1949

Further studies - heat transfer coefficients for liquids in forced convection normal to banks of staggered tubes

Terrell C. Claunch

FURTHER STUDIES--HEAT TRANSFER COEFFICIENTS FOR LIQUIDS IN FORCED CONVECTION NORMAL TO BANKS OF STAGGERED TUBES

By

TERRELL C. CLAUNCH

A

THESIS

submitted to the faculty of the

SCHOOL OF MINES AND METALLURGY OF THE UNIVERSITY

OF MISSOURI

in partial fulfillment of the work required for the

Degree of

MASTER OF SCIENCE, CHEMICAL ENGINEERING

Rolla, Missouri

1949

Approved by Frank H. Conrad
Professor of Chemical Engineering

ACKNOWLEDGMENT

The author wishes to express his appreciation to Dr. Frank H. Conrad for his fine guidance and cooperation in connection with the work presented herein.

Also to Dr. J. B. O'Hara for his interest and assistance so freely given.

CONTENTS

LIST OF ILLUSTRATIONS

FURTHER STUDIES--HEAT TRANSFER COEFFICIENTS FOR LIQUIDS IN FORCED CONVECTION NORMAL TO BANKS OF STAGGERED TUBES

INTRODUCTION:

The problem for these experiments may be divided into two sections. The first section of the problem consists of setting up procedure for duplicating data. The second section is that of studying and explaining characteristics of the resulting curves. Correlation of the data obtained will be necessary in order to obtain the curves.

In order that the first section of the problem could be solved it was necessary to establish a standard of cleanliness for the apparatus used. Establishment of procedures for the operation and cleaning of the apparatus was necessary.

Before Kenneth E. Rudert (1) began his design and construction of the apparatus for these experiments he made a literature survey on this particular phase of heat exchange. Work notes from this search were available at the beginning of this problem. Further search of the literature failed to reveal any data not already considered.

At the time the original literature survey was made for this problem there was very little pertinent data recorded. The only comprehensive experimental data found were those of Perrone (2) for a small commercial

(1) Rudert, Kenneth E., Thesis, School of Mines and Metallurgy of the University of Missouri, 1948. The Design, Construction, and Testing of a Heat Exchanger for Determining Heat Transfer Coefficients for Liquids in Forced Convection Normal to Banks of Staggered Tubes.

(2) Perrone, S. H., Oil Gas Journal, March 28, 1935, pp. 71

exchanger having cutout type baffles. Water was the only fluid that was studied. An equation of the dimensionless type was derived. This equation includes factors such as diameter of the shell, number of baffles, and pitch for the tubes that were not considered in this work.

It was found by Rudert (1) that data from the exchanger used in this problem could be correlated by a plot of the dimensionless groups $\frac{\text{Nusselt number}}{\text{Prandtl number}^{0.3}}$ vs. Reynold's number (modified). These groups were used in the correlation of the data of these experiments.

In his thesis Rudert (1) shows that the steam film offers negligible resistance to the flow of heat and that the temperature of the steam may be assumed to be the average water surface temperature without appreciable error. These calculations have been conducted on this basis.

Perrone (2) states that the losses from his apparatus by radiation were negligible. Due to the differences in the construction of the apparatus used in these experiments, radiation losses were checked.

DISCUSSION OF PROCEDURE USED IN THE OPERATION OF THE APPARATUS FOR DUPLICATION OF DATA:

Reference is made to the schematic diagram of the apparatus used in these experiments, Figure 1, in the following study of procedure for the operation of the heat exchanger so that reproducible data may be obtained.

Set the measuring drum 1, on the scales 10, and set the scales on the desired reading for the amount of water to be used in the computation of the rate of water flow. (400 pounds were used in these experiments.) Open the valves controlling the incoming water 16, slightly and turn on the pump 17. Open the valve controlling the incoming steam 8, cautiously at first. Blow air from steam line 11, and drain 15. Close valve at 15. Allow valve at 11 to remain open enough to insure an excess of steam at all times. Open the incoming water valve 16, to maximum. Open incoming steam valve 8, to maximum. (The maximum working pressure of the piece of equipment is 150 pounds per square inch.) Roll the scale under the outlet and clock at zero time. Mark time when the scales balance on 400 pounds. Read the temperatures of the thermometers 9, 13, 4, 5, and 3 when constant. Caution! This is a problem in the steady state of heat transfer. The temperatures must have reached a steady state before the data will be of value. Slow the water rate by use of the incoming water valve 16, judging the amount of decrease by use of the mercury manometer, not shown on the diagram. Allow the operation to become steady while measuring the water output by use of the platform scale and measuring drum again. Read all temperatures and record. Repeat for other points along the rate of water flow curve. When the lower rates of water flow have been reached, the more sensitive acetylene-tetra-bromide manometer may be placed on stream by opening the valve in the manometer line, not shown in the diagram.

When the lower rates of flow have been reached, it may be found necessary to cut the rate of steam input 8, in order to prevent the water from reaching its boiling point.

A comparison of data obtained by the above procedure with that of data obtained by Rudert (1) is shown in Figure 2.

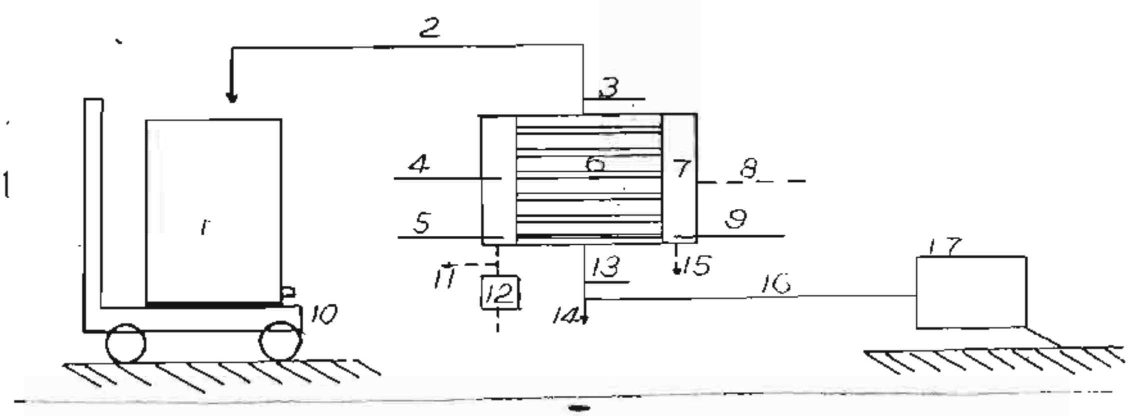

Figure 1

Schematic Diagram

1 Drum
2 Out Water
3 Out Water Thermometer
4 Out Steam Thermometer
5 Condensate Thermometer
6 Cross Flow Section
7 Steam Chest
8 Steam in Line
9 Steam in Thermometer
10 Platform Scales
11 Condensate Out Tap
12 Steam Trap
13 Water in Thermometer
14 Drain
15 Drain
16 Water in Line
17 Pump

