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AN INVESTIGATION OF THE FOULING OF 19-FIN-PER-INCH
ADMIRALTY TUBES IN THREE HEAT TRANSFER UNITS
LOCATED AT THE AURORA GASOLINE COMPANY REFINERY

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OBJECTIVE

The purpose of this investigation was to study the fouling rates of 3/4-inch, 19-fin-per-inch admiralty tubes in three heat-exchanger units located at the Detroit, Michigan, Refinery of the Aurora Gasoline Company. The units involved are identified as a debutanizer overhead condenser, a debutanizer bottoms cooler, and a high-pressure gas cooler.

ABSTRACT

Field test data collected over a period of nine months on a debutanizer overhead condenser, a debutanizer bottoms cooler, and a high-pressure gas cooler are presented. The heat transfer performance variations and fouling rates of the three units were computed from the test data and the results tabulated. The fouling results obtained on the debutanizer overhead condenser are inconclusive because of the effect of partial flooding and fluctuation in the amount of flooding on the overall heat transfer coefficients. The fouling rate of the debutanizer bottoms cooler was determined. The results on this unit indicate that the water-side fouling rate greatly affects the overall fouling of the tube. The fouling results obtained on the high-pressure gas cooler are inconclusive because this unit was operated as a partial condenser and it was not possible to determine the shell-side heat transfer coefficients from the test data. The composition of the shell-side vapor-liquid feed varied during the tests.

I. INTRODUCTION

Wolverine Trufin tubing is rapidly becoming accepted for certain oil refinery heat transfer applications. This report presents the results of an extensive field investigation of the fouling characteristics of Trufin tubes in a partial condenser, a total condenser, and a bottoms cooler in a local oil refinery.

The data for this report were obtained from three shell and tube heat-exchanger units located at the Aurora Gasoline Company refinery, Detroit, Michigan. The field tests were made on these units over a period of nine months, starting August 21, 1955. The tests were started the fourth day after the units were first placed on stream. Field test data were obtained on twenty-seven occasions. The first five tests were conducted at about three-day intervals. This was followed by seven tests taken at approximately weekly intervals. After the third month the tests were conducted at intervals depending upon the plant operating schedule and the observed fouling trend of the units.

II. DESCRIPTION OF UNITS

The three heat-exchanger units on which test data were obtained are located in the gas concentration unit of the Aurora refinery. All the exchangers were initially tubed with 3/4-inch (diameter over the fins) admiralty Trufin tubes having 19 fins per inch. The specification sheets for the units are reproduced in Appendix A. The Griscom-Russell design blueprints are shown in Appendix B. Line diagrams showing the flow arrangements in the units are presented in Figs. 1, 2, and 3.

A. HIGH-PRESSURE GAS COOLER

The high-pressure gas cooler consisted of two heat exchangers operating in series as indicated in Fig. 1. This unit operated as a partial condenser. Midway through the test period the severe corrosive conditions existing in the high-pressure gas cooler necessitated retubing the upper exchanger of this unit. To gain better corrosion resistance, the exchanger was retubed with 3/4-inch-OD steel Trufin tubes having 19 fins per inch (Wolverine Catalog No. 195065-63).

Thermowells were located in the following positions: (a) in the hydrocarbon inlet pipe line just above the inlet vapor nozzle of the upper condenser, (b) in the hydrocarbon outlet pipe line from the bottom condenser

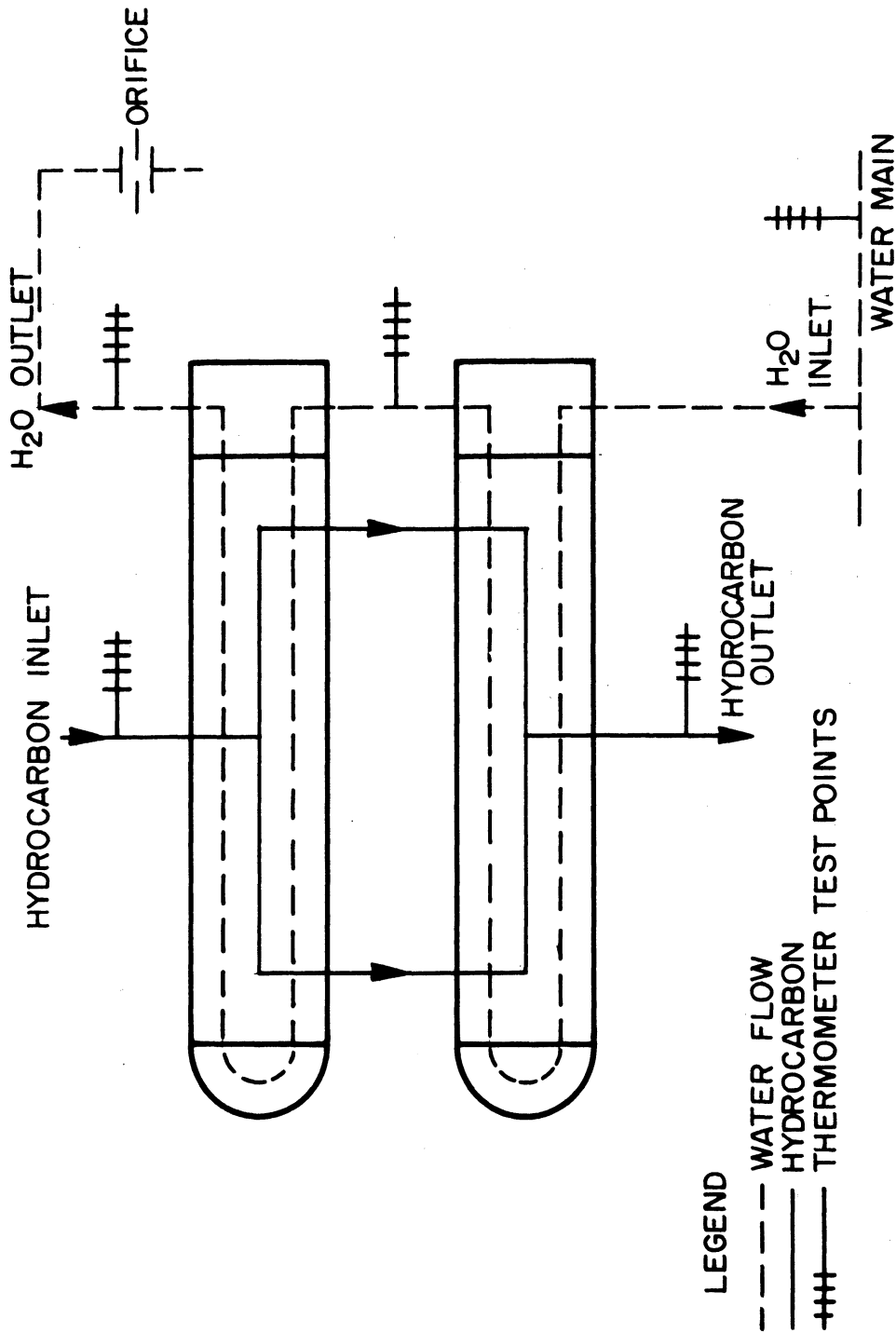


Fig. 1. Flow arrangement of the high-pressure gas cooler.

about ten feet downstream and (c) in the outlet water line from the top condenser. The piping arrangement on the water side was such that water could be bled from the line connecting the water headers of the exchangers (water flowed in series) and from the water main supplying the cooling water to all the exchangers which were tested. A sharp-edged orifice was located in the outlet water line for determining the water flow rate through the exchangers.

B. DEBUTANIZER OVERHEAD CONDENSER

The debutanizer overhead condenser also consisted of two exchangers, but with the exchangers arranged in parallel as indicated in Fig. 2. This unit operated as a total condenser. Thermowells were located (a) in the inlet hydrocarbon line approximately ten feet before the Tee split the stream between the two heat exchangers, (b) in the hydrocarbon line leaving the condensate receiver, and (c) in the water line downstream from the Tee at which the outlet water streams from the two heat exchangers were joined. The water leaving each heat exchanger could be bled from the system before the Tee in order to obtain the individual exit water temperatures. A sharp-edged orifice was located in the outlet water line for determining the total quantity of water passing through both heat exchangers.

C. DEBUTANIZER BOTTOMS COOLER

The debutanizer bottoms cooler consisted of a single liquid-to-liquid exchanger, having two passes on both the shell and tube sides as indicated in Fig. 3. Its function was to cool the bottoms product from the debutanizer column, with no change of phase being involved. Thermowells were located in (a) the hydrocarbon inlet pipeline to the cooler, (b) the hydrocarbon outlet pipeline leaving the cooler, and (c) the outlet water pipeline leaving the cooler. A sharp-edged orifice was located in the outlet water line for measuring the total quantity of cooling water flowing through the heat exchanger.

III. FIELD TEST PROCEDURE

The industrial thermometers provided in the thermowells were not sufficiently sensitive for the test measurements. It was therefore necessary to contrive some means by which laboratory-grade, mercury-in-glass thermometers could be used for temperature measurements. The technique which was used consisted of filling the thermowells with grease, followed by insertion of the thermometers into the wells. The comparatively low thermal conductivity of the grease tended to insulate the thermometer bulb from the effect of the outside air temperature. However, the grease also tended to damp out variations in the fluid temperature during a test run, due to the time lag required

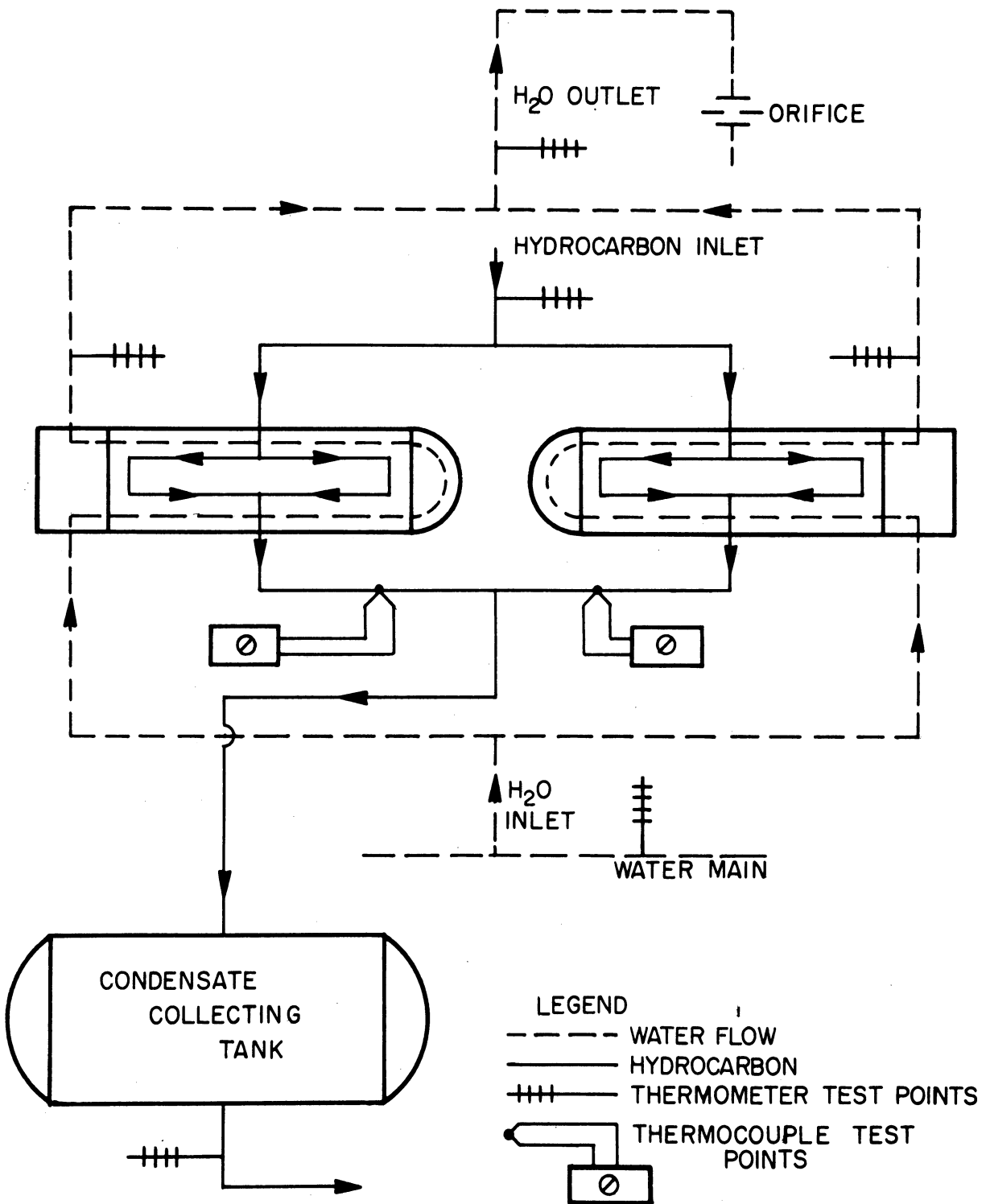


Fig. 2. Flow arrangement of the debutanizer overhead condenser.

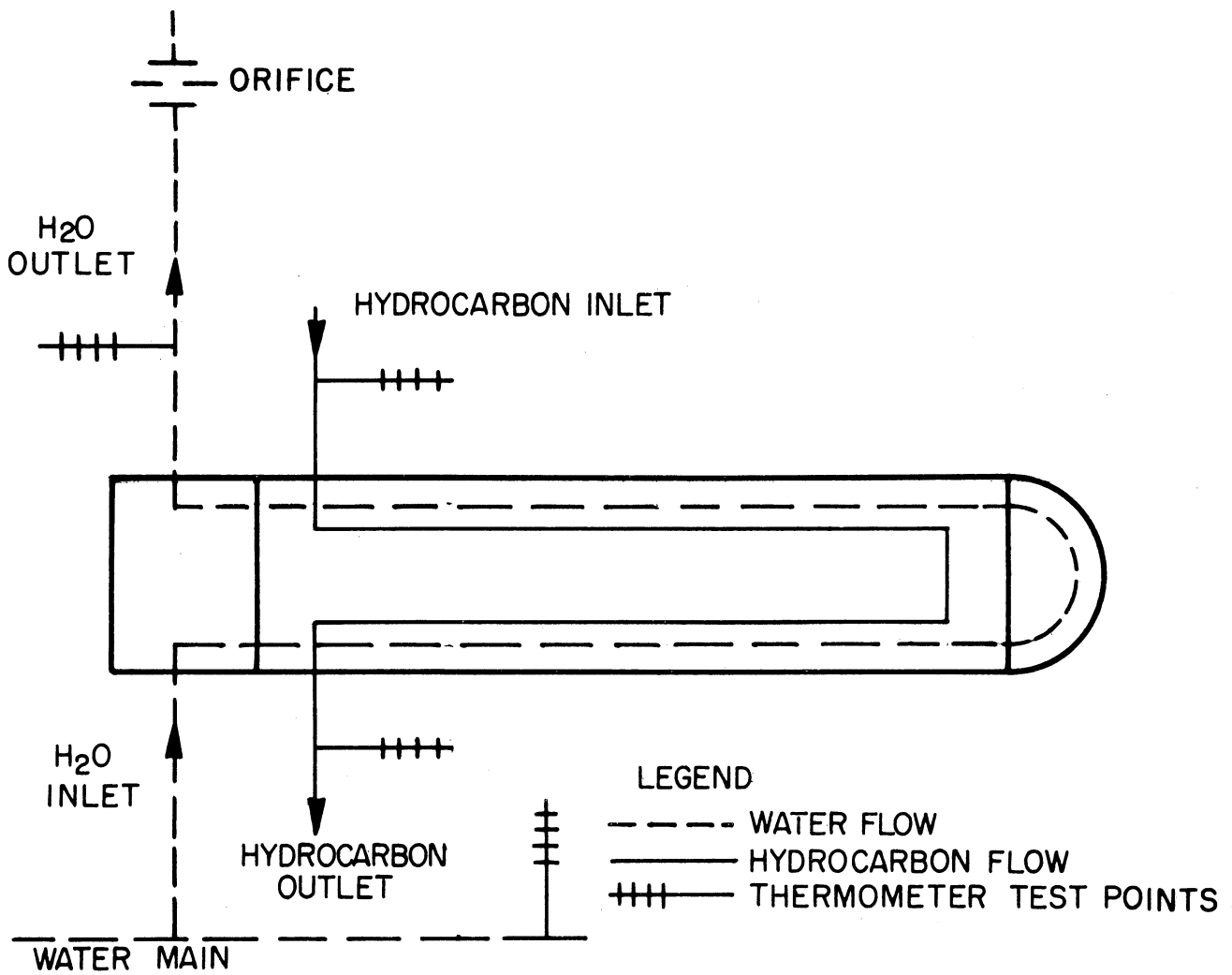


Fig. 3. Flow arrangement of the debutanizer bottoms cooler.

for the grease to reach a new stable temperature.

At some of the test points, such as the water intermediate between the two exchangers on the high-pressure gas cooler, water was bled from the line, with a mercury-in-glass thermometer being immersed in the water stream for a temperature measurement.

A problem was encountered on the debutanizer overhead condenser in that the thermowell for measurement of the outlet hydrocarbon temperature was located a considerable distance downstream from the unit, with a large condensate collecting tank being situated between the thermometer and the test unit. The condensate residence time in this tank had some effect on the fluid temperature, due to heat losses. Therefore, there existed some question as to the actual temperature of the hydrocarbon stream leaving the test unit. In an attempt to overcome this difficulty, thermocouples were embedded in the wall of the hydrocarbon outlet pipes from each exchanger, and a layer of insulation was placed over the thermocouples. However, due to various sources of error, the temperature measurements taken with these thermocouples did not appear to be reliable and were not used in the analysis of the test data.

The inlet water temperature for all units was measured by bleeding from the inlet water main feeding all three test units. This test point was located slightly upstream from all the units.

The water flow rate through the units was determined by measurement of the pressure drop across the orifices in the outlet water lines. This measurement was made with liquid-filled manometers. For most of the runs, mercury was used as the manometer fluid. However, to gain accuracy at the low flow rates such as were used in the Wilson Plot runs on the debutanizer bottoms cooler, dyed carbon tetrachloride was used as the manometer fluid.

During the first phase of the investigation all three units were tested. The temperatures and flow rates of the units were not adjusted, but were recorded as they occurred in normal plant operation.

It soon became apparent that the debutanizer bottoms cooler gave the most accurate indication of a fouling trend. It was therefore decided that the majority of the test data should be taken on this unit. Permission was received to vary the water flow rate through the exchanger for short periods of time so that Wilson Plot type data could be taken. This allowed a more accurate analysis of the fouling trend on the unit. No Wilson Plot test runs were made on the other two test units.

After 2-1/2 months of operation the test data indicated that the bottoms-cooler fouling resistance had exceeded the design fouling resistance. In order to improve the heat transfer performance of this and the other units, all the units were cleaned during the next plant shutdown early in November, 1955. Testing of this unit was resumed on November 18, 1955. After receiving

permission to vary the water flow rate through the bottoms cooler for short periods of time, the first Wilson Plot run was made on December 23, 1955.

At the time the units were cleaned an evaluation of the test procedure was undertaken. It was decided that since thermowells could not be installed in the hydrocarbon outlet lines leaving each of the debutanizer overhead condensers, sufficiently accurate test data for determining the fouling trend of this unit could not be obtained. Therefore, no further test data were taken on this unit after October 21, 1955.

Also at the time of cleaning of the units over fifty tubes in the top heat exchanger of the high-pressure cooler were plugged due to leakage caused by corrosion. No further test data were taken until after this heat exchanger was retubed with steel tubes. The next test data were taken on December 10, 1955.

Excessive fouling of the debutanizer bottoms cooler was apparent from the test data taken on February 1, 7, and 18, 1956. This fouling condition was discussed with the plant engineering office personnel on March 1, 1956. The project group was informed at this meeting that the high fouling resistance calculated from the field test data was due to oil contamination of the recirculated cooling tower water. On March 3, 1956, the tube side of this unit was cleaned by plant personnel. Test data were obtained just before and immediately after the cleaning. The testing was then continued at regular intervals until May 26, 1956, at which time all field tests were terminated.

Typical test data for the three units are presented in Tables I, II, and III.

