ENGINEERING RESEARCH INSTITUTE
THE UNIVERSITY OF MICHIGAN
ANN ARBOR

ANALYSIS OF THE PERFORMANCE OF FIVE FORCED-AIR COOLING UNITS AT THE DENTON OIL FIELD PLANT OF THE ATLANTIC REFINING COMPANY

Report No. 38

Edwin H. Young
Assistant Professor of Chemical Engineering

Dennis J. Ward
Luis O. Gonzalez
Research Assistants

Project 1592

WOLVERINE TUBE DIVISION
CALUMET AND HECLA, INC.
DETROIT, MICHIGAN

December 1955

ACKNOWLEDGEMENT

The University of Michigan research group wishes to acknowledge the excellent cooperation and assistance of Mr. Robert H. Fritz in conducting the field tests at the Denton oil field gasoline plant at Lovington, New Mexico.

TABLE OF CONTENTS

TABLE OF CONTENTS (Concl.)

LIST OF TABLES

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (Concl.)

ABSTRACT

This report contains an analysis of field test data taken on a jacket water cooler at the Denton oil field gasoline plant of the Atlantic Refining Co. Analyses of the proposed designs for a lean oil cooler, a debutanizer condenser, an evaporator condenser, and a still overhead partial condenser for the same plant are also included.

All the above units use air under forced convection as the cooling medium, with the air-side coefficients controlling the performance. The rating curve used by the designers greatly overrates the air-side coefficients, resulting in underperformance of the units.

I. DESCRIPTION OF UNITS AND TEST PROCEDURE

A. INTRODUCTION

The Denton oil field gasoline plant of the Atlantic Refining Co. had been on stream for about a year. During this period the plant production capacity was found to be considerably below minimum acceptance during the summer months. The cause of this reduced capacity was traced to several air-cooled units which were fabricated from Wolverine bimetal high-fin tubes. The most serious unit involved the cooling of recirculated compressor-jacket cooling water.

Field tests were made in order to obtain data which could be used to evaluate the performance of the jacket water cooler. The other field units involved were not tested but the specifications for these units were analyzed. This report contains a summary of the analysis of all the units involved.

B. UNITS TESTED ON THURSDAY, APRIL 28, 1955

The tests conducted on Thursday, April 28, were made on the jacket water cooler. This unit is an air-blast heat exchanger using forced air on the outside of the tubes to cool water flowing inside the tubes. The fans are located upstream from the tube bank.

The jacket water cooler tested consisted of two separate bays, the east bay and the west bay, each capable of operating independently from one another. Each bay contained 358 Wolverine Trufin tubes (Catalog No. 62-0916049-26), 24 ft long. Each bay had its own 16-ft, 4-blade, Moore pressure-blower-type fan for pushing the air vertically upward through the tube bank.

The east bay fan-tip clearance had been reduced by representatives of the Happy Co. prior to the tests, whereas the west bay fan-tip clearance had not been adjusted for these tests.

The tubes in both of the two bays were arranged with the bottom three rows in line and the fourth row staggered. A section of one tube had been removed from the west bay and shipped to The University of Michigan for testing. The results of these tests are summarized in Section VIII of this report.

C. UNITS TESTED ON MONDAY, MAY 2, 1955

From April, 29 to May 1, 1955, the gasoline-plant operations were shut down. During this time new tube banks fabricated by the Happy Co. were installed in place of the tube banks tested on the previous Thursday. The fan arrangements were not changed. The new east bay of the jacket water cooler contained the same three-in-line, fourth-row-staggered arrangement of the tubes as the tube bank it replaced. The tube arrangement of the new west bay tube bank was changed to a triangular pitch arrangement.

D. DESCRIPTION OF TEST METHODS AND PROCEDURE

The University of Michigan research group, assisted by Mr. Robert Fritz of Wolverine Tube, measured the following variables: (a) temperature of the inlet water; (b) temperature of the outlet water; (c) water flow rate; (d) air inlet temperature to the fans; (e) outlet air temperature profile; (f) outlet air velocity profile; and (g) fan-motor amperage. An attempt was also made to measure the static pressure profile between the fan and tube bank; however, it was unsuccessful for reasons discussed in Section II-A of this report.

