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The influence of surface characteristics on single phase liquid heat transfer

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THE INFLUENCE OF SURFACE CHARACTERISTICS ON SINGLE PHASE LIQUID HEAT TRANSFER

by

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ABSTRACT

A single tube steam condenser with water flowing inside was used to study inside heat transfer coefficients in tubes coated internally with Teflon, Paralene, Gold and Epoxy. These results are compared with those obtained for similar uncoated tubes. 90-10 Copper Nickel alloy, 18 BWG tubes with 7/8" O.D. were used. Linings were essentially thin - of the order of 1-2 mils thickness or less.

Teflon alone produced an improvement in the inside heat transfer coefficient. This effect was not, however, permanent and after a running time of 40-50 hours, the inside coefficient was similar to an unlined tube. A wetting agent added to the process water produced a similar decay very rapidly suggesting that the improvement in heat transfer was the result of a surface effect and also that the surface properties of Teflon were altered gradually in the conditions under which the experiment was carried out. An examination of Teflon surface at the end of 40-50 hours immersion in process water showed the distinct formation of a thin scale which considerably altered surface properties such as contact angle.

Fanning friction factors are also reported for flow in an unlined and a Teflon lined 7/8" O.D., 18 BWG, 90-10 Copper Nickel alloy tube. Pressure drop measurements were made over 6' sections of the tubes for water velocities ranging from 1.5 ft/sec to 13.2 ft/sec; the same velocity range used for the heat transfer measurements. Results indicate that the change in friction factors between the Teflon lined and the unlined tube could be attributed almost entirely to difference in surface roughness suggesting that axial slip at the wall of the conduit does not exist to any significant extent in non-wetted flow.

Heat transfer measurements indicated that a definite improvement in inside heat transfer coefficient occurred. Since a major portion of the resistance to heat transfer lies in the laminar sub layer, it is conceivable that this is in some way altered, possibly by deeper penetration of eddies into the laminar sublayer and perhaps even by interchange of material at the surface with water from the core.

INTRODUCTION

Studies of the rate of heat transfer from steam to cooling water flowing inside metal tubes have been fairly extensive. Improvements in the rate of heat transfer would mean a reduction in the size and hence cost of condensers in power plants and proposed desalination plants. Efforts to make such improvements require consideration of the detailed transfer mechanisms involved.

The total resistance to heat transfer may be broken up into the outside condensing film resistance, the conductive resistance of the wall and any scale present, and the inside film resistance. At normal water flow rates, more than half of the total heat transfer resistance resides in the inside heat transfer coefficient.

The present study examines the possibility of lowering the inside heat transfer resistance by using a thin coating of hydrophobic material on the inside of a copper alloy tube.

Though the influence of surface wetting has been investigated extensively for liquid metal heat transfer, similar work with aqueous systems has been limited.

Non wetting with liquid metals have been found by several investigators [2] [3] [4] to decrease heat transfer. These results however, cannot be extended to aqueous systems because the mechanisms of heat transfer differ in some important aspects. A significant portion of the heat in liquid metal systems is transfered by electron and molecular conduction. Gas entrainment is thought to be largely responsible for any decrease in non wetted liquid metal heat transfer.

The presence of small gas bubbles in water will not ordinarily affect heat transfer in turbulent flow to any great extent because high Prandtl Number heat transfer depends almost entirely on convective or eddy transport. In such a system, the flow patterns close to the wall where a significant portion of the heat transfer resistance lies will be of greater importance. A non wetting surface could affect the flow of water near the wall, hence changing the nature of the laminar sublayer and yielding a different heat transfer coefficient.

The performance of aluminum condenser tubes lined internally with Teflon was investigated by Kremer[1]. His report indicated an increase in the inside heat transfer coefficient. His investigation did not, however, consider the effect of prolonged exposure of the Teflon surface. The present study investigates the behavior of Teflon over a longer period of time. It also evaluates the performance of tubes lined with other hydrophobic materials, Epoxy, Paralene-N, and Gold.

THEORETICAL BACKGROUND

A. Wilson Plot Theory

The rate of heat transfer from steam condensing on the outside of a horizontal tube to water flowing inside is given by:

$$q = h_o A_o (T_s - T_{wo}) = \frac{k_w A_m}{x_w} (T_{wo} - T_{wi})$$

$$= h_i A_i (T_{wi} - T_i) \quad \ldots \ldots \quad (1)$$

Equations for the heat flow through the outside film, the tube wall and the inside film may be written

$$T_s - T_{wo} = \frac{q}{h_o A_o} \qquad T_{wo} - T_{wi} = \frac{q\, x_w}{k_w A_m}$$

$$T_{wi} - T_i = \frac{q}{h_i A_i}$$

From which

$$\Delta T = T_s - T_i = q \left[\frac{1}{h_o A_o} + \frac{x_w}{k_w A_m} + \frac{1}{h_i A_i} \right] \quad \ldots . \quad (2)$$

Defining an overall coefficient of heat transfer, U_o, based on A_o such that

$$q = U_o A_o \Delta T$$

$$\frac{q}{U_o A_o} = \Delta T = q \left[\frac{1}{h_o A_o} + \frac{x_w}{k_w A_m} + \frac{1}{h_i A_i} \right]$$

giving

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{x_w A_o}{k_w A_m} + \frac{A_o}{h_i A_i} \quad \ldots \ldots \ldots \ldots \quad (3)$$

If the tube is lined, the additional resistance of the lining must be included. If an outside scale resistance is also included Equation 3 becomes

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{x_w}{k_w}\frac{A_o}{A_m} + \frac{x_L A_o}{k_L A_i} + R_i + \frac{1}{h_i}\frac{A_o}{A_i} \quad \ldots\ldots (4)$$

If the inside flow is turbulent, the condensate film resistance is almost independent of water velocity and the second, third and fourth terms on the right hand side of Equation 4 may be considered constant.

The Dittus-Boelter Equations for the inside coefficient is

$$h_i = a.\left(\frac{k}{D}\right) \cdot \left(\frac{\rho v D}{\mu}\right)^{0.8} \cdot \left(\frac{C_p \mu}{k}\right)^{0.4} \quad \ldots\ldots\ldots\ldots (5)$$

where

a = 0.024 for clean tubes.

The water velocity v is the only quantity in Equation 4 that changes significantly, hence it can be rewritten as

$$\frac{1}{U_o} = B + \left[\frac{A_o/A_i}{a\left(\frac{k}{D}\right)\left(\frac{\rho D}{\mu}\right)^{0.8}\left(\frac{C_p\mu}{k}\right)^{0.4}}\right]\frac{1}{v^{0.8}} \quad \ldots\ldots (6)$$

or $$\frac{1}{U_o} = B + \frac{C}{v^{0.8}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (7)$$

where $$B = \frac{1}{h_o} + \frac{x_w}{k_w}\frac{A_o}{A_m} + \frac{x_L}{k_L}\frac{A_o}{A_i} + R_i \quad \ldots\ldots\ldots (8)$$

Thus $\frac{1}{U_o}$ plotted against $\frac{1}{V^{0.8}}$ should yield a straight line with slope C and intercept B.

Since the fluid is water at 80°F and the condenser tubes are of fixed geometry, C = .0000845/a. For the uncoated tube, the intercept becomes

$$B_u = \frac{1}{h_o} + \frac{x_w}{k_w}\frac{A_o}{A_m} + R_i \quad \ldots\ldots\ldots\ldots\ldots \quad (9)$$

Thus comparison of intercepts between lined and unlined tubes, B and B_u, may be used to obtain coating charactis-tics.

B. Calculation of U_o

With steam condensing on the outside surface of the tube at a known temperature, measurement of the flowing water temperature at the inlet and outlet of the tube, and the water flow rate, all taken at equilibrium conditions enable calculations of the overall heat transfer coefficient by the following relationship

$$q = U_o A_o \Delta T_{lm} = w\ C_p (T_2 - T_1) \qquad (10)$$

$$U_o = \frac{w\ C_p (T_2 - T_1)}{A_o \dfrac{(T_s - T_1) - (T_s - T_2)}{\ln\ (T_s - T_1)/(T_s - T_2)}}$$

$$\text{or } U_o = \frac{w\ C_p}{A_o} \ln \frac{(T_s - T_1)}{(T_s - T_2)} \qquad (11)$$

DESCRIPTION OF APPARATUS

A. Heat Transfer Apparatus

An equipment flow sheet is shown in Figure 1. The arrangement may be described in terms of two flow systems, one for water and the other for steam.

Water is pumped from a 55 gallon holding tank by a 25 gpm centrifugal pump to the test tube. Flow is regulated manually by a control valve upstream from the calibrated rotometer. Water then enters the calorimeter where baffles direct it past the inlet mercury thermometer and into the tube being tested. The water, heated in its passage through the calorimeter steam chest, is directed past the mercury outlet thermometer and through the calorimeter exit. This stream is now mixed with cold city water. Mixing ratios are automatically controlled so that the temperature in the holding tank is maintained at a constant level of 80°F (± 1°F).

Steam is generated in an evaporator with 150 p.s.i.g. heating steam. The 150 p.s.i.g. steam passes through reducing valves before entering the evaporator to provide heat for the generation of 100°F saturated steam. The low-pressure steam flows through an 8" pipe to the

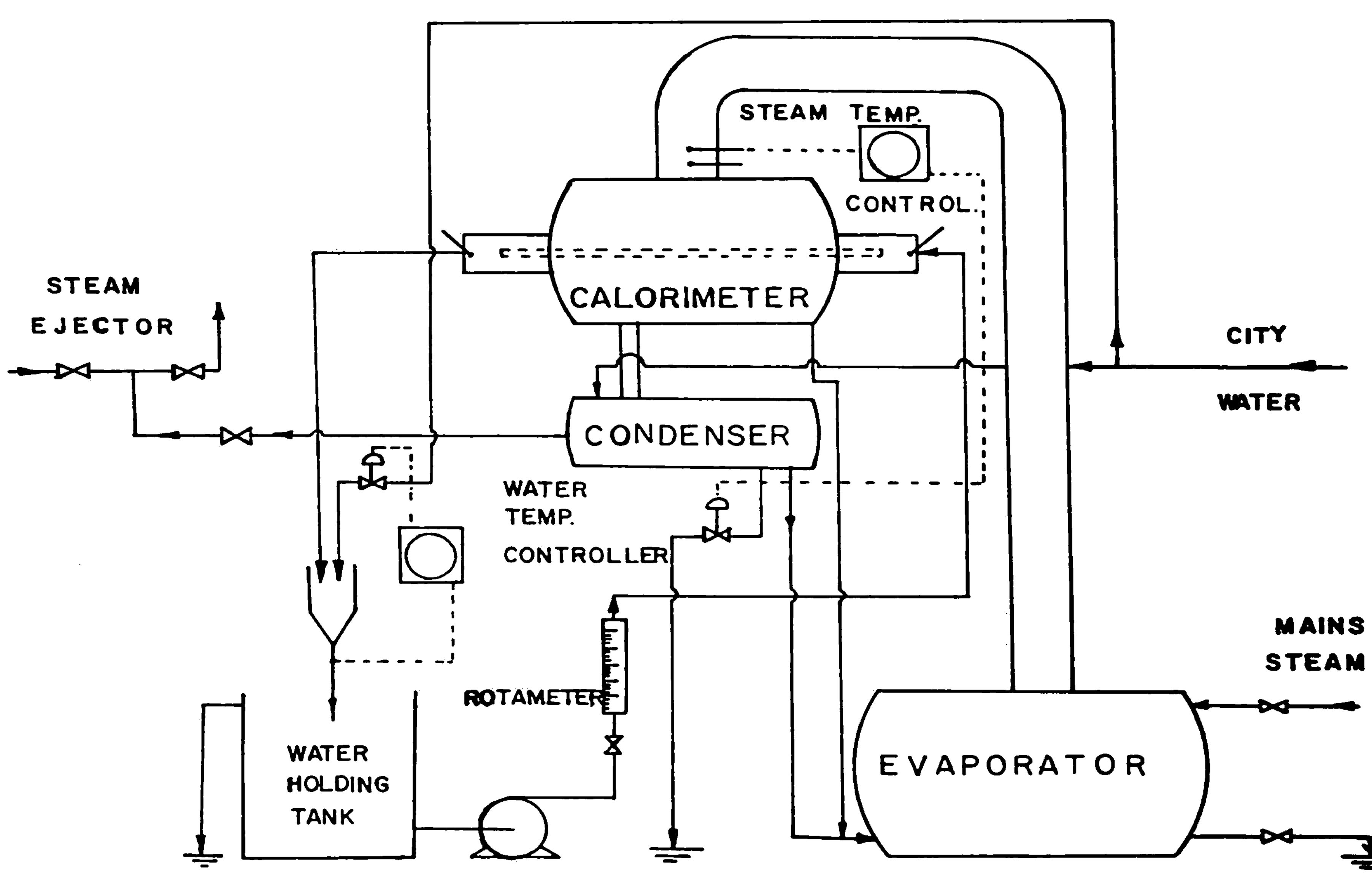


FIGURE I

FLOW DIAGRAM FOR HEAT TRANSFER APPARATUS

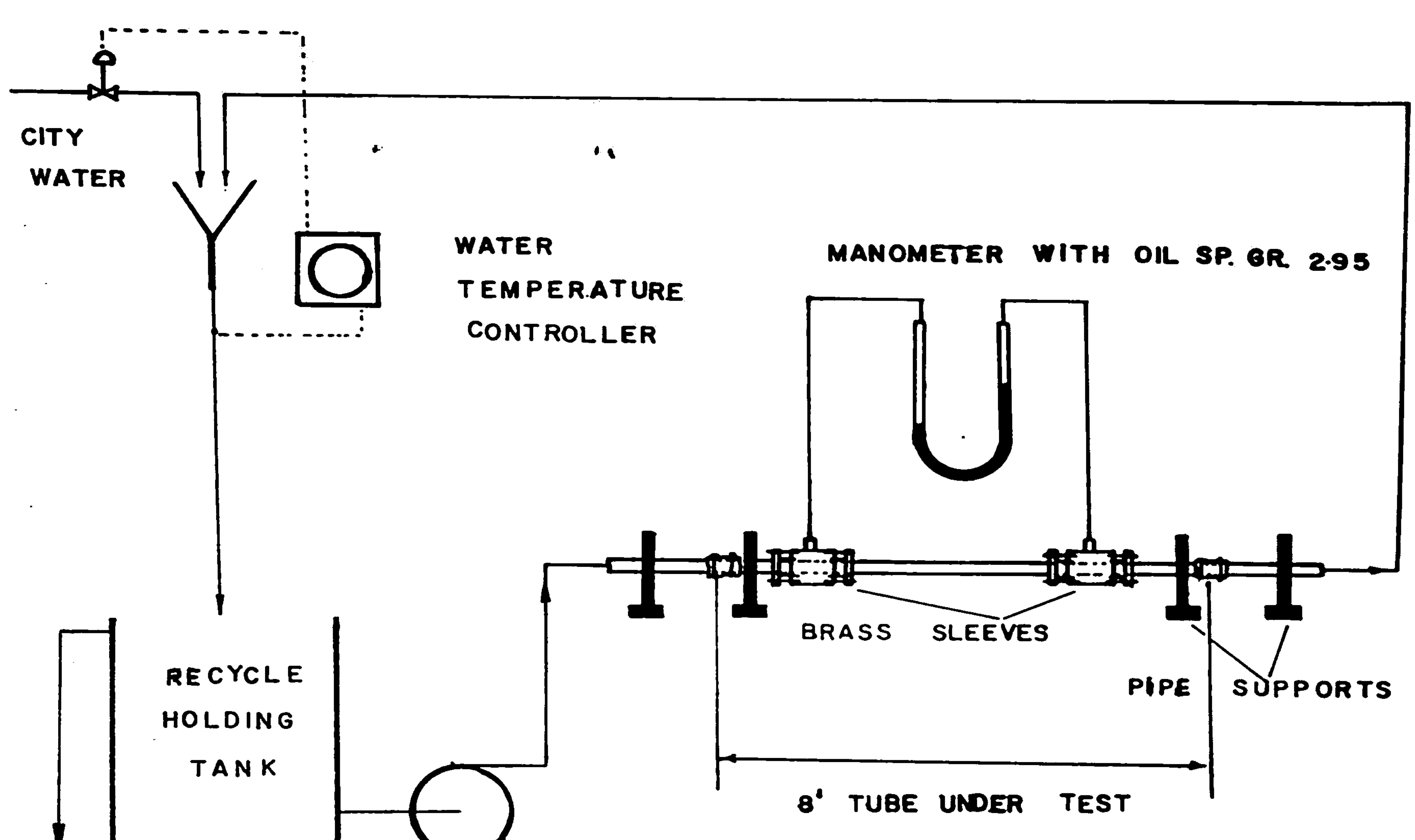


FIGURE 2
PRESSURE DROP APPARATUS

FIGURE 3

HEAT TRANSFER APPARATUS

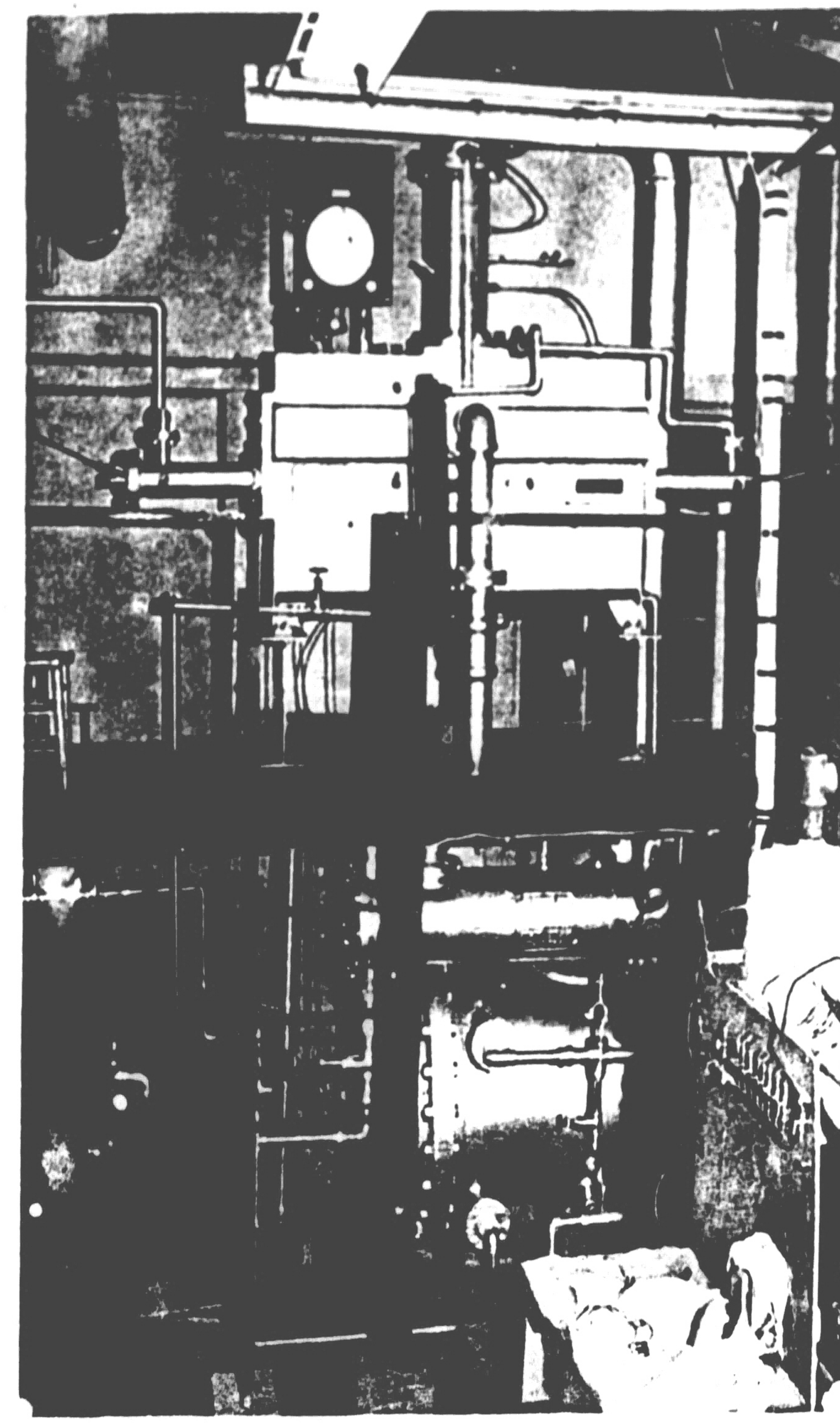
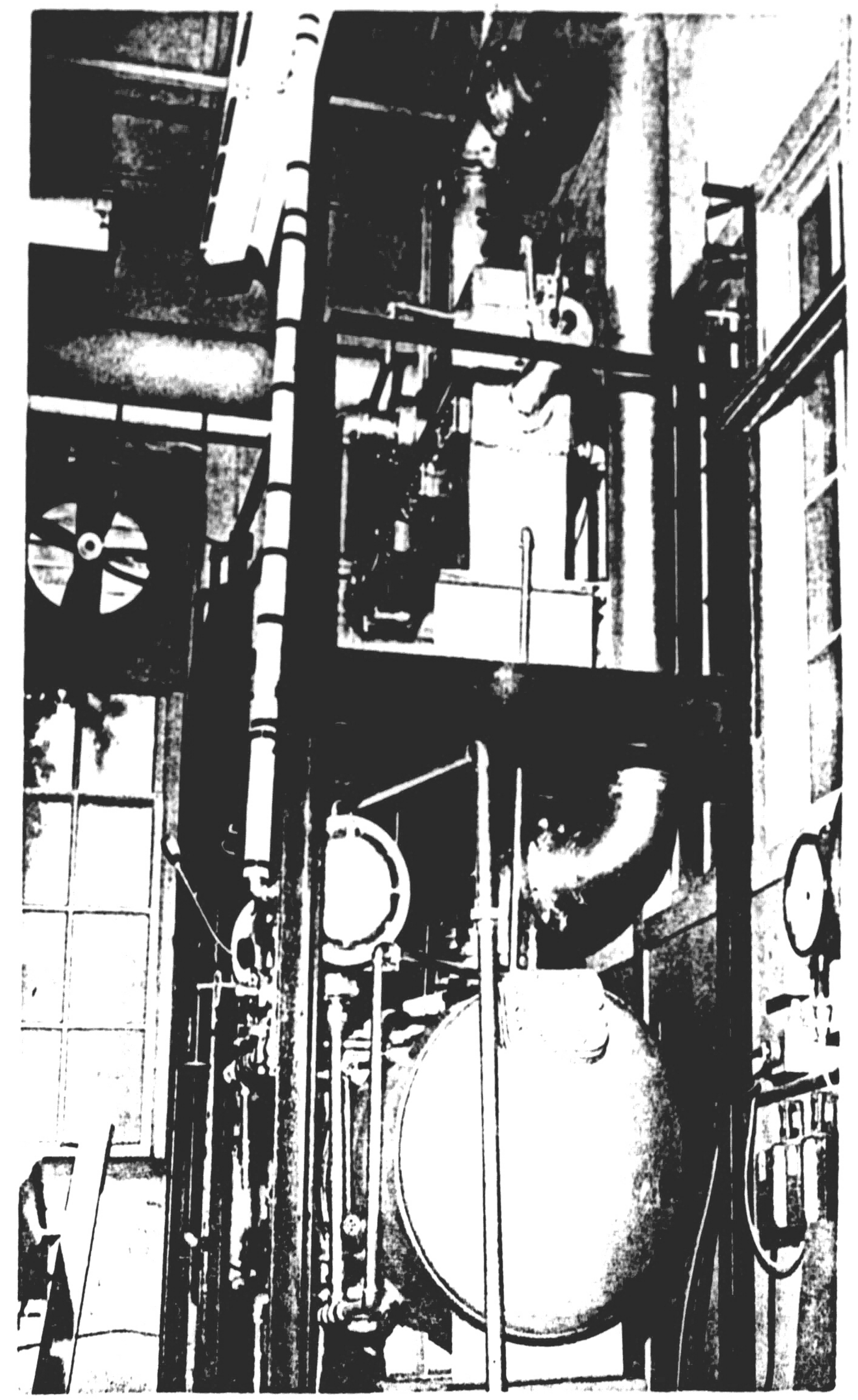