TABLE I

TYPICAL TEST DATA ON THE HIGH-PRESSURE GAS COOLER

Date: May 26, 1956

Run No.: 21

| Time (p.m.) | Hydrocarbon (°F) | | Water (°F) | | | Manometer (Inches of CCl ₄) | |
|----------------|---------------------|--------|------------|-------------------|--------|--|--------|
| | In | Out | In | Inter- Mediate | Out | Left | Right |
| 12:23 | 139.5 | 113.2 | 70.6 | 90.6 | 116.7 | +13.4 | -12.6 |
| 12:25 | 139.2 | 113.5 | 70.6 | 91.6 | 116.6 | +13.4 | -12.6 |
| 12:27 | 139.0 | 113.5 | 70.3 | 91.0 | 116.7 | +13.3 | -12.6 |
| 12:29 | 139.0 | 113.5 | 70.6 | 90.8 | 116.7 | +13.2 | -12.6 |
| 12:31 | 136.5 | 113.2 | 70.6 | 90.4 | 116.6 | +13.2 | -12.6 |
| 12:33 | 135.5 | 113.4 | 70.6 | 90.5 | 116.6 | +13.3 | -12.7 |
| 12:35 | 135.3 | 113.5 | 70.7 | 90.8 | 116.6 | +13.3 | -12.7 |
| 12:37 | 135.0 | 113.6 | 70.7 | 91.5 | 116.6 | +13.4 | -12.6 |
| Average | 137.38 | 113.43 | 70.59 | 90.83 | 116.66 | +13.31 | -12.63 |

Manometer Temperature = 68°F

TABLE II
TYPICAL TEST DATA ON THE DEBUTANIZER OVERHEAD CONDENSER

Date: October 14, 1955
Run No.: 10

| Time (p.m.) | Water In | | Water Out | | Mixed Mean (°F) | Hydrocarbon In | | Hydrocarbon Out | | Thermocouple Readings | | | Manometer | |
|----------------|----------|-------|---------------------|---------------------|--------------------|----------------|--------|-----------------|-------------------------------|-----------------------|-------------------|-------|-----------|-------|
| | (°F) | (°F) | Exchanger A (°C) | Exchanger B (°C) | | (°F) | (°F) | (°F) | After Collecting Tank (°F) | Exchanger A mv | Exchanger B °F | mv | Left | Right |
| 3:50 | 76.0 | 76.0 | 33.7 | 34.5 | 92.4 | 144.4 | 102.0 | 1.210 | 64.30 | 1.020 | 67.50 | -3.0 | +3.0 | |
| 3:52 | 75.9 | 75.9 | 33.7 | 34.3 | 92.3 | 144.2 | 101.0 | | | | | -3.0 | +3.0 | |
| 3:54 | 75.7 | 75.7 | 33.6 | 34.5 | 92.3 | 144.0 | 99.2 | | | | | -3.0 | +2.9 | |
| 3:56 | 75.7 | 75.7 | 34.2 | 35.0 | 93.0 | 144.0 | 98.0 | | | 1.020 | 67.50 | -2.9 | +2.8 | |
| 3:58 | 75.5 | 75.5 | 34.1 | 35.2 | 93.4 | 144.1 | 98.2 | | | | | -2.8 | +2.8 | |
| 4:00 | 75.5 | 75.5 | 33.9 | 35.1 | 93.3 | 144.2 | 99.2 | | | | | -2.9 | +2.8 | |
| 4:02 | 75.3 | 75.3 | 34.2 | 35.1 | 93.0 | 144.5 | 100.0 | 1.110 | 64.30 | | | -2.9 | +2.8 | |
| 4:04 | 75.5 | 75.5 | 34.3 | 35.0 | 93.4 | 144.6 | 100.9 | | | | | -3.1 | +2.8 | |
| 4:06 | 75.5 | 75.5 | 34.3 | 35.3 | 93.5 | 144.7 | 101.8 | | | | | -2.8 | +2.7 | |
| 4:08 | 75.2 | 75.2 | 34.0 | 35.0 | 92.8 | 144.3 | 102.0 | | | 1.130 | 67.50 | -2.5 | +2.4 | |
| 4:10 | 75.1 | 75.1 | 34.4 | 35.6 | 94.3 | 144.2 | 99.7 | | | | | -2.3 | +2.2 | |
| 4:12 | 75.0 | 75.0 | 35.0 | 35.7 | 94.5 | 144.5 | 98.6 | 1.200 | 66.00 | | | -2.3 | +2.2 | |
| 4:14 | 75.0 | 75.0 | 35.0 | 35.8 | 94.5 | 144.7 | 98.9 | | | | | -2.3 | +2.2 | |
| 4:16 | 74.8 | 74.8 | 34.8 | 35.8 | 94.5 | 145.0 | 100.0 | | | | | -2.5 | +2.4 | |
| 4:18 | 75.0 | 75.0 | 34.0 | 35.1 | 93.0 | 145.2 | 101.0 | | | | | -2.9 | +2.8 | |
| 4:20 | 75.0 | 75.0 | 33.9 | 34.8 | 92.5 | 145.2 | 101.2 | | | | | -3.1 | +3.1 | |
| Average | 75.37 | 75.37 | 34.19 | 35.11 | 93.29 | 144.49 | 100.10 | 1.170 | 64.87 | 1.070 | 67.50 | -2.77 | +2.68 | |

TABLE III

TYPICAL TEST DATA ON THE DEBUTANIZER BOTTOMS COOLER

Date: January 13, 1956

Run No.: 27

| Time (p. m.) | Hydrocarbon (°F) | | Water (°F) | | Manometer (inches of Hg) | |
|-----------------|---------------------|-------|---------------|-------|-----------------------------|-------|
| | In | Out | In | Out | Left | Right |
| 2:05 | 177.0 | 75.0 | 70.1 | 86.8 | -1.45 | -3.85 |
| 2:07 | 177.0 | 74.8 | 70.3 | 86.5 | -1.45 | -3.85 |
| 2:09 | 178.0 | 75.0 | 70.4 | 86.8 | -1.45 | -3.85 |
| 2:11 | 177.0 | 75.2 | 70.5 | 86.7 | -1.40 | -3.80 |
| 2:13 | 178.0 | 75.7 | 70.4 | 86.6 | -1.40 | -3.80 |
| Average | 177.40 | 75.14 | 70.37 | 86.68 | -1.43 | -3.83 |

A summary of all the test data collected on the high-pressure gas cooler, the debutanizer overhead condenser, and the debutanizer bottoms cooler is presented in Appendices C, D, and E, respectively. Example calculations of the test data given in Tables I, II, and III are presented in Appendices F, G, and H, respectively.

IV. THEORETICAL CONSIDERATIONS

The overall coefficient of heat transfer is defined by the equation

$$Q = U_o A_o \Delta T_M \quad , \quad (1)$$

in which

- Q = rate of heat transfer, Btu/hr,
- U_o = overall coefficient of heat transfer, Btu/hr-°F-ft² outside area
- A_o = outside heat transfer area, ft², and
- ΔT_M = mean temperature difference driving force, °F.

The overall coefficient of heat transfer is related to the individual heat transfer coefficients by

$$\frac{1}{U_o} = \frac{1}{h_o'} + r_o' + r_f + \frac{A_o}{A_i h_i} + \frac{A_o}{A_m} r_m + \frac{A_o}{A_i} r_i \quad , \quad (2)$$

in which

- h_o' = outside film coefficient of heat transfer for a finned tube, Btu/hr-°F-ft² outside area,
- r_o' = outside fouling film resistance for a finned tube, hr-°F-ft² outside area/Btu,
- r_f = fin resistance as defined by Equation 3, hr-°F-ft² outside area/Btu,
- h_i = inside film coefficient of heat transfer, Btu/hr-°F-ft² inside area,
- r_m = root-wall metal resistance, hr-°F-ft² mean metal area/Btu,
- r_i = inside fouling film resistance, hr-°F-ft² inside area/Btu,
- A_i = inside heat transfer area, ft², and
- A_m = logarithmic mean metal area, ft².

The fin resistance¹ is defined by the equation

$$r_f = \left[\frac{1}{h_o'} + r_o' \right] \left[\frac{1 - E_f}{\frac{A_r}{A_f} + E_f} \right], \quad (3)$$

in which E_f = fin efficiency of Gardner.² The fin resistance of the admiralty tubes used in the heat exchangers is given in Table IV. Examination of Table IV indicates that the fin resistance may be assumed constant over the range of coefficients encountered in the heat exchangers tested.

TABLE IV

FIN RESISTANCE OF A 3/4-INCH,
19-FINS-PER-INCH ADMIRALTY TUBE

| $\left[\frac{1}{\frac{1}{h_o'} + r_o'} \right]$ | r_f |
|--|----------|
| 0 | .0001128 |
| 50 | .0001127 |
| 100 | .0001125 |
| 500 | .0001112 |
| 1000 | .0001097 |

The heat transfer rate for a unit may be obtained from a heat balance on the water side of the unit, so that

$$Q = W C_p \Delta t_{H_2O}, \quad (4)$$

in which

W = water flow rate, lb/hr,
 C_p = heat capacity of water, Btu/lb-°F, and
 Δt_{H_2O} = temperature rise of the water, °F.

The inside coefficient for water inside tubes may be calculated from the equation³

$$h_i = \frac{150(1 + 0.011 t_w) V_t^{0.8}}{d_i^{0.2}}, \quad (5)$$

in which

t_w = average water temperature, °F,
 V_t = water velocity inside the tubes, ft/sec, and
 d_i = inside tube diameter, inches.

Since the flow rate of the shell-side fluids was not directly measured and the composition of the shell-side stream was unknown for the majority of the test points, the outside (fin side) coefficient could not be directly calculated.

In the case of the debutanizer bottoms cooler, assuming a constant heat capacity for the shell-side fluid, the shell-side flow rate is proportional to the ratio of the rate of heat transfer, Q, to the temperature drop of the hydrocarbon stream. Also, as indicated in the Williams-Katz report,⁴ the shell-side coefficient is proportional to the shell-side mass flow rate to the 0.65 power. Therefore, it was assumed that

$$h_o' = C \left[\frac{Q}{\Delta t_{\text{hydrocarbon}}} \right]^{0.65}, \quad (6)$$

where

h_o' = fin-side coefficient in the bottoms cooler and
 C = a constant (to be experimentally determined).

The constant was determined from analysis of the data taken with the tubes in a nonfouled condition.

In the case of the high-pressure gas cooler the shell-side coefficient could not be determined in the manner indicated above since the method is only applicable to units in which no phase changes occur on the shell side. The shell-side coefficients for this unit could be determined from the test data taken on the unit in a nonfouled condition by use of Equation 2 (assuming r_o' and r_i to be zero). This coefficient was assumed to be constant throughout the test period.

The shell-side coefficient on the high-pressure gas cooler could not be calculated due to reasons to be explained later in the report.

The fouling resistances for the bottoms cooler were obtained from the field test data by

1. computing U_o from the data, using Equation 1,
2. computing h_i from the data, using Equation 5,
3. computing h_o' from the data, using Equation 6, and
4. solving Equation 2 for the fouling resistance,

$$\left(r_o' + \frac{A_o}{A_i} r_i \right) .$$

The factors r_o' and r_i in this group cannot be separated and evaluated using the field test data.

The fouling resistances for the high-pressure gas cooler were obtained from the field test data by

1. computing U_o from the data using Equation 1,
2. computing h_i from the data using Equation 5,
3. assuming h_o' to be a constant equal to the value determined for the unit in the nonfouled condition, and
4. solving Equation 2 for the fouling resistance,

$$\left(r_o' + \frac{A_o}{A_i} r_i \right) .$$

The fouling resistances for the debutanizer overhead condenser could not be calculated (see Section VI-B).

V. PRELIMINARY CALCULATIONS FOR ANALYSIS OF TEST DATA

A. FINNED-TUBE SPECIFICATIONS

The dimensions of the admiralty tubes used in all three units are tabulated in Table V. These specifications were taken from the Wolverine Trufin Catalog C for tube No. 195065-26. This catalog number corresponds to the tube specified for the units on the blueprints shown in Appendix B.

TABLE V

ADMIRALTY TUBE DIMENSIONS

| | |
|-----------------------------|-------------------------------------|
| Catalog Number: | 195065-26 |
| Specified Outside Diameter: | 3/4 inch |
| Alloy: | Admiralty |
| Plain End | |
| Outside Diameter: | 0.747 - 0.753 inch |
| Wall Thickness: | 0.078 - 0.086 inch |
| Finned Section | |
| Diameter over Fins: | 0.732 - 0.742 inch |
| Root Diameter: | 0.633 - 0.647 inch |
| Wall Thickness: | 0.061 - 0.069 inch |
| Outside Area: | 0.438 ft ² /ft of length |

From correspondence with Wolverine Tube, the area ratio for the tubes was determined to be

$$A_0/A_i = 3.28$$

The following calculations present the determination of (a) the heat transfer areas, (b) the flow-rate equations, and (c) the fouling-resistance equations which were used in the analysis of the field test data.

B. EXTERNAL HEAT TRANSFER AREAS

1. High-Pressure Gas Cooler.—From the heat-exchanger blueprints (Appendix B),

tube length = 196 inches, finned length = 185-1/2 inches,
 number of tubes = 1300 (650 per shell),
 $A_0 = 0.438 \text{ ft}^2/\text{ft of length}$ (from Table V),

$$\text{total outside area} = \left(\frac{185.5}{12}\right) (1300)(0.438) = 8802 \text{ ft}^2.$$

The Griscom-Russell specification sheet (Appendix A) gives 5425 ft² of surface area per shell for a total area of 10,850 ft². The value of 8802 ft² computed on the basis of the blueprint notation calling for Wolverine Trufin No. 195065-26 tube was used in analyzing the test data taken before December 5, 1955.

The top heat exchanger in this unit was retubed on December 5, 1955, with steel finned tubes (Catalog No. 195065-63) having an A_0 (outside area per

foot of length) of 0.496 ft²/foot. The total external area of this unit after the top exchanger was retubed was calculated as

$$A = \left(\frac{185.5}{12}\right)(650)(0.438) + \left(\frac{185.5}{12}\right)(650)(0.496) = 4401 + 4984 = 9385 \text{ ft}^2.$$

This external heat transfer area was used in analyzing all data taken on this unit after December 5, 1955.

2. Debutanizer Overhead Condenser.—From the heat-exchanger blueprints (Appendix B),

$$\begin{aligned} \text{tube length} &= 196 \text{ inches, finned length} = 185\text{-}1/2 \text{ inches,} \\ \text{number of tubes} &= 1410 \text{ (705 per shell),} \\ A_o &= 0.438 \text{ ft}^2/\text{ft of length (from Table V),} \end{aligned}$$

$$\text{total outside area} = \left(\frac{185.5}{12}\right)(1410)(0.438) = 9570 \text{ ft}^2.$$

The Griscom-Russell specification sheet (see Appendix A) gives 5870 ft² of surface area per shell for a total area of 11,740 ft². The value of 9570 ft² computed on the basis of the blueprint notation calling for Wolverine Trufin No. 195065-26 tube was used in analyzing the test data.

3. Debutanizer Bottoms Cooler.—From the heat-exchanger blueprints (Appendix B),

$$\begin{aligned} \text{tube length} &= 192 \text{ inches, finned length} = 185\text{-}1/2 \text{ inches,} \\ \text{number of tubes} &= 370, \\ A_o &= 0.438 \text{ ft}^2/\text{ft of length (from Table V),} \end{aligned}$$

$$\text{total outside area} = \left(\frac{185.5}{12}\right)(370)(0.438) = 2505 \text{ ft}^2.$$

The Griscom-Russell specification sheet (see Appendix A) gives 3050 ft² of surface area for this unit. The value of 2505 ft² computed on the basis of the blueprint notation calling for Wolverine Trufin No. 195065-26 tube was used in analyzing the test data.

C. WATER FLOW-RATE RELATIONSHIPS

The outlet water pipe diameters and water orifice diameters for the three units are given in Table VI. The water flow rate can be computed from the pressure drop across the orifice by use of⁵

$$W = C_o A_o \sqrt{\frac{2g_c \rho (-\Delta P)}{1 - \left(\frac{A_o}{A_i}\right)^2}}, \quad (7)$$

TABLE VI

DIMENSIONS OF OUTLET WATER PIPES AND ORIFICES FOR UNITS TESTED

| Unit | Outlet Water Pipe | | Orifice Diameter (inches) |
|-------------------------------------|------------------------------|-----------------------------|------------------------------|
| | Nominal Diameter (inches) | Inside Diameter (inches) | |
| High-Pressure Gas Cooler | 8 | 7.981 | 5.147 |
| Debutanizer Over- head Condenser | 10 | 10.020 | 6.543 |
| Debutanizer Bottoms Cooler | 6 | 6.065 | 4.457 |

in which

- W = water flow rate, lb/hr,
- C_O = orifice coefficient, dimensionless,
- A_O = flow area of the orifice, ft²,
- g_c = conversion factor, (lb mass) ft/ (lb force) hr²,
- ρ = water density, lb/ft³,
- ΔP = pressure drop across the orifice, lb/ft², and
- A_i = inside flow area of the pipe, ft².

The orifice coefficient, C_O, in Equation 7 is a function of the Reynolds number in the orifice. For Reynolds numbers greater than 30,000, the coefficient can be assumed to have a constant value of 0.61.⁵ During the tests the orifice Reynolds numbers were all computed to be greater than 30,000.

1. High-Pressure Gas Cooler.—Substituting 0.61 for the value of the orifice coefficient and the dimensions of the orifice and pipe (for this unit) from Table VI into Equation 7 gives

$$W = 178,000 \sqrt{-\Delta P'} \quad , \quad (8)$$

where -ΔP' is the manometer reading in inches of mercury.

In the case of the test data which were taken with carbon tetrachloride as the manometer fluid,

$$-\Delta P' = \frac{-\Delta P''}{20} \text{ at } 40^\circ\text{F} \quad , \quad (9a)$$

and

$$-\Delta P' = \frac{-\Delta P''}{21} \text{ at } 70^\circ\text{F} \quad , \quad (9b)$$

where $-\Delta P''$ is the manometer reading in inches of carbon tetrachloride.

The water velocity inside the tubes was computed using the inside diameter obtained from the dimensions given in Table V. The average inside diameter is computed from the dimensions in Table V as

$$d_i = d_r - 2(x_{\text{wall}}) = 0.64 - 0.13 = 0.51 \text{ inch.}$$

$$\text{The inside flow area} = \frac{\pi(0.51)^2}{4 \times 144} N = 0.00142 N \text{ ft}^2, \quad (10)$$

where N is the average number of tubes per pass on the water side.

$$\text{Therefore, the total inside flow area} = 0.00142 \times \frac{1300}{4} = 0.462 \text{ ft}^2.$$

Assuming an average water density of 62 lb/ft^3 ,

$$V_t = \frac{W}{(62)(3600)(0.462)} = \frac{W}{103,000} \text{ ft/sec} \quad (11)$$

2. Debutanizer Overhead Condenser.—Substituting the value of the orifice coefficient and the dimensions given in Table VI for this unit into Equation 7,

$$W = 288,000 \sqrt{-\Delta P'} \quad (12)$$

where $-\Delta P'$ is the manometer reading in inches of mercury.

For the test data obtained on this unit using carbon tetrachloride as the manometer fluid, Equations 9a and 9b were used to obtain $-\Delta P'$.

The average number of tubes per pass on the tube side of this unit is (from the blueprints) $(1410/2)$ or 705 tubes.

The total water flow area is

$$\begin{aligned} \text{flow area} &= 0.00142 N = (0.00142)(705) \\ &= 1.002 \text{ ft}^2. \end{aligned}$$

Assuming an average water density of 62 lb/ft^3 ,

$$V_t = \frac{W}{(62)(3600)(1.002)} = \frac{W}{223,000} \text{ ft/sec.} \quad (13)$$

3. Debutanizer Bottoms Cooler.—Substituting the values of the orifice coefficient and the dimensions given in Table VI for this unit into Equation 7,

$$W = 144,000 \sqrt{-\Delta P'} \quad (14)$$

where $-\Delta P'$ is in inches of mercury. For the test data obtained on this unit with carbon tetrachloride in the manometer, Equations 9a and 9b were used to convert the pressure drop to inches of mercury.

The average number of tubes per pass for this exchanger is (from the blueprints) $(370/2) = 185$ tubes per pass.

Substituting in Equation 10,

$$\begin{aligned} \text{flow area} &= 0.00142 N = 0.00142 (185) \\ &= 0.262 \text{ ft}^2. \end{aligned}$$

Again assuming an average water density of 62 lb/ft^3 ,

$$V_t = \frac{W}{(62)(3600)(0.262)} = \frac{W}{58,500} \text{ ft/sec.} \quad (15)$$

D. FOULING-RESISTANCE EQUATIONS

1. High-Pressure Gas Cooler.—From the initial test data obtained on August 2, 1955, the overall heat transfer coefficient was determined as (see Appendix J)

$$U_o = 24.35 \text{ Btu/hr-}^\circ\text{F-ft}^2.$$

The computed average water velocity was

$$V_t = 3.60 \text{ ft/sec} ,$$

with an average water temperature of 101.92°F . Substituting the values of the tube inside diameter (see Section V-B-1), the average water velocity, and the average water temperature into Equation 5, the inside water film coefficient is computed as

$$\begin{aligned} h_i &= \frac{150[1 + 0.011(101.92)] (3.60)^{0.8}}{(0.51)^{0.2}} \\ &= 1013 \text{ Btu/hr-}^\circ\text{F-ft}^2. \end{aligned}$$

The corresponding inside water film resistance is

$$\frac{A_o}{A_i h_i} = \frac{3.28}{1013} = 0.00323.$$

Assuming negligible fouling present on August 2, 1955, the sum of the fin, root metal, and outside film resistance can be calculated from Equation 2 as

$$\frac{1}{h_o'} + r_f + r_m \frac{A_o}{A_m} = \frac{1}{U_o} - \frac{A_o}{A_i h_i} \quad (16a)$$

Substituting the above values of U_o and $A_o/A_i h_i$ into Equation 16a gives

$$\frac{1}{h_o'} + r_f + r_m \frac{A_o}{A_m} = \frac{1}{24.35} - 0.00323 = 0.0379. \quad (16b)$$

Assuming a constant outside film coefficient, the fouling resistance can be computed by use of the following relationships obtained from a rearrangement of Equation 2:

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + \frac{1}{h_o'} + r_f + r_m \frac{A_o}{A_i} \right], \quad (17a)$$

or, substituting from Equation 16b,

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + 0.0379 \right]. \quad (17b)$$

For the data obtained after the top exchanger had been retubed, the value of the constant given in Equation 17b was revised to include the added heat transfer resistance of the steel tubes. The resulting relationship was

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + 0.0382 \right]. \quad (17c)$$

The steel tubes had an (A_o/A_i) ratio of 3.86 compared to the value of 3.28 for the admiralty tubes. Therefore, an average value of $A_o/A_i = 3.57$ must be used to compute the value of $A_o/A_i h_i$ in Equation 17c.

2. Debutanizer Overhead Condenser.—No fouling results could be computed from the data taken on this unit (see Section VI-B).

3. Debutanizer Bottoms Cooler.—From the initial test data obtained on August 2, 1955, the overall heat transfer coefficient was determined as (see Appendix L)

$$U_o = 64.8 \text{ Btu/hr-}^\circ\text{F-ft}^2.$$

The average water velocity was computed as

$$V_t = 4.83 \text{ ft/sec},$$

with an average water temperature of 110.73°F.