SCALE EFFECTS AND PROCEDURE FOR CLEANING:

When the heat exchanger had been on stream for a total of about thirty hours, the cover was removed for inspection of tube surface. It was found that a scale had collected on the upper rows of tubes to a thickness of 1/32 inch. On the lower rows of tubes the thickness of the scale deposited was only a thin film. A comparison of data taken from clean tubes with data taken from tubes covered with scale is made in Figure 3, page 7.

When the apparatus is to be cleaned, it is necessary to remove the head. This is done so that the cleaning procedure may be checked for thoroughness each time. It was determined by experiment that a solution of near 0.05 N HCl would best clean the tube surface with a minimum of damage to the apparatus. Procedure used was as follows: Fill the exchanger with water. Drain into a container. Pour back into the exchanger about two gallons of water in order that the valves in the incoming water line may be protected. Carefully mix about 800 c. c. of 12 normal HCl with the water remaining in the container. Add the weak acid solution to the exchanger and allow to stand until the evolution of CO_2 ceases. Drain for inspection. Repeat if the tube surface is not clean. Wash out the exchanger well with water.

If the apparatus is cleaned at the end of each 20 hours operation little scale effect will be noted in the data obtained. Results of two runs made after each of two cleanings are shown in Figure 10. Agreement of points within experimental accuracy is indicated.

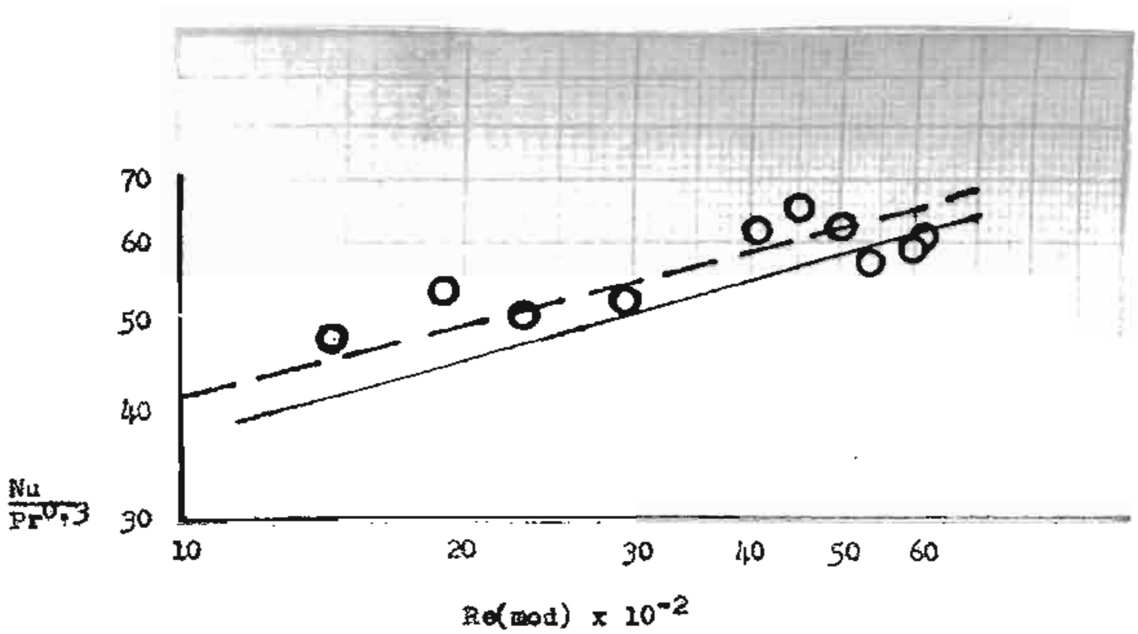

Figure 2

Duplication run. Points run for duplication of Rudert's data, - - -. Solid line shows data corrected for heat gain from tube sheet.

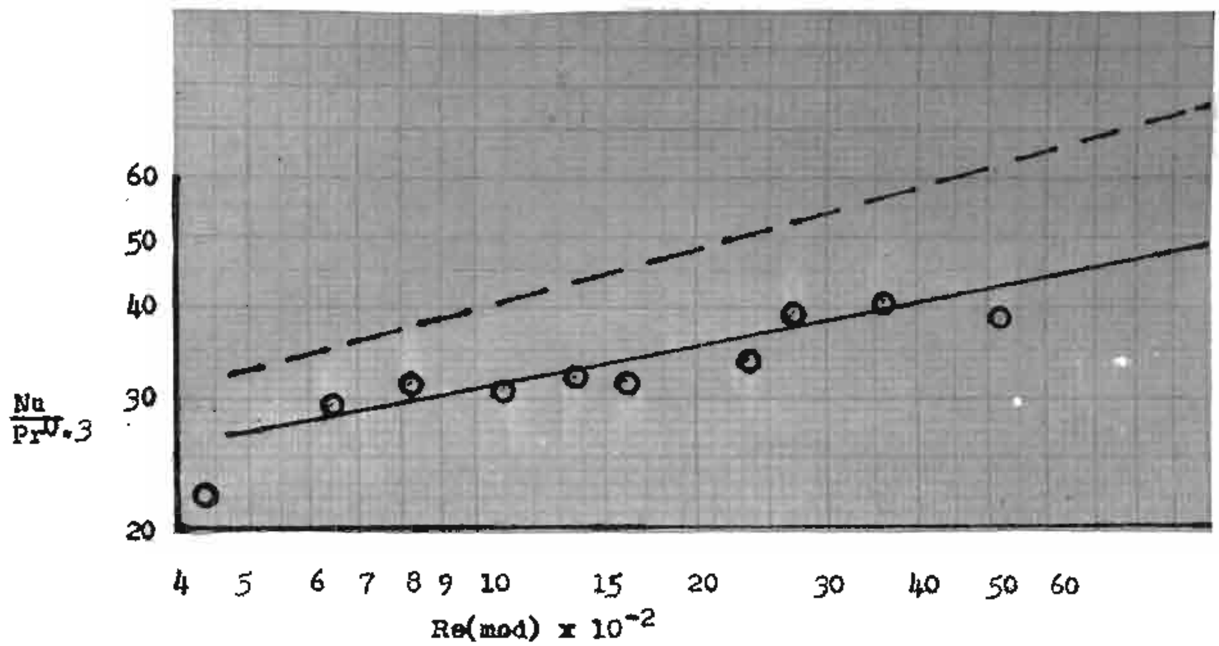

Figure 3

Comparison of clean to fouled tubes. Dash shows clean tube runs.