FIGURE 4

HEAT TRANSFER APPARATUS



calorimeter where baffles channel the steam over the tube being tested. Steam velocity past the tube is maintained at approximately 120ft/sec. Steam temperature is measured by a mercury thermometer located in the 8" pipe at its entrance to the calorimeter. Uncondensed steam goes to a shell and tube condenser and the condensate returns to the evaporator. Non-condensables are removed from the system by a 2-stage steam ejector which maintains the entire steam system at a pressure of approximately 1 p.s.i.a.

Temperature of the water at the inlet and outlet of the calorimeter and of the steam are measured with calorimeter grade mercury-in-glass thermometers with 0.1°F minimum divisions. Temperatures can be measured to a precision of about $\pm$ .02°F.

The steam temperature and the inlet water temperature are controlled automatically using a Minneapolis-Honeywell dual two-mode pneumatic recorder controller. The steam temperature sensing element is located in the 8" steam pipe at the entry to the calorimeter. Temperature control is achieved by regulating the flow of cooling water to the condenser.

The sensing element for the water system is located shortly after the exit from the calorimeter but prior to entry to the holding tank. The addition of cold city water is used to maintain the desired temperature.

B. Pressure Drop Apparatus

The pressure drop apparatus is shown in Figure 2.

Water was circulated using a centrifugal pump from a recycle storage tank through hoses to the test section. Water in the recycle storage tank was maintained at 80°F by the automatic addition of cold city water. Flow rate was measured by a calibrated rotometer.

One foot lengths of a 90-10 Copper Nickel alloy tube were attached to the 8 foot tube being tested at the entry and exit to minimize end disturbances. ¼ inch holes were drilled one foot from each end of the test section and brass couplings were slipped over these holes.

A water tight seal was effected by tightening down on the Teflon - Asbestos packing glands at each end of the coupling.

A hole drilled into each coupling and fitted with a nipple was placed over the hole in the test tube which was connected by Tygon tubing to one arm of a manometer filled with meriam oil of Specific Gravity 2.95.

PROCEDURE

A. Heat Transfer Measurements

Tubes tested were 7/8" OD x 18 BWG, 90-10 Copper Nickel alloy. Both lined and unlined tubes were treated identically. The tube to be tested was cleaned on the inside with a soft cloth dipped in trichlorethylene while the outer surface was cleaned with emery paper and steel wool till a polished and smooth appearance was obtained. The tube was washed with water, inserted into the calorimeter and the packing glands tightened. The calorimeter end section was bolted on and the steam ejector and water pump were started to check the seals. An ineffective seal was easily detected by water leaking into the steam chest.

The Air Compressor supplying air for the controller and control valves was started and this was followed by opening fully the valve supplying heating steam to the evaporator.

A start up time of 3-4 hours was allowed to purge the system of non condensable gases and to achieve steady state conditions.

The steam temperature and water inlet and outlet temperatures were measured at frequent intervals over a period of fifteen minutes. When successive readings showed good agreement thus indicating steady state had been reached these readings were recorded together with the rotometer readings.

The water flow rate was reset and a period of an hour allowed for the system to come to equilibrium before further measurements. This was continued until a sufficient number of data points in the 1.5ft/sec to 13.2ft/sec range were obtained.

B. Pressure Drop Measurements

The tube was mounted on a preconstructed support to prevent any movement during data taking, and the necessary hoses were connected. The brass sleeves were fitted into position with the hole in the sleeve aligned properly over the hole in the tube. Packing glands were tightened by bolts on both sides of the sleeve. The water pump was started and the packing checked for leaks. Air bubbles were purged and this was done carefully to prevent any displacement of oil from the manometer into the Tygon tubing.

When the water in the recycle storage drum reached 80°F measurements were started. Pressure drop was measured at flow rates ranging from 1.5ft/sec to 13.2 ft/sec.

The system was very sensitive to the presence of even small air bubbles. Detection of air bubbles was fairly easy with the transparent tygon and the system was pu..ged at frequent intervals. Nevertheless, flow rate readings were duplicated to ensure accuracy. Further, the tube with the brass sleeves was turned end for end to check that pressure drop was not influenced by burrs at the pressure taps. The mean of the pressure drop readings was used for any one flow rate.

DISCUSSION OF RESULTS

A. Pressure Drop Measurements

Tables I and II present pressure drop measurements for different water velocities in terms of cm. of meriam oil (Sp. Gr. 2.95) for the unlined and Teflon lined tubes respectively. Each value is the mean of three individual readings and set I differs from set II in that the direction of flow was reversed.

The Teflon lined tube shows a higher pressure drop throughout the flow range increasing from a 3% difference at the low velocity of 1.5ft/sec to 11% at the high velocity of 13.2ft/sec.

The following may be considered as possible major factors contributing to the difference in pressure drop between the unlined and lined tubes.

(a) Differences in internal pipe diameter.

(b) Differences in internal surface roughness.

(c) Differences in velocity gradient within the two tubes.

(d) The Blassius Equation generally applicable to turbulent flow at Reynold's Numbers up to 10^5 gives:

$$\frac{dP}{dz} = \frac{0.158\,\rho\,\bar{v}^2}{g_c D}\left(\frac{D\bar{v}\rho}{\mu}\right)^{-0.25}$$

At a given volumetric flow rate, a change in diameter would affect the velocity thus,

$$\bar{v} = \frac{Q}{D^2}$$

or $\bar{v} \propto \frac{1}{D^2}$ for a fixed Q

Therefore for a fixed Q,

$$\frac{dP}{dz} \propto \frac{1}{D^5}\left(\frac{1}{D}\right)^{-0.25}$$

or $\frac{dP}{dz} \propto \frac{1}{D^{4.75}}$

For the same volumetric flow or the same nominal velocity, the ratio of pressure drops in pipes of different diameter is

$$\frac{\Delta P_2}{\Delta P_1} = \left(\frac{D_1}{D_2}\right)^{4.75}$$

The Teflon coating of thickness .0005 in. - .001 in. will reduce the pipe diameter at the most by .002 in. With a tolerance in pipe diameter of .002 in, the internal diameter at worst will be 0.777 - 0.773 in.

Thus the effect on pressure drop will be

$$\frac{\Delta P_2}{\Delta P_1} = \frac{0.777}{0.773}^{4.75} = 1.0247$$

TABLE I

Pressure Drop Measurements for Unlined Tube

Water Temperature 80°F ± 0.3°F
Tube Specifications 90-10 Cu-Ni, 7/8" O.D. x 18 BWG

Water velocity ft/sec	Reynold's Number ($Dv\rho/\mu$)	Pressure Drop Reading cm. of meriam oil (sp.gr. 2.95)/6ft			Friction Factor f
		1 at Inlet End	2 at Inlet End	Mean	
1.5	1.04×10^4	1.67	1.65	1.66	0.033
2.7	1.89×10^4	4.68	4.75	4.72	0.0290
4.05	2.83×10^4	9.27	8.87	9.07	0.0247
5.4	3.77×10^4	15.30	14.48	14.89	0.0229
6.7	4.68×10^4	22.85	20.92	21.89	0.0218
8.0	5.59×10^4	31.87	30.07	30.97	0.0216
9.5	6.69×10^4	42.70	40.04	41.37	0.0205
10.9	7.61×10^4	52.75	51.40	52.53	0.0198
12.5	8.74×10^4	68.77	64.10	66.44	0.0190
13.2	9.20×10^4	78.77	73.30	74.51	0.0191

TABLE II

Pressure Drop Measurements for Teflon Lined Tube

Water Temperature $80^{0}F \pm 0.3^{o}F$
Tube Specifications 90-10 Cu-Ni, 7/8" O.D. x 18 BWG
Teflon Lining : Approximately .0005" thick

Water velocity ft/sec	Reynold's Number $(D\rho v/\mu)$	Pressure Drop Reading cm. of meriam oil (sp.gr. 2.95)/6ft			Friction Factor f
		1 at Inlet End	2 at Inlet End	Mean	
1.5	1.04×10^4	1.70	1.72	1.71	0.034
2.7	1.89×10^4	5.01	5.00	5.0	0.0306
4.05	2.83×10^4	9.71	9.61	9.69	0.0263
5.4	3.77×10^4	16.03	15.90	15.97	0.02455
6.7	4.68×10^4	24.25	23.95	24.10	0.02395
8.0	5.59×10^4	34.10	33.60	33.85	0.02365
9.5	6.69×10^4	45.40	44.67	45.10	0.0224
10.9	7.61×10^4	59.10	58.30	58.70	0.02218
12.5	8.74×10^4	74.35	73.30	73.78	0.0211
13.2	9.20×10^4	85.20	83.70	84.45	0.0217

FIGURE 5

PRESSURE DROP FOR WATER FLOW IN

7/8" OD × 18 BWG 90-10 COPPER-NICKEL

CONDENSER TUBES

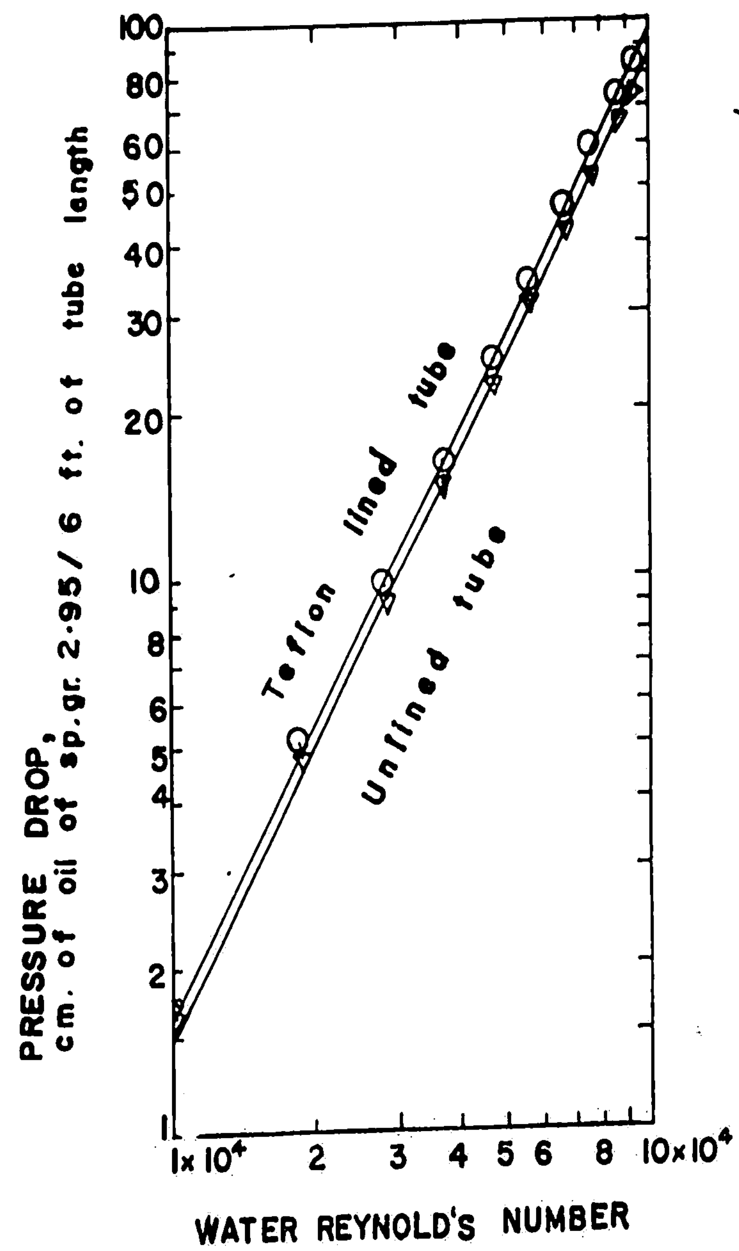
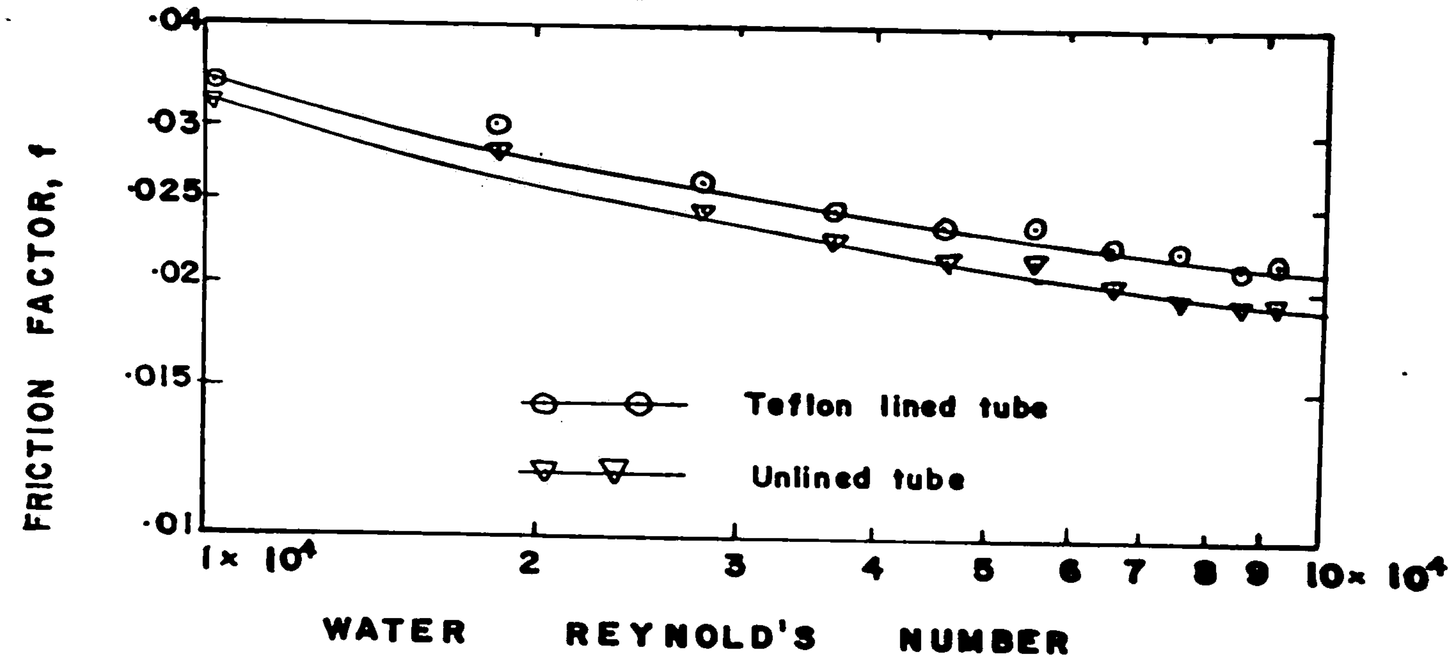


FIGURE 6

FRICTION FACTOR versus REYNOLD'S NUMBER
FOR WATER FLOW IN 7/8" OD × 18 BWG
90-10 COPPER-NICKEL CONDENSER TUBES



giving a constant increase of 2.5% throughout the flow range. Thus this effect, although significant, is not a major contribution to pressure drop, at least at higher flow rates.

(b) A Brush Instrument Co. surf-indicator was used to measure the relative roughness of the lined and unlined surfaces. This device measures the root mean square of the deviation of the peaks and valleys of the wall surface from the average. Knowing the diameter of the pipe D, the relative roughness parameter ϵ/D may thus be computed. The curved surface of the longitudinally sectioned samples of the tubes made precise roughness values difficult to obtain but the Teflon lined surface was 2 - 8 times rougher than the unlined surface. A plot of experimentally determined Friction Factor f versus Reynold's Number is shown in Figure 6. Comparison with the Moody Diagram [5] shows the Teflon lined tube as having a relative roughness approximately 3 times that of the unlined tube; of the same order as may be expected from surf-indicator tests.

(c) By virtue of its "non-wetting" characteristics the Teflon lining may be expected to allow slip at the tube wall. Thus the velocity would not approach zero at the wall and for the same volumetric flow rate would have a somewhat smaller center line velocity. An analysis of momentum transfer fundamentals indicates that the greatest pressure drop will occur in the case of zero velocity at the wall.

If a change in flow patterns results in a change in pressure drop, a greater pressure drop may be expected with the unlined tube. However, experimental values show otherwise and comparison with the Moody Diagram indicates the Teflon surface as being approximately 3 times rougher than the unlined surface. This agrees well with surfindicator tests which predict a pressure drop difference between the two tubes of the same order if it is assumed that difference in pressure drop is attributable wholly to difference in surface roughness.

We have concluded that the difference in surface roughness is the major factor in causing differing pressure drop values between lined and unlined tubes. Axial slip at the wall, if it occurs at all, must be small enough so as not to change markedly the friction factor.

B. Heat Transfer Measurements

Table III with Figures 7 and 8 present the results obtained on the first unlined tube tested. Also plotted is the curve representing results for the tube that would be expected from theoretical considerations. (See Appendix). Experimental results fall somewhat below theoretical predictions perhaps due to an experimental condensing film coefficient slightly different from literature values[6].

Tables IV and V with Figures 9 through 12 present results for the Teflon lined and Epoxy lined tubes. While the Epoxy lined tube shows an overall heat transfer coefficient that is consistently below that of the unlined tube, the Teflon performance shows an improvement over the unlined tube at water velocities less than 5ft/sec.

The Wilson Plot (see Theoretical Background) helps in readily interpreting the significance of these results. The slope of the straight line in the Wilson Plot is C = .0000845/a and the intercept is B. (Equations 6,7,8 in Theoretical Background).

