# THE EFFECT OF WATER-SIDE PACKING
# ON
# THE TRANSFER OF HEAT IN A SINGLE-PASS EXCHANGER

by

Clay H. Dickinson

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF
BACHELOR OF SCIENCE IN CHEMICAL ENGINEERING

The University of Tennessee

December, 1948

CONTENTS


| | Page |
|---|---|
| Summary | 1 |
| Introduction | 3 |
| Materials and Equipment | 6 |
| Schematic Diagram of Equipment | 8 |
| Procedure | 9 |
| Results | 11 |
| Tabulated Results | 12 |
| Figure 2, The Effect of Water Velocity On Heat-Transfer Coefficients | 16 |
| Figure 3, The Effect of Water-Side Packing On Heat-Transfer Coefficients | 17 |
| Discussion of Results | 18 |
| Conclusions and Recommendations | 20 |
| Bibliography | 21 |
| Appendix A - Sample Calculations | 22 |
| Appendix B - Rotameter Calibration Data and Curve | 26 |
| Appendix C - Original Data | 29 |

# THE EFFECT OF WATER-SIDE PACKING

# ON

# THE TRANSFER OF HEAT IN A SINGLE-PASS EXCHANGER

## SUMMARY

This investigation was undertaken to determine some of the quantitative effects of water-side packing on the overall heat-transfer coefficients of a single-pass tubular heat exchanger employing steam as the condensing vapor.

Copper-coated steel spheres were used as the packing medium. Tests were made to determine the coefficients with and without the packing, at various water velocities and steam pressures.

The results of this investigation indicate a decided improvement in coefficients during tests with packing over those in which the water tube was not packed. This increase was of the order of approximately 90% above the coefficients without packing. These results also indicate that the use of packing is restricted due to the high pressure drop across the packing material. This investigation also indicates the water-film resistance to be the controlling factor in determining the overall heat-transfer coefficient.

It is recommended that this investigation be extended to include the determination of overall coefficients with higher steam pressures and larger volumes of circulating water per unit of time. It is also recommended that this

extension include the investigation of the effect of the type of metal which is used as the packing material.

# THE EFFECT OF WATER-SIDE PACKING
# ON
# THE TRANSFER OF HEAT IN A SINGLE-PASS EXCHANGER

## INTRODUCTION

In heat exchangers, a vapor, usually steam, is being condensed on the outer surface of a tube while a liquid flows in turbulent motion inside the tube. In order that condensation take place, the latent heat of vaporization of the vapor must be transferred to the liquid within the tube. Resistance to this transfer of heat is afforded by the film of condensing vapor outside the tube, the metal wall and the thin laminar film of liquid moving along the inner surface of the pipe.

Consider the liquid film. Any condition which tends to disrupt this relatively stagnant film should result in a decrease in the resistance to heat flow. Possibly the most feasible method of obtaining this condition is the agitation of the film by packing the water tube. This packing material should be in the form of pellets or spheres of uniform shape and sufficiently small to permit thorough agitation. In addition to this function, the packing material would cause an increase in liquid velocity provided sufficient pressure is available to overcome the large pressure drop resulting from the packing and at the same time give a reasonably large volume of liquid per unit of

time. This increase in velocity should add to the increase in the coefficient of heat transfer.

According to Humphrey[1] a distinct improvement in heat transfer was observed as a result of the use of a copper spiral on the water side of each tube of a single-pass tubular heat exchanger.

One of the most convenient methods for correlating over-all coefficients is that introduced by Wilson[2] which employs graphical analysis. This method is applicable to series of tests at constant steam pressure and variable water velocities. It is assumed that the sum of the resistances of the vapor film and the metal wall are substantially constant. Actually, according to the equation of Nusselt the resistance on the vapor side depends upon the temperature difference and the temperature of the condensate film and should vary somewhat as the water velocity is changed. Also, changes in the wall temperature brought about by changes in water velocity would cause minor variations in the thermal conductivity of the tube wall. However, except where very high water velocities are used, the water-side resistance is the controlling factor in the heat transfer.

---

(1) Humphrey, A.J., Heat Transfer Improved, Southern Power and Industry, 63, 3, (March 1945)

(2) Wilson, E.E., Trans. Amer. Soc. Mech. Engr., 37, 47, (1915)

The water-side resistance is an inverse function of the water velocity through the tubes and, neglecting the effects of changes in the water temperature due to changes in velocity, the water-side resistance could be taken as a function of the water velocity alone. Thus, a plot of the reciprocal overall heat-transfer coefficient, $1/U$, against the reciprocal water velocity function, $1/\theta(V)$, should give a straight line when using rectangular coordinates. For turbulent flow of the water in a given apparatus, $\theta(V)$ may be taken as $A(V)^{0.8}$ (1), where A is an empirical constant and may be considered as the apparent individual coefficient of heat transfer from the tube to the water, based on the outside surface for a water velocity of 1 foot per second.

---

(1) Humphrey, op. cit.

## MATERIALS AND EQUIPMENT

### Heat-transfer Unit

The heat exchanger proper was composed of a 3/4-in. standard brass pipe enclosed in a 2-in. standard iron pipe, approximately 3 feet long. It was constructed to give horizontal countercurrent flow with the water in the inner tube. A steam inlet and a condensate outlet were provided in the form of short nipples welded to the steam jacket. A 1/4-in. bleed value was placed at the end of the steam jacket vertically opposite the condensate outlet to assure the presence of steam at all points along the water tube. All inlet and outlet lines were equipped with unions near the transfer unit to facilitate the packing of the water tube.

The condensate line contained a bucket-type steam trap and a by-pass line for flushing the cold steam line of accumulated condensate and prevent injury to the steam trap.

