv(P_volts)
1860 Ts(K)=FNTvsp(Pvap)
1870 Sum=0.
1880 FOR I=0 TO 3
1890! CONVERT VOLTAGE READINGS TO TEMPERATURE
1900 CALL Tvsv(T(I))
1910 IF I=0 THEN
1920 Tci(K)=FNTemp(T(0),0)
1930 ELSE
1940 M=I+2
1950! APPLY THERMOCOUPLE CORRECTIONS
1960 To=FNTemp(T(I),M)
1970 Sum=Sum+To
1980 END IF
1990 NEXT I
2000 Tco(K)=Sum/3.
2010 ELSE
2020 ENTER @File:Ts(K),Tci(K),Tco(K),Fm
2030 END IF
2040 Ta(K)=(Tci(K)+Tco(K))*0.5
2050 Md(K)=66.86*Fm/(100*60)
2060! COMPUTE WATER-SIDE VELOCITY
2070 Vw(K)=Md(K)/(FNRhow(Ta(K))*Ai)
2080 BEEP
2090 IF Im=1 THEN INPUT "ARE YOU TAKING MORE DATA (1=YES,0=NO)?",Go_on
2100 M=K+1
2110 PRINT USING "10X,DD.5(2X,DDD.DD)":M,Vw(K),Ts(K),Tci(K),Tco(K)
2120 IF Im=1 THEN OUTPUT @File:Ts(K),Tci(K),Tco(K),Fm
2130 K=K+1
2140 IF Im=1 THEN

```

```

2150 IF Go_on=1 THEN Repeat
2160 ELSE
2170 IF M<Nrun THEN Repeat
2180 END IF
2190 K=K-1
2200 J=0
2210 Jj=0
2220!
2230! PERFORM ITERATION TO COMPUTE SIEDER-TATE COEFFICIENT
2240!
2250 Ssq=0
2260 Sx=0.
2270 Sy=0.
2280 Sxs=0.
2290 Sxy=0.
2300 J=J+1
2310 PRINT " "
2320 PRINT "          Iteration number = ";J
2330 IF J=1 OR Jj=1 THEN PRINT "          X          1/HI          (TW-TB)          Q
          CF"
2340 FOR I=0 TO K
2350 Q=Md(I)*(Tco(I)-Tci(I))*4180.
2360 Lmtd=(Tco(I)-Tci(I))/LOG((Ts(I)-Tci(I))/(Ts(I)-Tco(I)))
2370 Un=Q/(Lmtd*PI*Do*L)
2380 Unr=1./Un
2390 Rei=FNRRhou(Ta(I))*Vw(I)*Di/FNMu(Ta(I))
2400 Prw=FNPrw(Ta(I))
2410 X=Rei*(-Xn)*Prw*(-.3333)
2420 IF It=2 THEN
2430 Hir=Unr*Di/Do
2440 Kw=FNKw(Ta(I))
2450 GOTO 2660
2460 END IF
2470 IF I=0 THEN
2480 Cf=1.
2490 Kw=FNKw(Ta(I))
2500 Hi=Kw/Di*Ci*Rei^Xn*Prw^.3333*Cf
2510 Dt=Q/(PI*Di*L*Hi)
2520 Cfc=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2530 IF ABS((Cf-Cfc)/Cfc)>.01 THEN
2540 Cf=(Cf+Cfc)/2.
2550 GOTO 2500
2560 END IF
2570 Ho=1./((Unr-Rw-Rfi*Do/Di-Do/(Di*Hi))
2580 Hir=1/Hi
2590 Qo=Q
2600 ELSE
2610! COMPUTE 1/HI
2620 Hir=(Unr-1/Ho*(Q/Qo)^.3333-Rw-Rfi*Do/Di)*Di/Do
2630 Dt=Q/(PI*Di*L*Hir)
2640 Cf=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2650 END IF
2660 IF It=2 THEN
2670 Cf=1.
2680 Hic=Kw*Ci/Di*Rei^Xn*Prw^.3333*Cf
2690 Dt=Q/(PI*Di*L*Hic)
2700 Cfc=(FNMu(Ta(I))/FNMu(Ta(I)+Dt))^.14
2710 IF ABS((Cf-Cfc)/Cfc)>.01 THEN
2720 Cf=(Cf+Cfc)*.5
2730 GOTO 2680

```