The water inlet temperatures and water outlet temperatures were measured with mercury-in-glass thermometers. The inlet air temperature and the outlet air temperature profiles were measured with iron-constantan thermocouples. The thermocouples used in the outlet air temperature profile were centered in six-in.-diameter stovepipe ducts approximately one ft high. This was necessary in order to avoid mixing of the air coming from the tube bank with the ambient air above the bank. The EMF readings were obtained with a potentiometer furnished by the University of Michigan Engineering Research Institute. The air velocity profile of the air leaving the units was determined by using a Taylor four-in., Model 3132, vane-type anemometer placed concentrically in a six-in. duct; the anemometer time period was measured with a stop watch. The electrical current flowing through the fan motor was measured, using a General Electric hook-on volt-ammeter, Type AK-1.

A discussion of the reliability of the test measurements is given in Section II-C of this report.

The procedure followed in testing a unit was as follows. Ten thermocouple assemblies were arranged on top of the unit in a grid form as indicated in Fig. 1. The grid areas are identified by letter-number combinations such as 2D, 3A, etc. One thermocouple assembly was placed directly under the fan. Water-thermometer, thermocouple, and the cold-junction temperature readings were obtained in succession until reasonably steady-state conditions were reached. The local barometric pressure was obtained by telephoning the local airport. The velocity profile was made over the same areas