### Packing Material

The water tube was packed with 1/8-in. copper-coated steel spheres, which were retained in place by a copper screen cup soldered to the inner periphery of the tube at the water outlet. A similar cup was inserted but not soldered in the tube at the water inlet to prevent shifting of

the packing material when changes in water velocity were made.

## Control Devices

The steam and water inlet lines each contained a precision Hancock flo-control valve for regulation of flow. By-pass lines with globe valves were constructed at each of the control valves to minimize effects of fluctuations in the main-line pressures.

A rotameter was included in the inlet water line for the determination and control of the water rate. The plummet in this instrument was chromium-plated steel to prevent corrosion by the water.

## Measuring Instruments

The steam pressure within the steam jacket was determined by a standard Bourdon gage for tests without packing and by a mercury manometer for tests with the water side packed. A steam calorimeter was utilized in the steam line for the determination of the quality or superheat, if any, of the entering steam.

The inlet and outlet water temperatures were determined by standard industrial thermometers.

The rate of steam consumption was obtained by weighing on platform scales the amount of condensate collected during a 20-minute interval as determined by a stop watch.

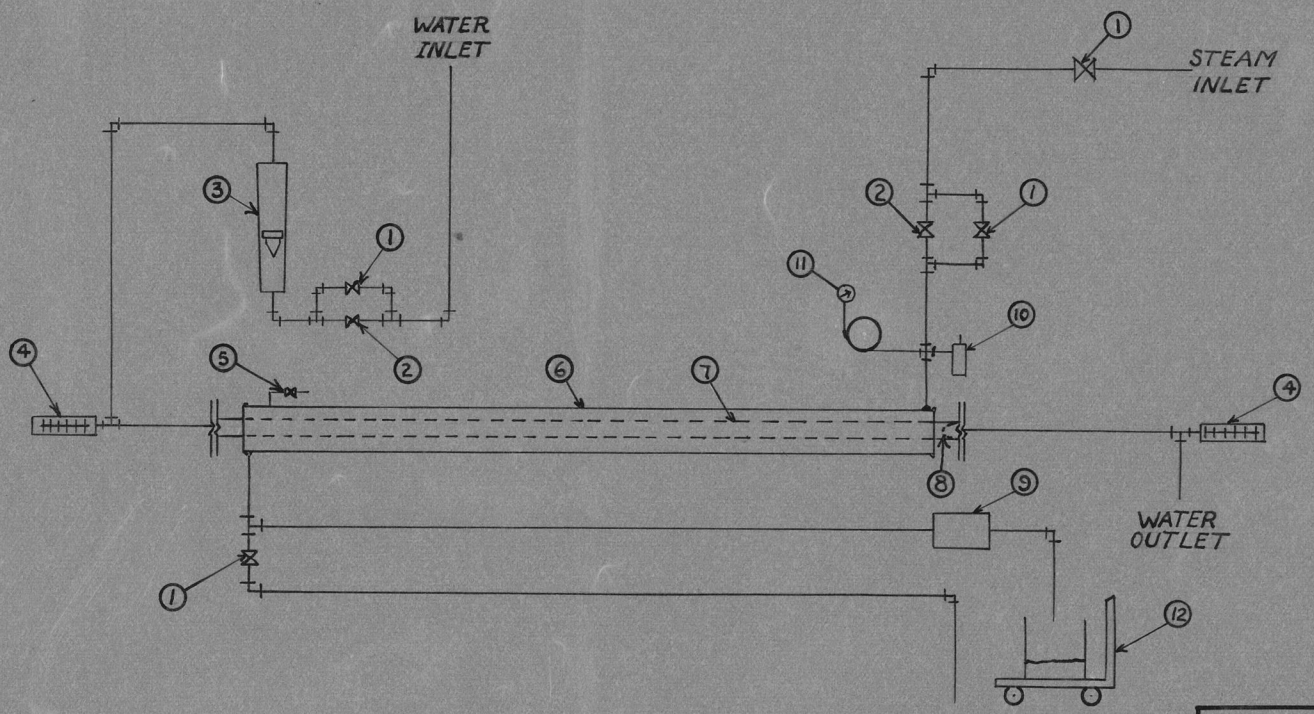



FIGURE 1

SCHEMATIC DIAGRAM OF HEAT TRANSFER UNIT

## PROCEDURE

The operational procedure of this investigation may best be described by assuming the unit to be at room temperature and all values closed. The steps followed in obtaining equilibrium are: (see Figure 1)

1) Open globe valve in the water line to give rotameter reading less than that desired.
2) Regulate control valve in water line to obtain desired rotameter reading.
3) Open globe valve in main steam line.
4) Open valve controlling flow to steam trap by-pass.
5) Open by-pass globe valve in steam line sufficiently to permit removal of accumulated condensate.
6) After condensate has been removed, close steam trap by-pass.
7) Regulate control valve in steam line to give desired steam pressure.
8) Regulate calorimeter and bleed valves.

After the system had attained equilibrium, as indicated by constant temperature readings, the weight of condensate collected during a 20-minute interval was determined and recorded, as well as the other pertinent data. (see Tables V and VI).

The metallic packing was then inserted in the water tube and the copper screen cup soldered in place. Due to

the high pressure drop across the unit with packing, it was necessary to remove the rotameter from the water line and weigh the amount of water flowing during a convenient interval of time for subsequent tests. It was also necessary to replace the Bourdan gage with a mercury manometer for measuring the low steam pressures for the tests made with the heat exchanger packed.

## RESULTS

From the observed data, energy balances were calculated for each test, the results of which are presented as Tables I and II.

The overall heat-transfer coefficients were calculated from the observed data of each test and are presented as Tables III and IV. Included in these tables are the water velocity functions and reciprocal overall heat-transfer coefficients. The effects of water velocity and steam pressure on the overall coefficient for the unit without packing are shown graphically in Figure 2. The values comprising this graph were taken from Table III.