Substituting the values of the tube inside diameter, the average water velocity, and the average water temperature in Equation 5, the inside water film is computed as

$$h_i = \frac{150[1 + 0.011(110.73)] (4.83)^{0.8}}{(0.51)^{0.2}}$$

$$= 1346 .$$

The inside water film resistance is then

$$\frac{A_o}{A_i h_i} = \frac{3.28}{1346} = 0.00244 .$$

The root metal resistance is computed as

$$r_{m \frac{A_o}{A_m}} = \frac{X}{k_m} \frac{A_o}{A_m} = \frac{0.065 (0.141)}{(12)(65)(0.438)}$$

$$= 0.00027 \text{ hr-}^\circ\text{F-ft}^2/\text{Btu} .$$

From Table IV, the fin resistance is

$$r_f = 0.000113 \text{ hr-}^\circ\text{F-ft}^2/\text{Btu} .$$

Assuming that the degree of fouling on August 2, 1955, was negligible, the outside film coefficient is computed as

$$\frac{1}{h_o'} = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + r_{m \frac{A_o}{A_m}} + r_f \right]$$

$$= \frac{1}{64.8} - [0.00244 + 0.00027 + 0.000113]$$

$$= 0.01272 ,$$

or

$$h_o' = \frac{1}{0.01272} = 78.5 \text{ Btu/hr-}^\circ\text{F-ft}^2 .$$

Using the values of Q (heat transfer rate) and $\Delta t_{\text{hydrocarbon}}$ (temperature drop of bottoms fluid) obtained on August 2, 1955, with the value of h_o' given above, the value of the constant C in Equation 6 can be evaluated:

$$h_o' = C \left[\frac{Q}{\Delta t_{\text{hydrocarbon}}} \right]^{0.65} ,$$

or

$$C = \frac{78.5}{\left[\frac{9.37 \times 10^6}{273.12} \right]^{0.65}} = \frac{78.5}{887} = 0.0886$$

Therefore, for the debutanizer bottoms cooler,

$$h_o' = 0.0886 \left[\frac{Q}{\Delta t_{\text{hydrocarbon}}} \right]^{0.65} \quad (18)$$

For this unit, the relationships used to determine the fouling resistance were

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + r_m \frac{A_o}{A_m} + r_f + \frac{1}{h_o'} \right], \quad (17a)$$

or substituting in the values of $r_m \frac{A_o}{A_m}$ and r_f given above ($r_m \frac{A_o}{A_m} = 0.00027$ and $r_f = 0.000113$),

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + \frac{1}{h_o'} + 0.00038 \right], \quad (17d)$$

where h_o' is determined from Equation 18.

The Wilson Plot test data were analyzed by rearranging Equation 17d to give

$$\left[\frac{1}{U_o} - \frac{1}{h_o'} \right] = \frac{A_o}{A_i h_i} + \left[\frac{A_o}{A_i} r_i + r_o' \right] + 0.00038 \quad (19)$$

A plot of the value of $\left[\frac{1}{U_o} - \frac{1}{h_o'} \right]$ vs $\frac{A_o}{A_i h_i}$ gives a straight line which has a slope of one and an intercept value equal to

$$\left[\frac{A_o}{A_i} r_i + r_o' \right] + 0.00038.$$

The intercept value can be used to obtain the combined fouling resistance.

4. Summary of Relationships.—The areas and relationships which were used in analyzing the field test data are summarized in Table VII.

TABLE VII
SUMMARY OF EQUATIONS USED IN ANALYSIS OF TEST DATA

| Unit | Total Outside Heat Transfer Area, ft ² | Water Flow Rate lb/hr | Water Velocity in the Tubes ft/sec | Fouling Resistance $\left[\frac{A_0}{A_i} r_i + r_o \right] - \text{hr-ft}^2\text{°F/Btu}$ |
|--|---|-----------------------------------|------------------------------------|---|
| High-Pressure Gas Cooler (before retubing) | 8802 | $W = 178,000 \sqrt{(-\Delta P')}$ | $V_t = \frac{W}{103,000}$ | $= \frac{1}{U_0} - \left[\frac{A_0}{A_i} \frac{h_i}{h_i} + 0.0379 \right]$ |
| High-Pressure Gas Cooler (after retubing) | 9385 | $W = 178,000 \sqrt{(-\Delta P')}$ | $V_t = \frac{W}{103,000}$ | $= \frac{1}{U_0} - \left[\frac{A_0}{A_i} \frac{h_i}{h_i} + 0.0382 \right]$ |
| Debutanizer Overhead Condenser | 9570 | $W = 288,000 \sqrt{(-\Delta P')}$ | $V_t = \frac{W}{223,000}$ | Cannot be computed from the test data |
| Debutanizer Bottoms Cooler | 2505 | $W = 144,000 \sqrt{(-\Delta P')}$ | $V_t = \frac{W}{58,500}$ | $= \frac{1}{U_0} - \left[\frac{A_0}{A_i} \frac{h_i}{h_i} + \frac{1}{h_{o'}} + 0.00038 \right]$ |

where

$$h_{o'} = 0.0886 \left[\frac{Q}{\Delta t_{\text{hydrocarbon}}} \right]^{0.65}$$

VI. DISCUSSION OF RESULTS

A. HIGH-PRESSURE GAS COOLER

A summary of the calculated results for the high-pressure gas cooler is presented in Appendix J. The fouling resistances computed for this unit are plotted in Fig. 4 vs the number of days of operation and are plotted in Fig. 5 vs the heat duty of the unit.

No test data were obtained on this unit during the two-month period from October 14, 1955, to December 10, 1955, because of tube failures. Approximately one-half of one percent of the tubes had been plugged by October 14, 1955. Thereafter, a number of additional tubes were plugged, and arrangements were made by Aurora Gasoline Company for retubing of the upper heat exchanger. No further test data were taken until the retubed bundle was installed and the unit was placed back on stream. Arrangements had been made to have two thermowells installed in the hydrocarbon outlet nozzles of the upper heat exchanger during this retubing period. This modification was not feasible because the nozzle-wall thicknesses were too thin to take screwed fittings and still meet code requirements. The testing of this unit was resumed on December 10, 1955.

The fouling resistance of this unit was computed from the field test data, using the assumption that the outside film coefficient remained constant and was equal to the value determined for the unit in the nonfouled condition (see Section IV). This assumption would be essentially correct if (1) the total mass flow rate of the hydrocarbon gas stream and the ratio of condensables to noncondensables remained constant, or (2) a fortuitous combination of the above two variables occurred, such that the shell-side coefficient remained constant. Examination of the field test data for this unit (see Appendix C) indicates that the shell-side heat transfer conditions varied widely during the course of the investigation. This indicates that the above assumption of a constant outside coefficient was not true for many of the test runs.

The outside coefficient in a partial condenser is a function of (1) the mass flow rate of the hydrocarbon noncondensable gas stream, (2) the amount of condensables present in the stream, (3) the physical properties of the fluids, and (4) the tube-wall temperature. An increase in the heat duty of a partial condenser (for approximately the same inlet hydrocarbon temperature) indicates either (1) an increase in the flow rate of noncondensables, (2) an increase in the flow rate of condensables, or (3) an increase in both of these items. Since all the above three conditions increase the shell-side film coefficient, it would be expected that for the runs in which the heat duty exceeded that of August 2, 1955 ($Q > 6.13 \times 10^6$ Btu/hr), the computed fouling resistance would be less than the actual fouling resistance. In the case of

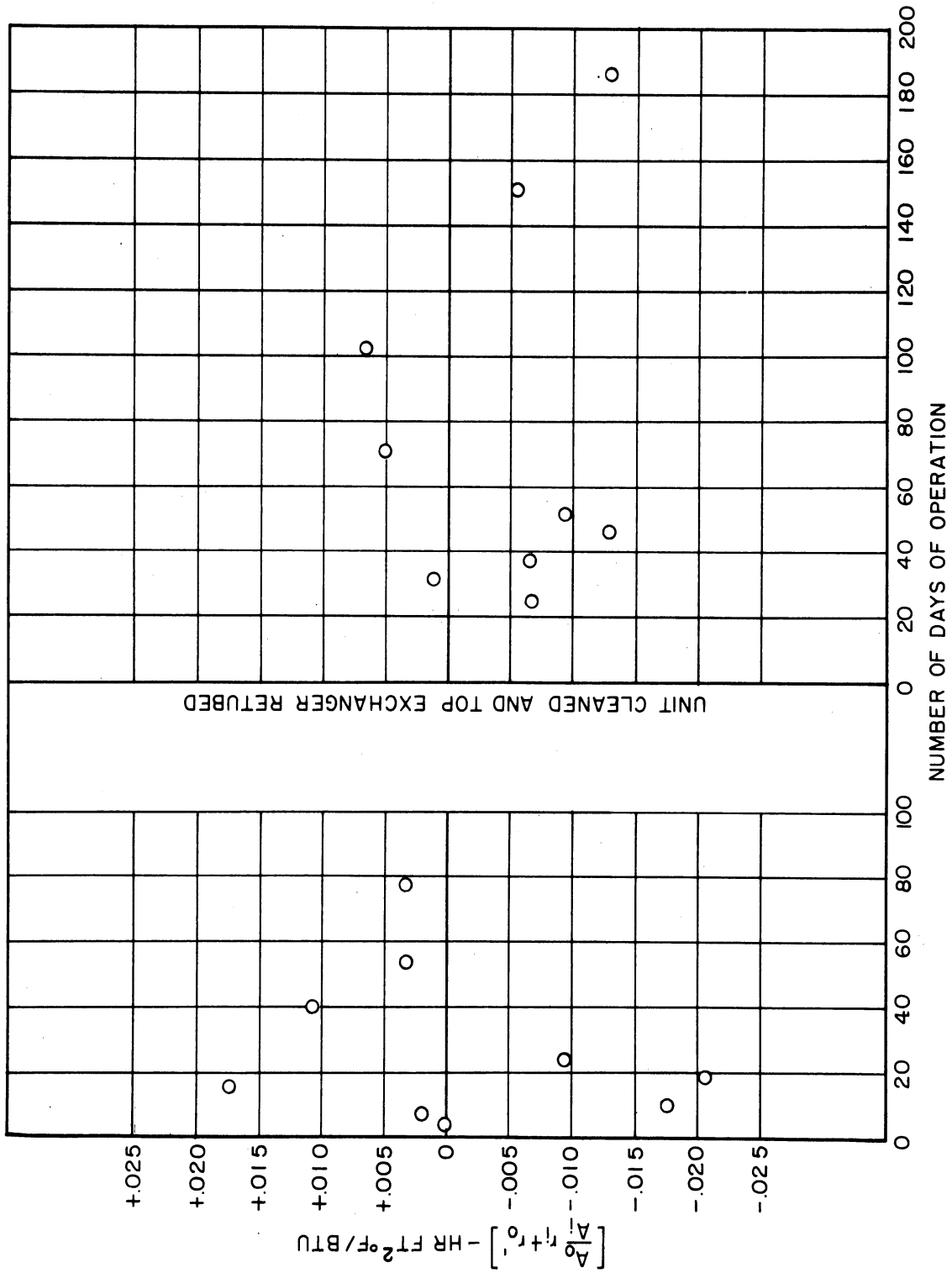


Fig. 4. High-pressure gas cooler; variation of the computed fouling resistance with days of operation.

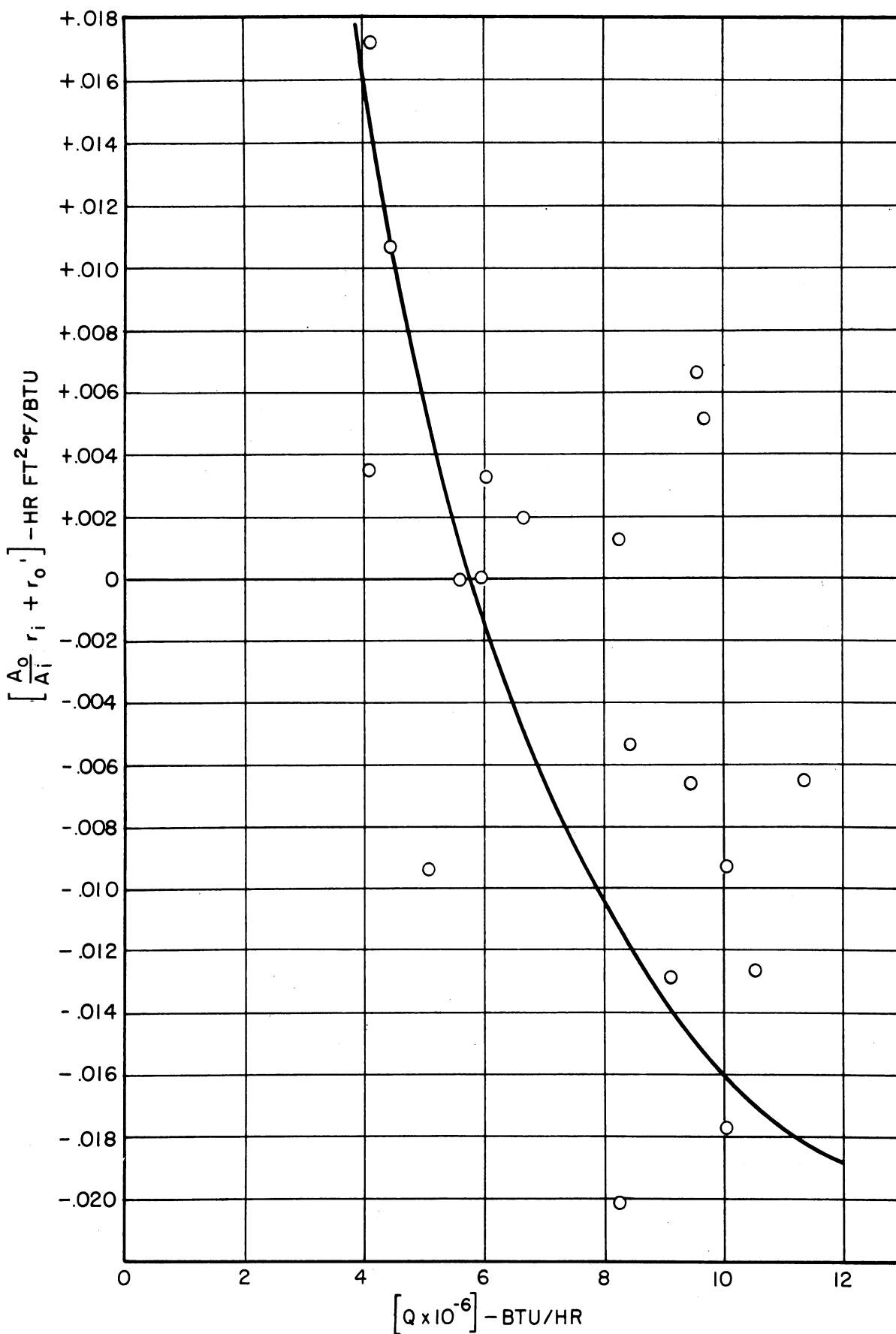


Fig. 5. High-pressure gas cooler; variation of the computed fouling resistance with heat duty.

the runs in which the heat duty of this unit was less than 6.13×10^6 Btu/hr, the computed fouling resistance would be expected to be above the actual fouling resistance. Thus, the variation in the computed fouling resistance shown in Fig. 5 includes a variation in the shell-side coefficient. The logarithmic mean temperature difference was used in computing all the overall coefficients given in Appendix J for this unit. The use of this temperature difference implies the assumption that the heat transferred from the hydrocarbon stream per degree of temperature drop is a constant. The composition of the hydrocarbon stream for this unit was not available to check this assumption.

Using the following approximate relationship for the shell-side heat transfer coefficient in a partial condenser,¹

$$h_{\text{eff.}} = h_o' \frac{Q_T}{Q_S}, \quad (20)$$

in which

- $h_{\text{eff.}}$ = effective outside coefficient for partial condensers,
- h_o' = gas film coefficient,
- Q_T = total heat duty, and
- Q_S = sensible heat duty (heat duty not associated with a change of phase),

and assuming a constant Q_S , sensible heat transfer, and a constant gas film coefficient, h_o' , the effective coefficient is related to the total heat duty of the exchanger by

$$h_{\text{eff.}} = KQ_T, \quad (21)$$

in which $K = \text{a constant} = \frac{h_o'}{Q_S}$.

A plot of Equation 20 on rectangular coordinates will result in a straight line passing through the origin. This is illustrated in Fig. 6, where the computed shell-side coefficients (assuming no fouling) are plotted vs the heat duty of the exchanger. The straight line given on this figure was drawn through the origin and through the test point computed from the data taken on the unit in a nonfouled condition (data of August 2, 1955).

It is significant to note that, although the experimental data points plotted in Fig. 6 scatter considerably, the majority of the test points lie below the line. If the inlet conditions for the high-pressure gas cooler were such that the assumption of a constant Q_S and h_o' proved to be valid, all the points computed from data taken with the tubes in a fouled condition would necessarily fall below the line. The scatter shown on the graph indicates that the inlet conditions varied sufficiently to cause Q_S and h_o' to change somewhat during the course of the investigation. The quantitative effect of the varying inlet conditions on the computed fouling resistance could not be computed due to the lack of the hydrocarbon stream analysis. However, the

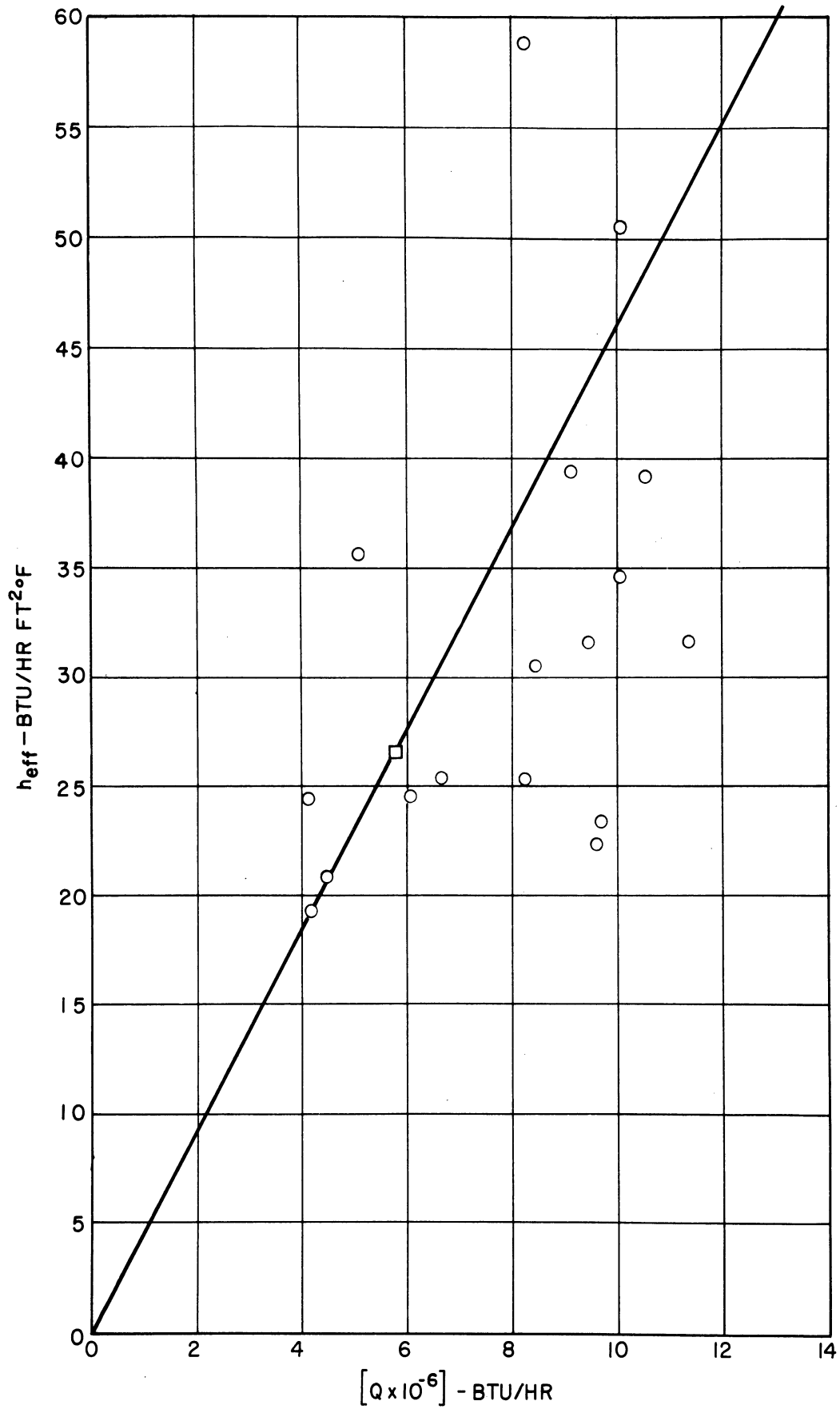


Fig. 6. High-pressure gas cooler; variation of the computed effective outside coefficient with heat duty.

fact that the majority of the test points lie below the line indicates that the tubes fouled somewhat during the course of the test investigation.

The equation relating h_{eff} and Q , which was plotted in Fig. 6, was also used to determine a reference line for Fig. 5. The line shown in Fig. 5 represents the curve which would be obtained if the variation in Q for the unit which was observed during the test period were due entirely to variations in the outside coefficient of the unit (no fouling of the tubes). Consequently, all points on this figure corresponding to tubes in the fouled condition should fall below the line. The scatter of the test points (with some points falling above the line) indicates that the assumption of a constant h_o' and Q was not entirely valid.

The fouling results obtained from the field test data on this unit are not significant. Therefore, no conclusions can be drawn in this case. The computed results for this unit are summarized in Appendix J and are useful for comparing the actual performance and load conditions against the design specifications for the unit. For convenience, the average values of the test data and computed results appear at the bottom of the summary sheets in Appendices C and J. The test values are summarized in Table VIII for comparison with specifications.