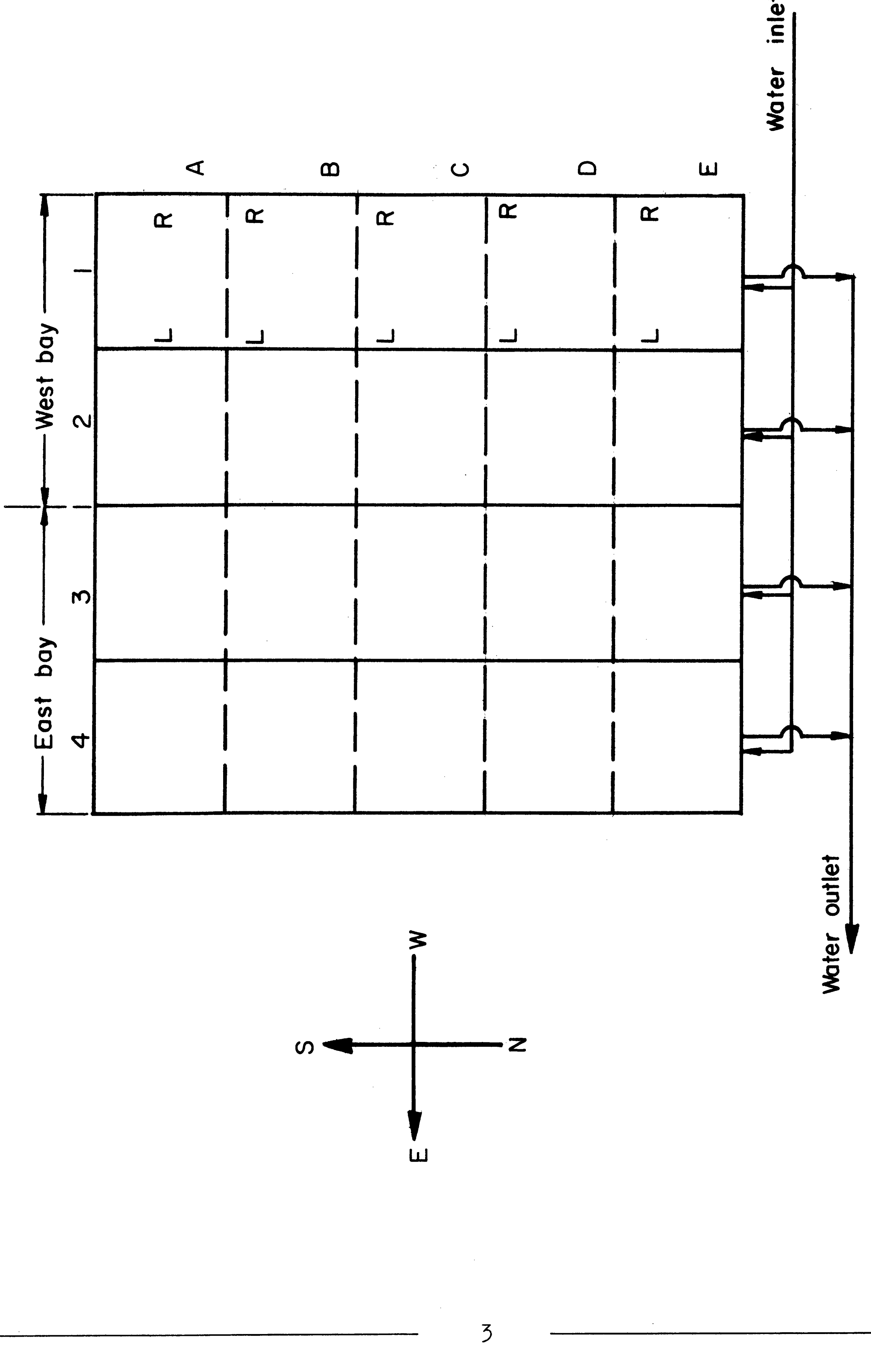

Fig. 1. Jacket water cooler layout.

occupied by the thermocouples during the period when the thermocouple readings were being made.

E. DESCRIPTION OF UNITS NOT TESTED

In addition to the jacket water cooler there are four other air-cooled units on stream at the gasoline plant. These units are identified as (a) Lean oil cooler, (b) Still overhead partial condenser, (c) Evaporation condenser, and (d) Debutanizer overhead condenser. The specifications for these units are given in Section XI of this report.

Evaluations of these units are presented in Section XII of this report.

II. FIELD TEST DATA ON JACKET WATER COOLER

A. DATA

The field test data obtained on the jacket water cooler are summarized in Tables I, II, and III. Table I presents the test data obtained on Thursday, April 28, 1955. These data were obtained on the unit during a period of fluctuating ambient-air temperatures and under fan-blade pitch control conditions such that fluctuations in the fan pitch resulted in varying throughput of air. Since these test data were not obtained under steady-state conditions, some questions arise as to their validity for determining the performance of the unit prior to the turnaround.

Table II summarizes the data obtained on the east bay of the jacket water cooler on Monday, May 2, 1955, for the period indicated. This bay contained 179 tubes on a three-rows-in-line, fourth-row-staggered pitch arrangement. This unit replaced one of the bays tested on the previous Thursday.

TABLE I

SUMMARY OF EXPERIMENTAL DATA OF THURSDAY, APRIL 28, 1955

Arrangement of Tubes: three rows in line, fourth row staggered

Data on West Bay

Position	EMF* (millivolts)			Anemometer Reading (ft)	Anemometer Time (sec)	Indicated Anemometer Velocity (ft/min)
	(1)	(2)	Avg			
1A R	--	--	--	446.5	58.6	456
1A L	--	--	--	351.5	58.9	358
1B R	--	--	--	494	58.3	508
1B L	--	--	--	575	60.1	573
1C R	1.48	1.38	1.43	590	58.2	608
1C L	1.60	1.60	1.60	419	62.8	400
1D R	1.48	1.45	1.465	539.5	58	558
1D L	1.50	1.47	1.485	623	59.5	628
1E R	1.47	1.47	1.47	496	60.0	496
1E L	1.50	1.48	1.49	549	59.9	550
2A R	--	--	--	366	59.8	367
2A L	--	--	--	398	60.0	398
2B R	--	--	--	466	59.1	473
2B L	--	--	--	258	60.0	258
2C R	1.37	1.38	1.375	546	60.0	546
2C L	1.60	1.60	1.60	430	60.2	429
2D R	1.47	1.45	1.46	531	59.1	539
2D L	1.42	1.47	1.445	413	60.0	413
2E L	1.52	1.57	1.545	464	59.5	467
2E R	1.60	1.60	1.60	377	60.2	376