The effect of packing on the overall heat-transfer coefficients is presented in Figure 3, in which the values for two groups of tests at the same steam pressure are plotted against the water velocity function. Due to the narrow limits of steam pressure during the tests with the water tube packed, variation in the coefficients with pressure was necessarily slight. For this reason only one constant pressure line is presented as being representative in the comparison of the coefficients of heat transfer with and without packing.

# TABLE I

## ENERGY BALANCES ON A SINGLE-PASS TUBULAR HEAT EXCHANGER WITHOUT PACKING

| Water Rate, lb/hr | Steam Pressure, psig | Heat Input, Btu/hr | Heat Output, Btu/hr | Heat Loss, Btu/hr | Heat Loss, % of Input |
|---|---|---|---|---|---|
| 2535 | 8.85 | 102,200 | 91,300 | 10,900 | 10.88 |
| 3675 | 8.85 | 120,200 | 106,500 | 13,700 | 11.40 |
| 4395 | 8.85 | 127,500 | 114,300 | 13,200 | 10.35 |
| 5280 | 8.85 | 140,300 | 121,700 | 18,600 | 13.35 |
| 1743 | 20 | 107,900 | 97,600 | 10,300 | 9.55 |
| 2955 | 20 | 129,800 | 124,000 | 5,800 | 4.47 |
| 4225 | 20 | 142,300 | 139,500 | 2,800 | 1.97 |
| 5450 | 20 | 155,800 | 150,000 | 5,800 | 3.72 |
| 2200 | 30 | 133,200 | 125,200 | 8,000 | 6.00 |
| 2955 | 30 | 144,200 | 141,800 | 2,400 | 1.67 |
| 4225 | 30 | 163,000 | 156,300 | 6,700 | 4.11 |
| 5450 | 30 | 177,000 | 171,800 | 5,200 | 2.94 |
| 2350 | 40 | 149,100 | 148,000 | 1,100 | 0.74 |
| 2955 | 40 | 156,000 | 153,300 | 2,700 | 1.73 |
| 4225 | 40 | 176,800 | 175,100 | 1,700 | 0.96 |
| 5450 | 40 | 190,500 | 185,200 | 5,300 | 2.78 |
| 2955 | 60 | 189,800 | 177,000 | 12,800 | 6.75 |
| 4225 | 60 | 205,000 | 198,500 | 6,500 | 3.17 |
| 2825 | 60 | 226,000 | 208,000 | 18,000 | 7.96 |
| 5450 | 60 | 236,000 | 218,000 | 18,000 | 7.63 |
| 3480 | 80 | 231,000 | 214,000 | 17,000 | 7.36 |
| 3900 | 80 | 242,000 | 222,000 | 20,000 | 8.25 |
| 4225 | 80 | 253,000 | 233,000 | 20,000 | 7.90 |
| 4825 | 80 | 238,000 | 236,000 | 2,000 | 0.84 |
| 5450 | 80 | 237,000 | 234,000 | 3,000 | 1.27 |

TABLE II

ENERGY BALANCES

ON A

SINGLE-PASS TUBULAR HEAT EXCHANGER

WITH PACKING

| Water Rate, lb/hr | Steam Pressure, in.Hg.gage | Heat Input, Btu/hr | Heat Output, Btu/hr | Heat Loss, Btu/hr | Heat, Loss, % of Input |
|---|---|---|---|---|---|
| 1485 | 4.1 | 153,700 | 138,100 | 15,600 | 10.16 |
| 1356 | 4.1 | 145,000 | 131,500 | 13,500 | 9.30 |
| 1020 | 4.1 | 121,900 | 109,100 | 12,800 | 10.50 |
| 795 | 4.1 | 104,300 | 90,600 | 13,700 | 13.10 |
| 1485 | 6.0 | 154,900 | 141,100 | 13,800 | 8.91 |
| 1356 | 6.0 | 149,000 | 134,200 | 14,800 | 9.93 |
| 1020 | 6.0 | 126,000 | 111,100 | 14,900 | 11.80 |
| 795 | 6.0 | 105,600 | 92,200 | 13,400 | 12.7 |
| 1485 | 8.0 | 157,300 | 144,000 | 13,300 | 8.45 |
| 1356 | 8.0 | 148,800 | 137,000 | 11,800 | 7.93 |
| 1020 | 8.0 | 127,900 | 113,300 | 14,600 | 11.40 |
| 795 | 8.0 | 112,800 | 94,600 | 18,200 | 16.10 |
| 1485 | 10.0 | 161,200 | 147,000 | 14,200 | 8.80 |
| 1356 | 10.0 | 150,000 | 139,000 | 11,000 | 7.34 |
| 1020 | 10.0 | 125,000 | 112,200 | 13,100 | 10.45 |
| 735 | 10.0 | 103,800 | 92,300 | 11,500 | 11.08 |
| 1356 | 18.0 | 157,500 | 149,100 | 8,400 | 5.33 |
| 1020 | 18.0 | 134,700 | 120,300 | 14,400 | 10.70 |
| 735 | 18.0 | 104,500 | 97,000 | 7,500 | 7.17 |