TABLE VIII

COMPARISON OF FIELD LOAD CONDITIONS WITH SPECIFICATIONS FOR THE HIGH-PRESSURE GAS COOLER

| <u>Item</u> | <u>Specifications</u> | <u>Field Test Average</u> |
|--------------------------|-------------------------------|---------------------------------|
| Inlet water | 85°F | 81.04°F |
| Outlet water | 110°F | 107.28°F |
| ΔT_{water} | 25°F | 26.24°F |
| Inlet hydrocarbon | 155°F | 144.67°F |
| Outlet hydrocarbon | 100°F | 110.01°F |
| $\Delta T_{hydrocarbon}$ | 55°F | 34.66°F |
| Heat duty, Q | 13,790,000 Btu/hr | 7,720,000 Btu/hr |
| Water velocity | 5.2 ft/sec | 2.90 ft/sec |
| U_o | 56 Btu/hr-ft ² -°F | 27.0 Btu/hr-ft ² -°F |
| MID (Uncorrected) | 27.3°F | 32.23°F |
| MID (G.R. correction) | 23.2°F | 27.40°F |
| A_o | 10,850 ft ² | 8,802 ft ² |

B. DEBUTANIZER OVERHEAD CONDENSER

A summary of the calculated results for the debutanizer overhead condenser is presented in Appendix K. The variation of the calculated overall heat transfer coefficient with days of operation is shown in Fig. 7. Due to

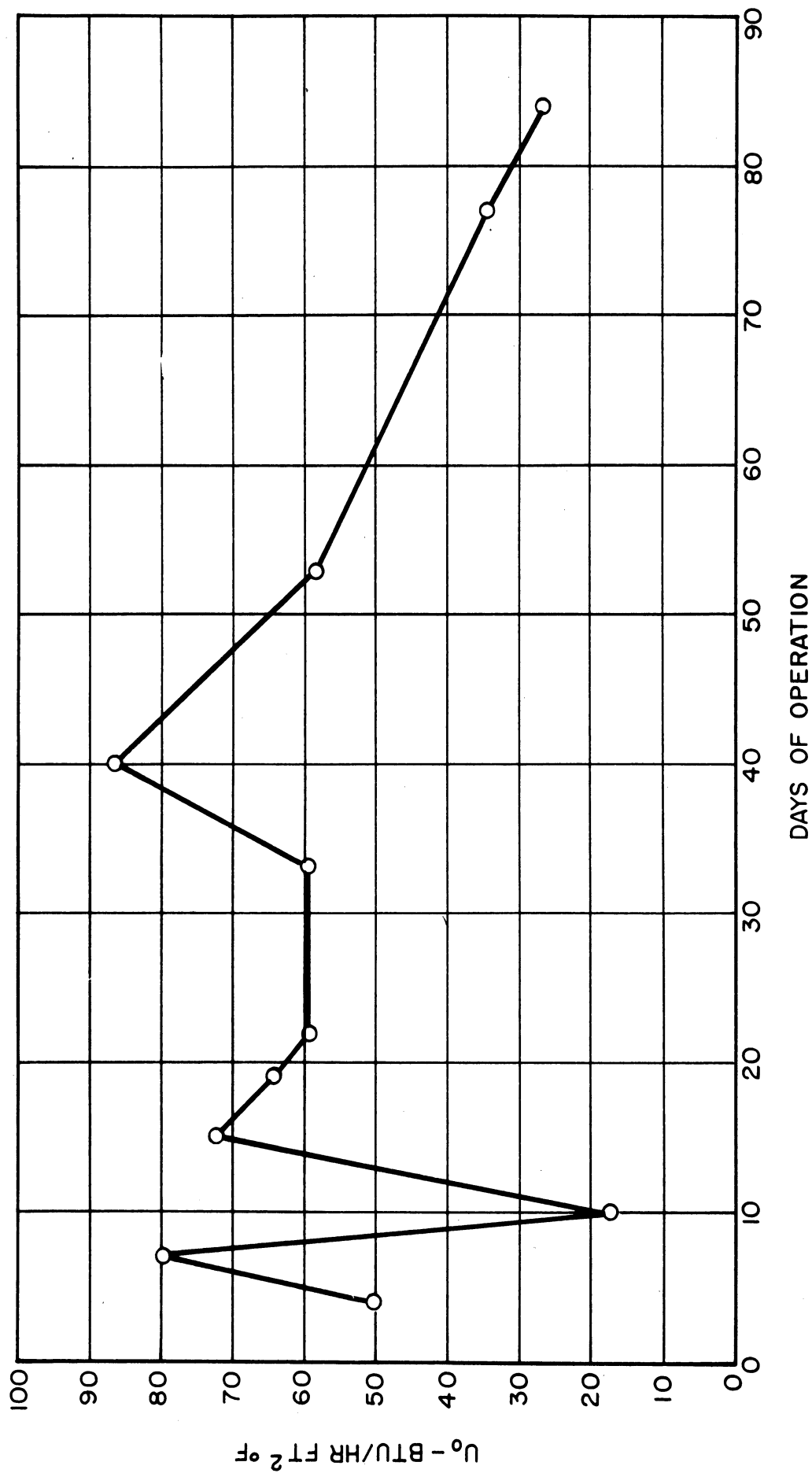


Fig. 7. Debutanizer overhead condenser; variation of the computed overall heat transfer coefficient with days of operation.

several factors which are discussed later in this section, it was impossible to compute any fouling results from the field test data obtained on this unit.

A total of 11 test runs were made on this unit from August 2, 1955, to October 21, 1955. At this time it was believed that the unit operated as a nonflooded total condenser. The test data were therefore used to compute a series of fouling resistances, assuming a constant outside coefficient of heat transfer. The results of this analysis showed no fouling trend, with several of the computed fouling resistances being negative. The lack of a fouling trend was at this time attributed to (1) the invalidity of the assumption of a constant outside heat transfer coefficient for the unit, and/or (2) the inaccuracy of the outlet hydrocarbon temperature measurement due to the position of the measuring thermometer (see Section III). Since it was impossible to install thermowells in the hydrocarbon outlet line leaving the exchangers (see Section III), it was decided that sufficiently accurate test data for a fouling analysis could not be obtained on this unit. Therefore, the tests on this unit were discontinued after approximately eighty-four days of operation.

A study of the specifications and the final designs was undertaken to ascertain why no fouling trend could be obtained from the field test data. A series of calculations, based on the specification sheet for the debutanizer overhead condenser, were made to determine the condensing range of the hydrocarbon stream for the unit. These calculations are presented in Appendix M. The calculations indicated that, at the design pressure, the condensing range of the hydrocarbon stream would be approximately 25°F. The specification sheet for the unit specifies a 25° temperature drop for the hydrocarbon stream. Calculations were also made to predict the overall heat transfer coefficient for the unit, assuming that all the tube area was used for condensing (no flooding of the exchanger). These calculations are presented in Appendix N. The predicted overall coefficient using this assumption checked within seven percent of the specified design coefficient. Therefore, since (1) the specification sheet allowed for no subcooling (25°F temperature drop of the hydrocarbon stream), and (2) the computed overall coefficient, assuming no flooding, checked reasonably well with the design value, it appeared that the unit was originally designed to operate with no flooding of the tubes. This seemingly confirmed our original thinking on this matter.

The average observed temperature drop of the hydrocarbon stream was 57.33°F (see Appendix K). This represented 32.33°F of subcooling if the calculated 25°F condensing range were correct and the hydrocarbon inlet stream had no superheat. This amount of subcooling could not be accounted for in the condenser itself unless the unit was flooded. However, the observed subcooling was assumed to be due to the condensate residence time in the overhead receiver since the measured outlet hydrocarbon temperature was always slightly higher than the ambient-air temperature at the time of the test. It was impossible to check this assumption since the actual hydrocarbon outlet temperature from the condensers could not be experimentally measured. This was one

of the more important reasons why thermowells in the outlet hydrocarbon lines leaving the exchangers would have been very valuable. In order to determine more accurately the actual condensing range of the hydrocarbon stream entering the unit, stream analysis data giving the composition of samples of the hydrocarbon stream were obtained from the Aurora Gasoline Company. Calculations of the condensing range, based on these compositions, checked closely with the value calculated indirectly from the specification sheets, i.e., between 20° and 25°F in all cases.

In industrial applications involving an overhead condenser operating off a fractionating tower, some method of control is necessary to prevent the condenser from affecting the operating pressure of the tower. The two most commonly used methods of control are (1) throttling the vapor line between the tower and the condenser and (2) flooding the condenser to the point where only sufficient surface is exposed to vapor condensation to give the desired total condensation. The first of these methods permits a fouling investigation to be made on the condenser. With the second case the actual operational heat transfer area used for condensing is unknown and is variable.

Due to the extreme scattering of the test data and the large apparent amount of subcooling, it seemed likely that, although the unit had apparently not been specifically designed for operating in a partially flooded condition, the control system used by the Aurora Gasoline Company might have resulted in partial flooding of the unit. Subsequent discussion with the Aurora Gasoline Company confirmed that the control system did involve partial flooding of both exchangers.

Since the operational condensing area of this flooded condenser cannot be determined, no meaningful fouling results could be computed from the field test data.

The overall coefficients (U_0) plotted in Fig. 7 are based on terminal temperatures, the total exchanger area, and an uncorrected log-mean temperature difference. This method of analysis would yield values of U_0 which would normally correspond to the design value of U_0 if the exchanger were not flooded. The computed overall coefficients in Fig. 7 do not represent either the condensing coefficient or the subcooling coefficient and serve only to demonstrate the scattering of the results computed in the above manner. Some of the factors causing errors in the analysis are

1. the heat transfer area of the exchanger, when flooded, is distributed between condensing and subcooling in an unknown manner,
2. the log-mean temperature difference computed using the terminal temperatures is not applicable when the area is split between condensing and subcooling, and
3. the log-mean temperature difference for a nonflooded condenser, in which case the terminal temperatures may be used in computing the correct

temperature difference, must be corrected according to the equation shown in Appendix O.

For convenience, the summaries of field test data (Appendix D) and computed results (Appendix K) have been averaged, with the average appearing at the bottom of the respective appendices. These averages are summarized in Table IX for comparison with the specifications for the unit.

TABLE IX

COMPARISON OF FIELD LOAD CONDITIONS WITH
SPECIFICATIONS FOR THE DEBUTANIZER OVERHEAD CONDENSER

| <u>Item</u> | <u>Specifications</u> | <u>Field Test Average</u> |
|---------------------------------|------------------------|---------------------------|
| Inlet water | 85°F | 86.00°F |
| Outlet water | 110°F | 103.71°F |
| ΔT_{water} | 25°F | 17.71°F |
| Inlet hydrocarbon | 135°F | 150.60°F |
| Outlet hydrocarbon | 110°F | 93.28°F |
| $\Delta T_{\text{hydrocarbon}}$ | 25°F | 57.32°F |
| Heat duty, Q | 21,360,000 Btu/hr | 9,830,000 Btu/hr |
| Water velocity | 7 ft/sec | 2.61 ft/sec |
| A_o | 11,740 ft ² | 9,570 ft ² |

C. DEBUTANIZER BOTTOMS COOLER

A summary of the calculated results for the debutanizer bottoms cooler is presented in Appendix L. The variation of the calculated fouling resistance with the number of days of operation is shown in Fig. 8. The Wilson Plots, from which the fouling resistance was determined during the later part of the test period, are shown in Fig. 9.

For discussion purposes, Fig. 8 will be divided into two sections— an initial period of 84 days (from August 2 to October 21, 1955) and a second period of 199 days (from November 8, 1955, to May 26, 1956) which occurred after the tubes had been cleaned on the inside during a plant shutdown. The division separates the data taken with an older catalytic cracking unit on stream (the initial period) from the data taken with a new catalytic cracker on stream.

The computed fouling resistances for the first period show a reasonable trend with the operating time, as indicated by the curve shown in Fig. 8. The fouling rate indicated by the data for this period was considerably greater than that normally encountered in fouling studies.⁸ The unit apparently reached the design fouling condition after about 70 days of operation, with the amount of fouling continuing to rise at a rapid rate. The rapid fouling

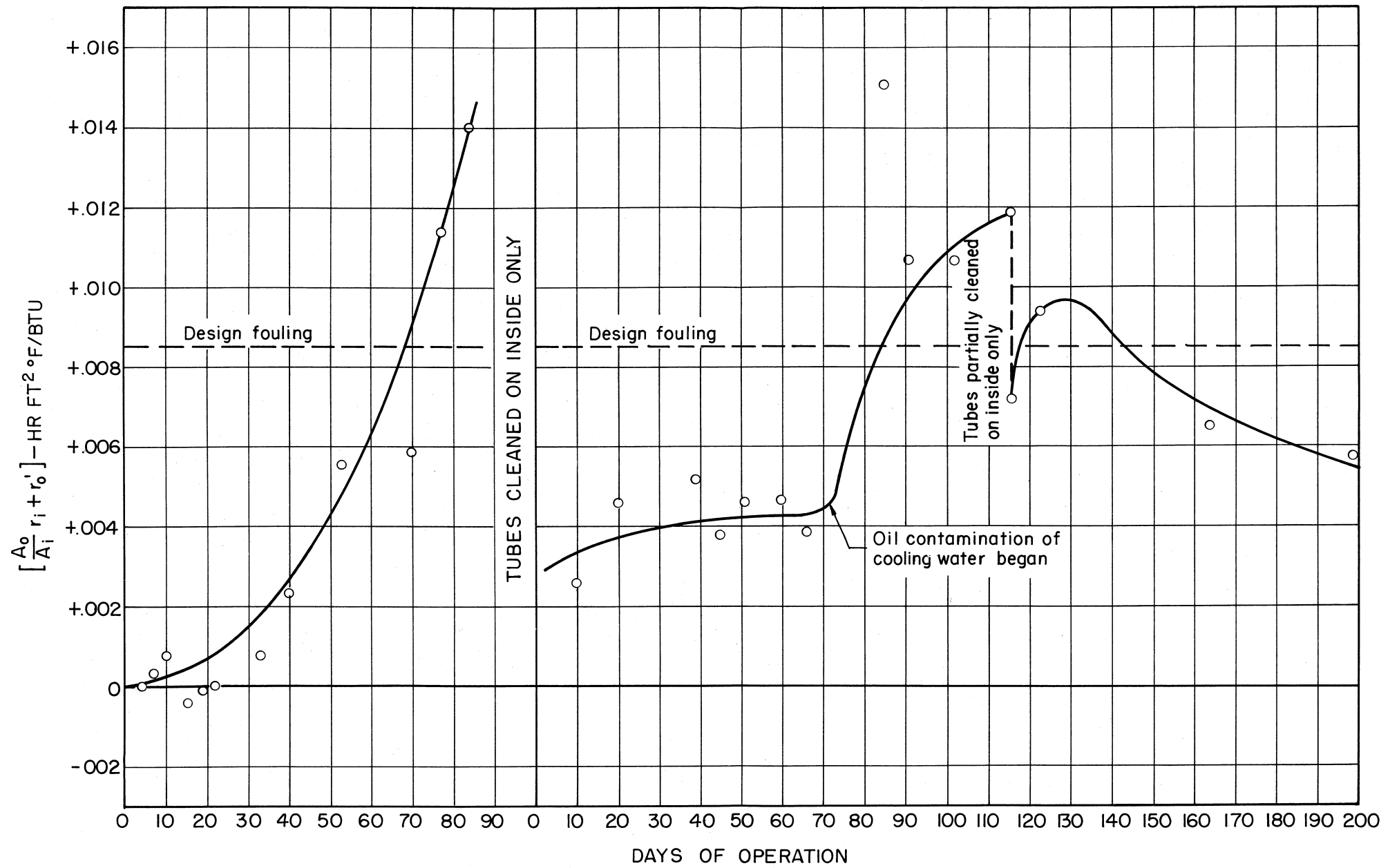


Fig. 8. Debutanizer bottoms cooler; variation of the computed fouling resistance with days of operation.

rate during this period was attributed to the high inlet hydrocarbon temperature, which ranged from 50° to 150°F above the design value. Such a high inlet hydrocarbon temperature causes a high tube-wall temperature, which in turn normally increases the precipitation rate of the hardness salts present in the cooling water. It seems apparent that the majority of the fouling accumulated during the first period was on the inside (water side) of the tubes; this fact is shown by the large decrease observed in the fouling resistance when the tubes were cleaned on the inside only. The large outside-to-inside area ratio of the finned tubes used in the exchanger tends to magnify the effect of fouling on the inside of the tubes.

The plot showing the second series of test data, after the tubes had been cleaned on the inside, has no data points within the first 10 days of operation. This is due to the fact that the project personnel were not informed that the unit had been cleaned and placed back on stream until nine days after the cleaning. Thus, the exact amount of cleaning could not be determined.

The first 70 days of the second period showed a low fouling rate, with the fouling remaining well below the design value. The hydrocarbon inlet temperature during this time was considerably lower than during the first period, falling from an average of 356°F during the first period to less than 200°F for the early part of the second period. This tends to corroborate the theory that the high fouling rate during the first period was due to an excessively high inlet hydrocarbon temperature.

The first Wilson Plot runs on the unit were taken on December 23, 1955. After this date, all data taken were of this type. The fouling resistances plotted in Fig. 8 may be obtained by subtracting 0.00038 (the sum of r_f and $\frac{A_o}{A_m} r_m$) from the corresponding day's intercept value in Fig. 9.

After approximately 70 days of operation, the fouling resistance of the unit was observed to rise sharply. This trend was first noticed on February 1, 1956, and was confirmed on February 7, 1956. Subsequent conversations with Aurora Gasoline Company plant engineers established that an oil leak had developed somewhere in the plant which was contaminating the recirculated cooling water with oil. Since the observed fouling was considerably higher than the design value, it was decided to attempt to clean the unit while leaving it on stream. By arrangement with the plant, project personnel were present to take data just before and immediately after the cleaning. The results of the cleaning are shown in Fig. 8 (at 116 days of operation). The cleaning succeeded in decreasing the fouling to below the design value. However, on the next occasion of obtaining test data (one week later) the fouling had again exceeded the design condition.

The remainder of the fouling-trend curve (from 123 to 200 days of operation) shows the results of the Aurora Gasoline Company's attempt to remove all oil contamination from the cooling water. Without actual cleaning

of the unit, the decrease in the amount of oil contamination of the water caused the observed fouling to decrease. Thus, the unit effectively seemed to clean itself during this period.

For convenience, the summaries of field test data (Appendix E) and computed results (Appendix L) have been averaged, with the average appearing at the bottom of the respective appendices. These averages are summarized in Table X for comparison with the specifications for the unit.

TABLE X
COMPARISON OF FIELD LOAD CONDITIONS WITH
SPECIFICATIONS FOR THE DEBUTANIZER BOTTOMS COOLER

| <u>Item</u> | <u>Specifications</u> | <u>Field Test Average</u> |
|--|-------------------------------|----------------------------------|
| Inlet water | 85°F | 75.75°F |
| Outlet water | 115°F | 108.87°F |
| ΔT_{water} | 30°F | 33.12°F |
| Inlet hydrocarbon | 230°F | 233.70°F |
| Outlet hydrocarbon | 100°F | 91.07°F |
| $\Delta T_{\text{hydrocarbon}}$ | 130°F | 142.63°F |
| Heat duty, Q | 9,790,000 Btu/hr | 5,780,000 Btu/hr |
| Water velocity, V_t | 6 ft/sec | 3.29 ft/sec |
| Overall heat transfer coefficient, U_o | 67 Btu/hr-ft ² -°F | 45.65 Btu/hr-ft ² -°F |
| MTD (ΔT_{LM}) | 48°F | 49.62°F |
| A_o | 3050 ft ² | 2505 ft ² |

VII. CONCLUSIONS

No significant conclusions could be drawn concerning the fouling rates of either the high-pressure gas cooler or the debutanizer overhead condenser. In the case of the high-pressure gas cooler the unit was operated as a partial condenser with a liquid-vapor feed containing noncondensables. It was not possible to separate the fouling resistance from the film resistances. The debutanizer overhead condenser was operated in a partially flooded manner. It was not possible to determine the fouling resistance because the extent of flooding was unknown.

Significant conclusions can be drawn from the fouling test data obtained on the debutanizer bottoms cooler. These are: (a) if the temperature of the hydrocarbon stream is maintained at the design level, the total fouling can be expected not to exceed the design level over a reasonable length of time, (b) the tube-side fouling (water side) greatly affects the overall foul-

ing, and (c) cleaning of the tube side restores the unit to below the design fouling level.

Discrepancies exist between the specified heat transfer areas and the actual heat transfer areas provided in all three units. The units have about 81% of the specified heat transfer area. In general, the three units were not operating with the design flow rates.

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APPENDIX A

THE GRISCOM-RUSSELL CO. HEAT EXCHANGER SPECIFICATIONS

THE GRISCOM-RUSSELL CO.
HEAT EXCHANGER SPECIFICATIONS G-R No.

| | | | | | |
|----|--|-------------------|----------------------------|---|-------------------------------|
| 1 | | | Job No. | 8235-A | |
| 2 | Customer | | Reference No. | AURORA GASOLINE COMPANY | |
| 3 | Address | | Inquiry No. | Detroit, Michigan. | |
| 4 | Plant Location | | Date | Gas Concentration Unit 11-11-51 | |
| 5 | Service of Unit | | Item No. | High Pressure Cooler E-2 | |
| 6 | Size and Type | | Two (2) #33-5K2-192 FSA | | |
| 7 | | | Horiz. - Vert. - Stat. Hd. | | |
| 8 | Surface per Unit | | Shells per Unit | Series | Parallel |
| 9 | 10,850 sq ft | | 2 | 2 | |
| 9 | PERFORMANCE OF ONE UNIT | | Shell Side | Tube Side | |
| 10 | Fluid Circulated | | See B. low | | Water |
| 11 | Total Fluid Entering | Water | 20,200 | lb per hr | 550,000 |
| 12 | Vapor | 109° A I-52 MW | 29,300 | lb per hr | |
| 13 | Liquid | 75° A I-2-3K | 205,100 | lb per hr | |
| 14 | Steam | | 700 | lb per hr | |
| 15 | Non-Condensables | | 25,700 | lb per hr | |
| 16 | Fluid Vaporized or Condensed | | 29,300 | lb per hr | |
| 17 | Steam Condensed | | 700 | lb per hr | |
| 18 | Gravity - Liquid | | | | |
| 19 | Density @ Ave. Cond. | 4.25# IN - 6# OUT | lb per cu ft | | lb per cu ft |
| 20 | Viscosity | | Centipoises at | °F | Centipoises at |
| 21 | Viscosity | * 0.8 | Centipoises at | 125 °F | Centipoises at |
| 22 | Molecular Wt. | | | | |
| 23 | Conductivity | 0.8 | | | |
| 24 | Specific Heat | .55 | Btu per lb | Btu per lb | |
| 25 | Latent Heat - Vapors | | Btu per lb | Btu per lb | |
| 26 | Temperature In | | 155 °F | 85 °F | |
| 27 | Temperature Out | | 100 °F | 110 °F | |
| 28 | Operating Pressure | | 210 psig | 75 psig | |
| 29 | Number of Passes | | Split Flow | | 2 |
| 30 | Velocity | | fps | | 5.2 fps |
| 31 | Pressure Drop | | 3 * | psi | 10 * psi |
| 32 | Heat Exchanged - Btu per hr | 13,790,000 | MTD (Corrected) | | 23.2 |
| 33 | Transfer Rate - Service | 56 | Clean | Fouling Resistance: Shell | .0005 Tubes .008 |
| 34 | CONSTRUCTION | | | | |
| 35 | Design Pressure | | 300 | psig | 75 psig |
| 36 | Test Pressure | | 450 | psig | 113 psig |
| 37 | Design Temp. (Max. Metal Temp.) | | 450 | °F | 250 °F |
| 38 | Tubes | Admiralty | No. 650 ea. | O.D. 5/8" BWG 16 | Length 192 Pitch 1" Square |
| 39 | Shell | 33 | I.D. Mat'l | Steel | Tube Fins Admiralty * |
| 40 | Shell Cover | Steel | Floating Head Cover | | |
| 41 | Channel | Steel | Channel Cover | | |
| 42 | Tube Sheets—Stationary | Steel | Floating | | |
| 43 | Baffles—Cross | Steel | Type Segment | Spacing 2 1/4 | Thickness 3/8" Hole Dia. |
| 44 | Baffle—Long | | Type | | Thickness Impact Baffle Steel |
| 45 | Tube Supports | Floating Head | No. 1 | Thickness 5/8" | |
| 46 | Gaskets | As Required | Gland Packing | | |
| 47 | Connections—Shell - In | 12" | Out 2 - 10" | Series 30 | Dome Yes |
| 48 | Channel - In | 8" | Out 8" | Series 15 FF | |
| 49 | Corrosion Allowance—Shell Side | 1/8" | Tube Side | 1/8" | |
| 50 | Weight | 18,000 | Code 1952 ASME | Customer's Specifications UOP 251, Rev. 7 | Tema Class R |
| 51 | Length | | Width | Height | |
| 52 | Ref. Print | | Inspection by whom | | |
| 53 | Based upon operating conditions as given, maximum temperature of tubes will not exceed °F | | | | |
| 54 | Note: Indicate after each part whether stress relieved (S.R.) and whether radiographed (X-R) | | | | |
| 55 | Remarks: S. S. No. LHW 5940TLR12CX | | | | |

* Total Pressure Drop for Two Shells
** Wolverine Trufin.