Wilson Plot values for the three tubes from Figures 8,10, and 12 are

TABLE III

<u>Heat Transfer Data for Unlined Tube 1</u>

Tube Data: Base - 7/8" O.D. x 18 BWG,
90-10 CuNi Condenser
Tube

Coating - None

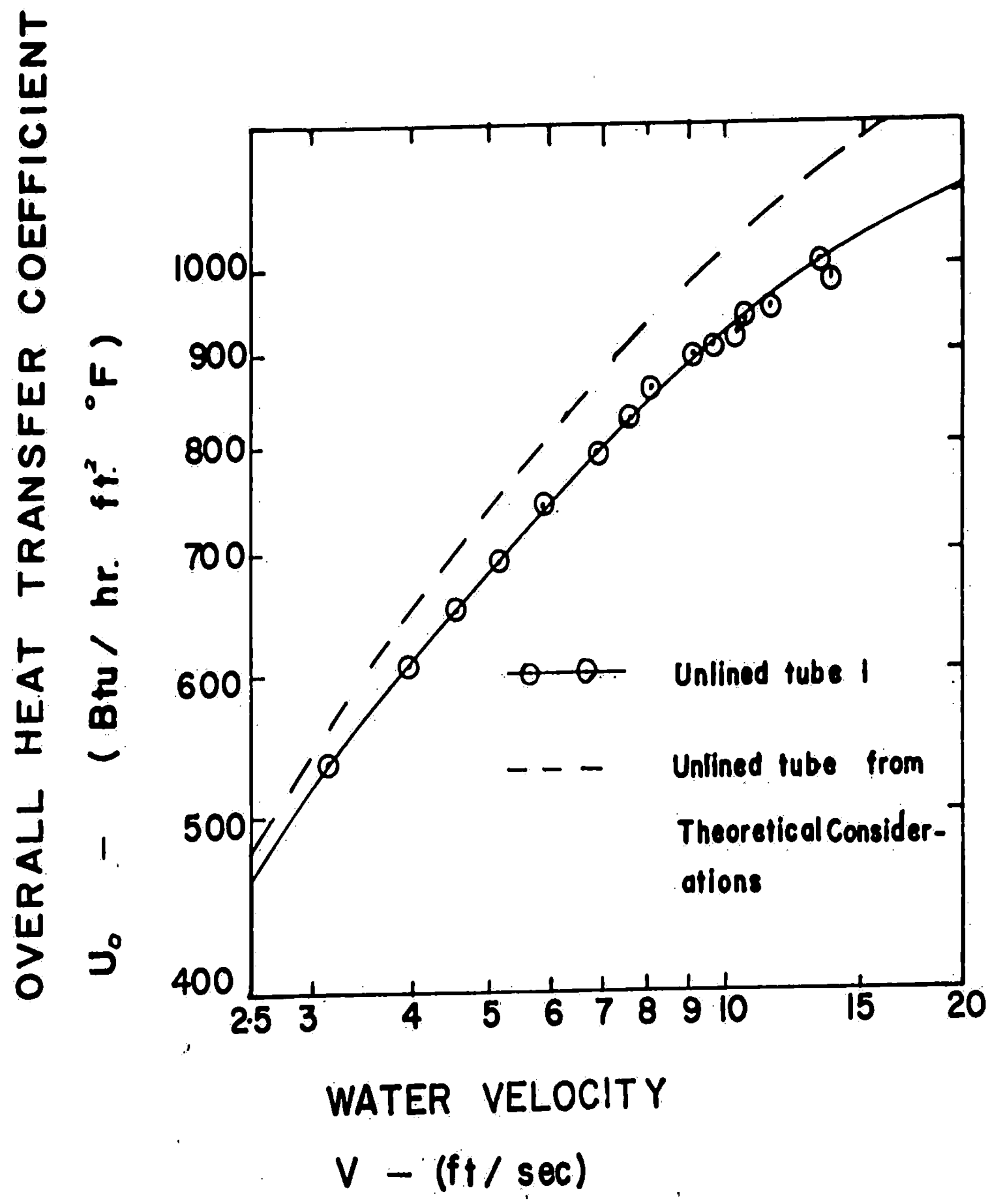
Run No.	Date	Water Velocity ft/sec	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Overall Coefficient U_o Btu/hr.ft^2 °F°
1	8/20/68	6.88	80.31	83.83	101.80	791.3
2		7.65	80.80	84.85	102.20	698.4
3		3.91	80.86	85.24	101.20	608.5
4		8.16	80.24	83.53	102.00	859.5
5		9.61	80 18	83.04	101.20	903.6
6		10.89	89.19	82.85	101.20	947.2
7		13.88	80.27	82.56	102.00	994.0
8	8/21/68	3.19	81.35	85.99	101.40	540.0
9		4.56	80.92	85.13	102.00	654.7
10		5.85	80.47	84.18	101.20	741.3
11		7.57	80.52	83.93	102.20	833.6
12		9.13	80.18	83.24	101.60	902.6
13		10.36	80.29	83.14	102.20	927.9
14		11.68	80.17	82.75	101.80	953.2
15		13.23	79.61	82.02	101.00	1015.2

FIGURE 7

PERFORMANCE OF UNLINED TUBE I

7/8" OD x 18 BWG 90-10 COPPER NICKEL

CONDENSER TUBE



WILSON PLOT FOR UNLINED TUBE I

FIGURE 8

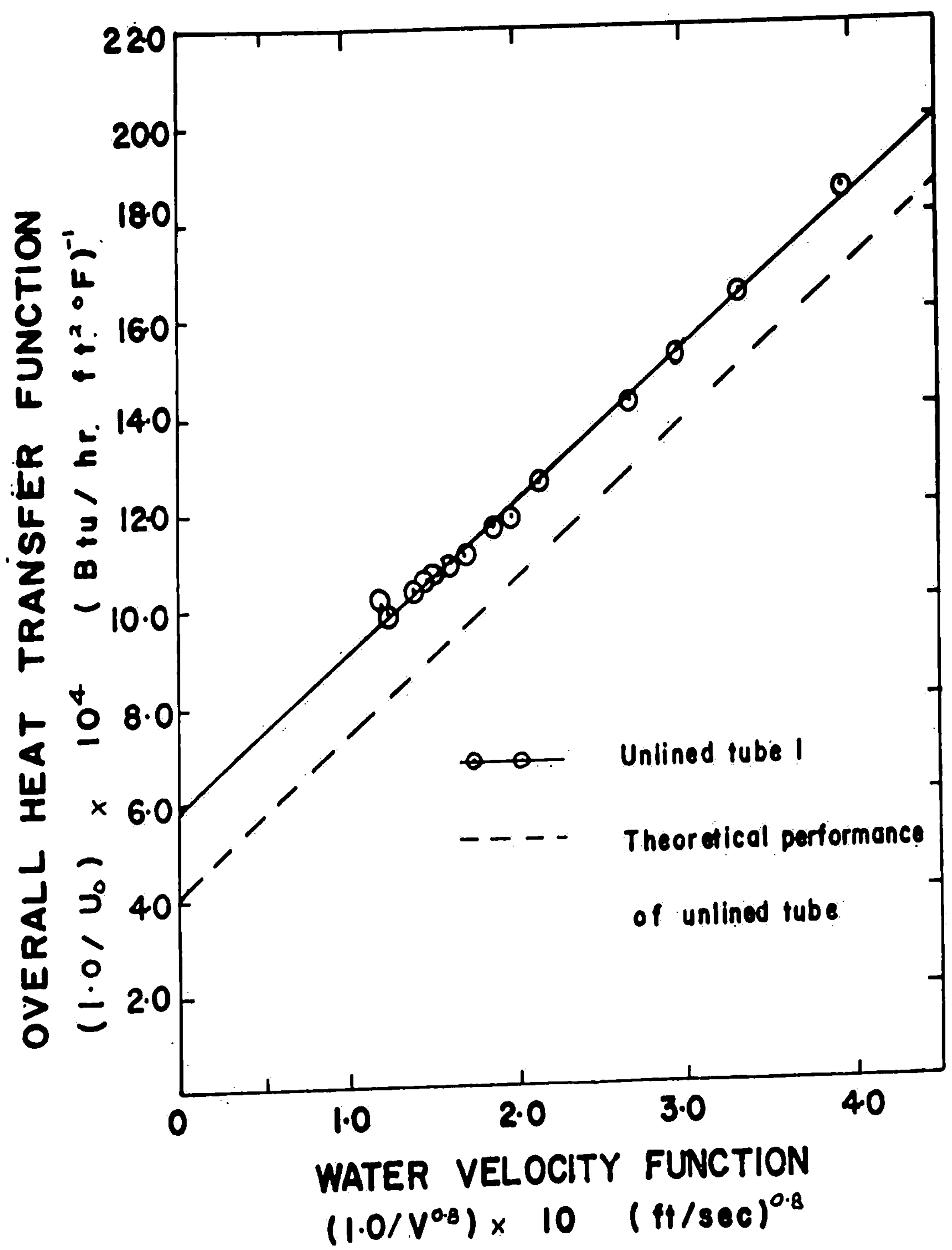


TABLE IV

<u>Heat Transfer Data for Teflon Lined Tube 1</u>

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon, 0.0005" thick on inside only

Run No.	Date	Water Velocity ft/sec	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Overall Coefficient U_o Btu/hr.ft^2 °F
16	8/26/68	3.30	80.81	85.59	100.90	639.8
17		5.32	80.65	84.45	101.20	699.0
18		6.71	80.54	84.04	100.80	776.0
19		8.08	80.29	83.25	101.30	787.5
20		9.50	80.30	82.96	100.80	848.1
21		11.09	80.14	82.45	100.80	846.8
22		12.44	80.07	82.16	100.80	849.7
23		13.85	80.09	81.99	100.80	857.3
24	8/27/68	4.65	80.57	84.77	101.10	678.4
25		7.40	80.37	83.57	101.10	790.0
26		10.40	79.60	81.99	101.60	810.5
27		8.79	79.85	82.52	100.80	769.4
28		3.10	80.90	85.57	100.90	530.7
29		11.88	79.79	81.99	100.90	840.0
30		14.54	79.69	81.62	100.80	895.5
31		12.85	79.75	81.87	100.80	876.7

FIGURE 9

PERFORMANCE OF TEFLON LINED TUBE I

Base: 7/8" OD x 18 BWG 90-10 Copper-Nickel

Condenser tube

Lining : Teflon ·0005" thickness

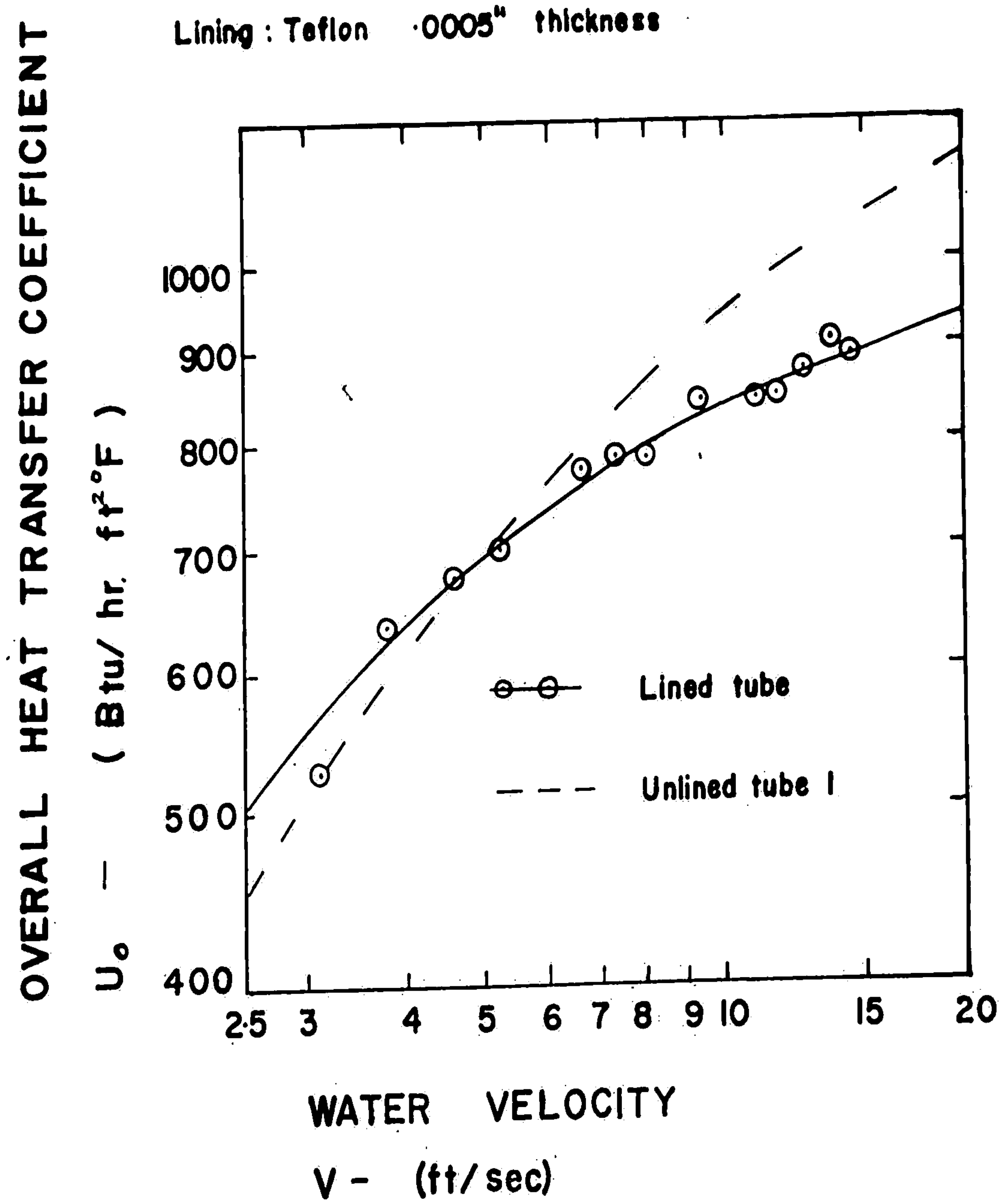


FIGURE 10

WILSON PLOT FOR TEFLON LINED TUBE I

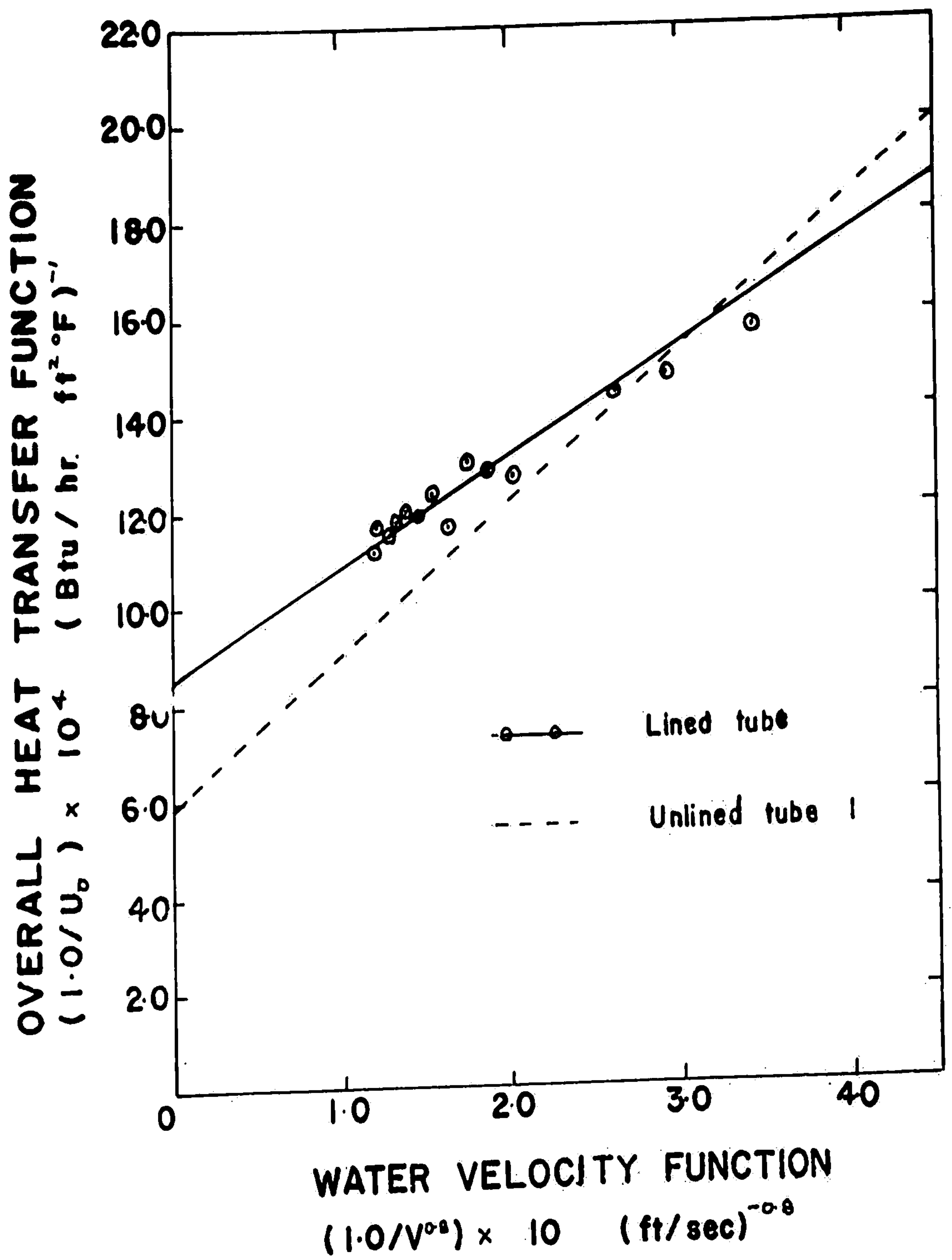


TABLE V

Heat Transfer for Epoxy Lined Tube

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Epoxy, 0.002" thick on inside only

Run No.	Date	Water Velocity v_1 ft/sec	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	U_o Overall Coefficient Btu/hr.ft^2°F
32	9/12/68	5.29	80.09	83.02	101.20	507.47
33		6.96	79.99	82.59	101.30	581.12
34		8.64	80.09	82.37	101.40	627.65
35		10.45	80.19	82.17	101.30	660.45
36		12.44	79.89	81.67	101.40	689.40
37		14.28	79.89	81.47	101.30	702.27
38	9/17/68	4.59	81.00	84.04	101.40	476.2
39		5.43	80.89	83.69	101.60	505.4
40		6.49	80.69	83.27	101.50	550.0
41		8.08	80.49	82.80	101.70	596.8
42		9.21	80.65	82.74	101.60	621.8
43		10.28	80.59	82.57	101.70	647.9

cont..

TABLE V (cont.)

Heat Transfer for Epoxy Lined Tube

Run No.	Date	Water Velocity v_1 ft/sec	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	U_o Overall Coefficient Btu/hr.ft^2 °F
44	9/17/68	11.24	80.75	82.57	101.60	656.8
45		12.32	80.69	82.36	101.70	652.8
46		13.35	80.42	82.05	101.20	698.3
47		14.69	80.57	82.08	101.50	701.6