## TABLE III

## OVERALL HEAT-TRANSFER COEFFICIENTS FOR A SINGLE-PASS TUBULAR HEAT EXCHANGER WITHOUT PACKING

| Steam Pressure, psig | Water Velocity, V, ft/sec | $1/(V)^{0.8}$ | Overall Coefficient, U, Btu/hr-ft$^2$-$^\circ$F | 1/U |
|---|---|---|---|---|
| 8.85 | 3.065 | 0.408 | 767 | 0.001303 |
| 8.85 | 4.44 | 0.303 | 881 | 0.001135 |
| 8.85 | 5.32 | 0.262 | 920 | 0.001088 |
| 8.85 | 6.39 | 0.225 | 954 | 0.001049 |
| 20 | 2.11 | 0.550 | 789 | 0.001268 |
| 20 | 3.58 | 0.360 | 941 | 0.001062 |
| 20 | 5.11 | 0.271 | 1039 | 0.000962 |
| 20 | 6.59 | 0.221 | 1090 | 0.000917 |
| 30 | 2.66 | 0.456 | 940 | 0.001063 |
| 30 | 3.58 | 0.360 | 1010 | 0.000993 |
| 30 | 5.11 | 0.271 | 1081 | 0.000925 |
| 30 | 6.59 | 0.221 | 1162 | 0.000860 |
| 40 | 2.84 | 0.434 | 1037 | 0.000965 |
| 40 | 3.58 | 0.360 | 1054 | 0.000947 |
| 40 | 5.11 | 0.271 | 1150 | 0.000870 |
| 40 | 6.59 | 0.221 | 1193 | 0.000837 |
| 60 | 3.58 | 0.360 | 1123 | 0.000888 |
| 60 | 5.11 | 0.271 | 1211 | 0.000825 |
| 60 | 5.84 | 0.244 | 1250 | 0.000800 |
| 60 | 6.59 | 0.221 | 1300 | 0.000769 |
| 80 | 4.21 | 0.316 | 1260 | 0.000790 |
| 80 | 4.72 | 0.289 | 1275 | 0.000785 |
| 80 | 5.11 | 0.271 | 1332 | 0.000750 |
| 80 | 5.84 | 0.244 | 1335 | 0.000749 |
| 80 | 6.59 | 0.221 | 1320 | 0.000750 |

## TABLE IV

## OVERALL HEAT-TRANSFER COEFFICIENTS FOR A SINGLE-PASS TUBULAR HEAT EXCHANGER WITH PACKING

| Steam Pressure in.Hg.gage | Water Velocity,V, ft/sec | $1/(V)^{0.8}$ | Overall Coefficient,U, Btu/hr-ft$^2$-$^\circ$F | 1/U |
|---|---|---|---|---|
| 4.1 | 4.71 | 0.290 | 1772 | 0.000564 |
| 4.1 | 4.30 | 0.316 | 1735 | 0.000576 |
| 4.1 | 3.24 | 0.390 | 1552 | 0.000644 |
| 4.1 | 2.52 | 0.477 | 1370 | 0.000730 |
| 6.0 | 4.71 | 0.290 | 1771 | 0.000565 |
| 6.0 | 4.30 | 0.316 | 1733 | 0.000577 |
| 6.0 | 3.24 | 0.390 | 1510 | 0.000662 |
| 6.0 | 2.52 | 0.477 | 1358 | 0.000736 |
| 8.0 | 4.71 | 0.290 | 1801 | 0.000555 |
| 8.0 | 4.30 | 0.316 | 1759 | 0.000568 |
| 8.0 | 3.24 | 0.390 | 1530 | 0.000654 |
| 8.0 | 2.52 | 0.477 | 1394 | 0.000717 |
| 10.0 | 4.71 | 0.290 | 1803 | 0.000554 |
| 10.0 | 4.30 | 0.316 | 1752 | 0.000570 |
| 10.0 | 3.24 | 0.390 | 1521 | 0.000657 |
| 10.0 | 2.33 | 0.509 | 1392 | 0.000718 |
| 18.0 | 4.30 | 0.316 | 1740 | 0.000575 |
| 18.0 | 3.24 | 0.390 | 1515 | 0.000660 |
| 18.0 | 2.33 | 0.509 | 1317 | 0.000760 |