EFFECTS OF CONDENSATE FORMATION:

The apparatus was operated to obtain the data illustrated in Figure 4, by a reverse procedure to that given previously. Constant heating surface temperature was maintained (within 10°F) for the run. Steam outlet valve was kept in the closed position. Condensate forming faster than it was being carried out of the tubes would account for the drop seen in the middle of the curve. Condensate forming on the inner surface and not being blown away would reduce the effective heat transfer area thus accounting for the lowering of the y-axis values. As the water rates were increased there would be an increase in the amount of condensate forming. This would tend to reduce the effective area of heat transfer even further with a resulting droop in the curve. At the same time the increasing water rates were increasing the rate of condensate formation, some of the adverse effects were compensated for by the thickness of the outside water film being reduced. This would account for the tendency of the curve to climb again in the higher x-axis values.

That the effect of the formation of condensate is most unpredictable is brought out by figure 5. Although the curve has the same droop and swell characteristics as Figure 4, these irregularities do not occur at the same values of Reynolds number. In these experiments many such sets of data were obtained. Varied attempts were made to find some method of correlating this data, as at the time it was thought that the droop and swells in the curves were due to some changes on the exterior of the tubing and not on the inside. After no method of satisfactory correlation was forthcoming for the unsmooth data, the procedure outlined above was tried. The relatively smooth data of Figure 2 was obtained.

The condensate condition present in this equipment may be sensitive to several factors. The temperature of the heating surface used for the particular point being read may have effect on the rate at which the condensate may be bourne away from the area of formation. Directly connected with this is, of course, the availability of the quantity of the steam used. If the steam consumption is low in the slow rates of water or cooling medium flow, there is plenty reserve steam available for use in forcing the condensate to the trap and out of the system. If the steam consumption is high as in the higher rates of cooling, the capacity of the incoming steam lines may be approached, resulting in little or no reserve steam for use in disposing of the condensate formed. As shown in Figure 5 at values of Reynolds number above 6000 the corresponding y-axis values may be affected in varied amounts. The seven points made reference to were taken at almost the same rates of water flow. The variation in the Reynolds number was caused by changes in viscosity of the film due to change in film temperatures. The amount of steam flowing in excess through the cracked discharge valve was, in all cases, not as much as was desired. Had the desired amount of steam been available, it is thought that a point for this rate of water flow could have been obtained which would have fit into the straight line pattern.

A test run was made on the bundle of one-half inch tubes installed in place of the 5/8 inch ones previously used. Results of this run are compared with data from the 5/8 inch bundle in Figure 6. The data from the two runs shown are on comparable basis as to procedure of operation and heating surface temperatures used. It seems that due to the decreased diameter and area of cross-section the accumulation of condensate was greater than in the larger tube bundle.

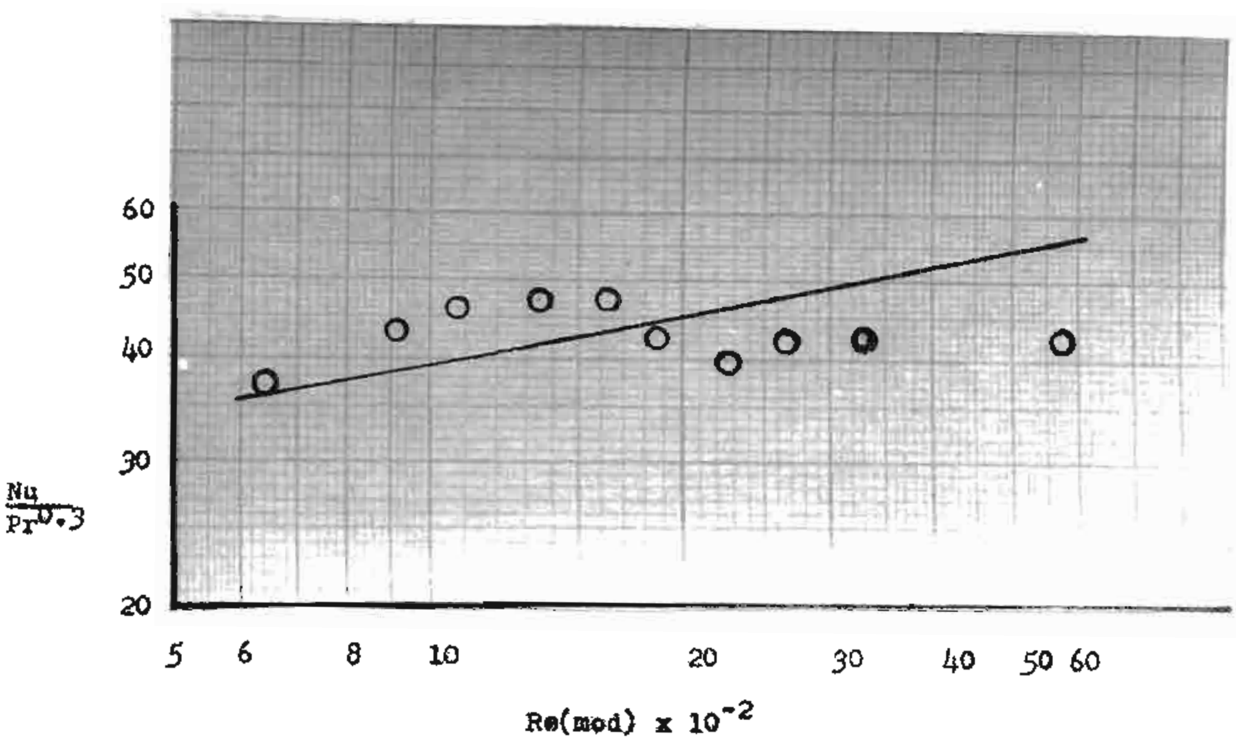

Figure 4

Illustration of distortion due
to condensate condition.