THE GRISCOM-RUSSELL CO.
HEAT EXCHANGER SPECIFICATIONS G-R No.

| | | | | | |
|----|--|------------------------------------|---------------------------|----------------------------|---|
| 1 | | | | Job No. | 8235-E |
| 2 | Customer | ALLORA GASOLINE CO. | | Reference No. | |
| 3 | Address | Detroit, Michigan. | | Inquiry No. | |
| 4 | Plant Location | Gas Condensation Unit | | Date | 11-10-54 |
| 5 | Service of Unit | Debutanizer Condenser | | Item No. | E-9 |
| 6 | Size and Type | Two (2) #35-512-192 TBA Condensers | | | |
| 7 | | | | Horiz. - Vert. - Stat. Hd. | |
| 8 | Surface per Unit | 11,740 sq ft | Shells per Unit | 2 ; Series | ; Parallel 2 ; Surface per Shell 5870 sq ft |
| 9 | PERFORMANCE OF ONE UNIT | | Shell Side | Tube Side | |
| 10 | Fluid Circulated | Hydrocarbon Vapors | | Water | |
| 11 | Total Fluid Entering | 142,000 | lb per hr | 835,000 | lb per hr |
| 12 | Vapor | 142,000 | lb per hr | | lb per hr |
| 13 | Liquid | | lb per hr | 835,000 | lb per hr |
| 14 | Steam | | lb per hr | | lb per hr |
| 15 | Non-Condensables | | lb per hr | | lb per hr |
| 16 | Fluid Vaporized or Condensed | 142,000 | lb per hr | | lb per hr |
| 17 | Steam Condensed | | lb per hr | | lb per hr |
| 18 | Gravity - Liquid | .56 - 121° API | | | |
| 19 | Density @ Ave. Cond. | | lb per cu ft | | lb per cu ft |
| 20 | Viscosity | 0.1 | Centipoises at 123 °F | Centipoises at | °F |
| 21 | Viscosity | | Centipoises at °F | Centipoises at | °F |
| 22 | Molecular Wt. | 50.8 | | | |
| 23 | Conductivity | .078 | | | |
| 24 | Specific Heat | | Btu per lb | | Btu per lb |
| 25 | Latent Heat - Vapors | | Btu per lb | | Btu per lb |
| 26 | Temperature In | 135 | °F | 85 | °F |
| 27 | Temperature Out | 110 | °F | 110 | °F |
| 28 | Operating Pressure | 132 | psig | 75 | psig |
| 29 | Number of Passes | 1 | | 2 | |
| 30 | Velocity | | fps | 7 | fps |
| 31 | Pressure Drop | | psi | 13.5 | psi |
| 32 | Heat Exchanged - Btu per hr | 22,200,000 | MTD (Corrected) | 215 | |
| 33 | Transfer Rate - Service | Clean | Fouling Resistance: Shell | .0005 | Tubes .008 |
| 34 | CONSTRUCTION | | | | |
| 35 | Design Pressure | 200 | psig | 75 | psig |
| 36 | Test Pressure | 300 | psig | 115 | psig |
| 37 | Design Temp. (Max. Metal Temp.) | 350 | °F | 250 | °F |
| 38 | Tubes | Admalty No. 705 | O.D. 1/2 | BWG 16 | Length 102 |
| 39 | Shell | 35 | I.D. Mat'l Steel | Tube Fins | Admalty |
| 40 | Shell Cover | Steel | Floating Head Cover | Steel | |
| 41 | Channel | Steel | Channel Cover | Steel | |
| 42 | Tube Sheets—Stationary | Steel | Floating | Steel | |
| 43 | Baffles—Cross | No | Type | Spacing | Thickness |
| 44 | Baffle—Long | No | Type | Thickness | Impact Baffle Steel |
| 45 | Tube Supports | (Conv) Steel | No. 7 | Thickness | 3/8" |
| 46 | Gaskets | As Rec'd | Gland Packing | | |
| 47 | Connections—Shell - In | 32 | Out 10 | Series 30 | Dome Yes |
| 48 | Channel - In | 12 | Out 12 | Series 15 | F.F. |
| 49 | Corrosion Allowance—Shell Side | 1/8" | Tube Side | 1/8" | |
| 50 | Weight | 17,300 | Code 1952 | ASME | Customer's Specifications No. 241, Rev. 7 |
| 51 | Length | | Width | | Height |
| 52 | Ref. Print | | Inspection by whom | | |
| 53 | Based upon operating conditions as given, maximum temperature of tubes will not exceed °F | | | | |
| 54 | Note: Indicate after each part whether stress relieved (S.R.) and whether radiographed (X-R) | | | | |
| 55 | Remarks: | S. S. No. GR 850111203 | | | |

* Pressure drop total for two shells in parallel performance, design and weight for one shell only.
Wolverine Trufin to be used.

THE GRISCOM-RUSSELL CO.
HEAT EXCHANGER SPECIFICATIONS G-R No.

| | | | |
|----|--|--|--|
| 1 | | Job No. 8235-G | |
| 2 | Customer <u>AMROA GASOLINE CO.</u> | Reference No. | |
| 3 | Address <u>DETROIT, MICHIGAN.</u> | Inquiry No. | |
| 4 | Plant Location <u>GAS CONCENTRATION UNIT</u> | Date <u>11-17-51</u> | |
| 5 | Service of Unit <u>DEBUTANIZER BOTTOMS COOLER</u> | Item No. <u>T-10</u> | |
| 6 | Size and Type <u>#25-532-192 EGA</u> | | |
| 7 | | Horiz - <u>Vert.</u> - Stat. Hd. | |
| 8 | Surface per Unit <u>3050</u> sq ft Shells per Unit <u>1</u> ; Series <u>1</u> ; Parallel ; Surface per Shell <u>3050</u> sq ft | | |
| 9 | PERFORMANCE OF ONE UNIT | | |
| 10 | Fluid Circulated | Shell Side | |
| 11 | Total Fluid Entering | Tube Side | |
| 12 | Vapor | Debutanized Gasoline | |
| 13 | Liquid | Water | |
| 14 | Steam | | |
| 15 | Non-Condensables | | |
| 16 | Fluid Vaporized or Condensed | | |
| 17 | Steam Condensed | | |
| 18 | Gravity - Liquid <u>11.75</u> | <u>562</u> API | |
| 19 | Density @ Ave. Cond. | lb per cu ft | |
| 20 | Viscosity | Centipoises at <u>165</u> °F | |
| 21 | Viscosity | Centipoises at °F | |
| 22 | Molecular Wt. | | |
| 23 | Conductivity | | |
| 24 | Specific Heat | Btu per lb | |
| 25 | Latent Heat - Vapors | Btu per lb | |
| 26 | Temperature In | °F | |
| 27 | Temperature Out | °F | |
| 28 | Operating Pressure | psig | |
| 29 | Number of Passes | | |
| 30 | Velocity | fps | |
| 31 | Pressure Drop | psi | |
| 32 | Heat Exchanged - Btu per hr <u>2,790,000</u> | MTD (Corrected) <u>40</u> | |
| 33 | Transfer Rate - Service <u>77</u> Clean | Fouling Resistance: Shell <u>.0005</u> Tubes <u>.008</u> | |
| 34 | CONSTRUCTION | | |
| 35 | Design Pressure | psig | |
| 36 | Test Pressure | psig | |
| 37 | Design Temp. (Max. Metal Temp.) | °F | |
| 38 | Tubes <u># Admiralty</u> No. <u>370</u> O.D. <u>5/8</u> BWG <u>14</u> Length <u>102</u> Pitch <u>1" Square</u> | | |
| 39 | Shell <u>25</u> I.D.; Mat'l <u>Steel</u> | | |
| 40 | Shell Cover <u>Steel</u> | | |
| 41 | Channel <u>Steel</u> | | |
| 42 | Tube Sheets—Stationary <u>Steel</u> | | |
| 43 | Baffles—Cross <u>Steel</u> Type <u>Seg.</u> Spacing <u>0"</u> Thickness <u>3/16"</u> Hole Dia. | | |
| 44 | Baffle—Long <u>Steel</u> Type <u>Flap</u> Thickness <u>1/4"</u> Impact Baffle <u>steel</u> | | |
| 45 | Tube Supports <u>Steel</u> No. <u>1</u> Thickness | | |
| 46 | Gaskets <u>As required</u> | | |
| 47 | Connections—Shell - In <u>6</u> Out <u>6</u> Series <u>30</u> Dome <u>None</u> | | |
| 48 | Channel - In <u>6</u> Out <u>6</u> Series <u>15</u> Tube Side <u>178"</u> | | |
| 49 | Corrosion Allowance—Shell Side | | |
| 50 | Weight <u>8800</u> Code <u>1952 ASME</u> Customer's Specifications <u>UOP 251, Rev. 7</u> Tema Class <u>R</u> | | |
| 51 | Length | Width | |
| 52 | Ref. Print | Inspection by whom <u>HSB</u> | |
| 53 | Based upon operating conditions as given, maximum temperature of tubes will not exceed °F | | |
| 54 | Note: Indicate after each part whether stress relieved (S.R.) and whether radiographed (X-R) | | |
| 55 | Remarks: | S. S. No. <u>LHW200VLR2CX</u> | |

* Wolverine Trufin.

APPENDIX B

GRISCOM-RUSSELL DESIGN BLUEPRINTS

APPENDIX C

SUMMARY OF TEST DATA ON THE HIGH-PRESSURE GAS COOLER

| Run No. | Date of Run | Cooling Water | | Hydrocarbon | | Manometer | | Manometer Fluid | Remarks | |
|-------------------------------|----------------|---------------|-----------------|-------------|----------|-----------|-------------|-----------------|------------------|---------------------------------------|
| | | Inlet °F | Intermediate °F | Outlet °F | Inlet °F | Outlet °F | Left Inches | | | Right Inches |
| 1 | Aug. 2, 1955 | 93.88 | 98.70 | 109.70 | 151.53 | 108.60 | +2.26 | -2.16 | Hg | |
| 2 | Aug. 2, 1955 | 94.46 | 98.92 | 109.65 | 154.73 | 109.90 | +2.20 | -2.10 | Hg | |
| 3 | Aug. 5, 1955 | 93.49 | 98.39 | 111.62 | 167.36 | 110.46 | +2.02 | -2.22 | Hg | |
| 4 | Aug. 8, 1955 | 84.18 | 88.60 | 110.91 | 149.86 | 101.52 | +0.78 | -3.69 | Hg | |
| 5 | Aug. 13, 1955 | 88.01 | 91.17 | 99.20 | 145.39 | 102.68 | +2.40 | -1.88 | Hg | |
| 6 | Aug. 17, 1955 | 90.82 | 96.52 | 107.82 | 127.18 | 110.24 | +1.73 | -1.23 | Hg | |
| 7 | Aug. 20, 1955 | 95.77 | 102.62 | 115.03 | 131.30 | 117.44 | +1.34 | -0.84 | Hg | |
| 8 | Sept. 7, 1955 | 82.57 | 93.06 | 103.48 | 132.46 | 112.66 | -2.75 | -3.75 | Hg | |
| 9 | Sept. 20, 1955 | 83.75 | 92.17 | 108.26 | 139.11 | 115.88 | -2.30 | -4.20 | Hg | |
| 10 | Oct. 14, 1955 | 75.15 | 80.88 | 91.46 | 105.75 | 106.19 | -1.06 | +0.93 | Hg | |
| 11 | Dec. 10, 1955 | 71.64 | — | 98.70 | 132.34 | 96.64 | +0.10 | -0.20 | Hg | Manometer readings questionable. |
| 12 | Dec. 17, 1955 | 70.89 | 82.88 | 101.56 | 140.40 | 105.65 | +0.55 | -2.45 | Hg | |
| 13 | Dec. 23, 1955 | 73.19 | 83.08 | 99.80 | 141.23 | 108.57 | +22.0 | -19.5 | Hg | Manometer reading in inches of water. |
| 14 | Dec. 29, 1955 | 73.37 | 87.90 | 109.80 | 163.77 | 109.03 | +1.35 | -1.71 | Hg | |
| 15 | Jan. 7, 1956 | 71.05 | 82.35 | 102.78 | 139.31 | 102.05 | +1.53 | -1.93 | Hg | |
| 16 | Jan. 13, 1956 | 69.78 | 81.62 | 102.55 | 144.83 | 100.13 | -1.15 | -4.13 | Hg | |
| 17 | Feb. 1, 1956 | 67.66 | 79.46 | 99.36 | 158.52 | 108.34 | -1.10 | -4.06 | Hg | |
| 18 | Mar. 3, 1956 | 78.32 | 90.50 | 111.48 | 167.70 | 122.94 | +1.40 | -1.40 | Hg | |
| 19 | April 11, 1956 | 82.80 | 100.00 | 122.20 | 150.40 | 124.40 | +0.57 | -0.88 | Hg | |
| 20 | May 26, 1956 | 70.59 | 90.83 | 116.66 | 137.38 | 113.43 | +13.31 | -12.63 | CCL ₄ | Manometer Temperature = 68°F. |
| Average (Omitting Run No. 11) | | 81.04 | 90.51 | 107.28 | 144.67 | 110.01 | | | | |

APPENDIX D

SUMMARY OF TEST DATA ON THE DEBUTANIZER OVERHEAD CONDENSER

DEBUTANIZER OVERHEAD CONDENSER

| Run No. | Date of Run | Cooling Water | | | | Hydrocarbon | | | | Manometer | | Manometer Fluid | | |
|---------|----------------|---------------|-------------|-------------|-----------------------|-------------|-----------|-----------|-----------|-----------|-----------------|-----------------|----------|-------------|
| | | Inlet °F | Outlet A °F | Outlet B °F | Outlet Mixed Mean, °F | Inlet °F | T.C. A mv | Ref. A °F | T.C. B mv | Ref. B °F | Outlet Tank, °F | | After °F | Left Inches |
| 1 | Aug. 2, 1955 | 94.03 | 109.95 | 103.69 | 107.20 | 150.90 | +0.023 | 95.33 | +0.027 | 96.33 | 100.50 | -3.03 | +3.07 | Hg |
| 2 | Aug. 5, 1955 | 93.78 | 111.95 | 104.79 | 107.48 | 155.62 | +0.041 | 90.80 | +0.039 | 90.80 | 96.11 | -4.28 | +4.28 | Hg |
| 3 | Aug. 8, 1955 | 84.53 | 97.89 | 94.01 | 96.53 | 154.45 | — | — | — | — | 87.10 | -4.61 | +4.61 | Hg* |
| 4 | Aug. 13, 1955 | 86.61 | 107.02 | 105.17 | 108.40 | 151.93 | — | — | — | — | 88.57 | -1.14 | +1.05 | Hg |
| 5 | Aug. 17, 1955 | 90.96 | 115.89 | 115.40 | 115.69 | 155.73 | +0.114 | 92.00 | +0.158 | 91.05 | 95.22 | -0.95 | +0.95 | Hg |
| 6 | Aug. 20, 1955 | 94.32 | 118.17 | 119.47 | 117.45 | 154.56 | +0.034 | 98.50 | +0.027 | 97.70 | 98.91 | -1.00 | +1.00 | Hg |
| 7 | Aug. 31, 1955 | 84.18 | 111.03 | 113.34 | 110.48 | 158.53 | +0.115 | 85.35 | +0.240 | 84.00 | 90.04 | -1.09 | +1.09 | Hg |
| 8 | Sept. 7, 1955 | 83.31 | 102.85 | 101.36 | 99.25 | 152.54 | — | — | — | — | 85.70 | -4.42 | +4.42 | Hg |
| 9 | Sept. 20, 1955 | 83.14 | 99.05 | 99.17 | 97.25 | 152.32 | — | — | — | — | 87.35 | -3.78 | +3.78 | Hg |
| 10 | Oct. 14, 1955 | 75.37 | 93.53 | 95.20 | 93.29 | 144.49 | +1.170 | 64.87 | +1.070 | 67.50 | 100.10 | -2.77 | +2.68 | Hg |
| 11 | Oct. 21, 1955 | 75.78 | 86.05 | 86.69 | 87.24 | 125.55 | +1.350 | 56.25 | +1.360 | 57.00 | 96.43 | -5.76 | -0.73 | Hg |
| Average | | 86.00 | 104.85 | 103.48 | 103.71 | 150.60 | | | | | 93.28 | | | |

* Manometer reading in inches of water gage.