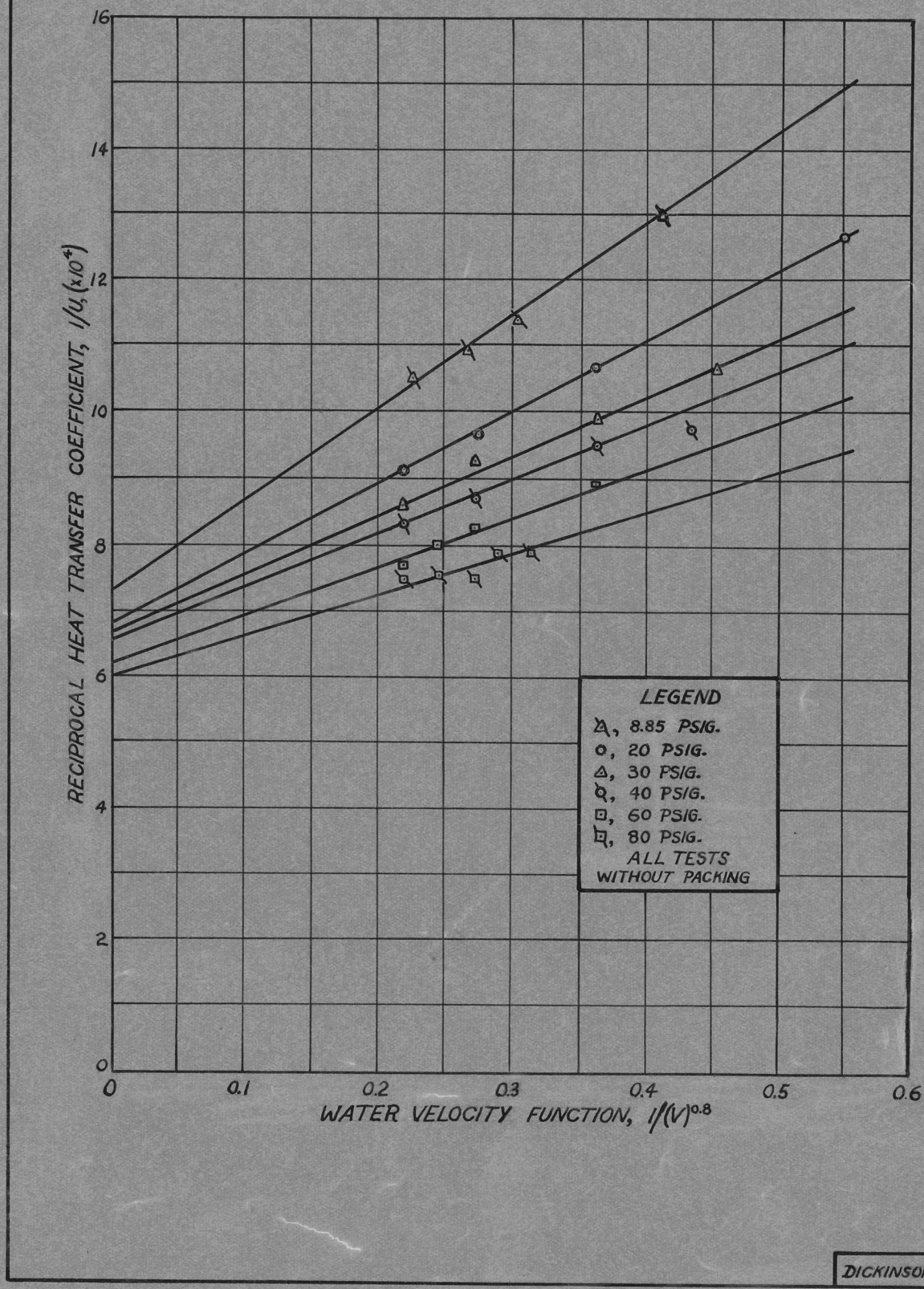
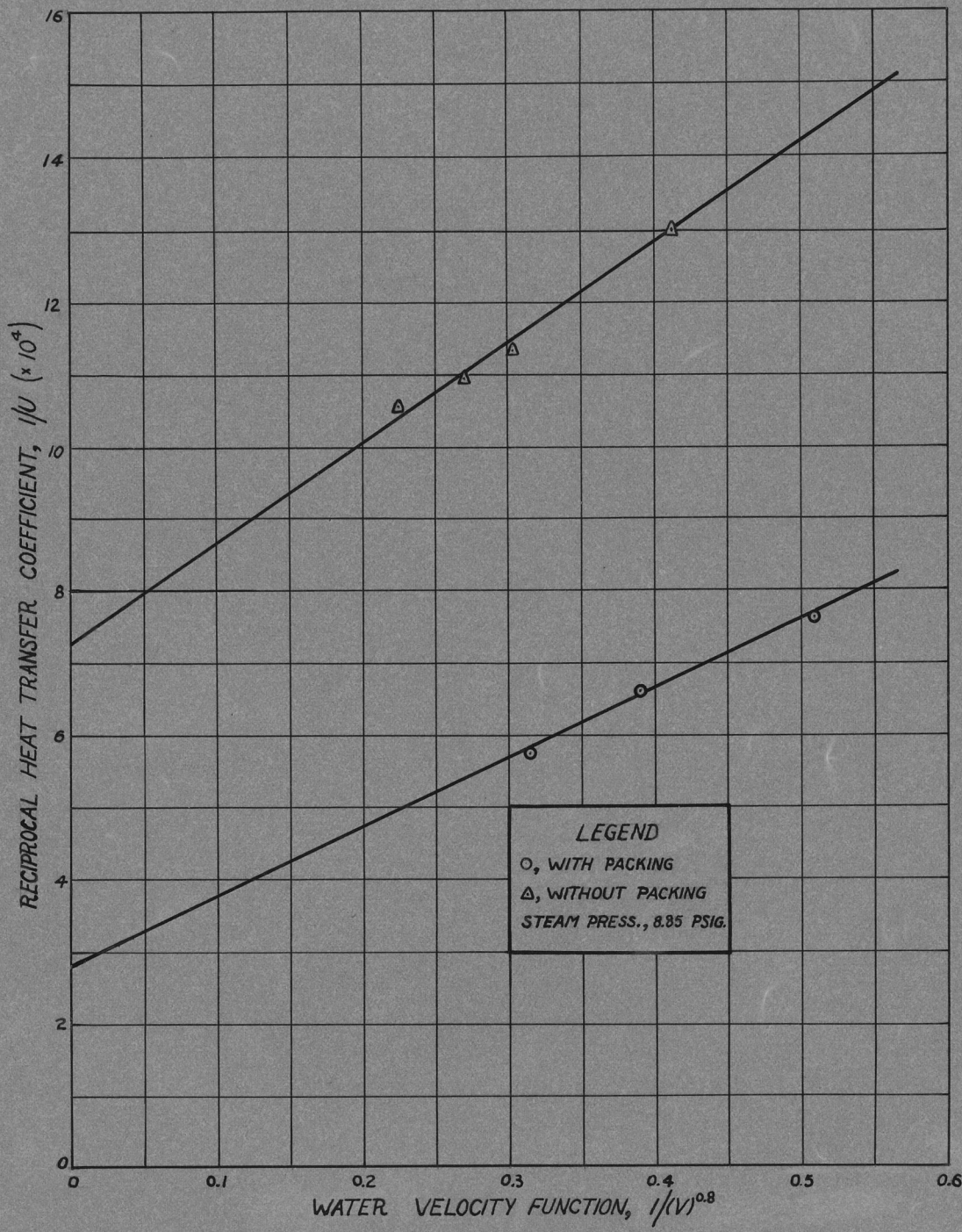



FIGURE 2

**THE EFFECT OF WATER VELOCITY ON HEAT TRANSFER COEFFICIENT**




DICKINSON 12/7/48

FIGURE 3

THE EFFECT OF WATER-SIDE PACKING
ON
HEAT TRANSFER COEFFICIENTS

## DISCUSSION OF RESULTS

A comparison of the reciprocal heat-transfer coefficients at various water velocities and steam pressures, as shown in Figure 2, indicates that an increase in the water velocity at constant steam pressure causes an increase in the coefficient of heat-transfer. Also, an increase in the steam pressure at constant water velocity results in an increase in the coefficient.