FIGURE 11

PERFORMANCE OF EPOXY LINED TUBE

Base: 7/8" OD × 18 BWG 90-10 Copper-Nickel

Condenser tube

Lining: Epoxy- ·002" thickness

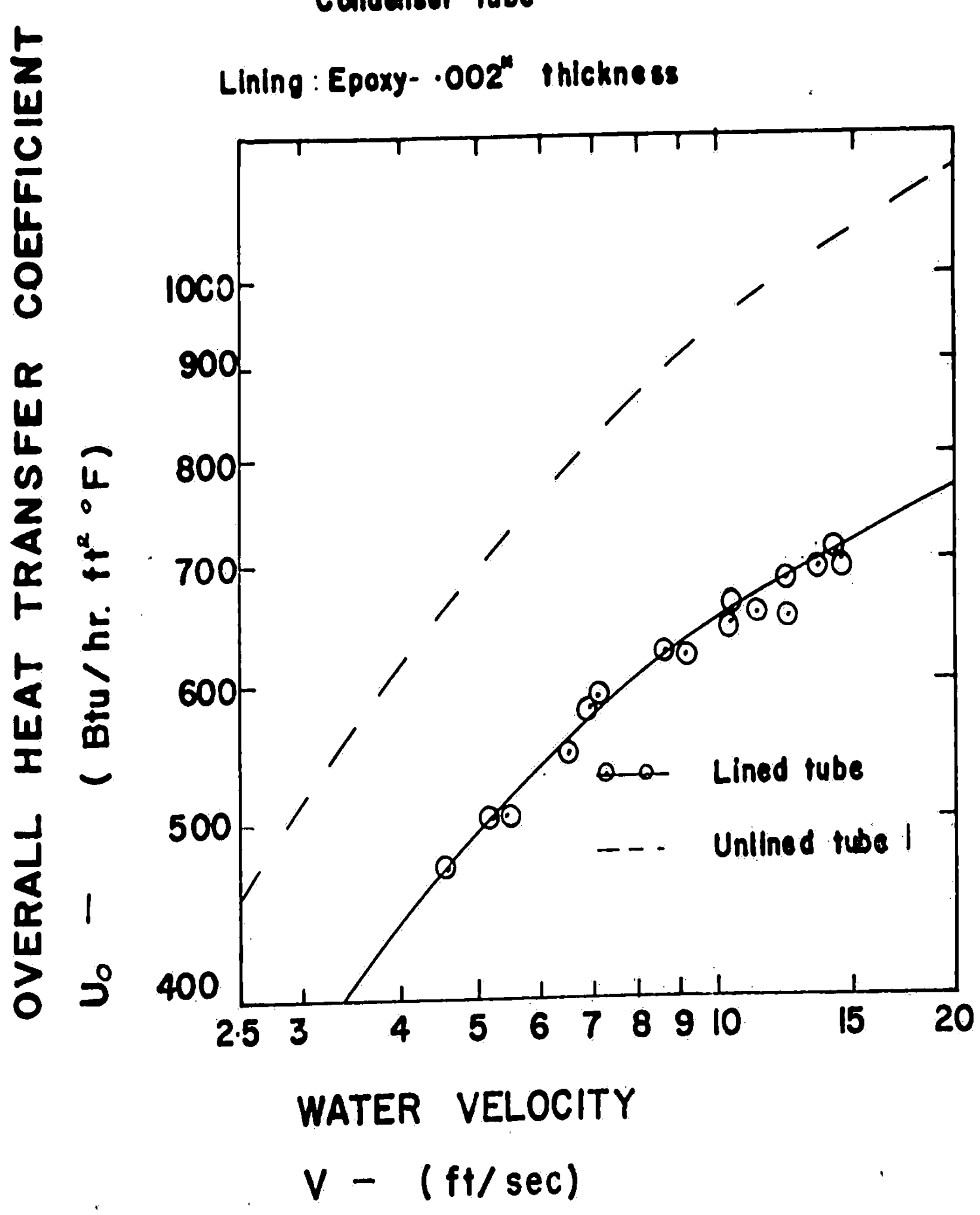
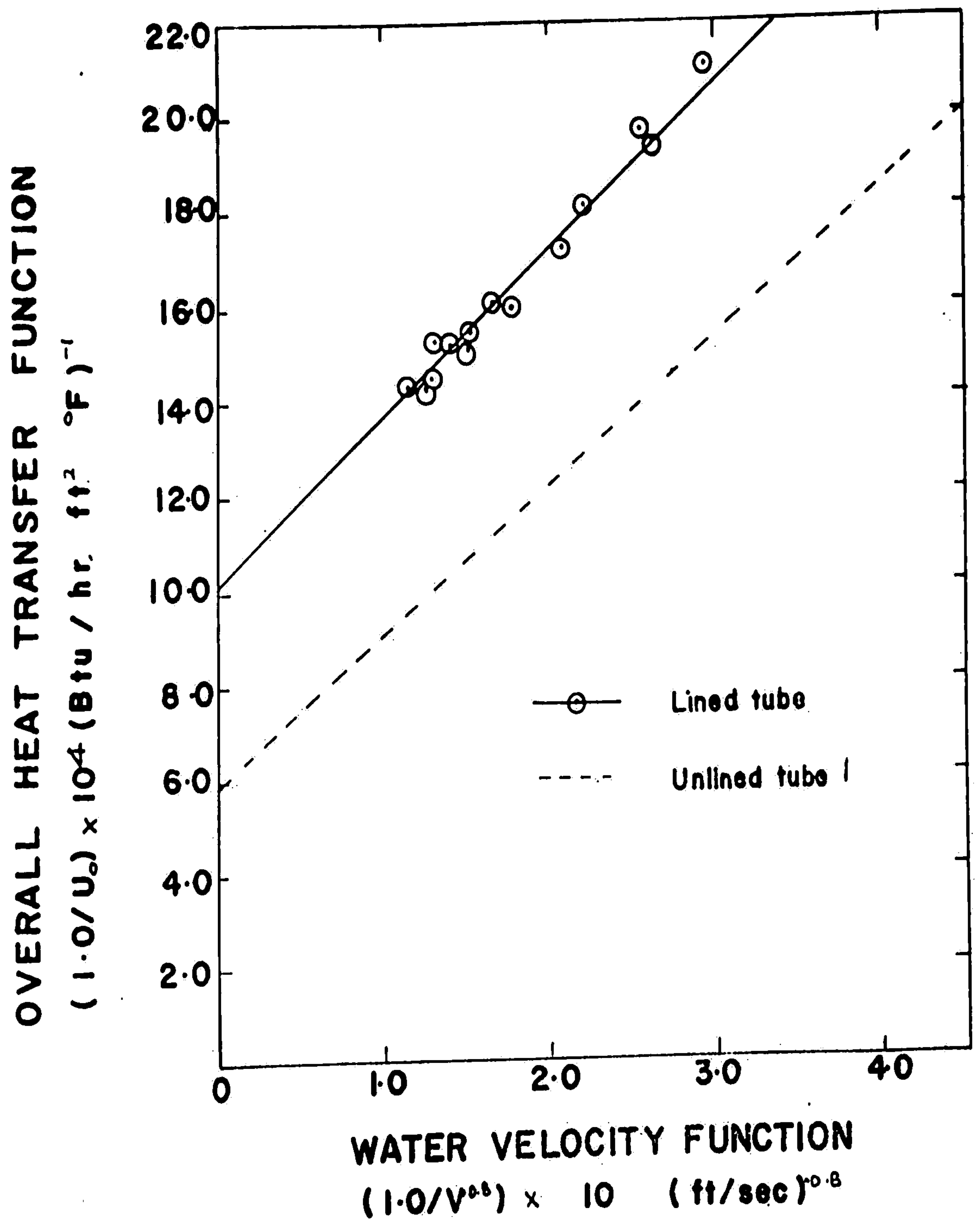


FIGURE 12

WILSON PLOT FOR EPOXY LINED TUBE



	Slope	Intercept
Uncoated Cu/Ni tube	.00316	.00058
Teflon lined Cu/Ni tube	.00227	.00085
Epoxy lined Cu/Ni tube	.00350	.00102

a (see Theoretical Background) for the three tubes is thus

a uncoated = .0267

a Teflon coated = .0383

a Epoxy coated = .0242

a is a direct measure of the inside convective heat transfer coefficient since it is the only factor in the Dittus-Boelter equation that can be affected by surface characteristics. A comparison of the values obtained indicates that the Teflon surface produces a 43.5% increase in inside heat transfer coefficient.

The thickness of the lining may be calculated using the difference in intercepts between lined and unlined tube Wilson Plots.

Material	$\frac{x_L A_o}{k_L A_i}$	k Btu/hr.ft^{o}F	x_L in.	x Nominal in.
Teflon	.00027	0.145	.00042	.0005
Epoxy	.00044	0.40	.0019	.002

Teflon and Epoxy thicknesses calculated above agree well with the specifications of the supplier, thus providing indirectly a check on the data.

The Teflon lining produced an improvement of 43.5% in inside heat transfer coefficient in contrast to the epoxy lining which showed no improvement at all.

This considerable increase could not be explained on the basis of diameter and roughness changes or experimental error. However, it was necessary to establish the validity of these results by tests on other Teflon lined tubes and further to examine the effect other hydrophobic linings may have on the inside heat transfer coefficient.

Tables VI and VII with Figures 13 through 16 show the results for Gold lined and Paralene-N lined tubes. No improvement is observed in the inside heat transfer coefficient.

(Note that for all tests starting with the Paralene-N lined tube, the reference tube is Unlined Tube 2 which has results almost identical to Unlined Tube 1).

The Wilson Plot slope for the Paralene lined tube is .00222 and for the Gold lined tube is .0031.

TABLE VI

<u>Heat Transfer Data for Paralene-N Lined Tube</u>

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi

Coating - Paralene-N coated on the inside to 10,000 to 25,000 Angstrom Unit Thickness

Run No.	Date	Water Temperature In. °F	Water Temperature Out. °F	Steam Temp. °F	V, ft/sec	Overall Heat Transfer Coefficients
521	11/6/68	81.29	86.16	101.38	3.12	565.5
522		80.84	84.47	101.50	5.51	683.4
523		80.39	83.30	101.50	8.24	786.8
524		80.55	83.73	101.30	6.76	724.1
525		80.37	82.77	100.85	10.94	875.1
526		80.67	83.34	101.65	9.45	828.3
527	11/8/68	80.22	82.17	101.25	14.90	932.9
528		81.03	84.88	101.40	4.72	635.0
529		80.53	83.47	101.13	7.60	751.4
530		80.39	82.58	101.20	12.29	878.5

FIGURE 13

PERFORMANCE OF PARALENE LINED TUBE

Base: 7/8" OD × BWG 90-10 Copper Nickel

Condenser tube

Lined with Paralene-N, 10000 - 25000 Angstrom thickness

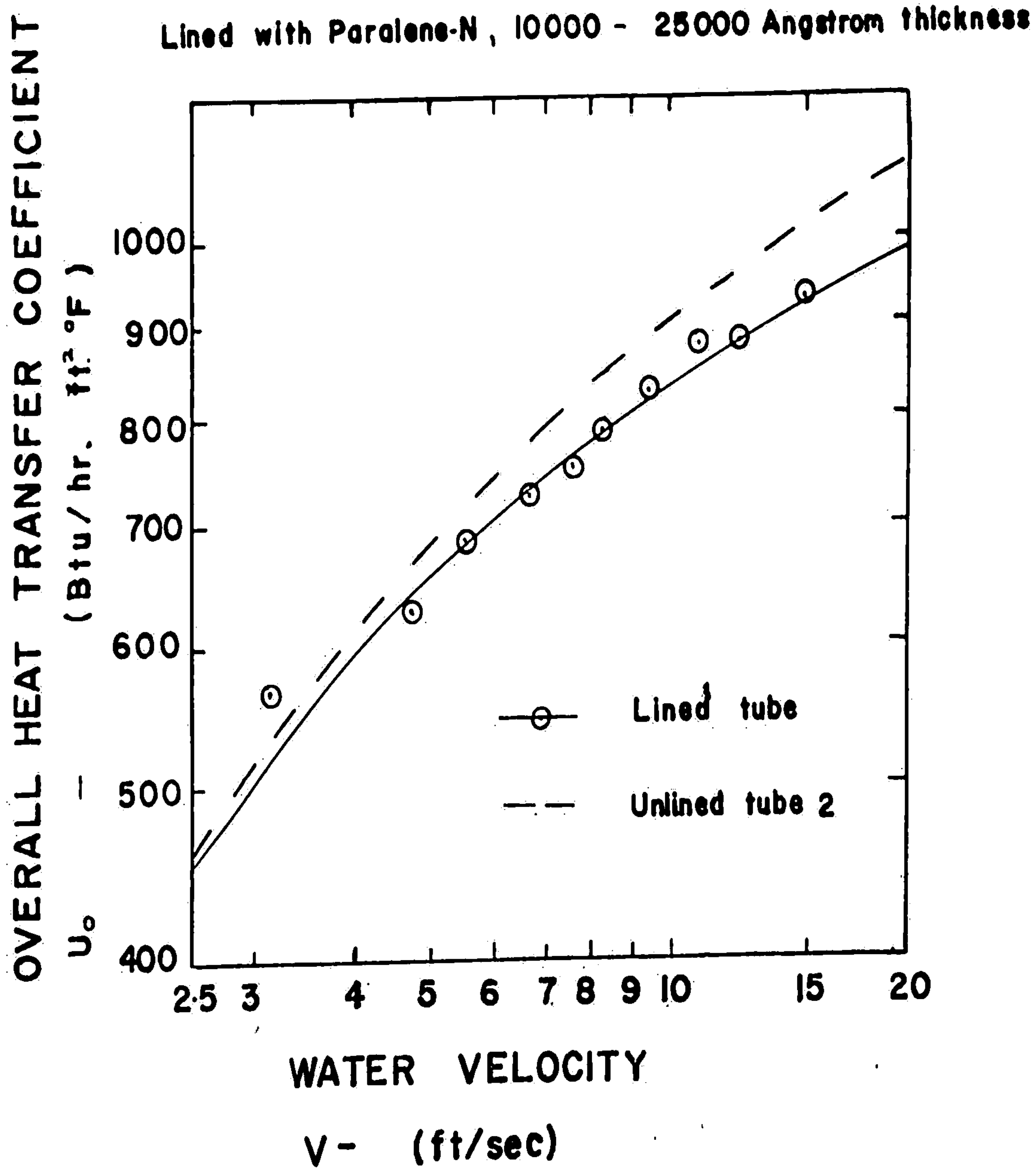


FIGURE 14

WILSON PLOT FOR PARALENE LINED TUBE

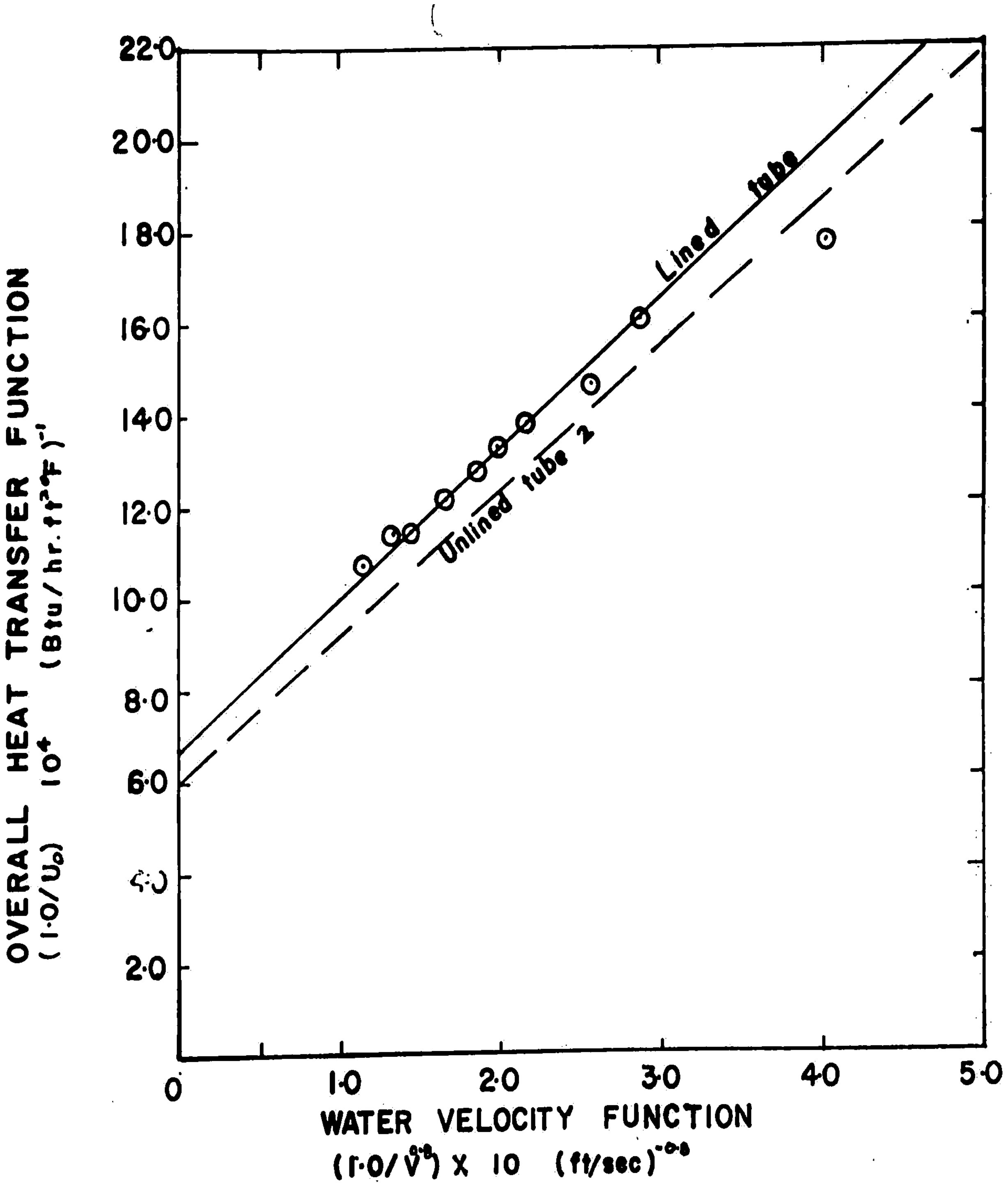


TABLE VII

<u>Heat Transfer Data for Gold Lined Tube</u>

Tube Data: Base - 7/8" O.D. x 18 BWG.
90-10 CuNi Condenser Tube

Coating - Gold, 5-10 micro-inch
Thick on inside only

Run No.	Date	Water Temperature Inlet°F	Water Temperature Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
531	11/14/68	80.36	84.51	101.30	5.15	732.2
532		79.94	82.63	101.35	11.09	957.8
533		80.29	83.38	100.95	8.10	845.4
534		80.22	83.04	100.87	9.55	903.5
535		80.41	83.79	100.90	6.70	777.1
536		80.05	82.25	101.18	13.90	981.3
537	11/15/68	80.62	84.41	100.95	4.36	578.7
538		79.99	82.48	101.30	11.70	936.8
539		80.29	83.21	101.40	8.67	831.2
540		80.31	82.55	101.60	13.29	952.8

FIGURE 15

PERFORMANCE OF GOLD LINED TUBE

Base: 7/8" OD × 18 BWG, 90-10 COPPER NICKEL CONDENSER TUBE

Lining: GOLD, 5-10 MICRO INCHES THICK

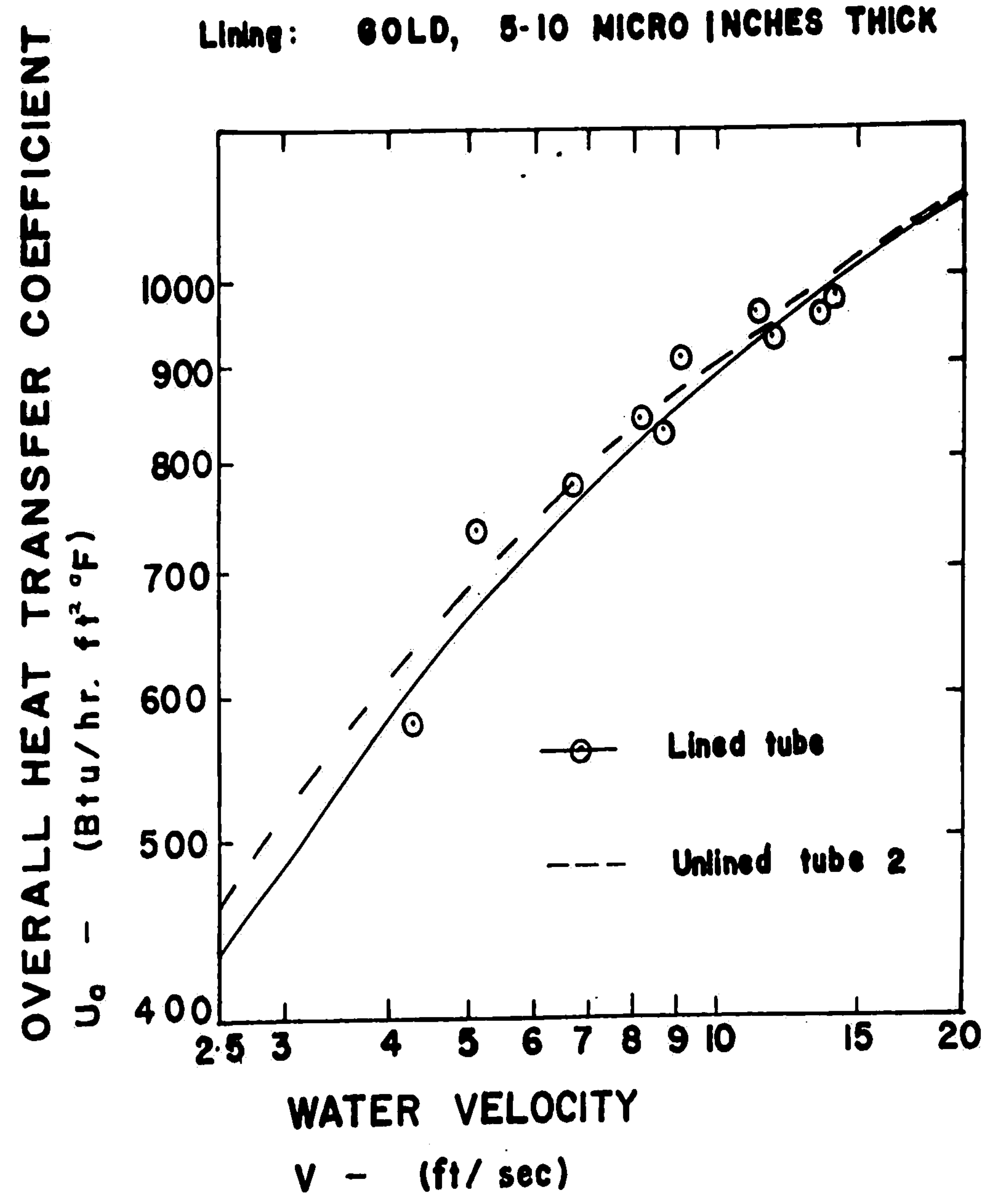
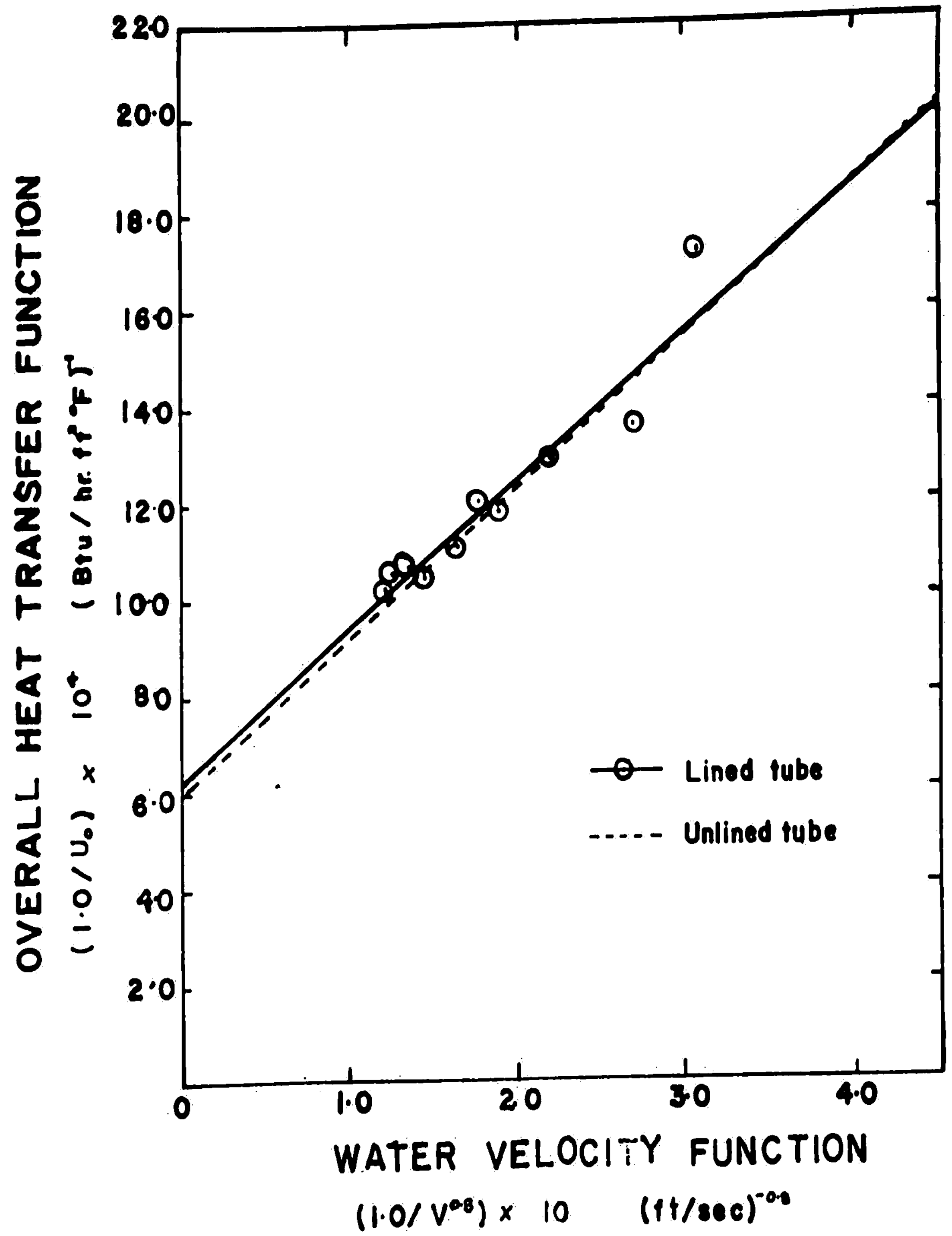


FIGURE 16

WILSON PLOT FOR GOLD LINED TUBE



These linings were thin and especially Gold on account of its high thermal conductivity offered negligible additional wall resistance (as seen from the intercept).