A consideration of the volume of the inside of the tubes shows:

$V\ (5/8") = (90)\ (13/12)\ (0.0521)^2\ (0.7852) = 0.209\ ft.^3$

$V\ (\frac{1}{2}") = (121)(13/12)(0.0417)^2\ (0.7852) = 0.179\ ft.^3$

The difference in the total inside volumes is 0.03 ft.^{3} or 14.3%. This difference in volume may be offered as a reason for the evidenced lag in the conduction of the condensate away from the point of formation.

As may be noted, above values of 3000 on the x-axis, the experimental points fall away sharply from a straight line, although the half inch outlet for the condensate was kept completely open throughout the run. Had the quantity of steam abailable been up to that desired, it is thought that the value of the point for Re = 5000 would have reached y-axis value of 50, which would have given a straight line.

The studies of heat transfer coefficients have been made general by considering the effects of many varying conditions on the values of $\frac{\text{Nusselt number}}{\text{Prandtl number}^{0.3}}$ rather than the effects on the individual coefficient. A record of typical values of h, the heat transfer coefficient with respect to the water film, are presented in Table Number 1.

Table Number 1

For 5/8 inch Tube Bundle		For 1/2 inch Tube Bundle	
h $\frac{BTU}{(hr)(ft)^2(^\circ F)}$	Reynold's No. modified	h $\frac{BTU}{(hr)(ft)^2(^\circ F)}$	Reynold's No. modified
593	6250	517	4510
590	5990	529	4460
593	6120	505	4215
557	5440	506	3930
615	5060	520	3560
648	4610	544	3370
615	4080	562	3235
572	3850	568	2920
511	2980	564	2645
483	2295	545	2340
504	1900	536	2115
444	1275	522	1680
		548	1152
		464	1014
		492	835

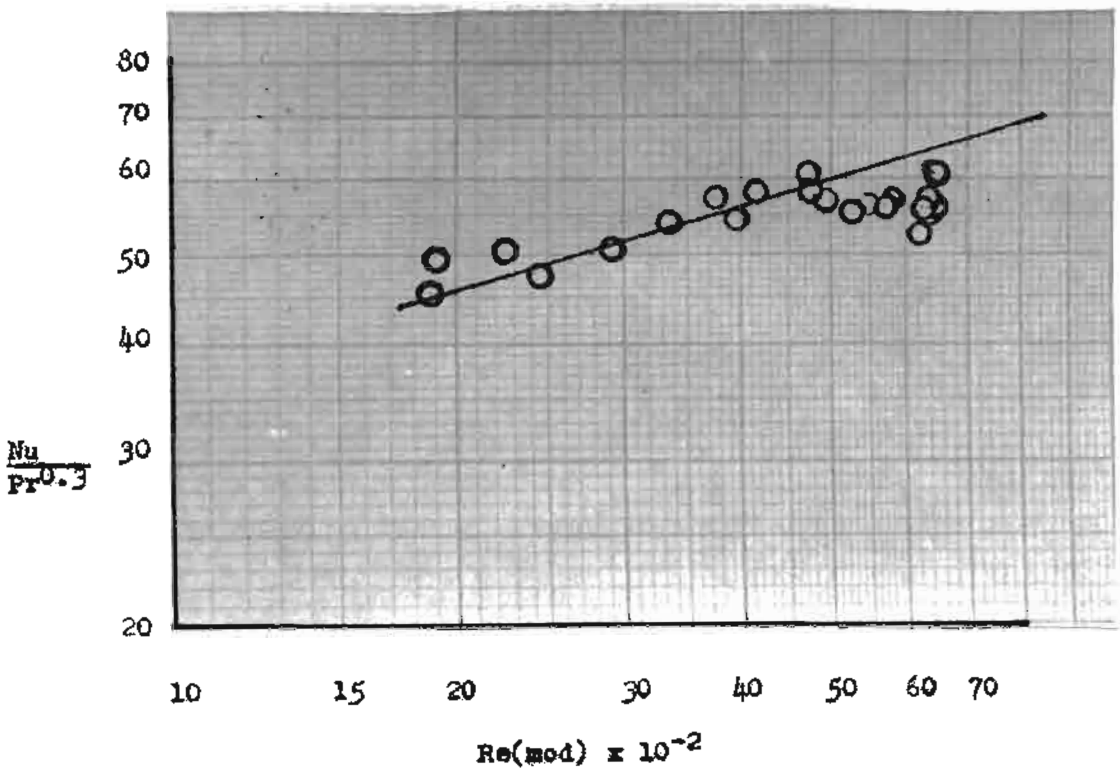

Figure 5

Effect of condensate gathering inside tube: at high water rates.

EFFECTS OF VARIATION OF HEATING SURFACE TEMPERATURE:

Runs were made for the purpose of studying the effects of varying heating surface temperatures. A water rate was established using a relatively low heating surface temperature. When the operation was steady the data was recorded. With the same water rate, the steam temperature was increased, and this data recorded. The same water rate was held and the steam temperature increased by increments of two or three degrees until the maximum steam temperature was reached. The steam temperature was decreased, the water rate changed, and data for another of the slashes seen in Figures 7 and 8 was taken in a like manner.

EFFECTS OF VARIATION OF TEMPERATURE OF WATER OUTLET:

It was found on further experiment that a change in the mechanism of heat transfer could account for the high points at the low values of Reynold's number shown in Figure 9. The change in mechanism was the change from the transfer of heat to a liquid in forced convection, to the transfer of heat to a boiling liquid.

In order that this change in mechanism could be observed, a small piece of apparatus was set up. A piece of 5/8 inch copper tube, of the type used in the first bundle of tubes tested, was inserted into a glass condenser. Steam connections were made for running steam through the tube and proper water attachments led cooling water through the condenser in the usual manner. When the small apparatus was put into operation and the discharge temperature of the water was brought to 187°F (steam temperature 220°F) three distinct areas of heat transfer could be seen on the surface of the copper tube.