The improvement in the coefficient due to increased water velocity is probably due to the added turbulence in the water and the resulting decrease in the waterfilm resistance. Also, the increase in water velocity results in an increased temperature difference between the steam and water, causing a larger total amount of heat to be transferred, which more than compensates for the increased temperature difference.

An increase in steam pressure results in a higher coefficient since the temperature difference between the steam and water is increased. This higher difference in temperature results from the fact that the saturation temperature of the steam increases with an increase in pressure.

A comparison of the heat-transfer coefficients obtained without water-side packing with those in which the water

tube was packed (see Figure 3) indicates an appreciable increase for the packed condition. This increase is of the order of 90% above the coefficient without packing. It will be noted that these two series of tests were conducted with the steam pressure essentially constant and water velocities at comparable values. The most probable explanation of this increase is that the packing functioned to disrupt the stagnant water film and decrease its resistance to heat flow.

From the standpoint of the total volume of water heated, the packing of the water tube is rather restrictive. This results from the high pressure loss across the packing. It was found that for a main-line water pressure of approximately 60 psi. the actual water velocity in the unit with packing was about 5 feet per second while the volume of water circulated was approximately 3 gallons per minute. For a corresponding velocity without packing, the volume of water was approximately 8-1/2 gallons per minute. Thus, the use of water-side packing is restricted to units in which the amount of cooling liquid available for a given amount of cooling is comparatively small.

## CONCLUSIONS AND RECOMMENDATIONS

It is concluded as a result of this investigation that the use of packing material in the water tube of a single-pass tubular heat exchanger is of definite benefit in improving the overall heat-transfer coefficient. Its use, however, is restricted to those cases in which the use of a large amount of cooling liquid is not a necessity. It is also concluded that, in the unit investigated, the resistance afforded by the liquid film was the controlling resistance.

It is recommended as a result of this investigation that further tests be conducted employing an auxiliary pump to obtain higher water pressure, which would result in an increase in the volume of water circulated. This larger volume of cooling water would permit the investigation of the overall heat-transfer coefficients for higher steam pressures. It is also recommended that the effect of various packing materials be investigated with all other conditions such as velocity and steam pressure constant.

# BIBLIOGRAPHY

Humphrey, A. J., Heat Transfer Improved, Southern Power and Industry, 63, 3, (March, 1945)

Keenan and Keys, Thermodynamic Properties of Steam, 1st. Ed., John Wiley and Sons, Inc., New York, 1946.

Wilson, E. E., Trans. Amer. Soc. Mech. Engr., 37, 47, 1915.

# APPENDIX A

## SAMPLE CALCULATIONS

## SAMPLE CALCULATIONS

(Test 1, without packing)

Steam pressure, psig. = 20

Steam pressure, psia. = 34.4

Heat of vaporization[1] $h_{fg}$, Btu/lb = 940

Steam saturation temperature, $^{o}F$ = 258

Rotameter reading, = 5.0

Water rate, gal/min (from Figure 4) = 3.48

Water rate, lb/hr =

3.48 gal/min x 8.34 lb/gal x 60 min/hr = 1743

I. Energy Balance:

Correction for flashing

Heat of vaporization at $210^{o}F$, Btu/lb = 972

Temperature difference, $^{o}F$ = 258 - 210 = 48

Actual weight of condensate, lb/20 min. = 36.45

Weight lost by flashing, lb = x

Total condensate, lb = 36.45 + x

$$x = \frac{36.45 + x}{972} = 1.80$$

Corrected steam consumption, lb/20 min =

36.45 + 1.80 = 38.25

Steam consumption, lb/hr = $38.25 \times \frac{60}{20} = 114.75$

Heat input, Btu/hr =

114.75 lb/hr x 940 Btu/lb = 107,900

---

(1) Keenan and Keyes, Thermodynamic Properties of Steam, 1st. Ed., John Wiley and Sons, Inc., New York, 1946.

Water inlet temperature, $^{o}F$ = 63

Water outlet temperature, $^{o}F$ = 119

Heat capacity of water, Btu/lb - $^{o}F$ = 1.0

Heat output, Btu/hr =

1743 lb/hr x (119 - 63)$^{o}F$ x 1.0 Btu/lb - $^{o}F$ = 97,600

Heat loss, Btu/hr = 107,900 - 97,600 = 10,300

Heat loss, % of input = $\frac{10,300}{107,900}$ x 100 = 9.55

II. Overall heat transfer coefficient:

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{2.3 \log \frac{\Delta t_1}{\Delta t_2}} = \frac{(258-63) - (258-119)}{2.3 \log \frac{(258-63)}{(258-119)}} = 166^{o}F$$

For 3/4-in. standard brass pipe:

outside radius, ft. = 0.0437

inside radius, ft. = 0.0342

Log mean radius, ft. = $\frac{r_1 - r_2}{2.3 \log \frac{r_1}{r_2}}$ =

$$\frac{0.0437 - 0.0342}{2.3 \log \frac{0.0431}{0.0342}} = 0.0388$$

Heat transfer length, ft. = 3.06

Heat transfer area, sq.ft. = 0.0388 x 2 x $\pi$ x 3.06 =

0.746

$$U = \frac{Q}{A(\Delta t_m)}$$

Overall heat transfer coefficient, U, Btu/hr - $ft^2$ -$^{o}F$ =

$$\frac{97,600}{0.746(166)} = 789$$

1/U = 1/789 = 0.001268

III. Water velocity function, $1/(V)^{0.8}$:

Internal volume water tube, $ft^3/ft$ =

$(0.0342)^2 \pi \ (1) = 0.00368$

Water velocity, V, ft/sec. =

$$\frac{1743 \text{ lb/hr.}}{3600 \text{ sec/hr} \times 0.0342 \text{ ft}^3/\text{ft} \times 62.4 \text{ lb/ft}^3} = 2.11$$