For these tubes:

a Paralene lined tube = .0262

a Gold lined tube = .0272

Teflon lined tube 2, Table VIII and Figures 17 and 18, was tested for the purpose of establishing that the change in inside heat transfer coefficient was primarily the result of a surface effect.

Initial data points (541-549) showed a slope of .00247 giving a 25.4% improvement in inside heat transfer coefficient. Detergent was added to the process water in the holding tank after data point 568 so that the teflon surface was wetted by the water-detergent solution. The concentration of wetting agent in the process water was maintained at 0.05%w.

The immediate effect of detergent addition was increased scatter with no sharply defined slope. After data taking was continued for a period of 10-12 hours a clearer trend was apparent with a slope of .00311 on the Wilson Plot, almost identical to the unlined tube slope.

Since diameter and roughness changes can only be negligible, the 25.4% change in slope as a result of

TABLE VIII

Heat Transfer Performance of Teflon Coated Tube 2

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon 0.0005" Thick on inside only

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
541	11/22/68	79.57	81.64	101.22	13.26	856.2
542		80.20	84.11	100.72	5.12	680.0
543		80.08	83.46	101.32	7.43	828.8
544		80.02	82.91	101.48	9.27	862.2
545		80.12	82.45	101.45	11.34	843.0
546		80.03	81.91	101.28	14.79	885.0
547	11/27/68	79.90	82.95	101.50	8.09	790.4
548		79.85	82.26	101.50	11.09	843.8
549		80.46	84.07	101.50	5.84	707.1
568	12/6/68	79.78	81.78	100.25	13.09	861.8
569		80.32	83.69	100.83	5.93	684.7
570		80.65	84.45	101.00	4.37	580.6
571		80.46	83.94	100.88	5.22	628.2
572		80.22	83.12	101.02	7.40	716.0
573		80.06	82.62	100.08	8.98	761.2

Detergent 0.05 wt % added after Run 551. Run 568 represents 18th data point after this time.

FIGURE 17

PERFORMANCE OF TEFLON LINED TUBE 2

Base: 7/8" OD × 18 BWG, 90-10 Copper-Nickel

Condenser tube

Lining: Teflon ·0005" thick

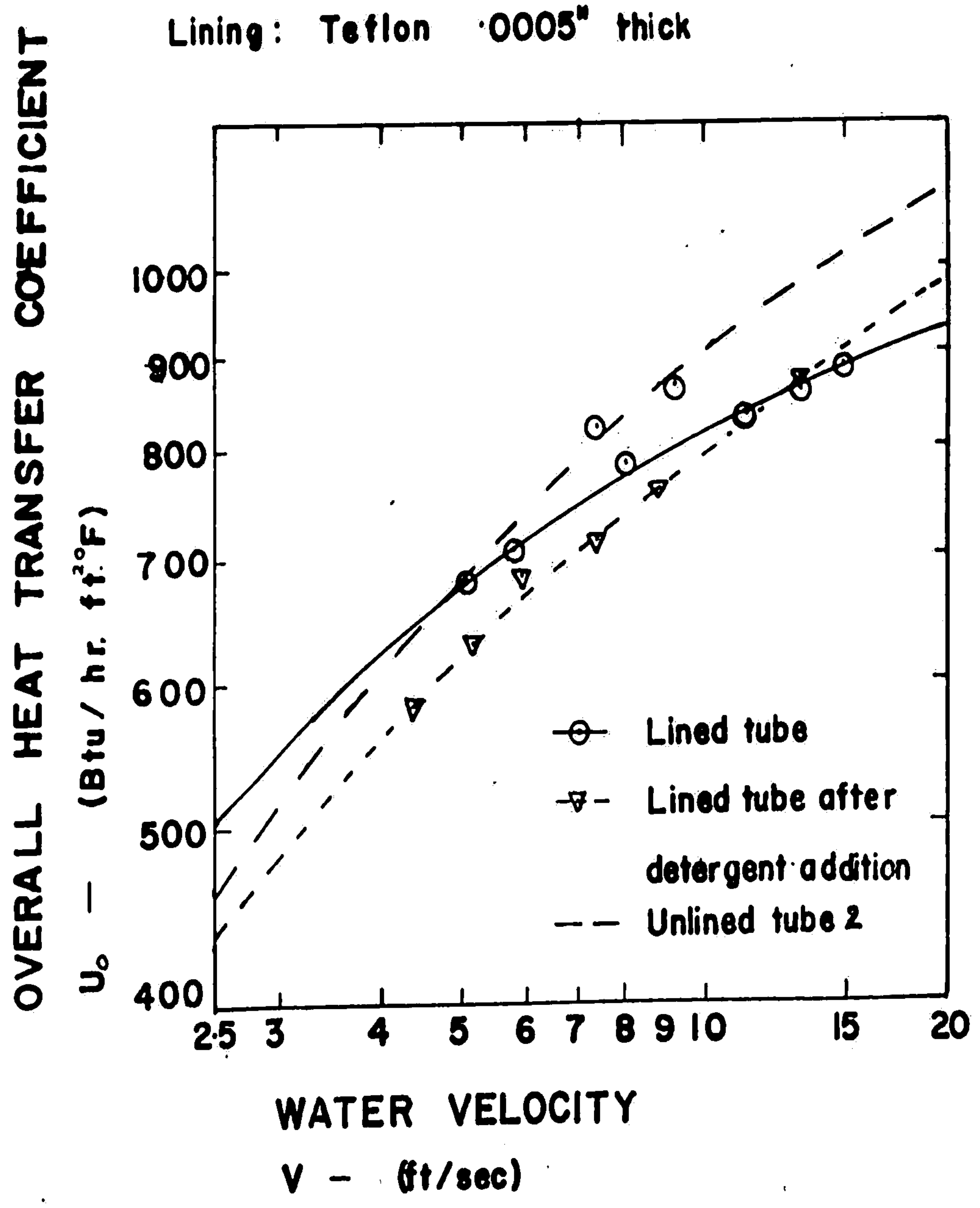
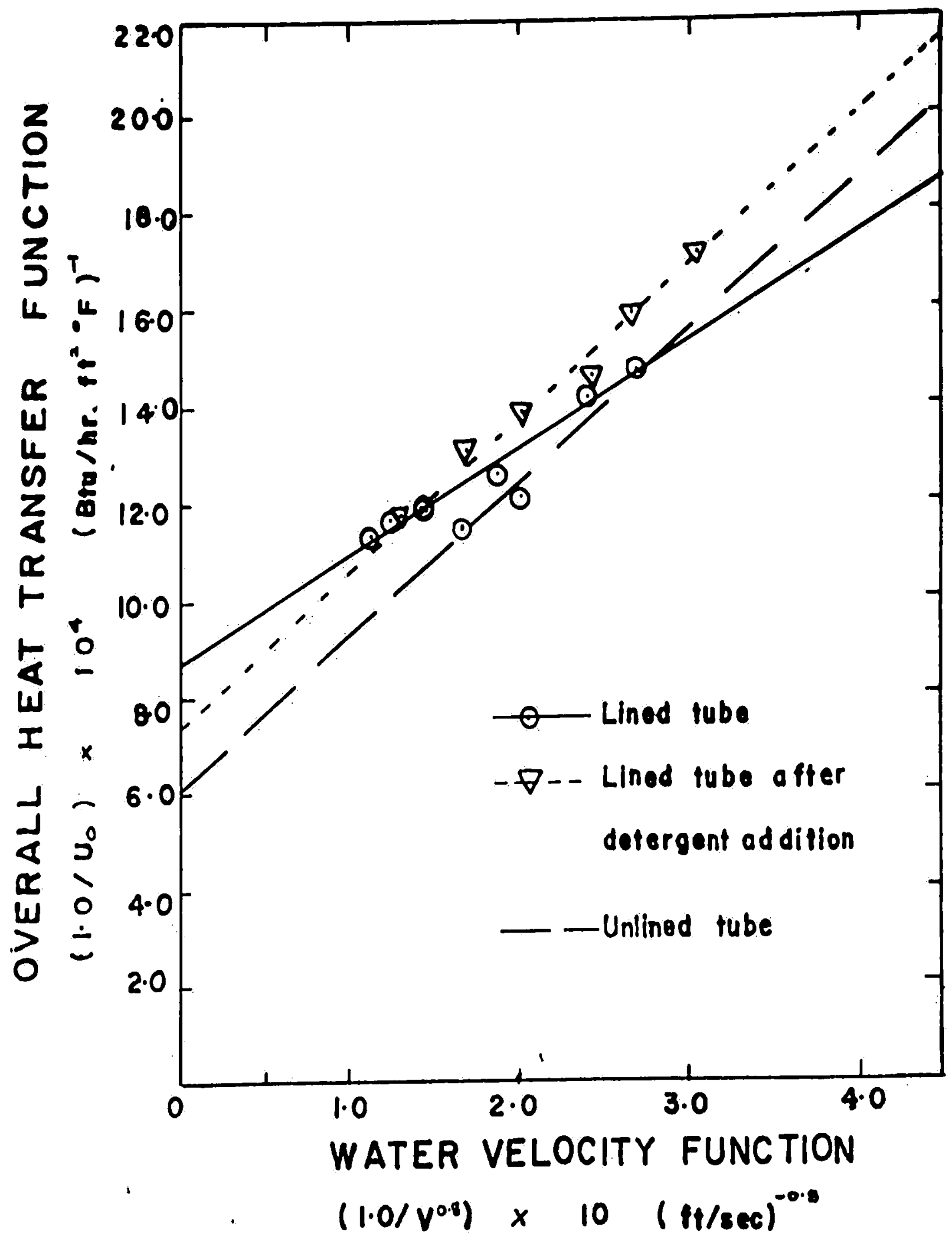


FIGURE 18

WILSON PLOT FOR TEFLON LINED TUBE 2



detergent addition strongly suggested that the increased inside heat transfer coefficient for non-wetted flow was due primarily to a surface effect.

Mention must be made of the scatter obtained in the results. Aside from the unlined tubes, Wilson Plots show scatter to a varying extent; scatter for the Teflon lined tubes being largest. Before drawing a representative straight line, one must consider the following factors (a) - (e) that might contribute to the scatter.

(a) Changing film condensing coefficient:

This might vary continuously to a small extent. However, the sharpest difference was observed between runs made on separate days i.e. yielding straight lines with clearly defined slopes but with different intercepts.

(b) First data point after start up:

The first data point taken after start up showed a greater overall heat transfer coefficient value than subsequent data points. The reason for this phenomenon is not clearly apparent. (See unlined Tube 2 (Figure 23 and Figure 24) where the first data points on 2/28/69 and 3/4/69 stand out markedly).

(c) **Experimental errors.**

(d) **Insufficient time allowed for achieving steady state.**

(e) Instability conditions at the liquid-solid interface.

Of the above factors, errors arising from (a) and (b) were easily recognisable and due consideration was given when drawing the best straight line for the Wilson Plot.

Every precaution was taken in minimising (c) and (d). It was difficult, however, to establish the extent to which scatter might be attributed to these factors.

The purpose of testing Teflon lined tubes A_1 and A_2 (Tables IX and X and Figures 19-22) was to confirm earlier findings and also to investigate the effect prolonged exposure might have on Teflon surface.

Tested over a period of several days, Teflon lined tube A_1 served if anything to confuse the issue. With surprisingly little scatter the slope of the Wilson Plot was 0.00285 with a = .0297 giving a mere 8.5% increase which could be within the bounds of experimental error.

TABLE IX

Heat Transfer Data on Teflon Lined Tube A1

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon, 0.0005" Thick on inside surface only

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
651	2/4/69	79.99	83.01	100.35	7.60	786.1
652		80.97	85.26	100.90	3.76	587.7
653		79.81	82.35	100.90	10.21	845.2
654		80.41	84.08	100.00	4.84	646.6
656		79.53	81.76	100.50	12.76	922.2
657		80.49	84.63	100.40	3.85	578.2
659		79.79	83.06	101.00	6.90	744.4
660		79.89	82.78	100.50	7.93	772.6
661		79.66	82.29	100.10	9.21	815.4
662		79.60	81.88	100.20	11.56	871.7
663	2/8/69	79.53	81.45	100.10	14.77	930.4
664		80.49	84.94	100.60	3.56	574.1
665		80.10	83.76	100.45	5.31	679.0
666		79.71	82.56	100.60	8.69	819.8
667		79.78	82.17	100.80	11.56	896.6

cont..

TABLE IX (cont.)

Heat Transfer Data on Teflon Lined Tube A1

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
668	2/8/69	79.78	81.86	100.80	13.97	935.5
669	2/11/69	79.58	82.72	100.40	7.68	807.6
670		80.46	84.79	100.65	3.75	582.3
671		80.20	84.04	100.00	4.61	638.9
672		80.08	83.69	100.50	5.62	702.9
673		79.82	82.99	100.10	6.88	750.9
674		79.69	82.52	100.60	8.75	817.9
675		79.99	82.40	100.20	10.71	873.7
676		79.56	81.86	100.20	12.11	917.4
677		79.48	81.60	99.80	13.35	942.4
678		79.61	81.59	100.20	14.85	962.0

FIGURE 19

PERFORMANCE OF TEFLON LINED TUBE A1

Base: 7/8" OD x 18 BWG 90-10 Copper-Nickel Condenser tube

Lining: Teflon ·0005" thick

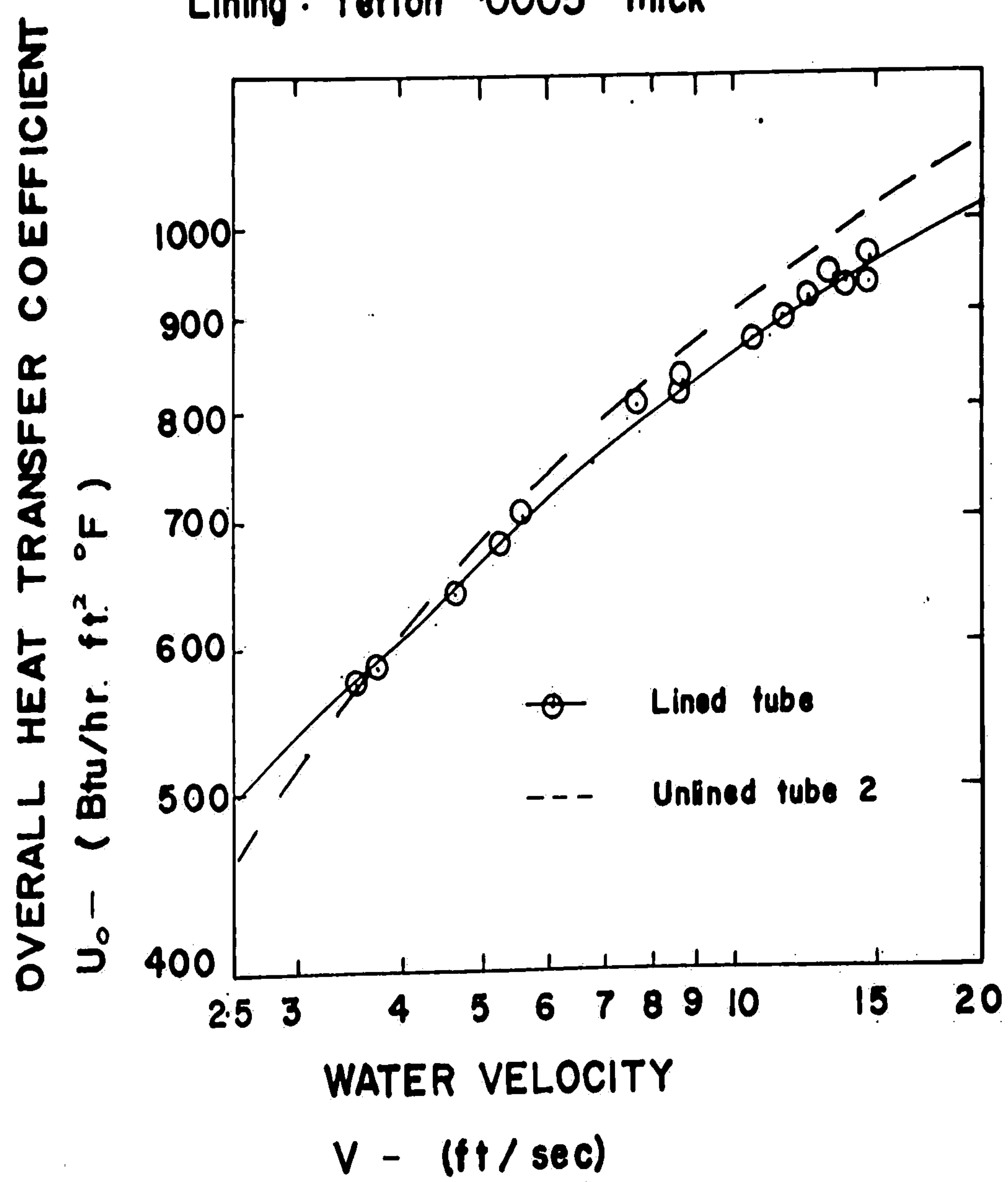


FIGURE 20

WILSON PLOT FOR TEFLON LINED TUBE A1

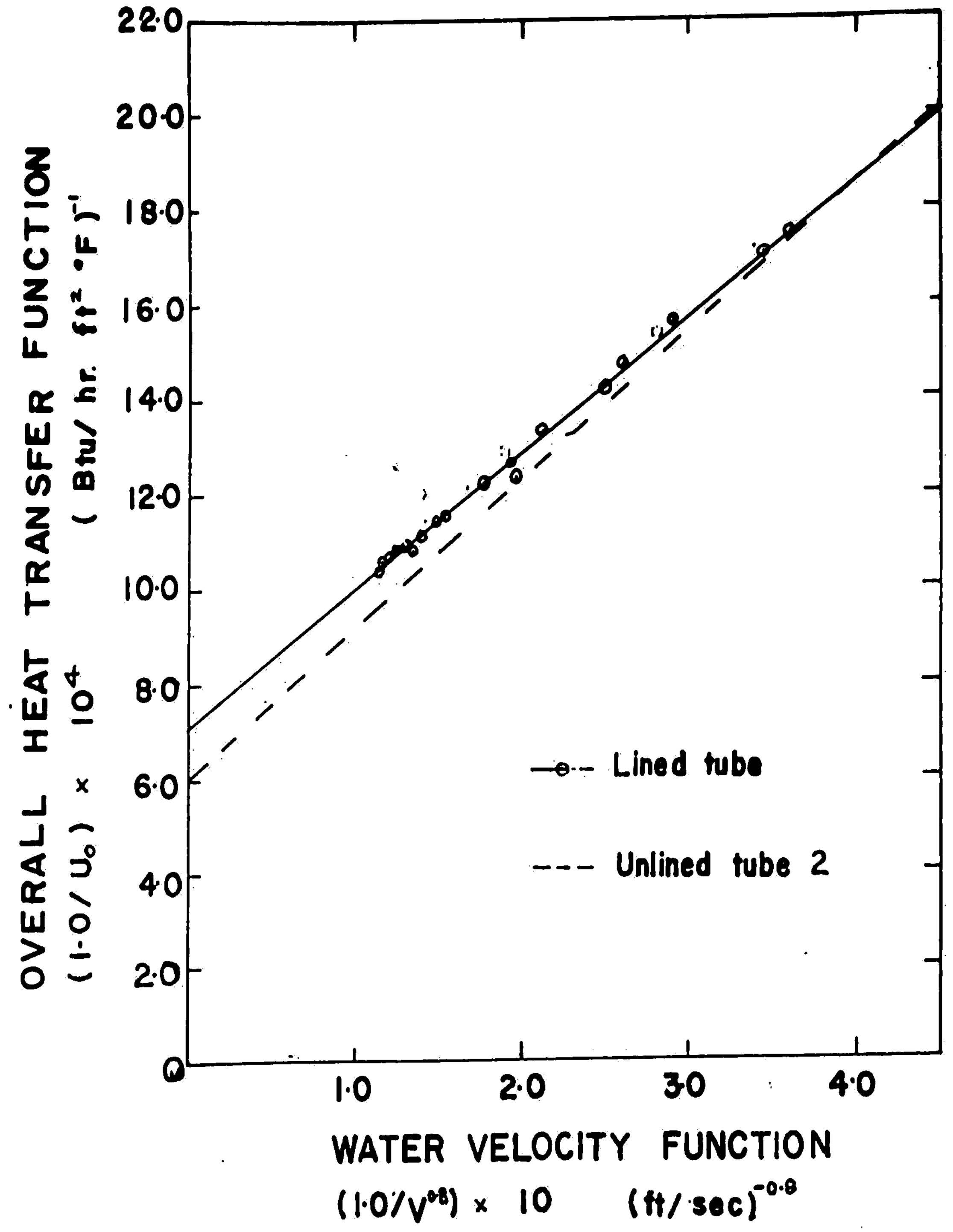


TABLE X

Heat Transfer Data for Teflon Lined Tube A2

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon, .0005" thick on inside surface only

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
701	2/17/69	79.78	83.28	100.28	7.15	837.1
702		80.88	85.59	100.80	3.30	573.8
703		80.51	84.76	100.40	4.02	622.6
704		80.26	84.06	100.00	4.88	672.3
705		80.07	83.63	100.80	5.90	716.2
706		79.76	82.47	100.15	8.61	789.2
707		79.89	82.37	100.80	10.27	836.3
708		80.65	82.78	101.00	12.17	866.6
709		79.66	81.64	100.40	13.40	864.5
710	2/18/69	79.79	83.06	100.40	7.15	796.0
711		80.09	83.80	100.45	5.81	753.8
712		80.19	84.26	100.60	4.81	689.6
713		80.59	85.14	100.60	3.53	587.4
714		80.99	85.76	100.40	2.95	536.4
715		79.99	82.95	100.30	8.00	812.6
716		80.03	82.80	100.80	9.15	844.4

cont..