```

2740 END IF
2750 END IF
2760 Qp=Q/(PI*Do*L)
2770 X=X/Cf
2780 IF Jj=1 THEN
2790 OUTPUT @Filep:X,Hir
2800 Hirc=A+B*X
2810 Ssq=Ssq+(Hir-Hirc)^2
2820 END IF
2830 IF J=1 OR Jj=1 THEN PRINT USING "10X,Z.7D,3X,Z.7D,4X,DD,DD,2X,D.3DE,X,Z.5D
":X,Hir,Dt,Qp,Cf
2840! COMPUTE COEFFICIENTS FOR LEAST-SQUARES SCHEME
2850 Sx=Sx+X
2860 Sy=Sy+Hir
2870 Sxs=Sxs+X^2
2880 Sxy=Sxy+X*Hir
2890 NEXT I
2900 N=N+1
2910! COMPUTE THE SLOPE OF THE LEAST-SQUARES LINE
2920! DEVELOPED FOR THE WILSON PLOT
2930 B=(N*Sxy-Sy*Sx)/(N*Sxs-Sx*Sx)
2940 C1c=D1/(B*Kw)
2950 PRINT USING "10X, ""Intermediate value of Sieder-Tate coefficient = "" ,Z.4D
":C1c
2960 IF ABS((C1-C1c)/C1c)>.01 THEN
2970 C1=C1c
2980 GOTO 2260
2990 END IF
3000 IF Jj=1 THEN
3010 PRINT USING "10X, ""Sieder-Tate coefficient = "" ,Z.4D":C1c
3020 A=(Sy-B*Sx)/N
3030 PRINT USING "10X, ""Estimated fouling factor = "" ,MZ.5DE, "" (m^2-K/W) "" :A
3040 PRINT " Least-squares line:"
3050 PRINT USING "13X, ""Slope = "" ,Z.5D":B
3060 PRINT USING "13X, ""Intercept = "" ,MZ.5DE":A
3070 ELSE
3080 Jj=1
3090 GOTO 2260
3100 END IF
3110 PRINT USING "10X, ""Sum of squares = "" ,D.5DE":Ssq
3120 ASSIGN @File TO *
3130 ASSIGN @Filep TO *
3140 IF Im=1 THEN
3150 BEEP
3160 PRINT USING "10X, ""NOTE: "" ,DD, "" Data runs were stored in file "" ,SA":M,D
_files
3170 END IF
3180 PRINT USING "10X, ""NOTE: "" ,DD, "" X-Y pairs were stored in file "" ,10A":K+
1,Dplots
3190 BEEP
3200 INPUT "ARE YOU RUNNING ANOTHER SERIES (1=YES,0=NO)?",Go_on
3210 IF Go_on=1 THEN Series
3220 END
3230 DEF FNRhow(I)
3240!
3250! THIS FUNCTION COMPUTES THE DENSITY OF WATER
3250!
3270 Ro=1006.35724-T*(.774489-T*(2.262459E-2-T*3.03304E-4))
3280 RETURN Ro
3290 FNEND

```