$1/(V)^{0.8} = 1/(2.11)^{0.8} = 0.550$

IV. Equivalent diameter, with packing:

Total volume of water pipe, $ft^3$ =

$0.00368 \text{ ft}^3/\text{ft} \times 3.06 \text{ ft} = 0.01128$

Volume of packing, $cm^3 = 230$

Volume of packing, $ft^3 = 230 \text{ cm}^3 \times (3.53 \times 10^{-5}) \text{ft}^3/\text{cm}^3$

$= 0.00812$

Length of tube containing packing, ft = 3.56

Volume of packing, $ft^3/ft$ pipe = $\frac{0.00812 \text{ ft}^3}{3.56 \text{ ft}}$ =

0.00228

Volume occupied by packing, $ft^3$ =

$0.00228 \text{ ft}^3/\text{ft} \times 3.06 \text{ ft} = 0.00698$

Volume annular space, $ft^3$ =

$0.001128 - 0.00698 = 0.00430$

Volume annular space, $ft^3/ft$ =

$$\frac{0.00430 \text{ ft}^3}{3.06 \text{ ft.}} = 0.001414$$

# APPENDIX B

## ROTAMETER CALIBRATION DATA AND CURVE

## TABLE VII

## DATA FOR ROTAMETER CALIBRATION

| Rotameter Reading | Time, Min. | Total Water Flow lbs. | Water Rate gal/min |
|---|---|---|---|
| 2.0 | 4.0 | 14.50 | 0.435 |
| 3.0 | 4.0 | 44.55 | 1.333 |
| 4.0 | 3.0 | 59.00 | 2.36 |
| 5.0 | 2.0 | 58.00 | 3.48 |
| 7.0 | 1.25 | 61.50 | 5.90 |
| 9.0 | 1.0 | 70.50 | 8.45 |
| 11.0 | 0.75 | 68.00 | 10.90 |