Where the water entered the tube the surface of heating area showed no turbulence. About a third of the way up the tube toward the water

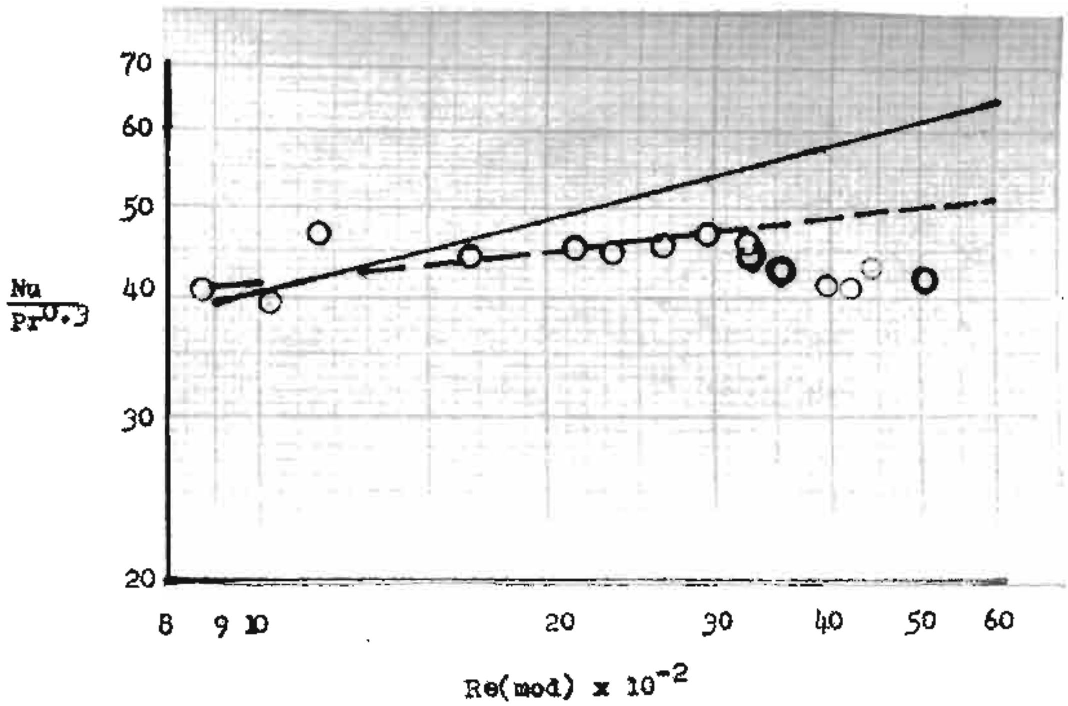

Figure 6

Comparison of 5/8 inch, solid line, with ½ inch tube bundles.