TABLE X (cont.)

Heat Transfer Data for Teflon Lined Tube A2

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
717	2/18/69	79.90	82.46	100.60	10.36	881.8
718		79.84	82.16	100.10	11.36	891.4
721	2/20/69	80.77	85.45	101.00	3.22	545.0
722		80.29	84.31	100.90	4.63	647.0
723		79.97	83.14	100.70	6.67	714.1
724		79.83	82.51	100.70	9.55	843.7
725		79.75	81.85	100.60	13.40	914.3
730	2/24/69	79.97	82.86	100.20	7.99	793.9
731		80.93	85.41	100.50	2.98	499.2
732		80.49	84.56	100.40	3.88	571.4
734		80.11	83.25	100.50	6.95	749.3
735		79.99	82.78	100.70	8.61	803.2
736		79.85	82.26	100.60	10.82	858.8
737		79.55	81.66	100.40	13.35	915.3
738	2/25/69	80.33	84.66	101.25	3.74	558.2
739		80.27	84.08	100.60	4.59	613.2
740		79.89	83.16	100.60	6.40	708.6

cont..

TABLE X (cont.)

Heat Transfer Data for Teflon Lined Tube A2

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
741	2/25/69	80.04	83.16	100.70	7.51	792.7
742		79.99	82.71	100.80	8.50	767.3
743		79.63	82.16	100.50	9.46	786.4
744		79.57	81.80	100.60	11.17	804.9
745	2/26/69	79.78	81.78	101.20	12.85	809.1
746		79.79	83.29	100.70	6.29	740.4
747		80.04	83.79	100.60	5.15	667.9
748		80.11	84.14	100.40	4.37	624.3
749		80.61	84.91	100.60	3.59	560.4
750		80.72	85.16	100.50	3.19	521.8

FIGURE 21

PERFORMANCE OF TEFLON LINED TUBE A 2

Base : 7/8" OD× 18 BWG 90-10 Copper-Nickel

Condenser tube

Lining : Teflon, ·0005" thick

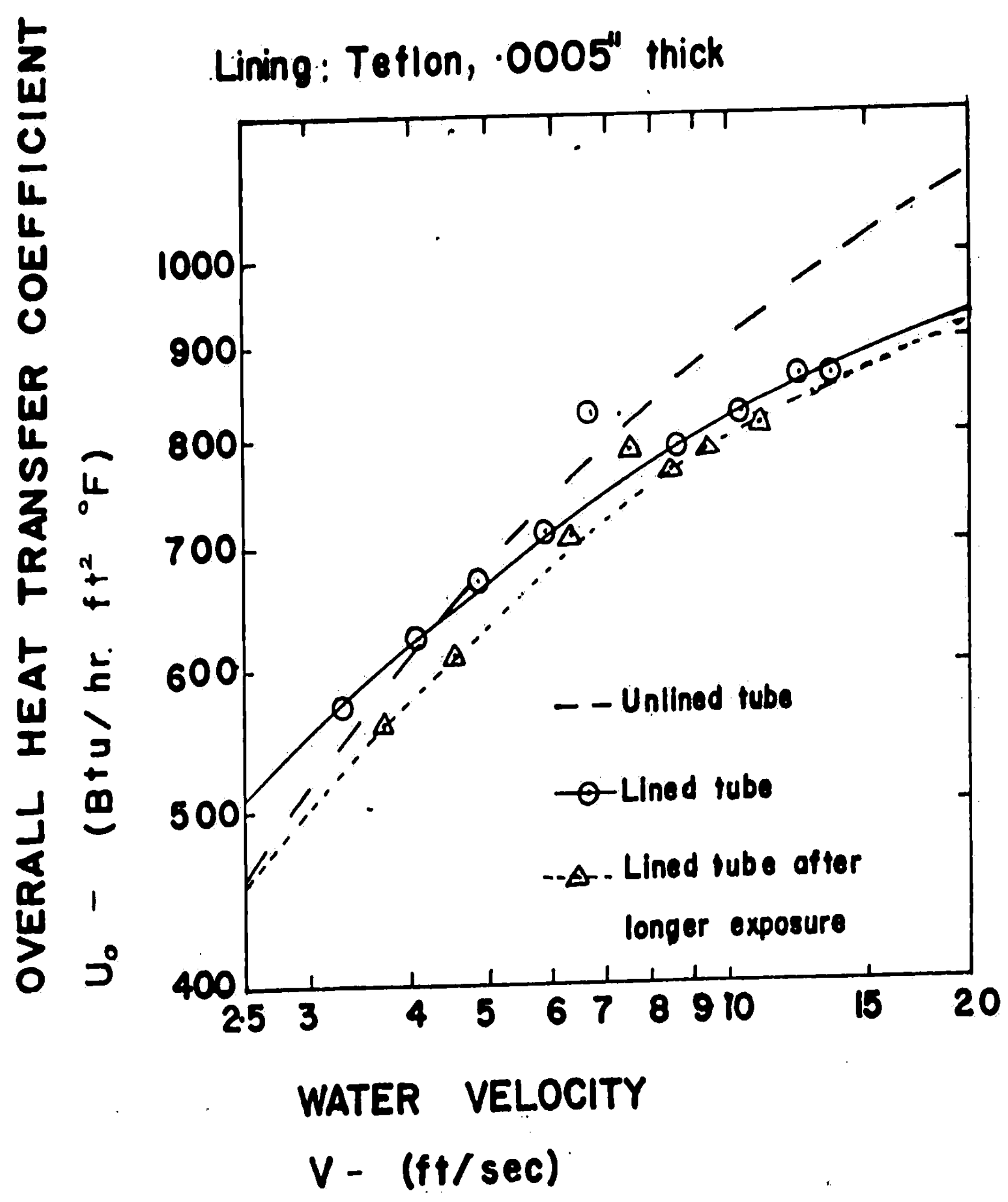
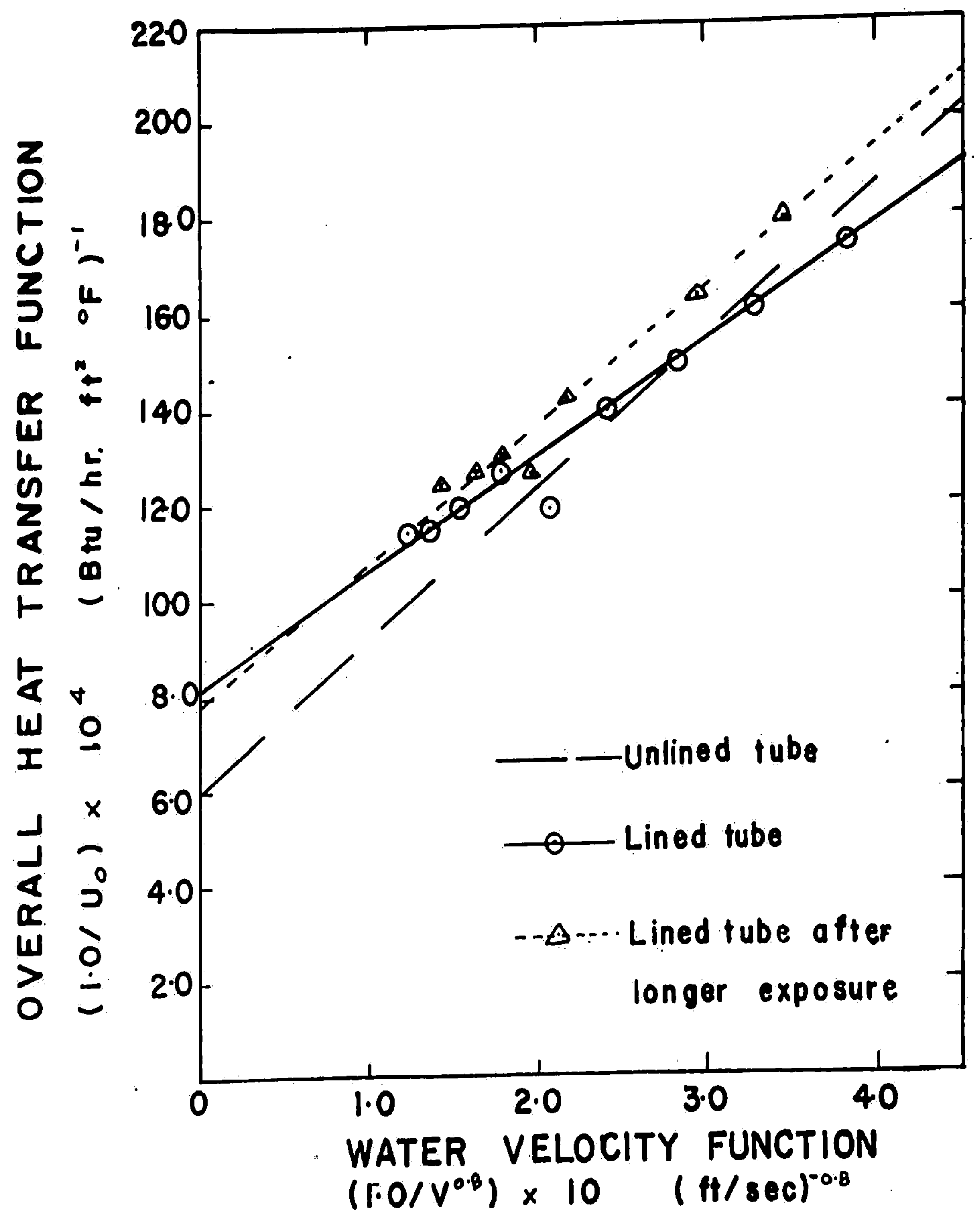


FIGURE 22

WILSON PLOT FOR TEFLON LINED TUBE A 2



It must be recalled that the previous tube (Teflon lined tube 2) was tested with detergent in the water. Though the system was flushed thoroughly the presence of detergent is known to be difficult to remove. Thus trace amounts of detergent could have remained in the heat transfer apparatus. Without proof however, results for Tube A_1 cannot be entirely discounted.

The behavior of Teflon lined Tube A_2 was in many ways enlightening. The first days run gave a slope of .00242 or a = .035 giving a 28% higher inside heat transfer coefficient. Results for the second day gave somewhat smilar results but with increased scatter. Runs on subsequent days showed scatter but it appeared that the slope was changing day by day. The run on 2/25/69 showed clearly a straight line with a slope almost identical to the unlined tube slope. Runs after 2/25/69 (not drawn here) showed little further change in slope.

A flat aluminum plate lined with Teflon .0015" thick was increased in the recycle tank during the course of the experiment on Tube A_2. Contact angle measurements before the experiment gave a value of 108° which agrees with values quoted in the literature(7). At the conclusion of the experiment, the contact angle was reduced to 84°.

The exposed and unexposed Teflon lined plate was also subjected to visual examination under a Microscope-Magnification 75x (Figure 29). The relatively clear surface before exposure was covered with a noticeable scale at the conclusion of the experiment. On the unexposed plate, water sprinkled on the surface formed clearly defined drops with a high contact angle. The behavior changed markedly after exposure; water tended to spread formlessly on the surface yielding a much lower contact angle. Treatment of the exposed surface with a stannous chloride/dilute hydrochloric acid solution restored the contact angle to its initial value. The scale was not entirely destroyed however (Figure 30).

Contact angle measurements made on the curved surface of the Teflon lined tube were not conclusive since the geometry and roughness of the surface made accurate determination difficult (Figure 31).

Comparison of heat transfer data with contact angle measurement strongly suggests that a change in surface properties during the course of the experiment with tube A_2 was responsible for the corresponding change in heat transfer results.

The scatter observed with lined tubes in general and tube A_2 in particular and the unexpected behavior of tube A_1 raised the question of the validity of the results. It was necessary to establish that the phenomenon of scatter was one associated with the hydrophobic lining and that it was not the result of inept data taking or faulty equipment. With control settings for the water and steam systems maintained exactly as for tube A_2 and with other conditions duplicated as far as possible, an unlined Copper-Nickel alloy tube was tested on 2/28/69 and 3/4/69.

Results are shown in Table XI and Figures 23 and 24. Apart from the first data points on 2/28/69 and 3/4/69, there is remarkably little scatter. The slope of .0[illegible]1 agrees well with that of unlined Tube 1. One must conclude that scatter is a phenomenon induced in some way by hydrophobic linings and by Teflon in particular.

Tubes B_1 and B_2 were tested with the aim of establishing a more clearly defined slope. The examination under a microscope of Tubes 1,2,A_1,A_2, with Teflon of .0005" thickness revealed a pitted and incomplete Teflon surface (Figure 31). Tubes B_1 and B_2 were lined with Teflon of .0015" thickness giving

TABLE XI

<u>Heat Transfer Data for Unlined Tube 2</u>

Tube Data: 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficient
761	2/28/69	79.63	82.93	100.80	7.92	862.9
762		79.82	83.29	100.80	6.94	806.7
763		80.69	85.26	100.25	3.01	516.1
764		80.61	85.02	100.60	3.63	583.7
765		80.31	84.48	100.55	4.30	639.8
766		80.37	84.32	100.80	4.91	679.4
767		80.08	83.71	100.60	5.84	733.0
768		79.83	82.66	100.60	9.41	886.3
769		79.68	82.24	100.70	11.31	944.0
770		79.48	81.64	100.40	14.38	1007.2
771	3/4/69	79.66	82.96	100.55	7.71	852.2
772		80.71	85.15	100.30	3.30	546.8
773		80.56	84.78	100.50	3.88	594.0
774		80.41	84.65	100.30	3.85	596.7
775		80.31	84.31	100.40	4.61	660.0

cont..

TABLE XI (cont.)

Heat Transfer Data for Unlined Tube 2

Tube Data: 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

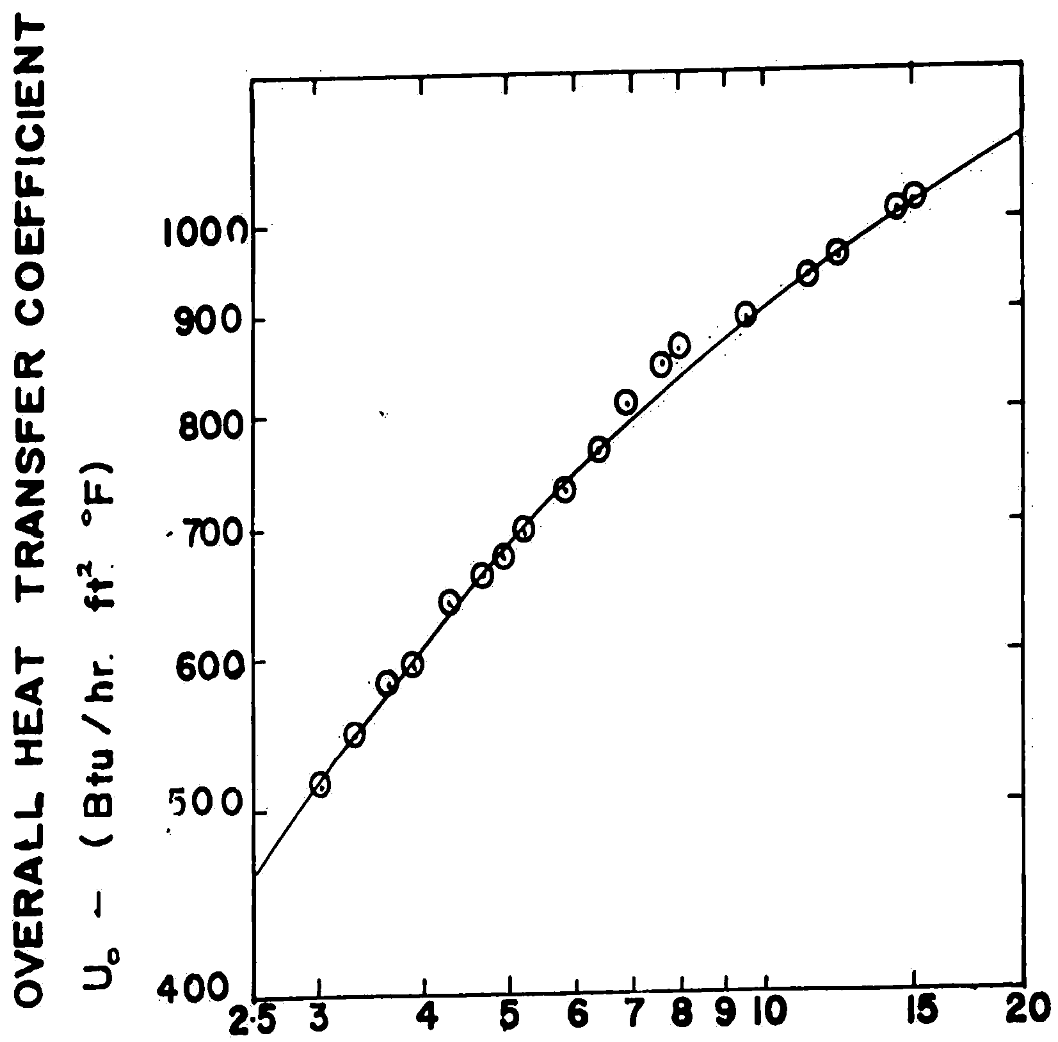
Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficient
776	3/4/69	80.21	84.01	100.40	5.20	698.9
777		79.89	83.31	100.20	6.42	763.6
778		79.50	81.90	100.30	12.17	959.8
779		79.55	81.69	100.70	14.97	1026.1

FIGURE 23

PERFORMANCE OF UNLINED TUBE 2

7/8" OD 18 BWG 90-10 Copper Nickel

Condenser tube



WATER VELOCITY

V - (ft/sec)