West Bay Avg = 470.1

Cold-Junction Temperature (Cell No. 1) = 27.80°C = 82.0°F
Cold-Junction Temperature (Cell No. 2) = 26.2°C = 79.2°F
Water Inlet Temperature = 163° and 164°F, avg = 163.5°F
Water Outlet Temperature, avg = 156.0°F
Air Inlet Temperature = 75.5°, 77.5°, and 74.1°F, avg = 75.7°F
Water-Main Pitot Tube Readings: varied widely (pulsating) from 3-1/2 to 8 in.

TABLE I (Concl.)

Data on East Bay

Position	EMF* (millivolts) (1)	(2)	Avg	Anemometer Reading (ft)	Anemometer Time (sec)	Indicated Anemometer Velocity (ft/min)
3A R	--	--	--	418	59.8	420
3A L	--	--	--	485	60.0	485
3B R	--	--	--	546	60.1	545
3B L	--	--	--	676	61.0	665
3C R	1.85	1.90	1.875	523	60.4	520
3C L	1.92	1.94	1.93	500	60.0	500
3D R	1.74	1.80	1.77	523	60.1	521
3D L	1.76	1.75	1.755	761	60.1	759
3E R	1.67	1.76	1.715	637	59.9	639
3E L	1.82	1.82	1.82	534	60.0	534
4A R	--	--	--	492	60.0	492
4A L	--	--	--	400	60.0	400
4B R	--	--	--	464	60.0	464
4B L	--	--	--	472	62.5	453
4C R	1.45	1.47	1.46	677.5	59.6	681
4C L	1.65	1.60	1.625	520	60.0	520
4D R	1.50	1.50	1.50	672.5	60.7	664
4D L	1.73	1.70	1.715	523	60.0	523
4E R	1.48	1.50	1.49	507	60.1	506
4E L	1.50	1.50	1.50	506	60.0	506

*Iron-constantan thermocouple.

East Bay Avg = 539.9

Cold-Junction Temperature (Cell No. 3) = 26.2°C = 79.2°F
Cold-Junction Temperature (Cell No. 4) = 27.8°C = 82.0°F
Water Inlet Temperature = 163° and 162°F, avg = 162.5°F
Water Outlet Temperature = 154.5° and 154.5°F, avg = 154.5°F
Air Inlet Temperature = 75.7° and 68.6°F, avg = 72.1°F

TABLE II

SUMMARY OF EXPERIMENTAL DATA OF EAST BAY
DATA OF MONDAY, MAY 2, 1955

Arrangement of Tubes: three in line, fourth row staggered

Time of test start: 2:00 P.M.
Time of test finish: 2:15 P.M.
Fan in Full-Pitch Position

Position	EMF* (millivolts)			Anemometer Reading (ft)	Anemometer Time (sec)	Indicated Anemometer Velocity (ft/min)
	(1)	(2)	Avg			
3A	1.45	1.38	1.415	780	60	780
3B	1.40	1.42	1.410	743	59.8	745
3C	1.35	1.38	1.365	622	60.0	622
3D	1.30	1.28	1.290	806	60.1	804
3E	1.30	1.30	1.300	767	60.8	756
4A	1.47	1.46	1.465	797	60.0	797
4B	1.18	1.18	1.180	857	60.0	857
4C	1.17	1.17	1.170	792	59.9	793
4D	1.25	1.26	1.255	891	60.2	889
4E	1.32	1.32	1.320	827	61.0	813
	Overall Avg	=	1.317			785.6

*Iron-constantan thermocouple

	Reading No. 1	2	3	4	Avg
Cold Junction Temperature, °C	29.2	29.0	--	--	29.1°C
Water Inlet, °F	153.0	153.5	152.0	152.5	152.7°F
Water Outlet, °F	144.5	143.5	143.5	143.0	143.6°F
Air Inlet Temperature, °F	81.0	81.5	80.0		80.8
Orifice Meter		66 inches water			66 in. water
Motor Amps	41.0	41.0	41.0		41.0 amps

(Barometric pressure 29.87 in. Hg, obtained from airport)

TABLE III

SUMMARY OF EXPERIMENTAL DATA OF WEST BAY
DATA OF MONDAY, MAY 2, 1955

Arrangement: triangular-pitch bay

Time of test start: 2:30 P.M.
Time of test finish: 2:50 P.M.