APPENDIX E

SUMMARY OF TEST DATA FOR THE DEBUTANIZER BOTTOMS COOLER

The University of Michigan • Engineering Research Institute

| Run No. | Date of Run | Cooling Water | | Hydrocarbon | | Manometer | | Manometer Fluid | Remarks |
|--|----------------|---------------|-----------|-------------|-----------|-------------|--------------|------------------|---|
| | | Inlet °F | Outlet °F | Inlet °F | Outlet °F | Left Inches | Right Inches | | |
| 1 | Aug. 2, 1955 | 94.17 | 127.29 | 371.10 | 97.98 | +29.30 | -22.75 | H ₂ O | |
| 2 | Aug. 5, 1955 | 94.29 | 124.10 | 367.64 | 97.38 | +26.70 | -19.70 | H ₂ O | |
| 3 | Aug. 8, 1955 | 84.79 | 117.15 | 371.53 | 88.88 | +26.80 | -20.80 | H ₂ O | |
| 4 | Aug. 13, 1955 | 86.64 | 118.88 | 370.45 | 91.35 | + 3.51 | -2.85 | Hg | |
| 5 | Aug. 17, 1955 | 90.85 | 131.73 | 379.27 | 96.90 | +2.12 | -1.70 | Hg | |
| 6 | Aug. 20, 1955 | 93.98 | 131.40 | 365.40 | 99.38 | +2.07 | -1.57 | Hg | |
| 7 | Aug. 31, 1955 | 83.75 | 120.43 | 334.07 | 90.55 | +1.96 | -1.47 | Hg | |
| 8 | Sept. 7, 1955 | 82.94 | 123.33 | 328.11 | 93.62 | +1.65 | -1.18 | Hg | |
| 9 | Sept. 20, 1955 | 83.20 | 124.95 | 368.30 | 100.60 | +1.80 | -1.30 | Hg | |
| 10 | Sept. 20, 1955 | 83.35 | 124.19 | 359.03 | 100.74 | +1.80 | -1.30 | Hg | |
| 11 | Oct. 7, 1955 | 85.10 | 125.24 | 337.27 | 103.82 | -1.77 | -4.76 | Hg | |
| 12 | Oct. 14, 1955 | 76.13 | 107.21 | 356.00 | 89.37 | -1.99 | -4.65 | Hg | |
| 13 | Oct. 21, 1955 | 75.77 | 102.60 | 317.95 | 95.07 | -1.83 | -4.78 | Hg | |
| Average of Runs 1-13 | | 85.77 | 121.42 | 355.86 | 95.82 | | | | |
| Tubes Cleaned on the Inside Only During Plant Shutdown | | | | | | | | | |
| 14 | Nov. 18, 1955 | 75.96 | 120.64 | 277.00 | 93.01 | +1.17 | -1.42 | Hg | |
| 15 | Nov. 28, 1955 | 69.19 | 121.72 | 283.09 | 99.17 | +1.15 | -1.27 | Hg | |
| 16 | Dec. 10, 1955 | — | — | — | — | — | — | — | Void (Incorrect manometer reading). |
| 17 | Dec. 17, 1955 | 70.99 | 101.29 | 190.33 | 85.85 | +0.61 | -0.75 | Hg | |
| 18 | Dec. 23, 1955 | 73.02 | 100.72 | 188.00 | 83.70 | +10.00 | -9.00 | H ₂ O | |
| 19 | Dec. 23, 1955 | 73.50 | 91.68 | 189.80 | 79.76 | +22.20 | -21.00 | H ₂ O | |
| 20 | Dec. 23, 1955 | 73.42 | 104.24 | 186.40 | 82.92 | +6.00 | -5.00 | H ₂ O | |
| 21 | Dec. 29, 1955 | 73.54 | 106.46 | 190.20 | 91.92 | +0.49 | -0.78 | Hg | |
| 22 | Dec. 29, 1955 | 74.08 | 95.44 | 196.60 | 83.10 | +1.51 | -1.87 | Hg | |
| 23 | Dec. 29, 1955 | 73.88 | 124.80 | 196.40 | 94.28 | +0.12 | -0.39 | Hg | |
| 24 | Jan. 7, 1956 | 70.72 | 99.04 | 181.00 | 84.30 | +13.93 | -13.58 | CCl ₄ | Manometer temperature ~ 40°F. |
| 25 | Jan. 7, 1956 | 70.44 | 114.50 | 181.00 | 91.02 | +4.85 | -4.72 | CCl ₄ | Manometer temperature ~ 40°F. |
| 26 | Jan. 7, 1956 | 70.52 | 90.58 | 184.00 | 78.94 | +1.66 | -1.82 | Hg | |
| 27 | Jan. 13, 1956 | 70.37 | 86.68 | 177.40 | 75.14 | -1.43 | -3.83 | Hg | |
| 28 | Jan. 13, 1956 | 70.38 | 83.18 | 176.40 | 74.30 | -0.90 | -4.46 | Hg | |
| 29 | Jan. 13, 1956 | 70.40 | 88.89 | 177.14 | 75.70 | +17.85 | -17.36 | CCl ₄ | Manometer temperature ~ 40°F. |
| 30 | Jan. 13, 1956 | 70.46 | 94.12 | 177.20 | 77.02 | +11.34 | -11.01 | CCl ₄ | Manometer temperature ~ 40°F. |
| 31 | Jan. 13, 1956 | 70.50 | 108.86 | 176.00 | 82.24 | +4.09 | -3.79 | CCl ₄ | Manometer temperature ~ 40°F. |
| 32 | Feb. 1, 1956 | 67.36 | 84.96 | 177.60 | 78.78 | +11.00 | -10.92 | CCl ₄ | Manometer temperature ~ 40°F. |
| 33 | Feb. 1, 1956 | 66.88 | 89.02 | 174.60 | 78.16 | +6.05 | -5.76 | CCl ₄ | Manometer temperature ~ 40°F. |
| 34 | Feb. 1, 1956 | 66.61 | 113.56 | 176.43 | 92.18 | +2.13 | -2.04 | CCl ₄ | Manometer temperature ~ 40°F. |
| 35 | Feb. 1, 1956 | 67.38 | 83.72 | 190.40 | 82.12 | -1.35 | -3.80 | Hg | |
| 36 | Feb. 1, 1956 | 67.46 | 82.48 | 178.60 | 77.28 | -1.80 | -3.30 | Hg | |
| 37 | Feb. 7, 1956 | 72.18 | 84.68 | 185.80 | 79.12 | +1.43 | -1.34 | Hg | |
| 38 | Feb. 7, 1956 | 72.20 | 87.88 | 187.00 | 80.68 | +1.00 | -0.95 | Hg | |
| 39 | Feb. 7, 1956 | 73.08 | 89.54 | 185.80 | 81.16 | +16.20 | -15.52 | CCl ₄ | Manometer temperature ~ 45°F. |
| 40 | Feb. 7, 1956 | 73.00 | 95.44 | 186.80 | 83.68 | +9.66 | -9.06 | CCl ₄ | Manometer temperature ~ 45°F. |
| 41 | Feb. 7, 1956 | 72.78 | 114.90 | 187.20 | 91.42 | +3.10 | -2.50 | CCl ₄ | Manometer temperature ~ 45°F. |
| 42 | Feb. 7, 1956 | 72.53 | 119.48 | 184.50 | 92.98 | +2.26 | -1.60 | CCl ₄ | Manometer temperature ~ 45°F. |
| 43 | Feb. 18, 1956 | 72.12 | 89.70 | 197.80 | 84.88 | +1.86 | -1.09 | Hg | |
| 44 | Feb. 18, 1956 | 72.18 | 93.74 | 198.00 | 86.70 | +1.37 | -0.57 | Hg | |
| 45 | Feb. 18, 1956 | 72.44 | 95.00 | 197.60 | 85.94 | +15.57 | -15.59 | CCl ₄ | Manometer temperature ~ 40°F. |
| 46 | Feb. 18, 1956 | 72.20 | 104.02 | 199.20 | 90.94 | +9.12 | -9.02 | CCl ₄ | Manometer temperature ~ 40°F. |
| 47 | Feb. 18, 1956 | 72.16 | 111.10 | 198.60 | 91.34 | +4.21 | -4.21 | CCl ₄ | Manometer temperature ~ 40°F. |
| 48 | Feb. 18, 1956 | 72.02 | 124.79 | 194.30 | 98.47 | +1.93 | -2.00 | CCl ₄ | Manometer temperature ~ 40°F. |
| 49 | March 3, 1956 | 78.50 | 98.26 | 213.00 | 94.44 | -1.33 | +1.36 | Hg | |
| 50 | March 3, 1956 | 78.48 | 104.56 | 213.60 | 97.34 | -0.74 | -0.76 | Hg | |
| Tubes Partially Cleaned on Inside Only | | | | | | | | | |
| 51 | March 3, 1956 | 79.08 | 101.00 | 217.00 | 92.26 | +1.64 | -1.64 | Hg | |
| 52 | March 3, 1956 | 79.46 | 107.14 | 245.00 | 92.20 | +1.06 | -1.06 | Hg | |
| 53 | March 3, 1956 | 79.16 | 115.38 | 245.00 | 98.32 | +16.68 | -16.08 | CCl ₄ | Manometer temperature ~ 45°F. |
| 54 | March 3, 1956 | 79.04 | 128.20 | 215.20 | 103.18 | +5.60 | -5.01 | CCl ₄ | Manometer temperature ~ 45°F. |
| 55 | March 3, 1956 | 78.60 | 142.88 | 214.40 | 110.00 | +2.79 | -2.30 | CCl ₄ | Manometer temperature ~ 45°F. |
| 56 | March 10, 1956 | 70.60 | 92.60 | 220.00 | 85.74 | +1.35 | -1.15 | Hg | |
| 57 | March 10, 1956 | 72.08 | 93.86 | 220.60 | 88.86 | +1.60 | -1.35 | Hg | |
| 58 | March 10, 1956 | 72.70 | 101.32 | 223.00 | 91.44 | +16.75 | -17.96 | CCl ₄ | Manometer temperature ~ 50°F. |
| 59 | March 10, 1956 | 72.16 | 113.10 | 219.00 | 94.70 | -8.83 | -10.28 | CCl ₄ | Manometer temperature ~ 50°F. |
| 60 | March 10, 1956 | 72.01 | 144.90 | 220.50 | 101.62 | +3.84 | -3.31 | CCl ₄ | Manometer temperature ~ 50°F. - Bubble in manometer line. |
| 61 | March 10, 1956 | 72.04 | 134.40 | 219.20 | 105.74 | +2.60 | -4.15 | CCl ₄ | Manometer temperature ~ 50°F. |
| 62 | March 10, 1956 | 72.00 | 107.94 | 219.20 | 93.02 | +8.84 | -10.82 | CCl ₄ | Manometer temperature ~ 50°F. |
| 63 | April 21, 1956 | 84.00 | 110.20 | 254.00 | 96.20 | +1.30 | -1.60 | Hg | |
| 64 | April 21, 1956 | 82.00 | 111.00 | 254.20 | 98.20 | +1.31 | -1.55 | Hg | |
| 65 | April 21, 1956 | 81.80 | 123.20 | 253.40 | 102.00 | +14.70 | -13.92 | CCl ₄ | Manometer temperature ~ 70°F. |
| 66 | April 21, 1956 | 82.20 | 158.60 | 252.60 | 114.60 | +3.68 | -3.14 | CCl ₄ | Manometer temperature ~ 70°F. |
| 67 | May 26, 1956 | 73.98 | 92.48 | 215.40 | 85.36 | -4.40 | -10.62 | Hg | |
| 68 | May 26, 1956 | 74.00 | 97.38 | 215.00 | 87.88 | -5.50 | -9.50 | Hg | |
| 69 | May 26, 1956 | 74.76 | 105.58 | 216.60 | 91.32 | -6.45 | -8.55 | Hg | |
| 70 | May 26, 1956 | 76.50 | 118.78 | 217.00 | 98.94 | +11.42 | -11.08 | CCl ₄ | Manometer temperature ~ 75°F. |
| 71 | May 26, 1956 | 76.52 | 142.96 | 217.00 | 111.72 | +3.66 | -3.48 | CCl ₄ | Manometer temperature ~ 75°F. |
| Average of Runs 14-71 (Omitting Run No. 16) | | 73.47 | 106.01 | 205.86 | 89.99 | | | | |
| Average of Runs 1-71 (Omitting Run No. 16) | | 75.75 | 108.87 | 233.70 | 91.07 | | | | |

APPENDIX F

EXAMPLE CALCULATION OF TEST DATA ON THE HIGH-PRESSURE GAS COOLER

Date of Run: May 26, 1956

Run No. 20

From the data,

$$\begin{aligned} (-\Delta P'') &= 13.31 - (-12.63) \\ &= 25.94 \text{ inches of CCl}_4 \end{aligned}$$

Manometer temperature = 68°F.

Substituting in Equation 9b,

$$\begin{aligned} (-\Delta P') &= \frac{(-\Delta P'')}{21} \\ &= \frac{25.94}{21} \\ &= 1.235 \text{ inches Hg.} \end{aligned}$$

From Equation 8,

$$\begin{aligned} W &= 178,000 \sqrt{(-\Delta P')} \\ &= 178,000 \sqrt{1.235} \\ &= 198,000 \text{ lb/hr.} \end{aligned}$$

From the data; the water temperatures were

$$\begin{aligned} t_{\text{H}_2\text{O out}} &= 116.66^\circ\text{F}, \\ t_{\text{H}_2\text{O in}} &= 70.59^\circ\text{F}, \\ \therefore \Delta t_{\text{H}_2\text{O}} &= 46.07^\circ\text{F} \end{aligned}$$

$$\text{and } t_w = 70.59 + \frac{46.07}{2} = 93.62^\circ\text{F.}$$

Substituting in Equation 4,

$$\begin{aligned} Q &= W C_p \Delta t_{\text{H}_2\text{O}} \\ &= (198,000)(1)(46.07) \\ &= 9.11 \times 10^6 \text{ Btu/hr.} \end{aligned}$$

From Equation 11,

$$\begin{aligned} V_t &= \frac{W}{103,000} \\ &= \frac{198,000}{103,000} \\ &= 1.92 \text{ ft/sec.} \end{aligned}$$

From the data, the hydrocarbon temperatures were

$$\begin{aligned} t_{\text{hydrocarbon in}} &= 137.38^\circ\text{F} \\ t_{\text{hydrocarbon out}} &= \underline{113.43^\circ\text{F}} \\ \therefore \Delta t_{\text{hydrocarbon}} &= 23.95^\circ\text{F.} \end{aligned}$$

Computing ΔT_{LM} ,

$$\begin{aligned} \Delta T_1 &= 137.38 - 116.66 = 20.72^\circ\text{F} \\ \Delta T_2 &= 113.43 - 70.59 = 42.84^\circ\text{F} \\ \Delta T_{LM} &= \frac{\Delta T_2 - \Delta T_1}{\ln \frac{\Delta T_2}{\Delta T_1}} \\ &= \frac{42.84 - 20.72}{\ln \frac{42.84}{20.72}} \\ &= 30.50^\circ\text{F.} \end{aligned}$$

Rearranging Equation 1,

$$U_o = \frac{Q}{A_o \Delta T_{LM}}$$

The outside heat transfer area for this unit is (from Table VII)

$$A_o = 9385 \text{ ft}^2.$$

Solving for the overall heat transfer coefficient,

$$\begin{aligned} U_o &= \frac{9.11 \times 10^6}{(30.50)(9385)} \\ &= 31.8 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F.} \end{aligned}$$

Then,

$$\begin{aligned} \frac{1}{U_o} &= \frac{1}{31.8} \\ &= 0.0314 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu.} \end{aligned}$$

From Equation 5,

$$\begin{aligned} h_i &= \frac{150[1 + 0.011(t_w)] (V_t)^{0.8}}{(d_i)^{0.2}} \\ &= \frac{150[1 + 0.011(93.62)] (1.92)^{0.8}}{(0.51)^{0.2}} \\ &= 586 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \end{aligned}$$

Then,

$$\begin{aligned} \frac{A_o}{A_i h_i} &= \frac{3.57}{586} \\ &= 0.00610 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \end{aligned}$$

Substituting in Equation 17c,

$$\begin{aligned} \left[\frac{A_o r_i + r_o'}{A_i} \right] &= \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + 0.0382 \right] \\ &= 0.0314 - [0.0061 + 0.0382] \\ &= -0.0129 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \end{aligned}$$

APPENDIX G

EXAMPLE CALCULATION OF TEST DATA ON THE
DEBUTANIZER OVERHEAD CONDENSER

Date of Run: October 14, 1955

Run No. 10

From the data,

$$(-\Delta P') = 5.45 \text{ inches of Hg.}$$

Substituting in Equation 12,

$$\begin{aligned} W &= 288,000 \sqrt{-\Delta P'} \\ &= 288,000 \sqrt{5.45} \\ &= 672,000 \text{ lb/hr} \end{aligned}$$

From the data, the water temperatures were

$$\begin{aligned} t_{\text{H}_2\text{O out}} &= 93.29^\circ\text{F} \\ t_{\text{H}_2\text{O in}} &= \underline{75.37^\circ\text{F}} \\ \therefore \Delta t_{\text{H}_2\text{O}} &= 17.92^\circ\text{F} \end{aligned}$$

Substituting in Equation 4,

$$\begin{aligned} Q &= W C_p \Delta t_{\text{H}_2\text{O}} \\ &= (672,000)(1)(17.92) \\ &= 12.05 \times 10^6 \text{ Btu/hr} \end{aligned}$$

From Equation 13,

$$\begin{aligned} V_t &= \frac{W}{223,000} \\ &= \frac{672,000}{223,000} \\ &= 3.01 \text{ ft/sec} \end{aligned}$$

From the data, the hydrocarbon temperatures were

$$\begin{aligned} t_{\text{hydrocarbon in}} &= 144.49^{\circ}\text{F} \\ t_{\text{hydrocarbon out (after tank)}} &= 100.10^{\circ}\text{F} \\ \therefore \Delta t_{\text{hydrocarbon}} &= 44.39^{\circ}\text{F} \end{aligned} .$$

Computing ΔT_{LM} ,

$$\begin{aligned} \Delta T_1 &= 144.49 - 93.29 = 51.20^{\circ}\text{F} \\ \Delta T_2 &= 100.10 - 75.37 = 24.73^{\circ}\text{F} \end{aligned}$$

$$\begin{aligned} \Delta T_{\text{LM}} &= \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \\ &= \frac{51.20 - 24.73}{\ln \frac{51.20}{24.73}} \\ &= 36.45^{\circ}\text{F} \end{aligned} .$$

The outside heat transfer area for this unit is (from Table VII)

$$A_o = 9570 \text{ ft}^2 .$$

Solving for the overall heat transfer coefficient,

$$\begin{aligned} U_o &= \frac{Q}{A_o \Delta T_{\text{LM}}} \\ &= \frac{12.05 \times 10^6}{(9570)(36.45)} \\ &= 34.5 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F} \end{aligned} .$$

APPENDIX H

EXAMPLE CALCULATION OF TEST DATA ON THE DEBUTANIZER BOTTOMS COOLER

Date of Run: January 13, 1956

Run No. 27

From the data,

$$(-\Delta P') = 2.40 \text{ inches of Hg} \quad .$$

Substituting in Equation 14,

$$\begin{aligned} W &= 144,000 \sqrt{-\Delta P'} \\ &= 144,000 \sqrt{2.40} \\ &= 223,500 \text{ lb/hr} \quad . \end{aligned}$$

From the data, the water temperatures were

$$t_{\text{H}_2\text{O out}} = 86.68^\circ\text{F}$$

$$t_{\text{H}_2\text{O in}} = \underline{70.37^\circ\text{F}}$$

$$\therefore \Delta t_{\text{H}_2\text{O}} = 16.31^\circ\text{F}$$

$$\text{and } t_w = 70.37 + \frac{16.31}{2} = 78.52^\circ\text{F}.$$

Substituting in Equation 4,

$$\begin{aligned} Q &= W C_p \Delta t_{\text{H}_2\text{O}} \\ &= (223,500)(1)(16.31) \\ &= 3.64 \times 10^6 \text{ Btu/hr} \quad . \end{aligned}$$

From Equation 15,

$$\begin{aligned} V_t &= \frac{W}{58,500} \\ &= \frac{223,500}{58,500} \\ &= 3.82 \text{ ft/sec} \quad . \end{aligned}$$

From the data, the hydrocarbon temperatures were

$$\begin{aligned} t_{\text{hydrocarbon in}} &= 177.40^{\circ}\text{F} \\ t_{\text{hydrocarbon out}} &= \underline{75.14^{\circ}\text{F}} \\ \therefore \Delta t_{\text{hydrocarbon}} &= 102.26^{\circ}\text{F} \quad . \end{aligned}$$

Computing ΔT_{LM} ,

$$\begin{aligned} \Delta T_1 &= 177.40 - 86.68 = 90.72^{\circ}\text{F} \\ \Delta T_2 &= 75.14 - 70.37 = 4.77^{\circ}\text{F} \end{aligned}$$

$$\begin{aligned} \Delta T_{LM} &= \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \\ &= \frac{90.72 - 4.77}{\ln \frac{90.72}{4.77}} \\ &= 29.30^{\circ}\text{F} \quad . \end{aligned}$$

The outside heat transfer area for this unit is (from Table VII)

$$A_o = 2505 \text{ ft}^2 \quad .$$

Solving for the overall heat transfer coefficient,

$$\begin{aligned} U_o &= \frac{Q}{A_o \Delta T_{LM}} \\ &= \frac{3.64 \times 10^6}{(2505)(29.30)} \\ &= 49.6 \text{ Btu/hr-ft}^2\text{-}^{\circ}\text{F} \quad . \end{aligned}$$

Then,

$$\begin{aligned} \frac{1}{U_o} &= \frac{1}{49.6} \\ &= 0.02015 \text{ hr-ft}^2\text{-}^{\circ}\text{F/Btu} \quad . \end{aligned}$$

From Equation 5,

$$\begin{aligned}
 h_i &= \frac{(150)[1 + 0.011(t_w)] (V_t)^{0.8}}{(d_i)^{0.2}} \\
 &= \frac{(150)[1 + 0.011(78.52)] (3.82)^{0.8}}{(0.51)^{0.2}} \\
 &= 937 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} .
 \end{aligned}$$

Then,

$$\begin{aligned}
 \frac{A_o}{A_i h_i} &= \frac{3.28}{937} \\
 &= 0.00350 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} .
 \end{aligned}$$

Substituting in Equation 18,

$$\begin{aligned}
 h_o' &= 0.0886 \left[\frac{Q}{\Delta t_{\text{hydrocarbon}}} \right]^{0.65} \\
 &= 0.0886 \left[\frac{3.64 \times 10^6}{102.26} \right]^{0.65} \\
 &= 80.3 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} .
 \end{aligned}$$

Then,

$$\begin{aligned}
 \frac{1}{h_o'} &= \frac{1}{80.3} \\
 &= 0.01248 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} .
 \end{aligned}$$

Substituting in Equation 17d,

$$\begin{aligned}
 \left[\frac{A_o}{A_i} r_i + r_o' \right] &= \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + \frac{1}{h_o'} + 0.00038 \right] \\
 &= 0.02018 - [0.00350 + 0.01245 + 0.00038] \\
 &= 0.00379 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} .
 \end{aligned}$$

APPENDIX J

SUMMARY OF CALCULATED RESULTS FOR THE HIGH-PRESSURE GAS COOLER

| Run No. | Date of Run | Days of Operation | ΔT_{H_2O} °F | $\Delta T_{hydro-carbon}$ °F | ΔT_{LM} °F | $Q \times 10^{-6}$ Btu/hr | V_t ft/sec | U_o Btu/hr-ft ² -°F | $A_o/A_i h_i$ hr-ft ² -°F/Btu | $\frac{A_o x_i + T_o'}{A_i}$ hr-ft ² -°F/Btu | Remarks |
|--|----------------|-------------------|-------------------------|---------------------------------|-----------------------|------------------------------|-----------------|-------------------------------------|---|--|--|
| 1 | Aug. 2, 1955 | 4 | 15.82 | 42.93 | 26.15 | 5.93 | 3.64 | 25.8 | .00319 | 0 | |
| 2 | Aug. 2, 1955 | 4 | 15.19 | 44.85 | 27.70 | 5.60 | 3.57 | 22.9 | .00321 | 0 | |
| 3 | Aug. 5, 1955 | 7 | 18.13 | 56.90 | 32.55 | 6.65 | 3.55 | 23.2 | .00323 | +0.0020 | |
| 4 | Aug. 8, 1955 | 10 | 26.73 | 48.36 | 26.85 | 10.05 | 3.65 | 42.6 | .00328 | -0.0177 | |
| 5 | Aug. 13, 1955 | 15 | 11.19 | 42.71 | 27.41 | 4.12 | 3.58 | 17.1 | .00340 | +0.0172 | |
| 6 | Aug. 17, 1955 | 19 | 27.00 | 16.94 | 19.89 | 8.29 | 2.98 | 47.4 | .00370 | -0.0205 | |
| 7 | Aug. 20, 1955 | 22 | 19.26 | 13.86 | 18.78 | 5.06 | 2.54 | 30.6 | .00421 | -0.0094 | |
| 8 | Sept. 7, 1955 | 40 | 24.91 | 19.80 | 27.50 | 4.44 | 1.73 | 18.3 | .00600 | +0.0107 | |
| 9 | Sept. 20, 1955 | 53 | 24.52 | 23.10 | 31.45 | 6.02 | 2.38 | 21.8 | .00465 | +0.0033 | |
| 10 | Oct. 14, 1955 | 77 | 16.31 | -0.47 | 21.60 | 4.10 | 2.44 | 21.6 | .00489 | +0.0035 | |
| Average number of days after cleaning of tubes | | | | | | | | | | | |
| 11 | Dec. 10, 1955 | 18 | 27.06 | 35.70 | 29.00 | 2.64 | 0.95 | 10.0 | .01142 | +0.0504 | Run No. 11 not plotted or included in averages due to questionable data. |
| 12 | Dec. 17, 1955 | 25 | 30.67 | 34.75 | 36.40 | 9.45 | 2.99 | 27.7 | .00449 | -0.0066 | |
| 13 | Dec. 23, 1955 | 31 | 26.61 | 32.66 | 38.50 | 8.26 | 3.02 | 22.8 | .00438 | +0.0013 | |
| 14 | Dec. 29, 1955 | 37 | 36.43 | 54.74 | 43.55 | 11.35 | 3.02 | 27.8 | .00425 | -0.0065 | |
| 15 | Jan. 7, 1956 | 46 | 31.73 | 37.26 | 33.45 | 10.51 | 3.19 | 33.5 | .00422 | -0.0126 | |
| 16 | Jan. 13, 1956 | 52 | 32.77 | 44.70 | 35.85 | 10.05 | 2.98 | 29.9 | .00446 | -0.0093 | |
| 17 | Feb. 1, 1956 | 71 | 31.70 | 50.18 | 49.70 | 9.70 | 2.97 | 20.8 | .00455 | +0.0052 | |
| 18 | Mar. 3, 1956 | 102 | 33.16 | 44.76 | 50.20 | 9.57 | 2.89 | 20.3 | .00437 | +0.0067 | |
| 19 | April 21, 1956 | 151 | 39.4 | 26.0 | 34.30 | 8.43 | 2.08 | 26.2 | .00541 | -0.0054 | |
| 20 | May 26, 1956 | 186 | 46.07 | 23.95 | 30.50 | 9.11 | 1.92 | 31.8 | .00610 | -0.0129 | |
| Average: (Omitting Run No. 11) | | | | | | | | | | | |
| | | | 26.24 | 34.66 | 32.23 | 7.72 | 2.90 | 27.0 | | | |