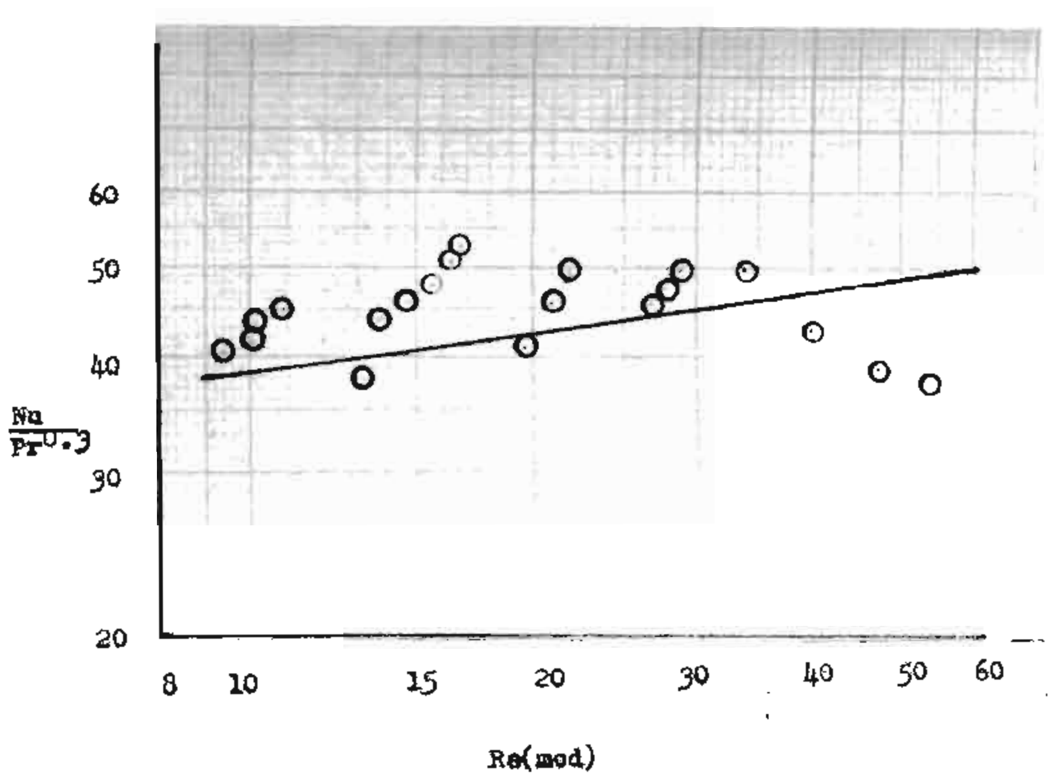

Figure 7

Steam temperature effects

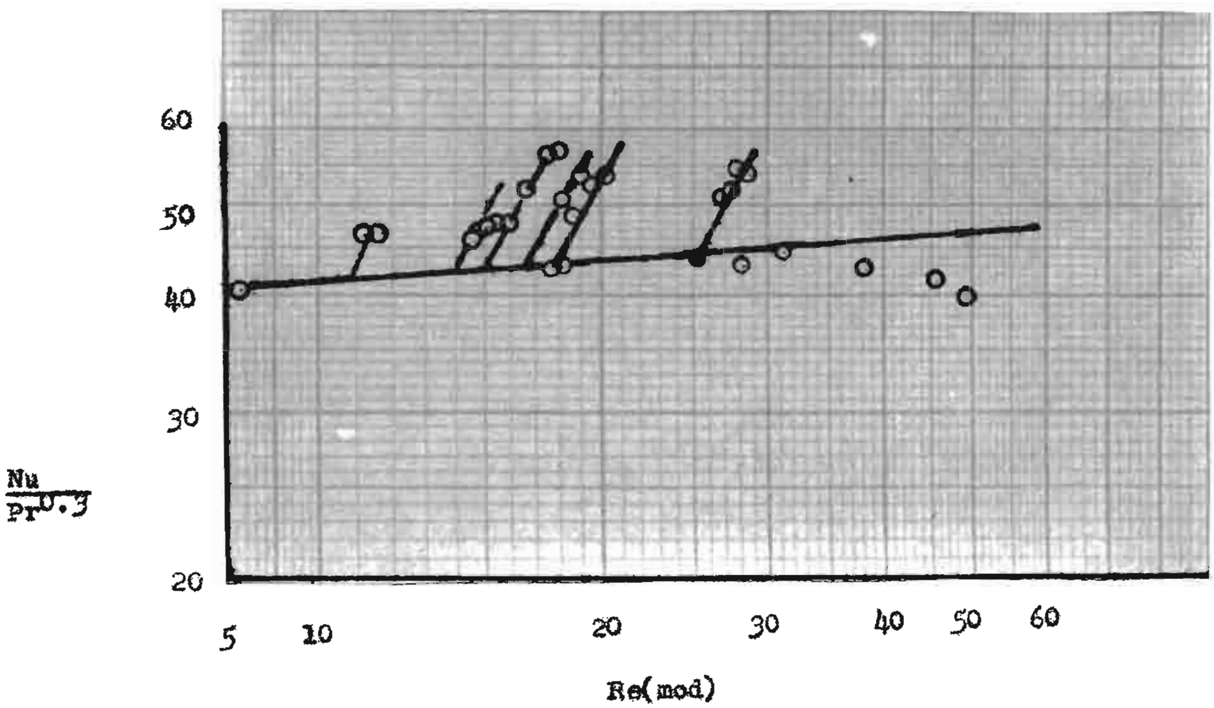

Figure 8

Further steam temperature effects

discharge end, there could be seen a definite change on the heating surface. The bits of air and vapor adhering to the heating surface could be seen to slowly break away causing a small amount of turbulence. The upper end of the tube nearer the water discharge end (about 1/3 of the copper tube inside of the condenser) seemed to be, as McAdams (3) states,

(3) McAdams, William H., Heat Transmission, 2nd Edition, 8th impression, pp. 296

in a state of "nuclear boiling". That is to say many bubbles of vapor formed on the heating surface were being discharged at a fast rate to the bulk of the water in passing. Although the discharge temperature of the water was several degrees below that of boiling, it appeared that the heat transfer in this upper section was actually a transfer to boiling liquid. The boiling in this case was on the immediate heating surface only.

As soon as the very small vapor bubbles,(approximately ½mm in diameter) separated from the tube, they again disappeared into the bulk of the liquid. These bubbles thus gave up their latent heat to increase the liquid temperature and condensed out of sight.

As the steam temperature was increased the size of the vapor bubbles increased. This increase in bubble size was leading to a true boiling liquid. McAdams (4) shows by use of a plot of heat flux vs. driving force

(4) McAdams, William H., Op. cit. pp. 296

that a maximum point in the heat transferred is reached for such a curve. In his explanation of the plot, he explains that after the maximum has been passed that a vapor layer is covering the heating surface in part, thus cutting down the total heat transferred by adding resistance to the path of flow. As the y-axis values used in these experiments are directly proportional to the total heat being transferred, we may assume that the vapor layer phenomena could be present in the cross flow apparatus of these experiments, and at very low values of Re(mod) (approaching zero) cause the experimental points to fall to a very low value. McAdams (5)

(5) McAdams, William H., Op. cit. pp.4

indicates that a value for the coefficient of heat transfer to vapor is 5 to $20 \frac{BTU}{(hr)(ft)^2(^\circ F)}$.

If $\frac{\text{Nusselt number}}{\text{Prandtl number}^{0.3}}$ were calculated from a coefficient of heat transfer of $12 \frac{BTU}{(hr)(ft)^2(^\circ F)}$ a value approaching zero would result

The above argument is offered in support of the extrapolated curve shown in Figure 10. This extrapolation, along with the solid line supported by experimental data, is offered as the type curve that may be used to correlate data from the horizontal tube heat exchanger for water in forced convection.

The dashed extrapolation is presented to show direction of linear trend only. Data of April 7, 1949, Figure 10, indicates the extrapolation may cut back to a downward direction too abruptly.

In summary the proposed curve begins, Re $=$ 0, at some value below $y = 10$, travels by continuous line (probably curved, concave down) to a maximum of $y = 50$ to 55 at $x = 1300$ to 1500, curves down to minimum of $y = 48$ to 52 where $x = 2250$, continues by straight line relation to $y = 60$ to 65 where $x = 6000$.

A report (6) of recent work done by Frank Krieth and Martin Sommerfield

(6) A Staff Report, Heat Transfer Coefficients Investigated, Chemical and Engineering News, July 11, 1949, pp. 1999

of the Jet Propulsion Laboratory, California Institute of Technology States:

"At high flux rates, it was possible to obtain heat transfer coefficients up to four times greater than with pure forced convection by reducing the pressure and thus cooling the tube in the surface bubble regime or the surface boiling regime." The fluids used by Krieth and Sommerfield were 96% aniline and n - butyl alcohol. At this writing, the type of equipment and details of experiment are not available.

In the quoted work the introduction of vacuum to the system had the effect of reducing resistance to the flow of the small bubbles from the heating surface to the bulk of the liquid which would be expected to multiply the surface boiling effect. As the work carried out with the horizontal tube heat exchanger was under some slight pressure above atmospheric pressure (one foot static head of water plus friction head of 8 feet of 2 inch diameter, plus fitting head of two elbows), it would be expected that the surface boiling effect would be reduced. This minimized effect is seen in the peak of the proposed curve at y = 50 to 55 where x = 1300 to 1500.

This curve will vary in y-axis values with the deposit of scale, with steam temperature used, with water outlet temperatures used, and is not offered for use when condensate is allowed to build up inside the tubes.

CORRELATION OF DATA BY USE OF RECTANGULAR COORDINATES:

A method of correlation of data from this type apparatus by use of rectangular coordinates has been found. The y-axis values have been modified in this method. See Figure 11. Many of the possibilities for the operation of this apparatus as to the steam temperatures and outlet water temperatures and condensate conditions have been considered in this cross-section of data taken.