Position	EMF* (millivolts) (1)	(2)	Avg	Anemometer Reading (ft)	Anemometer Time (sec)	Indicated Anemometer Velocity (ft/min)
1A	1.10	1.13	1.115	548	62.4	526
1B	0.90	0.90	0.900	761	60.0	761
1C	1.30	1.31	1.305	613	60.0	613
1D	1.05	1.07	1.060	666	60.1	664
1E	1.05	1.10	1.075	686	60.4	681
2A	1.15	1.13	1.140	554	60.0	554
2B	0.90	0.90	0.900	814	59.8	816
2C	0.92	0.96	0.940	535	60.2	532
2D	0.88	0.90	0.890	822	60.1	821
2E	1.02	1.00	1.010	650	60.0	650
	Overall Avg	=	1.0335			661.8

*Iron-constantan thermocouple

	Reading No. 1	2	3	4	Avg
Cold Junction Temperature, °C	35.4	34.6	35.1	--	35.03°C
Water Inlet, °F	150	150	147.5	147.5	148.7°F
Water Outlet, °F	142	141	140	139	140.5°F
Water Orifice		66 inches of water			66 in. water
Ambient-Air Thermometer	79.5	79.5	79.5	79.5	79.5°F
Motor Amps	43	43	42	42	42.5amps

Wind velocity - 935 ft/min
(Barometric pressure 29.87 in. Hg, obtained from airport)

Table III summarizes the field test data obtained on the new west bay of the jacket water cooler on Monday, May 2, 1955. This bay contained 178 tubes on a 2-1/16-in. triangular pitch arrangement.

It should be noted that the three tables also contain the overall average anemometer velocities in ft/min, average inlet and exit air temperatures, and water inlet and exit temperatures. A Pitot tube on the centerline of the 16-in. water main gave widely fluctuating readings of from 3-1/2 to 8 in. of meriman oil pressure on Thursday, April 28 (Table I). This Pitot tube was replaced by an orifice plate (11.25 in. diameter) which gave reliable pressure-drop readings on Monday, May 2, (Tables II and III).

A University of Michigan Pitot-static tube, five feet long, was taken along for making an air-side pressure-drop profile. The field units would not accommodate this Pitot tube. The units had a small hole which would accommodate a small-diameter tube (about 1/8 or 3/16 in.). Mr. Roberts had a makeshift arrangement whereby a copper tube was inserted through this hole for making a single-point pressure measurement. These data are contained under Section X of this report.

B. PRELIMINARY ANALYSIS OF THE DATA

1. Fouling on the Water Side and Air Side.—The problem of determining a suitable water fouling factor for the water side in analyzing Thursday's data was solved by using an experimentally determined fouling factor on a sample of a tube cut from the field unit. The experimentally determined value was found to be 0.0013, based on the liner area. The data for determining this value are given in Section VIII of this report.

The selection of a proper fouling factor for the test data taken on Monday is open to question. The units tested were new units shipped from the Happy Co. of Tulsa, Oklahoma, for this purpose. There is a strong argument in favor of assuming no fouling, since the units had been on stream less than two days. In order to show the influence of some fouling on the computed results, these data were analyzed under the assumed conditions of zero fouling and fouling of 0.001 for the water side. No fouling was assumed for the air side.