APPENDIX K

SUMMARY OF CALCULATED RESULTS FOR THE DEBUTANIZER OVERHEAD CONDENSER

| Run No. | Date of Run | Days of Operation | Δt_{H_2O} °F | $\Delta t_{hydro-carbon}$ °F | ΔT_{LM} °F | $Q \times 10^{-6}$ Btu/hr | V_t ft/sec | U_0 Btu/hr-ft ² -°F |
|---------|----------------|-------------------|-------------------------|---------------------------------|-----------------------|------------------------------|-----------------|-------------------------------------|
| 1 | Aug. 2, 1955 | 4 | 13.17 | 50.43 | 19.50 | 9.37 | 3.19 | 50.2 |
| 2 | Aug. 5, 1955 | 7 | 13.70 | 59.51 | 15.11 | 11.57 | 3.62 | 79.8 |
| 3 | Aug. 8, 1955 | 10 | 12.00 | 67.35 | 17.79 | 2.97 | 1.11 | 17.4 |
| 4 | Aug. 13, 1955 | 15 | 21.79 | 63.36 | 13.45 | 9.28 | 1.91 | 72.1 |
| 5 | Aug. 17, 1955 | 19 | 24.73 | 60.51 | 15.96 | 9.81 | 1.78 | 64.2 |
| 6 | Aug. 20, 1955 | 22 | 23.13 | 54.70 | 16.60 | 9.43 | 1.83 | 59.4 |
| 7 | Aug. 31, 1955 | 33 | 26.85 | 68.49 | 19.95 | 11.42 | 1.91 | 59.9 |
| 8 | Sept. 7, 1955 | 40 | 15.94 | 66.84 | 16.40 | 13.63 | 3.84 | 86.9 |
| 9 | Sept. 20, 1955 | 53 | 14.11 | 65.97 | 20.10 | 11.20 | 3.56 | 58.3 |
| 10 | Oct. 14, 1955 | 77 | 17.92 | 44.39 | 36.45 | 12.05 | 3.01 | 34.5 |
| 11 | Oct. 21, 1955 | 84 | 11.46 | 29.12 | 28.60 | 7.40 | 2.90 | 27.0 |
| Average | | | 17.71 | 57.32 | 19.99 | 9.83 | 2.61 | 55.4 |

APPENDIX L

SUMMARY OF CALCULATED RESULTS FOR THE DEBUTANIZER BOTTOMS COOLER

The University of Michigan • Engineering Research Institute

| Run No. | Date of Run | Δt_{H_2O} °F | $\Delta t_{hydro-carbon}$ °F | ΔT_{LM} °F | V_t ft/sec | $Q \times 10^{-6}$ Btu/hr | U_o Btu/hr-ft ² -°F | h_o' Btu/hr-ft ² -°F | $A_o/A_i h_i$ hr-ft ² -°F/Btu | $\frac{10^8}{U_o} - \frac{10^8}{h_o'}$ hr-ft ² -°F/Btu | $\frac{A_o}{A_i} + r_o'$ hr-ft ² -°F/Btu | Days of Operation |
|---|----------------|-------------------------|---------------------------------|-----------------------|-----------------|------------------------------|-------------------------------------|--------------------------------------|---|--|--|-------------------|
| 1 | Aug. 2, 1955 | 33.12 | 273.12 | 57.6 | 4.83 | 9.37 | 64.8 | 78.5 | 0.00244 | 2.82 | 0.00000 | 4 |
| 2 | Aug. 5, 1955 | 29.81 | 270.26 | 55.3 | 4.56 | 7.96 | 57.5 | 71.1 | 0.00265 | 3.33 | +0.00030 | 7 |
| 3 | Aug. 8, 1955 | 32.36 | 282.65 | 60.6 | 4.62 | 8.75 | 57.5 | 73.4 | 0.00267 | 3.77 | +0.00072 | 10 |
| 4 | Aug. 13, 1955 | 32.24 | 279.12 | 62.0 | 6.24 | 11.75 | 75.7 | 89.8 | 0.00210 | 2.06 | -0.00042 | 15 |
| 5 | Aug. 17, 1955 | 40.88 | 282.37 | 65.0 | 4.83 | 11.54 | 70.8 | 88.0 | 0.00246 | 2.74 | -0.00010 | 19 |
| 6 | Aug. 20, 1955 | 37.42 | 266.02 | 60.6 | 4.71 | 10.32 | 68.1 | 85.0 | 0.00251 | 2.91 | +0.00002 | 22 |
| 7 | Aug. 31, 1955 | 36.68 | 243.52 | 60.0 | 4.58 | 9.82 | 65.4 | 87.4 | 0.00270 | 3.83 | +0.00075 | 33 |
| 8 | Sept. 7, 1955 | 40.39 | 234.49 | 65.9 | 4.16 | 9.82 | 59.5 | 89.4 | 0.00291 | 5.61 | +0.00232 | 40 |
| 9 | Sept. 20, 1955 | 41.75 | 267.70 | 85.9 | 4.35 | 10.62 | 49.4 | 86.4 | 0.00275 | 8.66 | +0.00553 | 53 |
| 10 | Sept. 20, 1955 | 40.84 | 258.29 | 83.8 | 4.35 | 10.38 | 49.5 | 87.0 | 0.00275 | 8.70 | +0.00557 | 53 |
| 11 | Oct. 7, 1955 | 40.14 | 233.45 | 79.8 | 4.26 | 10.00 | 49.9 | 90.8 | 0.00278 | 9.03 | +0.00587 | 70 |
| 12 | Oct. 14, 1955 | 31.08 | 266.63 | 80.5 | 4.01 | 6.59 | 32.6 | 65.4 | 0.00313 | 14.90 | +0.01139 | 77 |
| 13 | Oct. 21, 1955 | 26.83 | 222.88 | 81.8 | 4.24 | 6.66 | 32.4 | 71.8 | 0.00304 | 16.92 | +0.01350 | 84 |
| Average of Runs 1-13 | | | | | | | | | | | | |
| 35.65 260.04 69.1 4.60 9.51 56.4 81.7 0.00268 | | | | | | | | | | | | |
| Unit Cleaned on Tube-Side Only | | | | | | | | | | | | |
| Days After Cleaning | | | | | | | | | | | | |
| 14 | Nov. 18, 1955 | 44.68 | 183.99 | 63.1 | 3.97 | 10.41 | 65.8 | 108.8 | 0.00305 | 6.00 | +0.00257 | 10 |
| 15 | Nov. 28, 1955 | 52.53 | 183.92 | 78.2 | 3.84 | 11.80 | 60.2 | 118.1 | 0.00318 | 8.14 | +0.00458 | 20 |
| 16 | Void | | | | | | | | | | | |
| 17 | Dec. 17, 1955 | 30.30 | 104.48 | 41.5 | 2.88 | 5.11 | 49.1 | 98.6 | 0.00420 | 10.22 | +0.00564 | 39 |
| 18 | Dec. 23, 1955 | 27.70 | 104.30 | 36.5 | 2.92 | 4.74 | 51.8 | 94.1 | 0.00415 | 8.68 | +0.00415 | 45 |
| 19 | Dec. 23, 1955 | 18.18 | 110.04 | 33.5 | 4.41 | 4.68 | 55.9 | 90.4 | 0.00306 | 6.82 | +0.00338 | 45 |
| 20 | Dec. 23, 1955 | 30.82 | 103.48 | 33.7 | 2.22 | 4.01 | 47.5 | 85.0 | 0.00510 | 9.27 | +0.00379 | 45 |
| 21 | Dec. 29, 1955 | 32.92 | 104.28 | 45.0 | 2.78 | 5.36 | 47.4 | 101.8 | 0.00420 | 11.26 | +0.00668 | 51 |
| 22 | Dec. 29, 1955 | 21.36 | 113.50 | 38.1 | 4.55 | 4.55 | 59.3 | 100.4 | 0.00294 | 6.92 | +0.00360 | 51 |
| 23 | Dec. 29, 1955 | 50.92 | 102.12 | 40.7 | 1.76 | 5.24 | 51.3 | 101.8 | 0.00577 | 9.66 | +0.00351 | 51 |
| 24 | Jan. 7, 1956 | 28.32 | 96.70 | 38.0 | 2.90 | 4.80 | 50.3 | 99.3 | 0.00420 | 9.87 | +0.00529 | 60 |
| 25 | Jan. 7, 1956 | 44.06 | 89.93 | 39.1 | 1.71 | 4.40 | 44.9 | 98.5 | 0.00624 | 12.13 | +0.00551 | 60 |
| 26 | Jan. 7, 1956 | 20.06 | 105.06 | 35.3 | 4.61 | 5.42 | 61.1 | 102.2 | 0.00307 | 6.57 | +0.00312 | 60 |
| 27 | Jan. 13, 1956 | 16.31 | 102.26 | 29.3 | 3.82 | 3.64 | 49.6 | 80.3 | 0.00350 | 7.67 | +0.00379 | 66 |
| 28 | Jan. 13, 1956 | 12.80 | 102.10 | 28.2 | 4.65 | 3.49 | 49.3 | 78.3 | 0.00303 | 7.52 | +0.00411 | 66 |
| 29 | Jan. 13, 1956 | 18.49 | 101.44 | 29.5 | 3.27 | 3.54 | 47.3 | 79.3 | 0.00392 | 8.29 | +0.00399 | 66 |
| 30 | Jan. 13, 1956 | 23.66 | 100.18 | 30.1 | 2.61 | 3.61 | 47.8 | 81.0 | 0.00465 | 8.35 | +0.00352 | 66 |
| 31 | Jan. 13, 1956 | 38.36 | 93.76 | 31.7 | 1.55 | 3.48 | 43.3 | 82.5 | 0.00675 | 10.78 | +0.00365 | 66 |
| 32 | Feb. 1, 1956 | 16.34 | 108.28 | 46.5 | 3.87 | 3.69 | 31.7 | 78.1 | 0.00353 | 18.75 | +0.01484 | 85 |
| 33 | Feb. 1, 1956 | 15.02 | 101.32 | 37.8 | 3.02 | 2.66 | 23.0 | 65.9 | 0.00432 | 20.51 | +0.01581 | 85 |
| 34 | Feb. 1, 1956 | 17.60 | 98.82 | 38.9 | 2.59 | 2.66 | 27.2 | 66.9 | 0.00490 | 21.30 | +0.01652 | 85 |
| 35 | Feb. 1, 1956 | 22.14 | 96.44 | 36.8 | 1.90 | 2.46 | 26.6 | 64.6 | 0.00620 | 22.12 | +0.01554 | 85 |
| 36 | Feb. 1, 1956 | 46.95 | 84.25 | 41.5 | 1.12 | 3.10 | 29.7 | 82.1 | 0.00876 | 21.47 | +0.01233 | 85 |
| 37 | Feb. 7, 1956 | 12.50 | 106.68 | 35.3 | 4.12 | 3.01 | 44.1 | 69.0 | 0.00333 | 14.85 | +0.01114 | 91 |
| 38 | Feb. 7, 1956 | 15.68 | 106.32 | 36.9 | 3.44 | 3.17 | 34.3 | 71.6 | 0.00379 | 15.17 | +0.01100 | 91 |
| 39 | Feb. 7, 1956 | 16.46 | 104.64 | 35.7 | 3.11 | 3.00 | 33.5 | 69.8 | 0.00407 | 15.03 | +0.01058 | 91 |
| 40 | Feb. 7, 1956 | 22.44 | 103.12 | 37.6 | 2.39 | 3.15 | 33.4 | 72.7 | 0.00455 | 16.19 | +0.01086 | 91 |
| 41 | Feb. 7, 1956 | 42.12 | 95.78 | 39.6 | 1.31 | 3.22 | 32.5 | 77.6 | 0.00755 | 17.91 | +0.00998 | 91 |
| 42 | Feb. 7, 1956 | 46.95 | 91.52 | 38.6 | 1.08 | 2.97 | 30.7 | 76.0 | 0.00871 | 19.38 | +0.01029 | 91 |
| 43 | Feb. 18, 1956 | 17.58 | 112.92 | 44.7 | 4.21 | 4.33 | 38.6 | 84.5 | 0.00719 | 14.07 | +0.01050 | 102 |
| 44 | Feb. 18, 1956 | 21.56 | 111.30 | 45.5 | 3.41 | 4.29 | 37.7 | 84.3 | 0.00376 | 14.75 | +0.01058 | 102 |
| 45 | Feb. 18, 1956 | 22.56 | 111.66 | 44.0 | 3.08 | 4.06 | 36.3 | 81.5 | 0.00403 | 14.92 | +0.01051 | 102 |
| 46 | Feb. 18, 1956 | 31.82 | 108.26 | 46.9 | 2.24 | 4.17 | 35.4 | 84.0 | 0.00510 | 16.43 | +0.01095 | 102 |
| 47 | Feb. 18, 1956 | 38.94 | 107.26 | 44.9 | 1.60 | 3.64 | 32.4 | 77.3 | 0.00650 | 17.99 | +0.01111 | 102 |
| 48 | Feb. 18, 1956 | 52.77 | 95.83 | 42.5 | 1.09 | 3.38 | 31.7 | 79.3 | 0.00355 | 19.03 | +0.01010 | 102 |
| 49 | March 3, 1956 | 19.76 | 118.56 | 50.2 | 4.05 | 4.68 | 37.2 | 85.9 | 0.00319 | 15.22 | +0.01165 | 116 |
| 50 | March 3, 1956 | 26.08 | 116.26 | 51.4 | 3.02 | 4.61 | 35.8 | 86.2 | 0.00392 | 16.35 | +0.01205 | 116 |
| Tubes Partially Cleaned on Inside Only | | | | | | | | | | | | |
| 51 | March 3, 1956 | 21.92 | 124.74 | 47.2 | 4.46 | 5.72 | 48.4 | 95.0 | 0.00290 | 10.17 | +0.00689 | 116 |
| 52 | March 3, 1956 | 27.68 | 152.80 | 52.8 | 3.60 | 5.82 | 44.1 | 83.6 | 0.00338 | 10.68 | +0.00682 | 116 |
| 53 | March 3, 1956 | 36.22 | 146.68 | 58.0 | 3.16 | 6.70 | 46.2 | 94.7 | 0.00369 | 11.03 | +0.00696 | 116 |
| 54 | March 3, 1956 | 49.16 | 112.02 | 48.9 | 1.80 | 5.16 | 42.2 | 95.4 | 0.00556 | 13.20 | +0.00726 | 116 |
| 55 | March 3, 1956 | 64.28 | 104.40 | 48.8 | 1.24 | 4.66 | 38.0 | 93.1 | 0.00719 | 15.58 | +0.00801 | 116 |
| 56 | March 10, 1956 | 22.00 | 134.26 | 52.8 | 3.90 | 5.02 | 37.9 | 83.0 | 0.00339 | 14.35 | +0.01058 | 123 |
| 57 | March 10, 1956 | 21.78 | 131.74 | 52.2 | 4.23 | 5.41 | 41.3 | 88.3 | 0.00318 | 12.88 | +0.00932 | 123 |
| 58 | March 10, 1956 | 28.62 | 131.55 | 55.0 | 3.25 | 5.44 | 39.5 | 85.7 | 0.00381 | 13.67 | +0.00948 | 123 |
| 59 | March 10, 1956 | 40.94 | 124.30 | 53.6 | 2.43 | 5.77 | 42.9 | 95.5 | 0.00466 | 12.87 | +0.00783 | 123 |
| 60 | March 10, 1956 | 72.89 | 118.88 | 49.2 | 1.67 | 7.11 | 57.5 | 112.2 | 0.00580 | 8.49 | +0.00231 | 123 |
| 61 | March 10, 1956 | 62.36 | 113.46 | 55.5 | 1.43 | 5.23 | 37.6 | 95.4 | 0.00670 | 16.11 | +0.00903 | 123 |
| 62 | March 10, 1956 | 35.94 | 126.18 | 54.2 | 2.45 | 5.15 | 37.9 | 87.8 | 0.00468 | 15.00 | +0.00994 | 123 |
| 63 | April 21, 1956 | 26.20 | 157.80 | 55.5 | 4.21 | 6.45 | 46.4 | 87.9 | 0.00294 | 10.22 | +0.00690 | 164 |
| 64 | April 21, 1956 | 31.00 | 155.00 | 60.1 | 4.17 | 7.57 | 50.1 | 98.6 | 0.00294 | 9.81 | +0.00649 | 164 |
| 65 | April 21, 1956 | 43.40 | 150.40 | 60.8 | 2.89 | 7.32 | 48.0 | 98.6 | 0.00383 | 10.71 | +0.00650 | 164 |
| 66 | April 21, 1956 | 78.40 | 137.00 | 59.1 | 1.41 | 6.43 | 43.3 | 96.2 | 0.00629 | 12.70 | +0.00603 | 164 |
| 67 | May 26, 1956 | 18.50 | 130.04 | 47.0 | 6.15 | 6.66 | 56.5 | 101.9 | 0.00234 | 7.48 | +0.00476 | 199 |
| 68 | May 26, 1956 | 23.38 | 127.12 | 48.5 | 4.94 | 6.75 | 55.5 | 104.1 | 0.00271 | 8.42 | +0.00533 | 199 |
| 69 | May 26, 1956 | 30.82 | 125.28 | 49.6 | 3.57 | 6.45 | 51.8 | 102.1 | 0.00346 | 9.51 | +0.00567 | 199 |
| 70 | May 26, 1956 | 42.28 | 118.06 | 51.4 | 2.56 | 6.32 | 49.1 | 104.8 | 0.00433 | 10.80 | +0.00609 | 199 |
| 71 | May 26, 1956 | 66.44 | 105.28 | 51.9 | 1.44 | 5.58 | 43.0 | 103.9 | 0.00650 | 13.61 | +0.00673 | 199 |
| Average of Runs 14-71 (Omitting Run No. 16) | | | | | | | | | | | | |
| 32.65 115.93 45.17 2.99 4.93 43.24 88.90 0.00455 | | | | | | | | | | | | |
| Average of Runs 1-71 (Omitting Run No. 16) | | | | | | | | | | | | |
| 33.12 142.63 49.62 3.29 5.78 45.65 87.56 0.00421 | | | | | | | | | | | | |

APPENDIX M

DEBUTANIZER OVERHEAD CONDENSER—
COMPUTATION OF THE CONDENSING RANGE OF THE HYDROCARBON STREAM

Assume that the hydrocarbon stream consists of 50 mol% n-butane and 50 mol% propane. The average molecular weight is then

$$\begin{array}{rcl} \text{n-Butane } 0.50 \times 58 & = & 29 \\ \text{Propane } 0.50 \times 44 & = & \underline{22} \\ & & 51 \end{array}$$

This checks closely with the specified value of 50.8, indicating that the assumption is reasonable.

To check the assumption further, the overall heat duty, assuming the above mixture, may be computed. The heat duty may be divided into the latent heat of condensation at 110°F and the sensible heat of cooling the vapor from 135° to 110°F.

$$\begin{array}{rcl} \text{n-Butane} & = & \frac{29}{51} \times 100 = 56.8 \text{ wt\%} \\ \text{Propane} & = & \frac{22}{51} \times 100 = 43.2 \text{ wt\%} \end{array}$$

At 110°F (data from Maxwell⁶):

$$\begin{array}{rcl} \text{Vapor pressure of propane} & = & 14.5 \text{ atm} \\ \text{Vapor pressure of n-butane} & = & 4.0 \text{ atm} \\ \text{Latent heat } (\lambda) \text{ of propane} & = & 130 \text{ Btu/lb} \\ \text{Latent heat } (\lambda) \text{ of n-butane} & = & 146 \text{ Btu/lb} \end{array}$$

In the range 135° to 110°F:

$$\begin{array}{rcl} C_p \text{ of propane} & = & .43 \text{ Btu/lb } ^\circ\text{F} \\ C_p \text{ of n-butane} & = & .43 \text{ Btu/lb } ^\circ\text{F} \\ \\ W \lambda \text{ propane} & = & 143,000 \times .432 \times 130 = 8,050,000 \text{ Btu/hr} \\ W \lambda \text{ butane} & = & 143,000 \times .568 \times 146 = 11,850,000 \text{ Btu/hr} \\ W C_p \Delta t \text{ propane} & = & 143,000 \times .432 \times 25 \times .43 = 665,000 \text{ Btu/hr} \\ W C_p \Delta t \text{ butane} & = & 143,000 \times .568 \times 25 \times .43 = \underline{875,000 \text{ Btu/hr}} \\ \text{Total } Q & = & 21,440,000 \text{ Btu/hr} \end{array}$$

From the NGSMA⁷ enthalpy charts the value of 21,130,000 Btu/hr may be obtained. The average of these figures checks closely with the specified value of 21,360,000 Btu/hr. Thus, a mixture of 50 mol% propane and 50 mol% butane will be used to compute the condensing range.

The procedure used in computing the dew-point and bubble-point curves for the stream is explained in Perry,⁹ p.587. It involves a trial-and-error solution and uses the equilibrium constants (K) of the components. These values of K were obtained from the NGSMA data book.⁷ The calculations are presented below.

I. BUBBLE-POINT CALCULATIONS.

A. At 200 psia.—

| | Mol% (M) | K _{140°F} | M x K _{140°F} | K _{143°F} | M x K _{143°F} |
|----------------------------------|----------|--------------------|------------------------|--------------------|------------------------|
| C ₃ H ₈ | 0.50 | 1.38 | .69 | 1.40 | 0.70 |
| n-C ₄ H ₁₀ | 0.50 | 0.57 | <u>.28</u> | 0.60 | <u>0.30</u> |
| | | | 0.97 | | 1.00 |

∴ Bubble point at 200 psia = 143°F.