```

3300 DEF FNPvsv(V)
3310!
3320! THIS FUNCTION CONVERTS THE PRESSURE TRANSDUCER
3330! READING FROM VOLTS TO PRESSURE IN MM HG
3340!
3350 Y=1.1103462+163.36413*V
3360 RETURN Y
3370 FNEND
3380 DEF FNTvsp(P)
3390!
3400! THIS FUNCTION COMPUTES THE SATURATION TEMPERATURE
3410! CORRESPONDING TO PRESSURE IN MM HG
3420!
3430 IF P<600 THEN
3440 T=31.8776158+P*(.235854929-P*(3.6613664E-4-P*2.41652372E-7))
3450 ELSE
3460 T=59.36562+P*(.0737946/-P*(3.15662E-5-P*6.27246E-9))
3470 END IF
3480 RETURN T
3490 FNEND
3500 DEF FNTemp(T,I)
3510!
3520! THIS FUNCTION APPLIES THERMOCOUPLE CORRECTIONS
3530!
3540 DIM A(5),B(5)
3550 DATA 0.640533,0.573054,0.593101,0.57298,0.567384,0.569577
3560 DATA 11.8744,8.63163,9.39412,8.570246,8.299436,8.36677
3570 READ A(*),B(*)
3580 T=T-(A(I)-B(I)*.001*T)
3590 RETURN T
3600 FNEND
3610 SUB Tvsv(T)
3620 COM /Cc/ C(7)
3630 Sum=0.
3640 FOR I=0 TO 7
3650 Sum=Sum+C(I)*T^I
3660 NEXT I
3670 T=Sum
3680 SUBEND
3690!
3700! THIS FUNCTION COMPUTES PRANDTL NUMBER FOR WATER
3710!
3720 DEF FNPrw(T)
3730 Pr=10*(1.09976605-T*(.013749326-T*(3.968875E-5+3.45026E-7*T)))
3740 RETURN Pr
3750 FNEND
3760 DEF FNMu(T)
3770!
3780! THIS FUNCTION COMPUTES THE VISCOSITY OF WATER
3790!
3800 Mu=1.5087546575E-3-T*(3.025732489E-5-T*(2.626439826E-7-T*8.18601937E-10))
3810 RETURN Mu
3820 FNEND
3830 DEF FNKw(T)
3840!
3850! THIS FUNCTION COMPUTES THE THERMAL CONDUCTIVITY OF WATER
3860!
3870 Kwa=.572183504477+1.52770121209E-3*T
3880 RETURN Kwa
3890 FNEND

```

C. PLOTTING PROGRAM

```

1000! FILE NAME: PLOT
1010 PRINTER IS 705
1020 BEEP
1030 INPUT "ENTER MINIMUM AND MAXIMUM X-VALUES".Xmin.Xmax
1040 BEEP
1050 INPUT "ENTER MINIMUM AND MAXIMUM Y-VALUES".Ymin.Ymax
1060 BEEP
1070 INPUT "ENTER STEP SIZE FOR X-AXIS".Xstep
1080 BEEP
1090 INPUT "ENTER STEP SIZE FOR Y-AXIS".Ystep
1100 BEEP
1110 PRINT "IN:SPI:IP 2300.1800.8300.6800:"
1120 PRINT "SC 0.100,0.100:TL 2,0:"
1130 Sfx=100/(Xmax-Xmin)
1140 Sfy=100/(Ymax-Ymin)
1150 PRINT "PU 0.0 PD"
1160 FOR Xa=Xmin TO Xmax STEP Xstep
1170 X=(Xa-Xmin)*Sfx
1180 PRINT "PA":X,".0: XT:"
1190 NEXT Xa
1200 PRINT "PA 100.0:PU:"
1210 PRINT "PU PA 0.0 PD"
1220 FOR Ya=Ymin TO Ymax STEP Ystep
1230 Y=(Ya-Ymin)*Sfy
1240 PRINT "PA 0.":Y."YT"
1250 NEXT Ya
1260 PRINT "PA 0.100 TL 0 2"
1270 FOR Xa=Xmin TO Xmax STEP Xstep
1280 X=(Xa-Xmin)*Sfx
1290 PRINT "PA":X,".100: XT"
1300 NEXT Xa
1310 PRINT "PA 100.100 PU PA 100.0 PD"
1320 FOR Ya=Ymin TO Ymax STEP Ystep
1330 Y=(Ya-Ymin)*Sfy
1340 PRINT "PD PC 100.":Y."YT"
1350 NEXT Ya
1360 PRINT "PA 100.100 PU"
1370 PRINT "PA 0.-2 SR 1.5,2"
1380 FOR Xa=Xmin TO Xmax STEP Xstep
1390 X=(Xa-Xmin)*Sfx
1400 PRINT "PA":X,".0:"
1410 PRINT "CP -2,-1:LB":Xa:""
1420 NEXT Xa
1430 PRINT "PU PA 0.0"
1440 FOR Ya=Ymin TO Ymax STEP Ystep
1450 Y=(Ya-Ymin)*Sfy
1460 PRINT "PA 0.":Y.""
1470 PRINT "CP -4,-.25:LB":Ya:""
1480 NEXT Ya
1490 BEEP
1500 INPUT "ENTER X-LABEL".Xlabels
1510 BEEP
1520 INPUT "ENTER Y-LABEL".Ylabels
1530 PRINT "SR 1.5,2:PU PA 50,-10 CP":-LEN(Xlabels)/2:"0:LB":Xlabels:""
1540 PRINT "PA -11.50 CP 0.":-LEN(Ylabels)/2*5/6:"DI 0.1:LB":Ylabels:""
1550 PRINT "CP 0.0 DI"

```