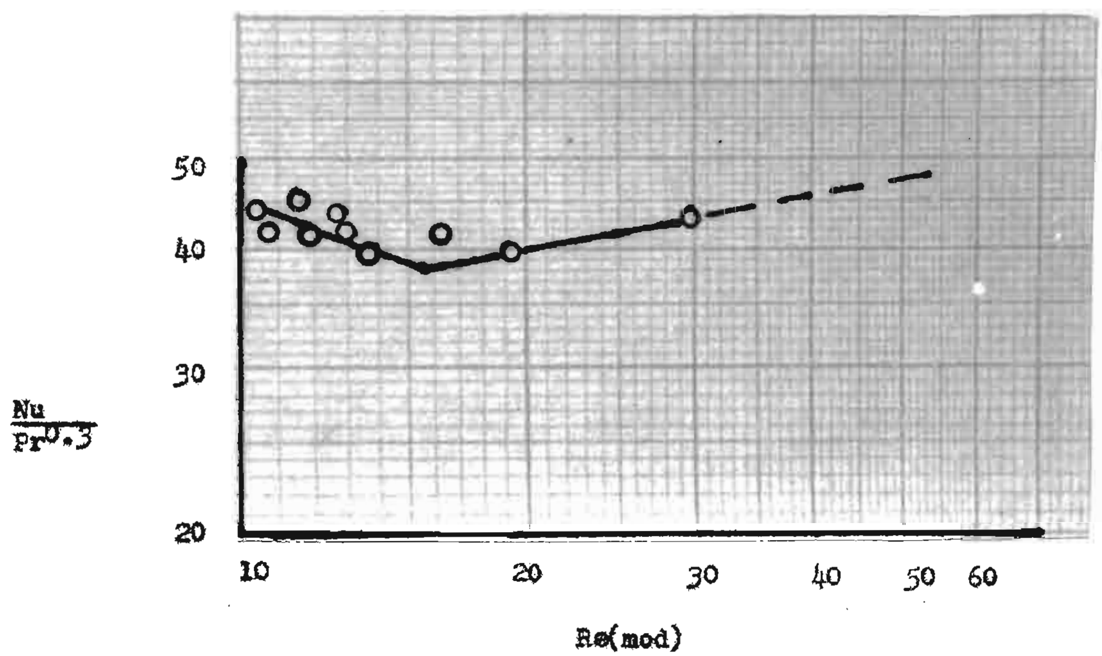

Figure 9

Effect of outlet water temperature

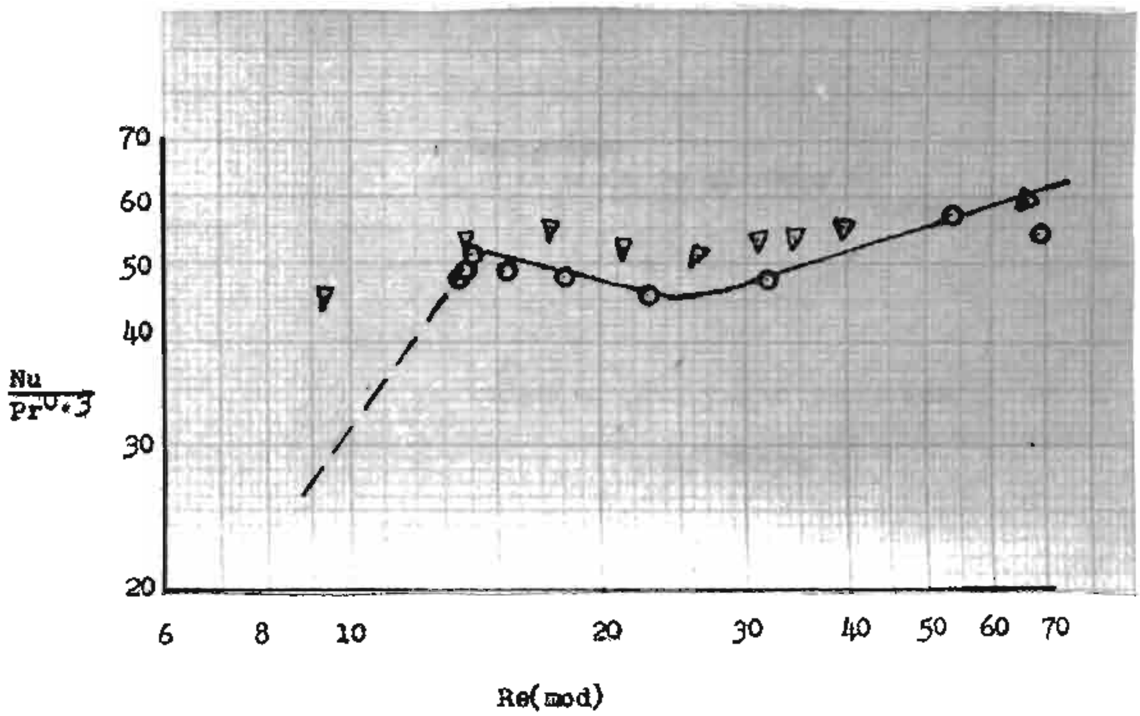

Figure 10

Proposed correlation curve for horizontal tube heat exchanger. Triangles illustrates duplication of data based on standard of cleanliness.