B. At 175 psia.—

| | Mol% (M) | K _{130°F} | M x K _{130°F} |
|----------------------------------|----------|--------------------|------------------------|
| C ₃ H ₈ | 0.50 | 1.430 | .715 |
| n-C ₄ H ₁₀ | 0.50 | .565 | <u>.283</u> |
| | | | 0.998 |

∴ Bubble point at 175 psia = 130°F.

C. At 150 psia.—

| | Mol% (M) | K _{120°F} | M x K _{120°F} | K _{118°F} | M x K _{118°F} |
|----------------------------------|----------|--------------------|------------------------|--------------------|------------------------|
| C ₃ H ₈ | 0.50 | 1.50 | 0.75 | 1.46 | 0.73 |
| n-C ₄ H ₁₀ | 0.50 | 0.55 | <u>0.28</u> | 0.54 | <u>0.27</u> |
| | | | 1.03 | | 1.00 |

∴ Bubble point at 150 psia = 118°F.

D. At 125 psia.—

| | Mol% (M) | $K_{100^{\circ}\text{F}}$ | $M \times K_{100^{\circ}\text{F}}$ | $K_{105^{\circ}\text{F}}$ | $M \times K_{105^{\circ}\text{F}}$ |
|-----------------------------|----------|---------------------------|------------------------------------|---------------------------|------------------------------------|
| C_3H_8 | 0.50 | 1.42 | 0.71 | 1.50 | 0.75 |
| $\text{n-C}_4\text{H}_{10}$ | 0.50 | 0.48 | <u>0.24</u> | 0.53 | <u>0.26</u> |
| | | | 0.95 | | 1.01 |

∴ Bubble point at 125 psia = 104°F.

E. At 100 psia.—

| | Mol% (M) | $K_{90^{\circ}\text{F}}$ | $M \times K_{90^{\circ}\text{F}}$ | $K_{88^{\circ}\text{F}}$ | $M \times K_{88^{\circ}\text{F}}$ |
|-----------------------------|----------|--------------------------|-----------------------------------|--------------------------|-----------------------------------|
| C_3H_8 | 0.50 | 1.55 | .775 | 1.52 | 0.76 |
| $\text{n-C}_4\text{H}_{10}$ | 0.50 | 0.49 | <u>.245</u> | 0.48 | <u>0.24</u> |
| | | | 1.020 | | 1.00 |

∴ Bubble point at 100 psia = 88°F.

F. At 70 psia.—

| | Mol% (M) | $K_{60^{\circ}\text{F}}$ | $M \times K_{60^{\circ}\text{F}}$ | $K_{62^{\circ}\text{F}}$ | $M \times K_{62^{\circ}\text{F}}$ |
|-----------------------------|----------|--------------------------|-----------------------------------|--------------------------|-----------------------------------|
| C_3H_8 | 0.50 | 1.50 | 0.75 | 1.55 | 0.775 |
| $\text{n-C}_4\text{H}_{10}$ | 0.50 | 0.42 | <u>0.21</u> | 0.43 | <u>0.215</u> |
| | | | 0.96 | | 0.990 |

∴ Bubble point at 70 psia = 63°F.

II. DEW-POINT CALCULATIONS.

Since both C_3H_8 and $\text{n-C}_4\text{H}_{10}$ are 50 mol% of the total, the trial and error may be simplified so that the value of $(1/K_{\text{C}_3\text{H}_8} + 1/K_{\text{n-C}_4\text{H}_{10}})$ must equal 2.00 for a check.

A. At 200 psia.—

| | $K_{165^{\circ}\text{F}}$ | $1/K_{165^{\circ}\text{F}}$ | $K_{162^{\circ}\text{F}}$ | $1/K_{162^{\circ}\text{F}}$ |
|-----------------------------|---------------------------|-----------------------------|---------------------------|-----------------------------|
| C_3H_8 | 1.67 | 0.60 | 1.62 | 0.617 |
| $\text{n-C}_4\text{H}_{10}$ | 0.73 | <u>1.37</u> | 0.71 | <u>1.408</u> |
| | | 1.97 | | 2.025 |

∴ Dew point at 200 psia = 164°F.

B. At 175 psia.—

| | $K_{152^{\circ}\text{F}}$ | $1/K_{152^{\circ}\text{F}}$ |
|-----------------------------|---------------------------|-----------------------------|
| C_3H_8 | 1.71 | 0.585 |
| $n\text{-C}_4\text{H}_{10}$ | 0.71 | <u>0.408</u> |
| | | 1.993 |

∴ Dew point at 175 psia = 152°F.

C. At 150 psia.—

| | $K_{140^{\circ}\text{F}}$ | $1/K_{140^{\circ}\text{F}}$ | $K_{138^{\circ}\text{F}}$ | $1/K_{138^{\circ}\text{F}}$ |
|-----------------------------|---------------------------|-----------------------------|---------------------------|-----------------------------|
| C_3H_8 | 1.78 | 0.562 | 1.74 | 0.575 |
| $n\text{-C}_4\text{H}_{10}$ | 0.70 | <u>1.425</u> | 0.69 | <u>1.450</u> |
| | | 1.987 | | 2.025 |

∴ Dew point at 150 psia = 139°F.

D. At 125 psia.—

| | $K_{125^{\circ}\text{F}}$ | $1/K_{125^{\circ}\text{F}}$ | $K_{127^{\circ}\text{F}}$ | $1/K_{127^{\circ}\text{F}}$ |
|-----------------------------|---------------------------|-----------------------------|---------------------------|-----------------------------|
| C_3H_8 | 1.80 | 0.555 | 1.85 | 0.54 |
| $n\text{-C}_4\text{H}_{10}$ | 0.68 | <u>1.470</u> | 0.69 | <u>1.45</u> |
| | | 2.025 | | 1.99 |

∴ Dew point at 125 psia = 127°F.

E. At 100 psia.—

| | $K_{110^{\circ}\text{F}}$ | $1/K_{110^{\circ}\text{F}}$ | $K_{113^{\circ}\text{F}}$ | $1/K_{113^{\circ}\text{F}}$ |
|-----------------------------|---------------------------|-----------------------------|---------------------------|-----------------------------|
| C_3H_8 | 1.90 | 0.526 | 1.95 | 0.513 |
| $n\text{-C}_4\text{H}_{10}$ | 0.65 | <u>1.539</u> | 0.68 | <u>1.470</u> |
| | | 2.065 | | 1.983 |

∴ Dew point at 100 psia = 112°F.

F. At 70 psia.—

| | $K_{90^{\circ}\text{F}}$ | $1/K_{90^{\circ}\text{F}}$ | $K_{86^{\circ}\text{F}}$ | $1/K_{86^{\circ}\text{F}}$ |
|-----------------------------|--------------------------|----------------------------|--------------------------|----------------------------|
| C_3H_8 | 2.12 | 0.471 | 2.04 | 0.49 |
| $n\text{-C}_4\text{H}_{10}$ | 0.68 | <u>1.470</u> | 0.64 | <u>1.56</u> |
| | | 1.941 | | 2.05 |

∴ Dew point at 70 psia = 88°F.

The calculated dew-point and bubble-point curves are shown in Fig. 10. The condensing range, which is the vertical distance between the curves in Fig. 10, is shown in Fig. 11.

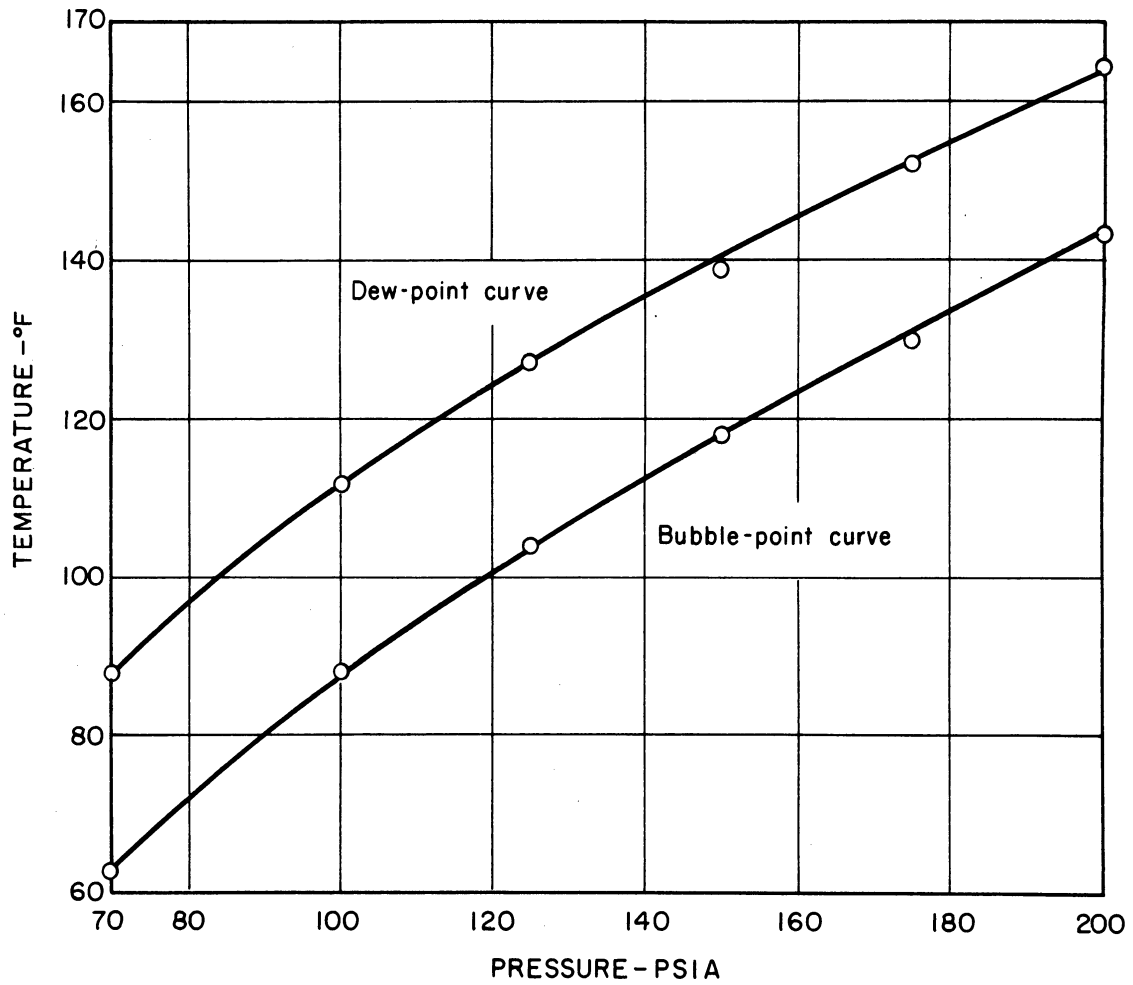


Fig. 10. Dew-point and bubble-point curves for a mixture of 50 mol% propane and 50 mol% n-butane.

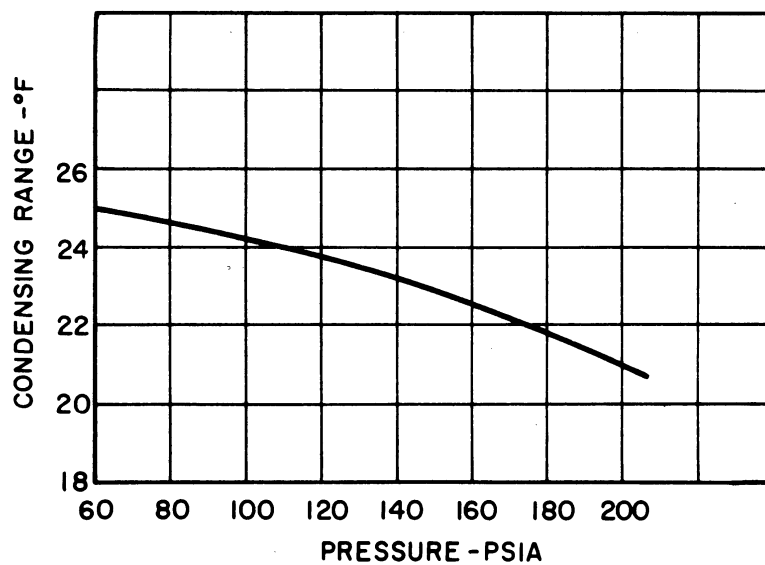


Fig. 11. Condensing range of a mixture of 50 mol% propane and 50 mol% n-butane.

APPENDIX N

DEBUTANIZER OVERHEAD CONDENSER —
CHECK ON THE OVERALL HEAT TRANSFER COEFFICIENT
ASSUMING A NONFLOODED TOTAL CONDENSER

From the specification sheet:

| | | |
|-------------------------------------|---|---------------|
| Water inlet temperature | = | 85°F |
| Water outlet temperature | = | 110°F |
| Water flow rate | = | 855,000 lb/hr |
| Hydrocarbon inlet temperature | = | 135°F |
| Hydrocarbon outlet temperature | = | 110°F |
| Operating pressure (shell side) | = | 133 psig |
| Hydrocarbon specific gravity | = | 121° API |
| Hydrocarbon molecular weight | = | 50.8 |
| Fouling resistance: | | |
| Shell side (r_o) | = | 0.0005 |
| Tube side ($\frac{A_o r_i}{A_i}$) | = | 0.008 |

Water velocity:

$$V_t = \frac{W}{223,000} = \frac{855,000}{223,000}$$

$$= 3.83 \text{ ft/sec.}$$

(Note: this disagrees with the specified value of 7.0 ft/sec.)

Inside heat transfer coefficient:

$$h_i = \frac{150(1 + 0.011t_w) V_t^{0.8}}{d_i^{0.2}}$$

$$t_w = \frac{85 + 110}{2} = 97.5^\circ\text{F}$$

$$d_i = 0.51 \text{ inch}$$

$$\therefore h_i = \frac{(150)(1 + 0.011 \times 97.5)(3.83)^{0.8}}{(0.51)^{0.2}}$$

$$= \frac{(150)(2.072)(2.93)}{(0.874)}$$

$$= 1041 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

$$\text{Then } \frac{A_o}{A_i h_i} = \frac{3.28}{1041} = .00315 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}.$$

Metal resistance:

$$\frac{A_o}{A_m} r_m = 0.00027 \quad (\text{see Section V-D}).$$

Fin resistance:

$$r_f = 0.00011 \quad (\text{see Section V-D}).$$

The condensing coefficient may be evaluated from the equation

$$h_o' = 0.725 \left[\frac{K_f^3 \rho_f^2 g_c}{\mu_f} \right]^{1/4} (\lambda)^{1/4} \left(\frac{1}{D_{eq}} \right)^{1/4} \left(\frac{1}{\Delta t_c} \right)^{1/4} \left(\frac{CN}{N} \right)^{1/4} .$$

From the specification sheet for the unit:

$$\begin{aligned} \mu_f &= 0.1 \text{ centipoise at } 123^\circ\text{F} \\ K_f &= 0.078 \text{ Btu/hr-ft}^2 \text{ (}^\circ\text{F/ft)} \\ \rho_f &= 0.56 \text{ gm/cc.} \end{aligned}$$

As shown in Appendix M the hydrocarbon stream has a latent heat closely approximating a mixture of 50 mol% propane and 50 mol% n-butane. Therefore, the value of latent heat to be used in calculating the condensing coefficient is the average of propane and n-butane.

Trial No. 1

$$\text{Assume } h_o' = 1160 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\begin{aligned} \therefore 1/h_o' &= .00085 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ A_o/A_i h_i &= .00315 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ A_o r_m/A_m &= .00027 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ r_f &= .00011 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ r_o &= .00800 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ A_o r_i/A_i &= \underline{.00050} \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \\ \therefore 1/U_o &= .01288 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu} \end{aligned}$$

$$\begin{aligned}
 \text{Then} \qquad U_o &= 77.6 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \\
 \Delta t_c &= \frac{U_o}{h_o'} (\Delta T_{LM}) = \frac{77.6}{1160} \times 25 \\
 &= 1.67^\circ\text{F} \\
 (1/\Delta t_c)^{1/4} &= 0.880 \\
 t_f &= t_b - (\Delta t_c/2) = \left(\frac{135 + 110}{2} \right) - (1.67/2) \\
 &\cong 122^\circ\text{F}.
 \end{aligned}$$

From Maxwell⁶ at 122°F:

$$\begin{aligned}
 \text{Vapor pressure propane} &= 16 \text{ atm} \\
 \text{Vapor pressure butane} &= 4.6 \text{ atm} \\
 \lambda \text{ propane} &= 128 \text{ Btu/lb} \\
 \lambda \text{ butane} &= \underline{142} \text{ Btu/lb} \\
 \lambda \text{ average} &= 135 \text{ Btu/lb} \\
 \therefore (\lambda)^{1/4} &= 3.40.
 \end{aligned}$$

At $h_o' = 1160 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$:

$$(1/D_{eq})^{1/4} = 3.34.$$

From analysis of the blueprints:

$$C_N/(N)^{1/4} = 0.94.$$

Solving for the physical property group:

$$\begin{aligned}
 \left[\frac{K_f^3 \rho_f^2 g_c}{\mu_f} \right]^{1/4} &= \left[\frac{(0.078)^3 (0.56 \times 62.4)^2 (32.2)(3600)^2}{(0.1 \times 2.42)} \right]^{1/4} \\
 &= [9.97 \times 10^8]^{1/4} \\
 &= 177.7.
 \end{aligned}$$

Therefore,

$$\begin{aligned}
 h_o' &= (0.725)(177.7)(3.40)(3.34)(0.88)(0.94) \\
 &= 1210 \text{ (no check with assumed value of 1160)}.
 \end{aligned}$$

Trial No. 2

$$\text{Assume } h_o' = 1220 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$1/h_o' = .00082 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$$

$$\therefore 1/U_o = .01285 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$$

and

$$U_o = 77.8 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

$$\Delta t_c = \frac{77.8}{1220} \times 25 = 1.59^\circ\text{F}$$

$$(1/\Delta t_c)^{1/4} = 0.890.$$

Since the other factors in the h_o' equation remain essentially constant from Trial No. 1,

$$h_o' = \frac{0.890}{0.880} \times 1210 = 1223 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

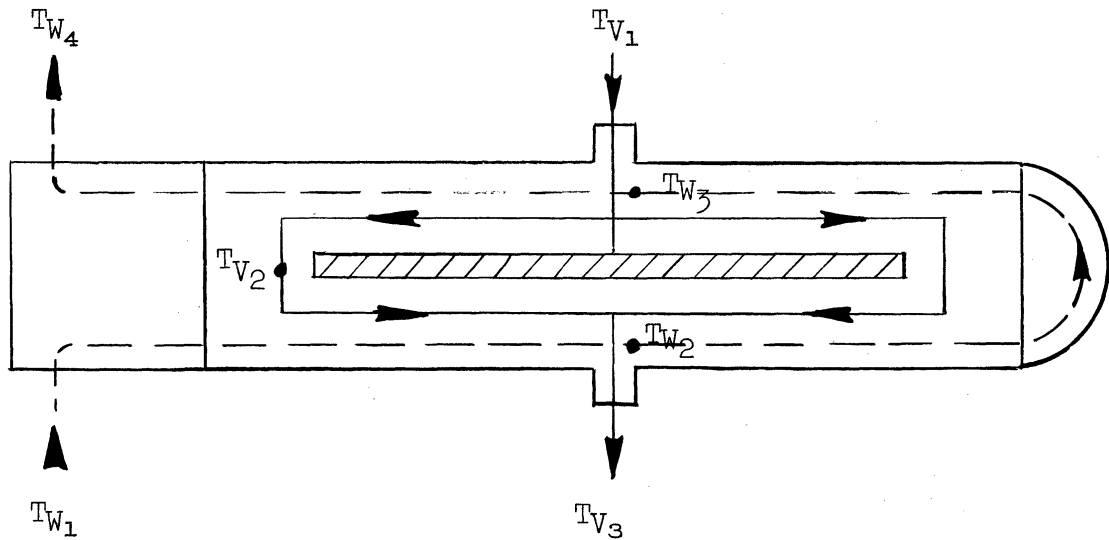
This checks with the assumed value of $h_o' = 1220$.

$$\text{Then } U_o = 77.8 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

This is $\frac{4.8}{73} \times 100 = 6.6\%$ higher than the specified value of $U_o = 73$.

APPENDIX O

CORRECTION FACTOR TO BE APPLIED TO THE LOG-MEAN TEMPERATURE DIFFERENCE IN A NONFLOODED, SPLIT FLOW CONDENSER



Legend

--- Water Flow
 — Vapor Flow

$$\text{Let } (\Delta T_{LM})_{\text{corrected}} = (F) \left[\frac{(T_{V1} - T_{W4}) - (T_{V3} - T_{W1})}{\ln \frac{(T_{V1} - T_{W4})}{(T_{V3} - T_{W1})}} \right] \cdot$$

Then the equation to be used in computing the value of the correction factor (F) is

$$F = \frac{1}{4} \left\{ \frac{[(T_{V2} - T_{W4}) - (T_{V1} - T_{W3})] \ln \frac{(T_{V1} - T_{W4})}{(T_{V3} - T_{W1})}}{[(T_{V1} - T_{W4}) - (T_{V3} - T_{W1})] \ln \frac{(T_{V2} - T_{W4})}{(T_{V1} - T_{W3})}} + \frac{[(T_{V3} - T_{W2}) - (T_{V2} - T_{W1})] \ln \frac{(T_{V1} - T_{W4})}{(T_{V3} - T_{W1})}}{[(T_{V1} - T_{W4}) - (T_{V3} - T_{W1})] \ln \frac{(T_{V3} - T_{W2})}{(T_{V2} - T_{W1})}} \right.$$

$$\frac{2 [(T_{V_3} - T_{W_2}) - (T_{V_1} - T_{W_3})] \ln \frac{(T_{V_1} - T_{W_4})}{(T_{V_3} - T_{W_1})}}{[(T_{V_1} - T_{W_4}) - (T_{V_3} - T_{W_1})] \ln \frac{(T_{V_3} - T_{W_2})}{(T_{V_1} - T_{W_3})}} \cdot$$