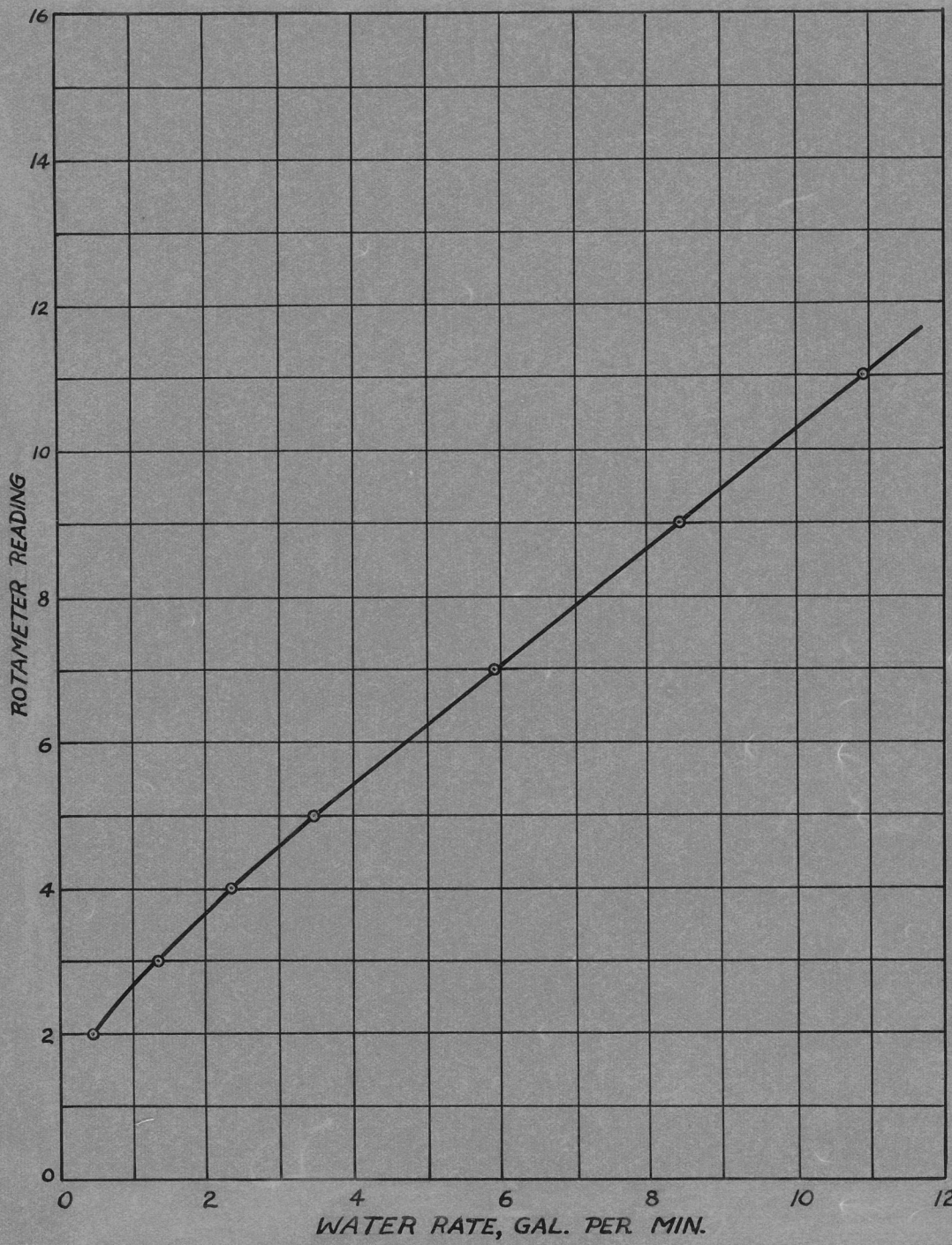



FIGURE 4

ROTAMETER CALIBRATION CURVE

# APPENDIX C

## ORIGINAL DATA

TABLE V

ORIGINAL DATA

FOR

HEAT EXCHANGER WITHOUT PACKING

| Steam Pressure, psig. | Calorimeter Temp., °F | Water Temperature °F | | Rotameter Reading | Condensate, wt. lb. (corrected) | Time min. |
|---|---|---|---|---|---|---|
| | | Inlet | Outlet | | | |
| 8.85 | 210 | 56 | 92 | 6.3 | 45.00 | 20 |
| 8.85 | 210 | 55 | 84 | 8.1 | 42.00 | 20 |
| 8.85 | 210 | 55 | 81 | 9.3 | 44.50 | 20 |
| 8.85 | 210 | 55 | 78 | 10.7 | 49.00 | 20 |
| 20 | 210 | 63 | 119 | 5.0 | 38.25 | 20 |
| 20 | 210 | 60 | 102 | 7.0 | 46.00 | 20 |
| 20 | 210 | 60 | 93 | 9.0 | 50.50 | 20 |
| 20 | 210 | 60 | 87.5 | 11.0 | 55.25 | 20 |
| 30 | 210 | 60 | 91.5 | 11.0 | 63.50 | 20 |
| 30 | 210 | 61 | 98 | 9.0 | 58.50 | 20 |
| 30 | 210 | 60 | 108 | 7.0 | 51.75 | 20 |
| 30 | 210 | 62 | 123 | 5.5 | 47.75 | 20 |
| 40 | 210 | 62 | 125 | 6.0 | 54.00 | 20 |
| 40 | 210 | 62 | 114 | 7.0 | 56.50 | 20 |
| 40 | 210 | 61 | 102.5 | 9.0 | 64.00 | 20 |
| 40 | 210 | 61 | 95 | 11.0 | 69.00 | 20 |
| 60 | 210 | 62 | 126 | 7.0 | 70.00 | 20 |
| 60 | 210 | 61 | 108 | 9.0 | 75.50 | 20 |
| 60 | 210 | 62 | 105 | 10.0 | 83.25 | 20 |
| 60 | 210 | 61 | 101 | 11.0 | 87.00 | 20 |
| 80 | 210 | 62 | 123.5 | 7.8 | 86.50 | 20 |
| 80 | 210 | 62 | 119 | 8.5 | 90.50 | 20 |
| 80 | 210 | 62 | 117 | 9.0 | 94.50 | 20 |
| 80 | 210 | 62 | 110 | 10.0 | 89.00 | 20 |
| 80 | 210 | 62 | 105 | 11.0 | 88.50 | 20 |

TABLE VI

ORIGINAL DATA

FOR

HEAT EXCHANGER WITH PACKING

| Steam Pressure in.Hg.gage* | Calorimeter Temp.* °F | Water Temperature °F | | Water Rate, lb/min. | Condensate, wt. lb. (corrected) | Time min. |
|---|---|---|---|---|---|---|
| | | Inlet | Outlet | | | |
| 4.1 | 210 | 60 | 153 | 24.75 | 53.00 | 20 |
| 4.1 | 210 | 60 | 157 | 22.60 | 50.00 | 20 |
| 4.1 | 210 | 60 | 167 | 17.00 | 42.00 | 20 |
| 4.1 | 210 | 60 | 174 | 13.25 | 36.00 | 20 |
| 6.0 | 210 | 60 | 155 | 24.75 | 53.50 | 20 |
| 6.0 | 210 | 60 | 159 | 22.60 | 51.50 | 20 |
| 6.0 | 210 | 58 | 167 | 17.00 | 43.50 | 20 |
| 6.0 | 210 | 60 | 176 | 13.25 | 36.50 | 20 |
| 8.0 | 210 | 60 | 157 | 24.75 | 54.50 | 20 |
| 8.0 | 210 | 60 | 161 | 22.60 | 51.50 | 20 |
| 8.0 | 210 | 58 | 169 | 17.00 | 44.25 | 20 |
| 8.0 | 210 | 60 | 179 | 13.25 | 39.00 | 20 |
| 10.0 | 210 | 60 | 159 | 24.75 | 56.00 | 20 |
| 10.0 | 210 | 60 | 162.5 | 22.60 | 52.00 | 20 |
| 10.0 | 210 | 62 | 172 | 17.00 | 43.50 | 20 |
| 10.0 | 210 | 57 | 182.5 | 12.25 | 36.00 | 20 |
| 18.0 | 210 | 60 | 170 | 22.60 | 55.00 | 20 |
| 18.0 | 210 | 60 | 178 | 17.00 | 47.00 | 20 |
| 18.0 | 210 | 57 | 189 | 12.25 | 36.50 | 20 |

* Referred to atmospheric pressure of 743 mm. Hg.

## APPROVAL

The foregoing thesis is hereby approved as a creditable study of an engineering subject, carried out and presented in a manner sufficiently satisfactory to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is to be understood that by this approval the undersigned does not necessarily endorse or approve any statement made, opinions expressed, or conclusions drawn therein, but approves the thesis only for the purpose for which it is submitted.

Name H. J. Garber

Title Prof. of Chem. Engg.

Date Dec. 16, 1948.