2. Thursday's Data.—The water-side flow-rate measurements obtained by means of a Pitot tube were unreliable due to rapid fluctuations of the manometer over a wide range (3-1/2 to 8 in.). An attempt was made to determine the performance of the unit, even though the fan pitch was known to have been varying during the test run. It was assumed that the air velocity and temperature profile (see Table I) would give an indication of the performance of the unit. The calculations are presented in Appendix A.

The results of the analysis are summarized below.

Air face velocities (indicated):
East Unit = 508 std. ft/min
West Unit = 445 std. ft/min
Average temperature rise of the air:
East Unit = 65.7°F
West Unit = 56.1°F
Apparent-mass flow rate of air:
East Unit = 825,000 lb/hr
West Unit = 723,000 lb/hr
Apparent heat duty:
East Unit = 13,000,000 Btu/hr
West Unit = 9,730,000 Btu/hr
Total 22,730,000 Btu/hr

The jacket water cooler was supposed to handle 21,492,000 Btu/hr. It was quite evident that the above figure of 22,730,000 was in error, since it was a known fact that the unit was actually operating considerably below design load as specified by the specification sheet for the unit. This is further emphasized by the quantity of water that would have had to be flowing in order to transfer the indicated load. This calculation is also given in Appendix A and is summarized below:

Required water flow rate:
East Unit = 1,625,000 lb/hr
West Unit = 1,390,000 lb/hr
Total = 3,015,000 lb/hr

Computed water flow rates:
East Unit = 9.33 ft/sec
West Unit = 8.00 ft/sec

(The design water flow rate was approximately 5.5 ft/sec.)

On the basis of 3,015,000 lb/hr of water flowing through the tubes and a heat duty of 22,730,000 Btu/hr, an overall coefficient of 121 was obtained on the east unit, whereas a value of 82.8 was obtained on the west unit, based on the liner area. The water-side coefficients as computed were 2520 and 2230, respectively, based on the inside area. The corresponding h_o values were 154.5 and 97.0, based on liner area. The water-film resistance is negligible in comparison with the air-side resistance. The fouling resistance used in the calculations was determined experimentally on a sample tube by measuring the overall performance of the tube before and after cleaning out the inside of the tube with steam plus a hot-water jet. The value determined was 0.0013 for inside fouling, based on liner area (see Section VIII of this report).

The computed air-side coefficients are obviously incorrect, for if such coefficients were actually realized, the units should have performed considerably better than indicated. It is quite apparent that one or more inconsistencies in the data are influencing the computed results. A discussion of the inconsistencies of the Thursday and Monday data is given in Section II-C. The resolution of the inconsistencies is given in Section II-D. This is followed by an analysis of the revised data in Section II-E.

3. Monday's Data.—

a. Three-in-line, fourth-row-staggered unit.

The data obtained on this unit are given in Table II and are of such a nature that both air-side and water-side heat loads may be determined. The calculations are presented in Appendix B and are summarized below.

Apparent air-face velocity	=	747 std. ft/min
Air-side heat duty	=	13,900,000 Btu/hr
Apparent air-side coefficient	=	166.5 based on liner area ($r_i = 0$)
Water-side heat duty (assuming one-half the water passed through this unit)	=	9,380,000 Btu/hr

A comparison of the water-side heat duty with the air-side heat duty indicates a discrepancy of 4,520,000 Btu/hr. The air-side heat duty is 48% higher than the water-side heat duty. The design heat load for the unit is 21,492,000 Btu/hr for the two bays, or 10,746,000 Btu/hr per bay. The air-side coefficient of 166.5 (see Appendix B Section 6) was based on the air-side heat duty and therefore is questionable in view of the lower water-side heat duty. Another air-side coefficient can be computed, assuming the water-side heat duty is correct and that the measured air temperatures are correct. This assumes that the air velocity profile is incorrect. Such a calculation is presented in Appendix B Section 9. The computed necessary air velocity is 504 ft/min (face velocity) at standard conditions, and the corresponding air-side coefficient is 110, based on the liner area. The computed velocity of 747 ft/min based on air-side data, is 48% greater than the value of 504 computed from the water-side heat load. Also, the air-side coefficient of 166.5 is 51.5% higher than the corresponding value of 110 computed from the water-side heat duty. In all the above cases the water-side fouling was assumed to be zero.