```

1560 Repeat:
1570 BEEP
1580 INPUT "DO YOU WANT TO PLOT DATA FROM A FILE (1=YES,0=NO)?",Ok
1590 IF Ok=1 THEN
1600 BEEP
1610 INPUT "ENTER THE NAME OF THE DATA FILE".D_files
1620 ASSIGN @File TO D_files
1630 BEEP
1640 INPUT "ENTER THE BEGINNING RUN NUMBER".Md
1650 BEEP
1660 INPUT "ENTER THE NUMBER OF X-Y PAIRS STORED".Npairs
1670 BEEP
1680 INPUT "SELECT A SYMBOL FOR THE PLOTTER (1=+,2=*,3=c,4=o,5=.)",Sy
1690 PRINT "PU DI"
1700 IF Sy=1 THEN PRINT "SM+"
1710 IF Sy=2 THEN PRINT "SM*"
1720 IF Sy=3 THEN PRINT "SMc"
1730 IF Sy=4 THEN PRINT "SMo"
1740 IF Sy=5 THEN PRINT "SM."
1750 BEEP
1760 INPUT "SELECT MODE (1=HN/H1,2=HN(avg)/H1)",Jm
1770 IF Md>Npairs THEN
1780 FOR I=1 TO (Md-1)
1790 IF Jm=1 THEN ENTER @File:Xa,Ya,Yy
1800 IF Jm=2 THEN ENTER @File:Xa,Yy,Ya
1810 NEXT I
1820 END IF
1830 FOR I=1 TO Npairs
1840 IF Jm=1 THEN ENTER @File:Xa,Ya,Yy
1850 IF Jm=2 THEN ENTER @File:Xa,Yy,Ya
1860 X=(Xa-Xmin)*Sfx
1870 Y=(Ya-Ymin)*Sfy
1880 PRINT "PA".X,Y,""
1890 NEXT I
1900 BEEP
1910 ASSIGN @File TO *
1920 INPUT "DO YOU HAVE MORE DATA TO BE PLOTTED (1=YES,0=NO)?",Go_on
1930 IF Go_on=1 THEN Repeat
1940 END IF
1950 BEEP
1960 INPUT "DO YOU LIKE TO PLOT THE NUSSELT RELATION (1=YES,0=NO)?",Go_on
1970 PRINT "PU:SM"
1980 IF Go_on=1 THEN
1990 FOR Xa=Xmin TO Xmax STEP Xstep/50
2000 X=(Xa-Xmin)*Sfx
2010 IF Jm=1 AND Xa>Xmin THEN Ya=Xa.75-(Xa-1).75
2020 IF Jm=2 AND Xa>Xmin THEN Ya=Xa(-.25)
2030 IF Xa=Xmin THEN Ya=1
2040 Y=(Ya-Ymin)*Sfy
2050 PRINT "PA".X,Y,"PD"
2060 NEXT Xa
2070 BEEP
2080 PRINT "PU"
2090 INPUT "MOVE THE PEN TO LABEL THE NUSSELT LINE".Ok
2100 PRINT "LBNusselt"
2110 END IF
2120 BEEP
2130 INPUT "DO YOU LIKE TO PLOT KERN RELATIONSHIP?".Yes
2140 IF Yes=1 THEN
2150 FOR Xa=Xmin TO Xmax STEP Xstep/20

```