FIGURE 24

WILSON PLOT FOR UNLINED TUBE 2

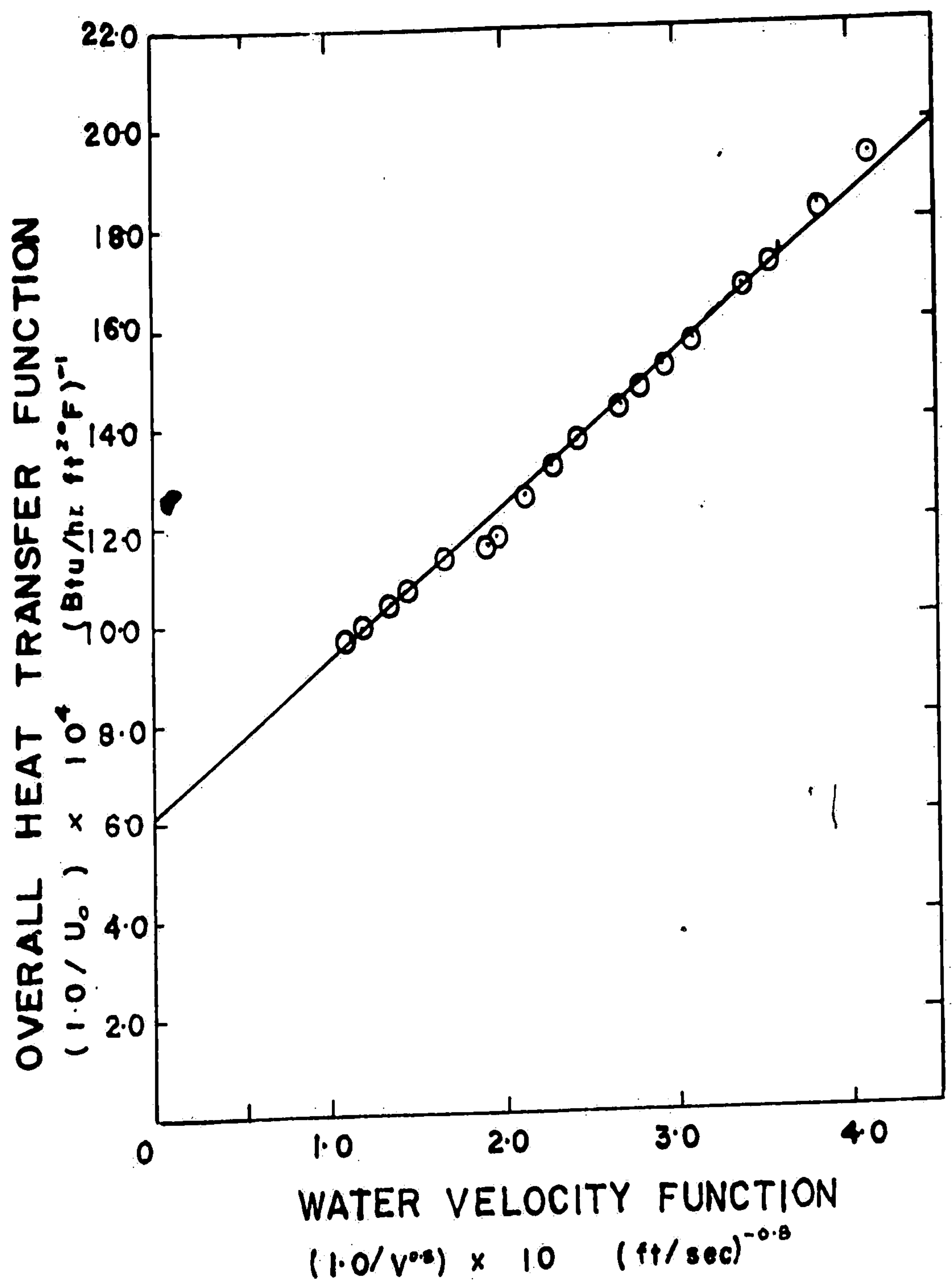


TABLE A-XII

Heat Transfer Performance of Teflon Lined Condenser Tubes B_1

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon, 0.0015" Thick on inside surface only

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficients
783	3/18/69	81.09	85.16	100.80	3.19	474.6
784		81.04	85.18	100.80	3.01	455.9
785		80.89	84.66	100.90	3.62	436.8
786		80.69	84.18	100.90	4.30	525.6
787		80.39	83.66	100.90	4.87	544.7
788		79.63	81.78	100.80	9.75	671.4
789		79.84	81.76	101.05	11.17	684.3
790		79.67	81.32	100.80	13.49	705.0
791		79.74	82.08	100.85	8.52	643.9

FIGURE 25

PERFORMANCE OF TEFLON LINED TUBE B1

Base 7/8" OD × 18 BWG 90-10 Cu-Ni

Condenser tube

Lining: Teflon ·0015" thick

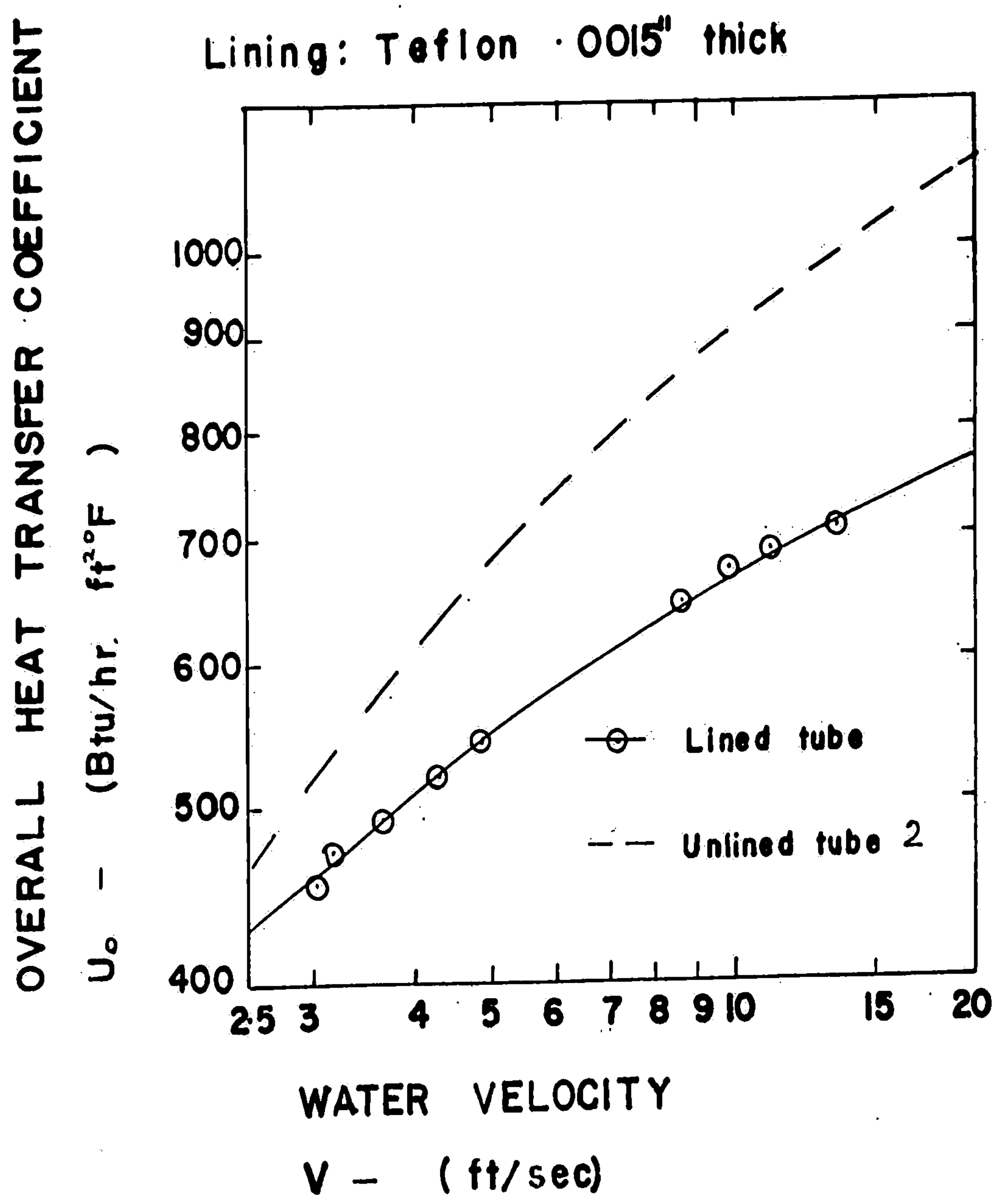


FIGURE 26

WILSON PLOT FOR TEFLON LINED TUBE B1

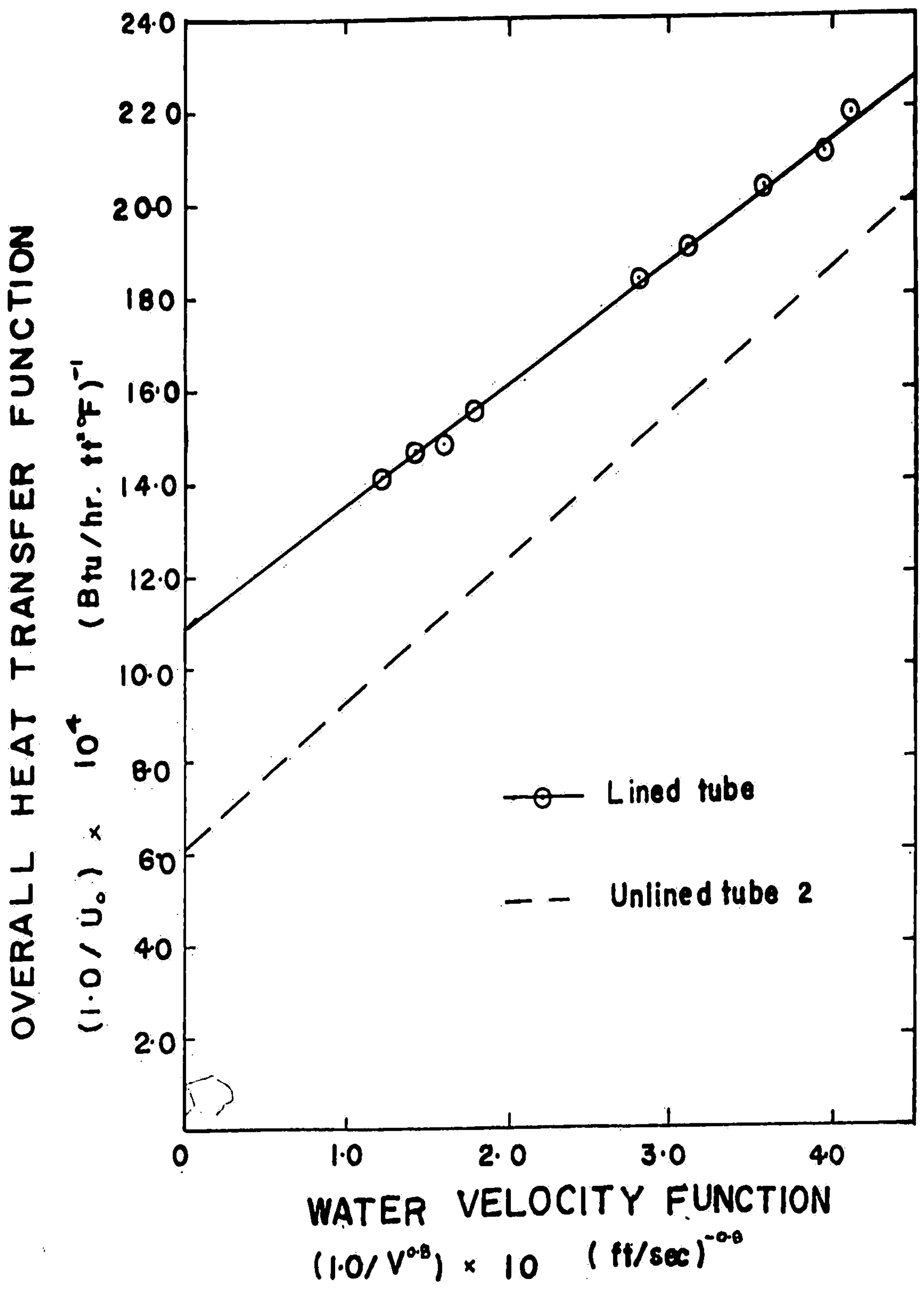


TABLE XIII

Heat Transfer Performance of Teflon Coated Condenser Tube B_2

Tube Data: Base - 7/8" O.D. x 18 BWG, 90-10 CuNi Condenser Tube

Coating - Teflon coating 0.0015" Thick on inside only

Run No.	Date	Water Temperature Inlet°F	Outlet°F	Steam Temp. °F	Water Velocity ft/sec	Overall Heat Transfer Coefficient
810	4/1/69	79.76	82.56	100.80	7.74	710.1
811		80.84	85.03	100.50	3.16	487.4
812		80.59	84.46	100.60	3.59	497.3
813		80.51	84.15	100.57	4.11	529.7
814		80.22	83.55	100.55	4.81	554.5
815		80.00	83.03	100.50	5.70	588.1
816		79.99	82.76	100.60	6.59	613.5
817		79.58	81.82	100.45	9.24	673.9
818		79.54	81.54	100.55	10.74	693.1
819		79.54	81.37	100.50	12.29	721.6
820		79.73	81.28	100.65	14.88	735.7
821		79.75	81.66	100.45	11.44	711.4
822		79.88	82.06	100.60	9.58	684.4
823		79.89	82.41	100.70	7.79	648.6
824		80.24	83.11	100.80	6.17	598.4

FIGURE 27

PERFORMANCE OF TEFLON LINED TUBE B2

Teflon thickness ·0015"

Base: 7/8" OD× 18 BWG 90-10 Cu-Ni

Condenser tube

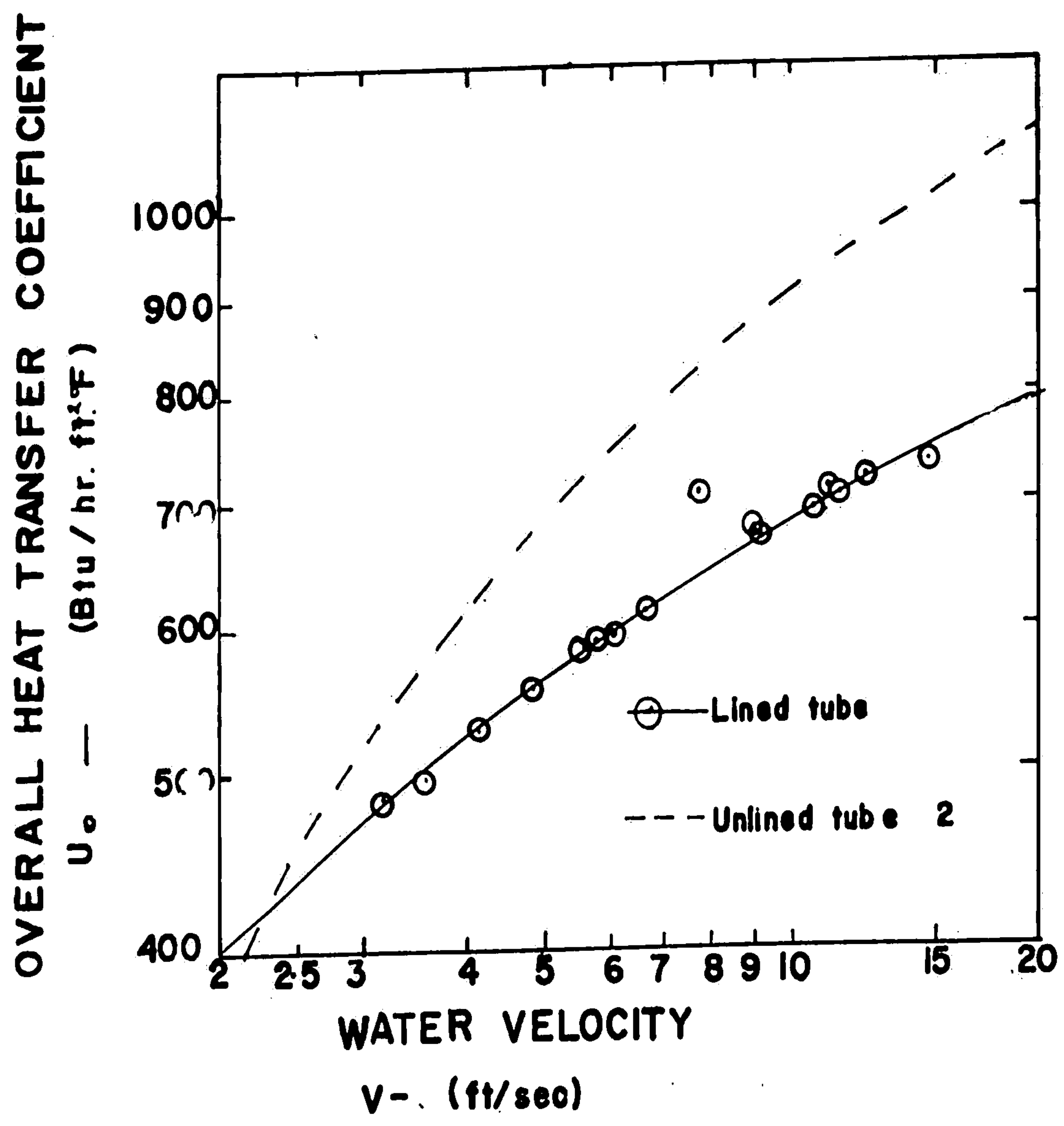


FIGURE 28

WILSON PLOT FOR TEFLON LINED TUBE B2

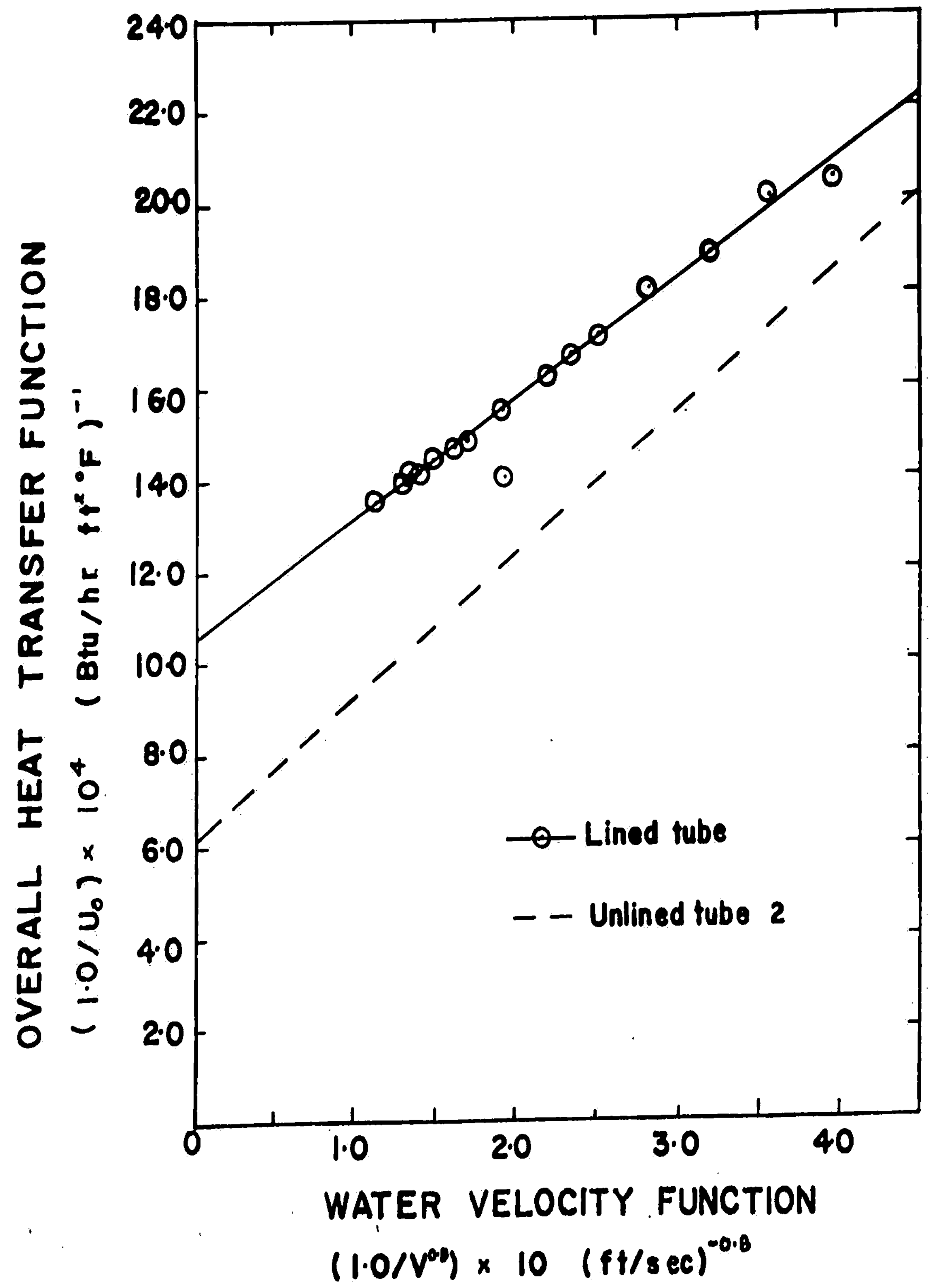
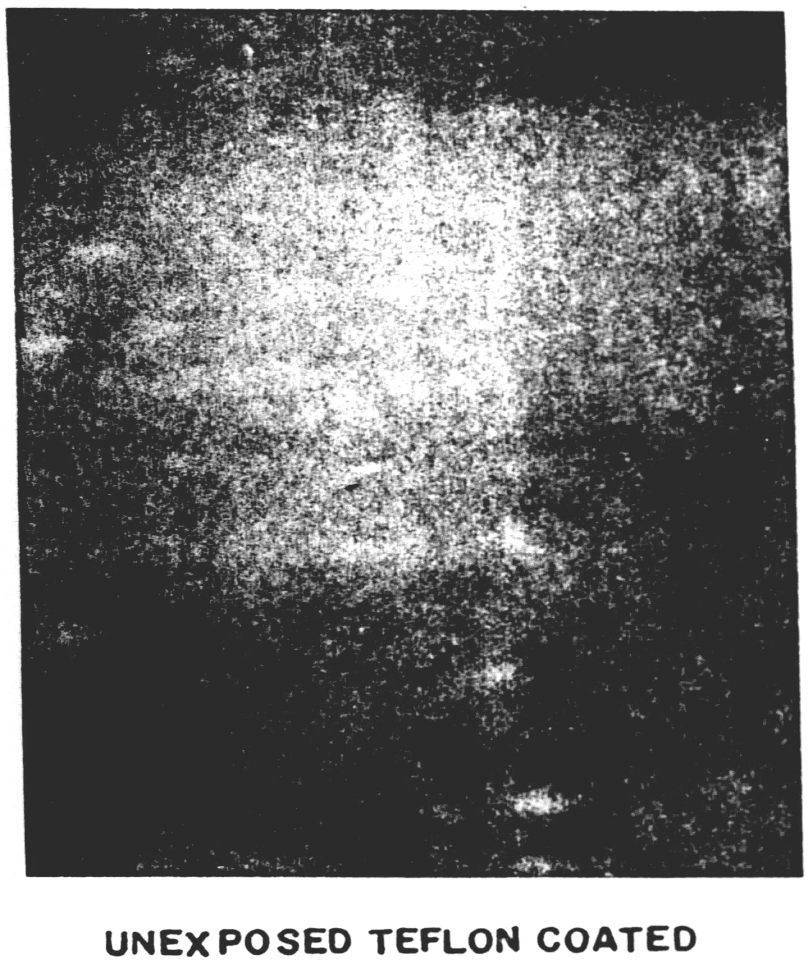