If the water-side fouling is assumed to be equal to 0.001, based on the water-side area, the h_o computed from the water-side data is 126, based on the liner area. The outside coefficient, h_o, based on the air-side test data, is 205, based on liner area. The air-coefficient value of 205 is 62.8% greater than the corresponding value of 126 (see Appendix B, Section 6).

Since the water-side heat duty is simple to determine accurately with an orifice plate, and water-temperature drop is easy to measure, there is every reason to believe that the lower air velocity and lower air coefficient computed on the basis of the water-side heat duty are representative of the air-side performance. The resolution of this point is presented in Section II-D of this report.

b. Triangular-pitch unit.

The data obtained on this unit are given in Table III and are of such a nature that both air-side and water-side heat loads may be determined. The calculations are presented in Appendix C and are summarized below.

Apparent air-face velocity	=	626 ft/min
Air-side heat duty	=	12,410,000 Btu/hr
Apparent air-side coefficient	=	174 based on liner area ($r_i = 0$)
Water-side heat duty (assuming one-half the water passed through this unit)	=	8,450,000 Btu/hr

A comparison of the water-side heat duty with the air-side heat duty indicates a discrepancy of 3,960,000 Btu/hr. The air-side heat duty is 47% higher than the water-side heat duty. An air-side coefficient can be computed on the basis of the water-side heat duty, assuming the measured air temperatures are correct. This assumes that the measured air velocities are incorrect. Such a calculation is given in Appendix C, Section 7. The air-side-face velocity so computed is 428 ft/min and the corresponding air-side coefficient based on liner area is 117. The computed air-side velocity of 626 from the air velocity profile is 47% higher than the corresponding velocity of 428 computed from the water-side heat duty. The apparent air-side coefficient of 174 is 49% higher than the corresponding coefficient of 117 computed from the water-side heat duty. In the above discussion the water-side fouling factor was assumed to be zero.

The water-side heat duty appears more reliable than the air-side duty, and the lower air velocity and lower air coefficient give a more reliable indication of the performance of the unit. The discrepancy is resolved in Section II-D of this report.

C. DISCUSSION OF INCONSISTENCIES

The analysis of Thursday's data, presented in Section II-B of this report, indicated that there was reason to believe that the air-side heat duty was too high. Because of the poor method of measuring the water-main velocity, no indication of the discrepancy between the air-side and water-side heat loads could be made.

The analysis of Monday's data on the new three-in line, fourth-row-staggered unit indicated a discrepancy of 48% in heat duty between the air side and water side. The analysis of Monday's data on the new staggered-pitch unit indicated a discrepancy of 47% between the air-side and water-side heat duties. It is therefore presumed that Thursday's air-side heat duty is about 48% greater than the water-side heat duty at the time of the run.

The problem of resolving this dilemma resulted in considerable difficulty. A survey of the situation indicates that a number of possible measurements could have been made in error. These are as follows:

1. The exit air temperature profile was measured, using iron-constantan thermocouples and a Minneapolis-Honeywell potentiometer.

2. The inlet air was measured, using several ASTM distillation thermometers and an iron-constantan thermocouple.

3. The inlet and exit water temperatures were measured, using ASTM distillation thermometers in thermowells packed with grease.

4. The water flow rate was measured using a new sharp-edged orifice, having an ID of 11.25 in. placed in the 16-in., schedule-20-pipe water line.

5. The air velocities were measured, using a Taylor anemometer (in a six-in. stove-pipe duct) and a stop watch.

The above items will be discussed in detail. The twelve thermocouples were made up at The University of Michigan and checked against a Bureau of Standards platinum -platinum rhoduim thermocouple and against Bureau of Standards glass thermometers, using a Leeds and Northrup precision potentiometer as well as the Minneapolis-Honeywell potentiometers used in field testing at Lovington, N. M. (before the field tests were made). Several of the thermocouples were rechecked after returning to the University. The potentiometer was also checked and found still to be operating satisfactorily. In addition to the above, the reference junction temperature was measured at Lovington with a calibrated Sargent thermometer. The air thermocouples were checked against ambient-air thermometers at the side of the tests and were found to be reading correctly.