```

2160 Ya=Xa-1/5
2170 X=(Xa-Xmin)*Sfx
2180 Y=(Ya-Ymin)*Sfy
2190 PRINT "PA",X,Y,"PD"
2200 NEXT Xa
2210 PRINT "PU"
2220 BEEP
2230 INPUT "MOVE THE PEN TO LABEL KERN RELATIONSHIP".Ok
2240 PRINT "LBKern:PU"
2250 END IF
2260 PRINT "PU PA 0.0"
2270 BEEP
2280 INPUT "OK TO PLOT CURVE FOR NEW DESIGN (1=OK,0=NO)?".Ok
2290 IF Ok=1 THEN
2300 FOR Xa=Xmin TO Xmax STEP Xstep/2
2310 IF Xa=1 THEN Ya=1
2320 IF Xa>1 THEN Ya=((Xa*5).75-(Xa*5-1).75+5)/6
2330 X=(Xa-Xmin)*Sfx
2340 Y=(Ya-Ymin)*Sfy
2350 PRINT "PA",X,Y,"PD"
2360 NEXT Xa
2370 PRINT "PU"
2380 END IF
2390 BEEP
2400 INPUT "DO YOU LIKE TO PLOT EISSENBERG RELATION?".Go_on
2410 IF Go_on=1 THEN
2420 FOR Xa=Xmin TO Xmax STEP Xstep/10
2430 Ya=.6+.42*Xa(-.25)
2440 X=(Xa-Xmin)*Sfx
2450 Y=(Ya-Ymin)*Sfy
2460 PRINT "PA",X,Y,"PD"
2470 NEXT Xa
2480 PRINT "PU"
2490 BEEP
2500 INPUT "MOVE THE PEN TO LABEL THE EISSENBERG LINE".Ok
2510 PRINT "LBEissenberg:PU"
2520 END IF
2530 PRINT "PU"
2540 BEEP
2550 INPUT "DO YOU LIKE TO PLOT THE EXPTL CURVE (1=Y,0=NO)?".Go_on
2560 IF Go_on=1 THEN
2570 BEEP
2580 INPUT "ENTER THE EXPONENT".Ex
2590 FOR Xa=Xmin TO Xmax STEP Xstep/10
2600 Ya=Xa(-Ex)
2610 X=(Xa-Xmin)*Sfx
2620 Y=(Ya-Ymin)*Sfy
2630 PRINT "PA",X,Y,"PD"
2640 NEXT Xa
2650 PRINT "PU"
2660 BEEP
2670 INPUT "MOVE THE PEN TO LABEL THE EXPTL CURVE".Ok
2680 PRINT "LBHN(avg)/H1=N"
2690 PRINT "PR 1.1"
2700 PRINT "LB":-Ex;"
2710 END IF
2720 BEEP
2730 INPUT "DO YOU LIKE TO DRAW A STRAIGHT LINE?".Go_on
2740 IF Go_on=1 THEN
2750 BEEP
2760 INPUT "ENTER THE SLOPE".S1

```

```

2770 BEEP
2780 INPUT "ENTER THE INTERCEPT",Ac
2790 FOR Xa=Xmin TO Xmax STEP (Xmax-Xmin)
2800 Ya=Ac+S1*Xa
2810 Y=(Ya-Ymin)*Sfy
2820 X=(Xa-Xmin)*Sfx
2830 IF Y<0 THEN
2840 Xam=(Ymin-Ac)/S1
2850 X=(Xam-Xmin)*Sfx
2860 Y=0
2870 END IF
2880 IF Y>100 THEN
2890 Xam=(Ymax-Ac)/S1
2900 X=(Xam-Xmin)*Sfx
2910 Y=100
2920 END IF
2930 PRINT "PA".X.Y."PD"
2940 NEXT Xa
2950 END IF
2960 PRINT "PU SP0"
2970 END

```

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