FIGURE 29

PHOTOMICROGRAPHS OF TEFLON COATED PLATE, MAGNIFICATION 75X



UNEXPOSED TEFLON COATED SAMPLE PLATE

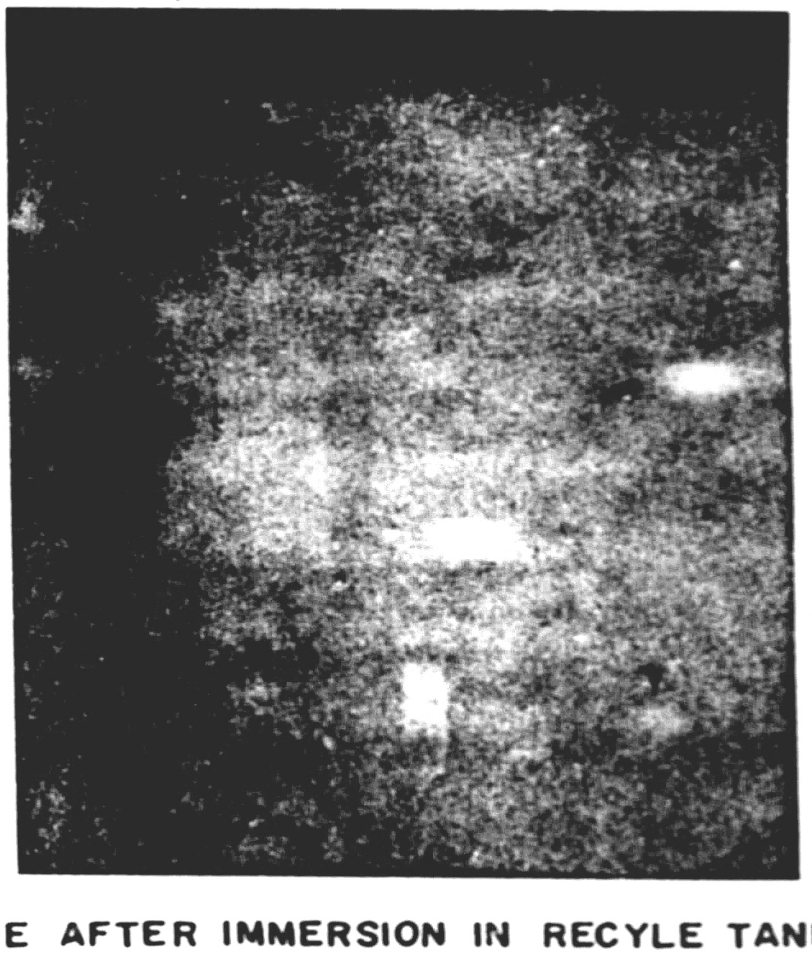


PLATE AFTER IMMERSION IN RECYLE TANK WATER FOR ONE WEEK

FIGURE 30

PHOTOMICROGRAPH OF TEFLON COATED PLATE, MAGNIFICATION 75 X

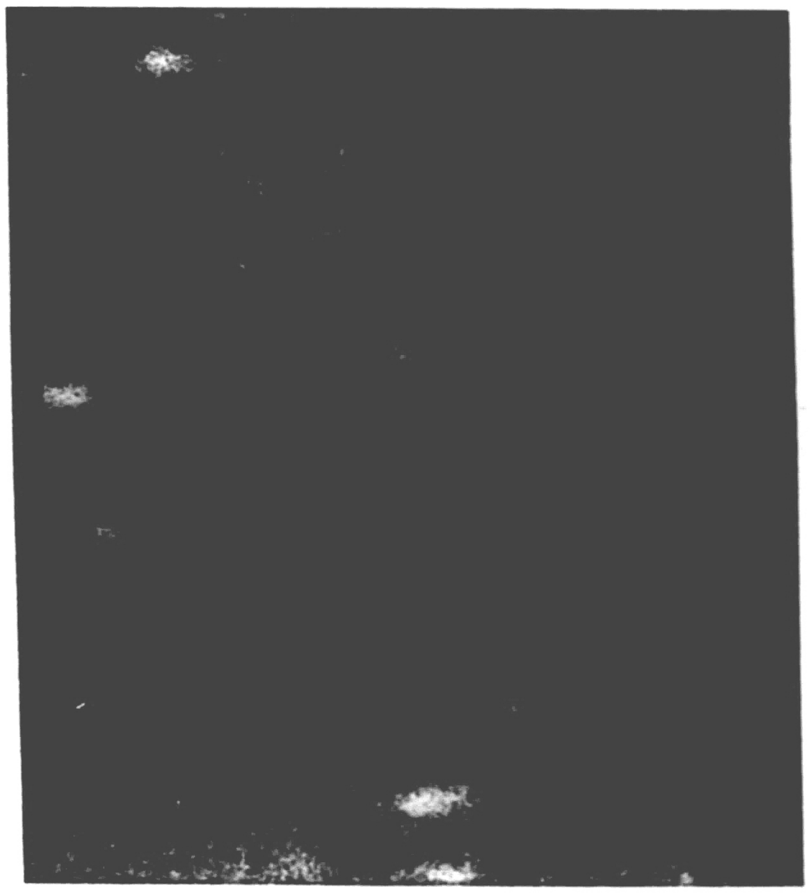
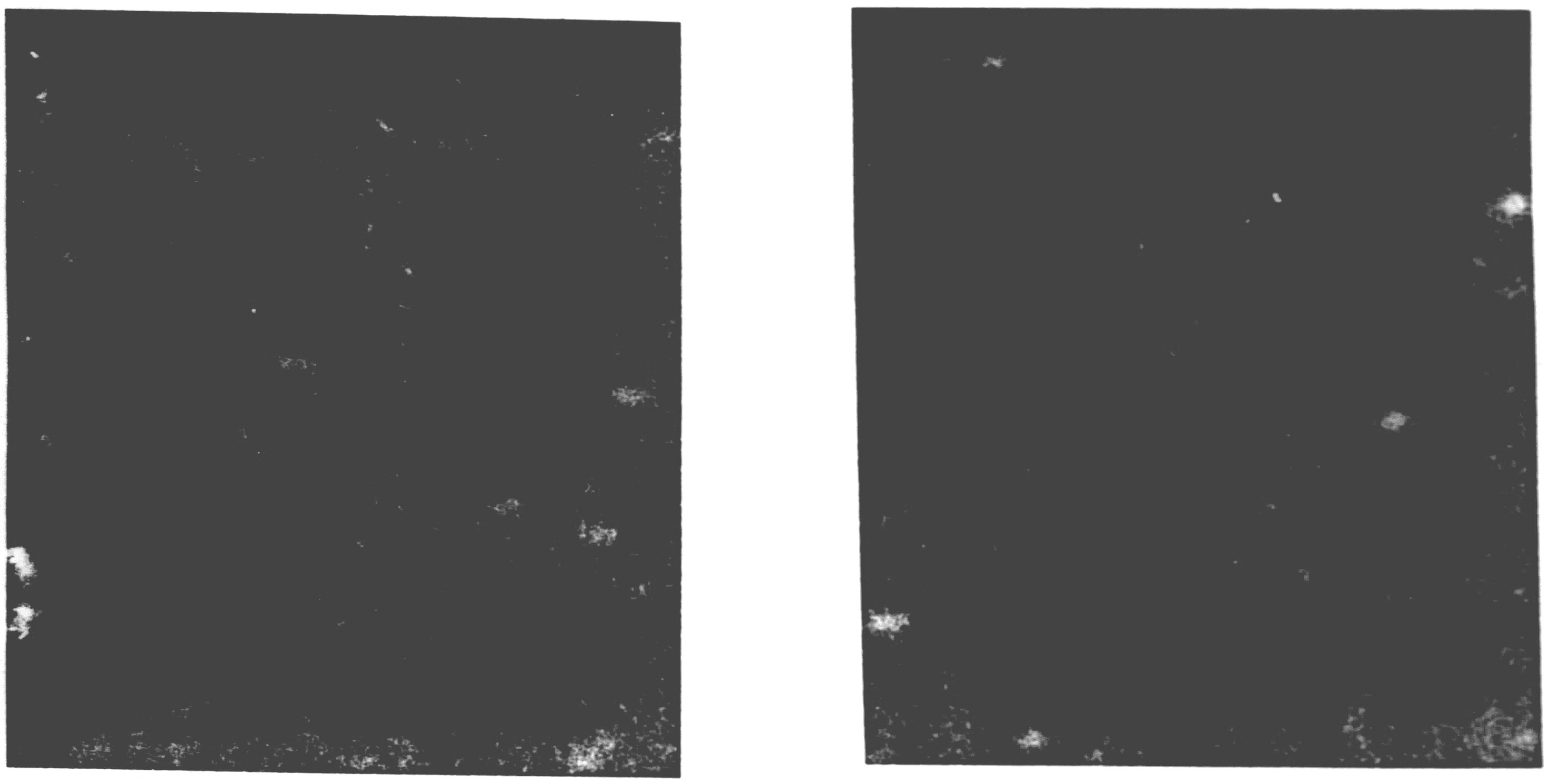


PLATE PREVIOUSLY IMMERSED IN RECYCLE TANK WATER AFTER TREATMENT WITH $SnCl_2$/HCl CLEANING SOLUTION

FIGURE 31

PHOTOMICROGRAPHS OF TEFLON COATED INTERNAL CONDENSER TUBE SURFACE

MAGNIFICATION 75 X



UNUSED TUBE A2

TUBE A2 AFTER CALORIMETER TESTING

a more even and consistent surface.

The effect of changing outside coefficient was minimized by carrying out the runs continuously (no shut downs). Results are indicated on Tables 12 and 13 and Figures 25 to 28.

Tube B_1 has a slope of .00262 (a = .0322) and Tube B_2 has a slope of .00268 (a = .0324) giving an average increase in slope of 18.8%.

The agreement of the above results is very satisfying though it must be acknowledged that since scatter was prevalent, the extreme closeness of the slopes was to some extent a coincidence. Though no quantitative measurements were made, examination under a microscope revealed that the inner surfaces of Tubes B_1 and B_2 were smoother and more even than earlier Teflon lined tubes. The slope increase for Tubes B_1 and B_2 is 18.8% as compared to the 26.9% increase for earlier Teflon lined tubes. The relatively smaller percentage increase in inside heat transfer coefficient for Tubes B_1 and B_2 could be due to decreased surface roughness.

SUMMARY

The effect hydrophobic materials may have on liquid phase heat transfer was investigated using 7/8" O.D. x 18 BWG, 90-10 Copper Nickel alloy condenser tubes lined on the inside with Teflon, Epoxy, Paralene-N and Gold.

Apart from Teflon, no other hydrophobic lining tested appeared to change the inside heat transfer coefficient. The Teflon lined tubes tested gave an average inside coefficient increase of 24%. This increased performance decayed with time until the tube performed similarly to an unlined tube of the same wall resistance.

Pressure drop measurements indicated that any change in friction factors between "wetted" and "non-wetted" flow may be attributed almost entirely to surface roughness.

The fact that the change in inside heat transfer coefficient was a surface rather than roughness phenomenon was illustrated by the use of detergent. A reduction of the inside heat transfer coefficient to the value obtained with the unlined tube was observed.

One Teflon lined tube gave results only slightly different from those of an unlined tube. This result

may have been caused by traces of detergent remaining in the system. Such a possibility does not seem too unreasonable in view of the fact that surface specialists confirm that detergent can be extremely persistent and even trace amounts can effect surface phenomena.

Teflon, among the hydrophobic linings, was singular in producing a change in inside heat transfer coefficient.

The term "non-wetting" or "hydrophobic" is somewhat misleading. While the hydrophobic linings tested had an air/water/lining contact angle greater than the air/water/Cu-Ni surface, no other material had so distinctly high a contact angle as did Teflon (108°) or for that matter one greater than 90°. If the increased heat transfer coefficient was indeed the result of a surface effect then only with Teflon might one expect a sharp change.

The apparent decay of performance with time is at first unexpected since Teflon is quoted in literature as being remarkably inert. However, it is not clear how much study has been carried out to investigate the absorptive properties of Teflon i.e. the tendency of Teflon by virtue of its surface characteristics to attract a surface layer of material in the form of scale.

Teflon has in some ways been known to behave strangely. Fox and Zisman [7] report that Teflon after exposure to air for several days has at times given a contact angle of less than 90°. Certain surface specialists now claim that the presence of inorganic oxides (iron oxide and copper oxide were abundant in our system) have been known to alter the surface properties of Teflon.

The restoration of the initial surface properties with the stannous chloride/hydrochloric acid treatment reinforces the theory that the alteration of surface properties after a period of time is not because of a chemical change but more as a result of scaling material adhered to the Teflon superficially but in sufficient concentration to change the surface properties.

CONCLUSIONS

1. Fanning friction factors for the Teflon lined tube are what one might expect for "wetted" flow in a tube of the same roughness. Axial slip at the wall of the conduit, if it exists, must be small enough so as not to affect the friction factor significantly.

2. Teflon lined tubes produce an increase in heat transfer performance initially. Paralene-N, Gold and Epoxy produce no similar effect.

Flow patterns past the wall are possibly altered, hence changing the nature of the laminar sub-layer where a significant portion of the heat transfer resistance lies.

3. The enhancement of the heat transfer performance decays with time, perhaps due to an alteration in the surface properties as a result of buildup of scaling material.

4. Ways of preventing scale buildup may be investigated by:

(a) use of water with little or no inorganic oxides

(b) use of additives in the process water which might inhibit scale buildup.

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APPENDIX

NOTATION

A. Symbols

A = Area in ft^2

C_p = Specific heat at constant pressure in Btu/lbm ^oF

D = Diameter in ft.

f = Fanning Friction Factor

h = Individual film heat transfer coefficient in Btu/hr ft^2 ^oF

k = Thermal conductivity in Btu/hr.ft ^oF

L = Tube length in ft.

Pr = Prandtl Number = $\frac{C_p\mu}{k}$ in dimensionless form

R = Resistance to heat transfe in hr.ft^2 ^oF/Btu

Re = Reynolds Number = $\frac{\rho v D}{\mu}$ in dimensionless form

s = Inside cross sectional area of tube in ft^2

T = Temperature in ^oF

U = Overall heat transfer coefficient in Btu/hr. ft^2 ^oF

v = Linear velocity in ft/sec

w = Mass rate of flow in lbm/hr

μ = Viscosity in lbf/hr.ft

ρ = Density in lbm/ft^3

B. Subscripts

b = Property evaluated at bulk temperature

m = Property evaluated at mean value of inside and outside of wall of tube

o = Property evaluated at outside wall of tube

s = Property of steam

w = Property evaluated at wall of tube

lm = Logarithmic mean value

i = Property evaluated on inside wall of tube

SAMPLE CALCULATIONS

Calculation of the Overall Heat Transfer Coefficient.

The following calculations were made from the data of Run 39, Table V.

Data:

T_1	= Inlet water temperature	= 80.89°F
T_2'	= Outlet water temperature	= 83.69°F
T_s	= Steam condensing temperature	= 101.60
v	= Water velocity	= 5.43 ft/sec
I.D.	= of tube	= 0.775 in.
O.D.	= of tube	= 0.875 in.
Length of tube in calorimeter		= 5.0 ft
C_p	for water	= 1.0Btu/lbm °F
ρ	for water	= 62.3lbm/ft³

Calculations:

A_o = Outside tube area $= \pi D_o L$

$$= \frac{(0.875 \text{ in.})(5.0 \text{ ft})}{(12 \text{ in/ft})}$$

$= 1.145 \text{ ft}^2$

s = Inside cross sectional area $= \dfrac{\pi D_i^{\ 2}}{4}$

$$= \frac{\pi (0.777 \text{ in})^2}{(4)(12 \text{ in/ft})^2}$$

$= .00327 \text{ ft}^2$

$$T_s - T_1 = 101.6\text{-}80.89 = 20.71^{\circ}F$$

$$T_s - T_1 = 101.6\text{-}83.69 = 17.91^{\circ}F$$

$$\frac{T_s - T_1}{T_s - T_2} = \frac{20.71^{\circ}F}{17.91^{\circ}F}$$

$$= 1.155$$

$$w = (v \text{ ft/sec})(s \text{ ft}^2)(\quad \text{lbm/ft}^3)(3600 \text{ sec/hr})$$

$$= (5.43 \text{ ft}^3/\text{sec})(.00327 \text{ ft}^2)(62.3\text{lbm/ft}^3)(3600 \text{ sec/hr})$$

$$= 3980 \text{ lbm/hr}$$

$$U_o = \frac{w\, C_p}{A_o} \ln \frac{T_s - T_1}{T_s - T_2}$$

$$= \frac{(3980\text{lbm/hr})(1 \text{ Btu/lbm}^{\circ}F)}{1.145\text{ft}^2} \ln 1.155$$

$$= \frac{(2.303)(3980\text{lbm/hr})(1 \text{ Btu/lbm}^{\circ}F)}{1.145\text{ft}^2} \log_{10} 1.155$$

$$= 505.4\text{Btu/hr}^{\circ}F \text{ ft}^2$$

A theoretical estimation of overall heat transfer coefficient for different water velocities for a 7/8" O.D. x 18 BWG, 90-10 Copper Nickel alloy tube

Consider Equation (3) in Theoretical Background

$$\frac{1}{U_o} = \frac{1}{h_o} + \frac{x_w A_o}{k_w A_m} + \frac{A_o}{h_1 A_1}$$

Representing $\frac{1}{h_o}$ by R_o

$\frac{x_w A_o}{k_w A_m}$ by R_w

and $\frac{A_o}{h_1 A_1}$ by R_1

$$R_w = x_w A_o / k_w A_m \qquad \frac{A_o}{A_m} = \frac{(0.875\text{in.})}{(0.826\text{in.})} = 1.060$$

$$R_w = \frac{0.0149\text{in.} \times 1.060}{(12\text{in./ft}) \times 27\ (\text{Btu/hr.ft}^{o}\text{F})}$$

$$= .00160\ (\text{Btu/hr.ft}^2\ {}^{o}\text{F})^{-1}$$

$$R_o = \frac{1}{h_o} = .002286$$

(Perry, Chem. Eng. Handbook)

$$R_1 = \frac{1}{h_1} = \frac{A_o/A_1}{\frac{k}{D_1}(0.023)(\text{Re})^{0.8}(\text{Pr})^{0.4}}$$

$$\frac{A_o}{A_1} = \frac{(0.875\text{in.})}{().777\text{in.})} = 1.126$$

$$Pr = \frac{C_p \mu}{k}$$

$$= \frac{(1.0\text{Btu/lbm}^o\text{F})(0.85 \times 2.42\text{lbf/ft})(g_c/g \ \text{lbm/lbf})}{(0.358\text{Btu/hr.ft}^o\text{F})}$$

$$= 5.75$$

$$(Pr)^{0.4} = 2.014$$

$$\frac{k}{D_1} = \frac{(0.358\text{Btu/hr.ft}^o\text{F})(12\text{in./ft})}{(0.777\text{in.})}$$

$$= 5.52$$

$$Re = \frac{D_1 \rho v}{\mu}$$

$$= \frac{(0.777\text{in.})(v \text{ ft/sec})(62.3\text{lbm/ft}^3)}{(12\text{in./ft})(.00672\text{lbf/ft.sec.})}$$

$$= 7040v$$

$$Re^{0.8} = 1200v^{0.8}$$

Therefore

$$R_1 = \frac{1.126}{(5.52)(.023)(1200)(v^{0.8})(2.014)}$$

$$R_1 = \frac{0.00351}{v^{0.8}}$$

V ft/sec	$V^{0.8}$	R_i	R_o+R_w	$R_T=\frac{1}{U_o}$	U_o	$\frac{1}{V^{0.8}}$
2	1.74	.00202	.00389	.002409	415	0.575
5	3.63	.000968	.00389	.001357	738	0.276
8	5.28	.000665	.00389	.001054	948	0.189
12	7.30	.000481	.00389	.000870	1150	0.137

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