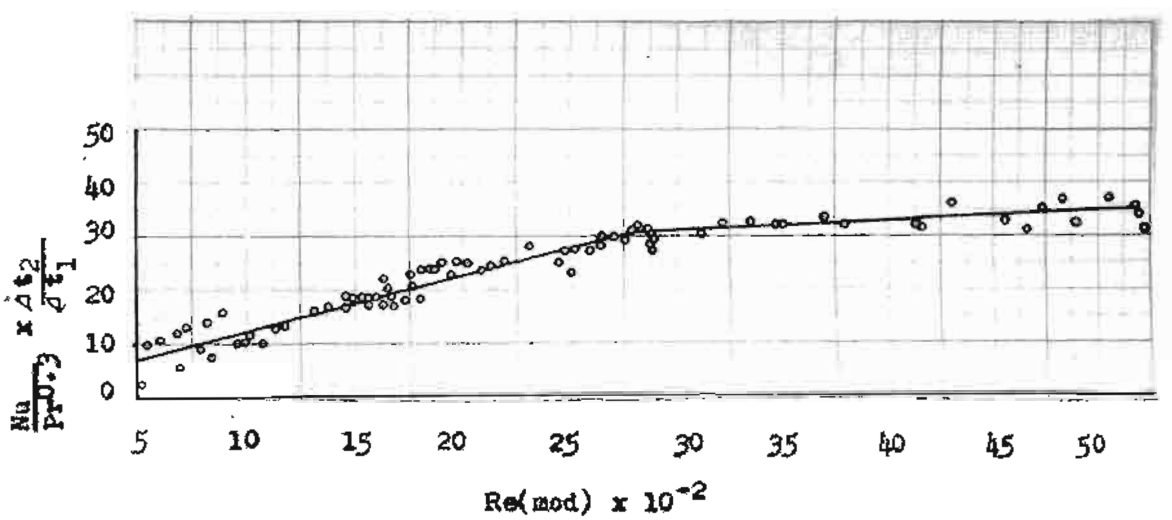

Figure 11

Correlation of data by Rectangular Coordinates.

Δt_2 = average steam temperature -- outlet water temp.

Δt_1 = average steam temperature -- inlet water temp.

CHECK OF RADIATION LOSSES:

Making use of the radiation equation given by McAdams (7) the loss from this apparatus by radiation has been checked and found to be negligible.

(7) McAdams, William H., Op. cit. pp. 52

$$q(net) = 0.173\ A_1\ E_1 \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4 \right]$$

A_1 = radiating area in square feet

E_1 = emissivity of the surface

T_1 = temperature of the emitting body

T_2 = temperature of the surroundings

Temperatures in degrees Rankine.

$$q(net) = (.173)(7.2)(0.8) \left[\left(\frac{672}{100}\right)^4 - \left(\frac{535}{100}\right)^4 \right]$$

$$= 1225\ \frac{BTU}{hr}$$

Even if the heat transfer rate were as low as 500,000 $\frac{BTU}{hr}$ would be only 0.25%. So, the assumption that the heat given up by the steam, is all taken up by the water, is justified.

HEAT GAINED FROM THE TUBE SHEET:

In order that a correction may be applied to the finished curve for the heat transferred through the header plates (tube sheets), some method of calculating an effective area of tubes as represented by the tube sheet must be used. A method of successive approximations has been used. It was found that the entire curve, see Figure 2, should be lowered by 4.25%.

Take point shown at Re(mod) = 6,250.

Average steam temperature............229.1°F

Average water temperature............ 83.7

Average driving force................145.4

Average water film temperature.......156.4

Thickness of tube sheet................0.0417 ft.

Thermal conductivity of tube sheet

$27 \dfrac{(\text{BTU})(\text{ft})}{(\text{hr})(\text{ft})^2(^\circ\text{F})}$

Coefficient of heat transfer for the tube sheet is

$$\frac{27}{0.0417} = 648 \frac{\text{BTU}}{(\text{hr})(\text{ft})^2(^\circ\text{F})}$$

It may be shown by the usual methods that the temperature drop through the tube sheet is less than one degree fahrenheit.

Free area, tube sheet $= 2 - (90)(0.521)^2(0.785)(2) = 1.232$ ft.^{2}

Area of tube surface $= 15.331$ ft.^{2}

Heat transfer rate $= 1{,}322{,}000 \dfrac{\text{BTU}}{(\text{hr})}$

Make first approximation:

$$h = \text{water film coefficient} = \frac{1{,}322{,}000}{(145.4)(15.331 + 1.232)}$$

$$= 550 \frac{\text{BTU}}{(\text{hr})(\text{ft})^2(^\circ\text{F})}$$

Overall coefficient for tube sheet and water film:

$$U = \frac{1}{\frac{1}{550} + \frac{1}{648}} = 296 \frac{\text{BTU}}{(\text{hr})(\text{ft})^2(^\circ\text{F})}$$

Effective area fraction $= \dfrac{296}{550} = 0.538$

Effective area $= 1.232\ (0.538) = 0.66$ ft.^{2}

A second approximation shows that the figure for the area is sufficiently accurate.

$$h\ (\text{water}) = \frac{1{,}322{,}000}{(145.4)(15.331 + 0.66)} = 570 \frac{\text{BTU}}{(\text{hr})(\text{ft})^2(^\circ\text{F})}$$

$$U = \frac{1}{\frac{1}{570} + \frac{1}{648}} = 306 \; \frac{(BTU)(ft)}{(hr)(ft)^2(^\circ F)}$$

Effective area fraction $= \frac{306}{570} = 0.537$

Effective area $= (1.232)(0.537) = 0.663$ ft.^{2}

So h for point is corrected from 593 to 570 $\frac{BTU}{(hr)(ft)^2(^\circ F)}$

Other points of the run were corrected similarly.

SUMMARY:

1. A procedure for the operation of the apparatus of these experiments has been established so data may be reproduced by any operator.

2. Procedure for maintaining standard of cleanliness for this apparatus has been established. Method and regularity of cleaning is proposed.

3. A curve for the correlation of data by use of log-log paper has been produced though not completely supported by experimental data.

4. A method for the correlation of data by the use of rectangular coordinates has been offered.

5. Explanations for irregularities found in the curves have been brought forward.

BIBLIOGRAPHY

1. McAdams, W. H., Heat Transmission, 2nd ed. N.Y. McGraw-Hill, 1942, 459p.

2. Perrone, S. H., Oil Gas Journal, March 28, 1935, pp. 71.

3. Rudert, K. E., The Design, Construction, and Testing of a Heat Exchanger for Determining Heat Transfer Coefficients for Liquids in Forced Convection Normal to Banks of Staggered Tubes, Thesis, Missouri School of Mines and Metallurgy, 1948, Rolla, Missouri.

4. A Staff Report, Heat Transfer Coefficients Investigated, Chemical and Engineering News, July 11, 1949, pp. 1999.

VITA

Terrell C. Claunch, born January 10, 1918, at Collins, Mississippi. Graduated two years junior college from HSJ Junior College, June 1937. Employed four years as shipping clerk by South Mississippi Lumber Company. Served five years in United States Army. Bachelor of Science, Chemical Engineering, degree received June, 1948. Employed by the United States Geological Survey while attending university at Rolla, Missouri.