The temperatures of the air leaving the tube bank was measured, using the thermocouples and potentiometer described above. Stovepipe ducts, six in. in diameter and placed on top of the unit being tested, were used to direct the flow of the air from the tube bank past the thermocouples without mixing with ambient air. There exists the possibility that the readings of the thermocouples were high due to radiation from the stovepipes to the thermocouples. This possibility is discussed in more detail in Section II-D.

The inlet and outlet water temperatures were measured by means of ASTM distillation thermometers. This particular measurement is the most sensitive of the series. Small errors in the difference between the inlet and outlet temperatures make a large difference in the water-side heat duty. This particular type of thermometer was designed for use in laboratory distillation tests and in general is thought to be quite reliable to within $\pm$ 0.25°F.

During the test, the thermometer positions were occasionally reversed (outlet to inlet), with no apparent change in reading. Thus, the inlet and exit water temperatures are believed to be reliable.

A new sharp-edged-orifice plate was installed in the outlet water line of the jacket water cooler during the "turnaround." Sharp-edged orifices are used extensively in industrial flow measurements and are capable of producing precise measurements of the quantity of flow through a pipe. Two possible sources of error can arise in the use of this device. The first can be an incorrect reading of the pressure drop occurring across the orifice. The second could be the use of a rounded-edge-orifice plate. The orifice plate was newly fabricated for this particular test and had sharp edges; the pressure-drop recording instrument was also new and had been installed for the test period. This instrument had been zeroed in on the morning of the test and should be considered reliable. The computed water velocity from the orifice pressure-drop measurements are therefore considered quite reliable.

The air velocity profiles were made by using a Birams type vane anemometer manufactured by the Taylor Instrument Co. The instrument has a calibration card furnished by the company, the calibration in a wind tunnel having been made against a Bureau of Standards calibrated anemometer which had also been certified by the National Physical Laboratory of England. A possible source of error could arise as a result of damage of the instrument in transportation to Lovington, N.M. This possibility of error was eliminated by comparison of the performance of the instrument against a new instrument of the same type after its return to the University. The performance-check runs indicated that the instrument used in the field tests still was working properly. The stop watch used was also checked and found to be keeping correct time. It was therefore concluded that the instrument readings and corresponding time intervals as measured in the field were correct.

In making the air velocity- and temperature-profile measurements it was essential that the ambient-air wind flowing across the top of the table bank be kept away from the anemometer and the thermocouples. In order to accomplish this, stovepipe sections six in. in diameter and twelve in. long were obtained from a local hardware dealer. These sections were used as ducts to conduct the air leaving the top row of tubes to the anemometer and to the thermocouples. The ducts were long enough to permit mixing of the air before reading the anemometers or the thermocouples. The possibility existed that the duct influenced the anemometer readings. An investigation of the influence of duct diameters on anemometer readings was undertaken after the University research group had taken up the problem with the Taylor Instrument Co. and found that the manufacturer of the instrument knew that there was such an influence factor but did not know the magnitude of the correction factor. The determination of this correction factor is discussed in Section II-D of this report.

D. RESOLUTION OF INCONSISTENCIES

1. Review.—In the previous section of this report it was pointed out that the water-side data were apparently more reliable than the air-side data. Examination of the air-side data and test technique indicated that either the air-side temperatures were incorrect or the air-side velocities were incorrect. Two possible influence factors could have caused the inconsistent computed heat load obtained from the air-side data. These were (a) radiation affecting the thermocouple readings and (b) a duct factor affecting the anemometer readings. Either one or both of these factors could have affected the data. These factors will be discussed in detail.

2. Radiation Correction Factor.—During the summer a number of experiments were conducted on the roof of the East Engineering Building, using a 6-in.-stovepipe duct and one of the thermocouples used in the field tests. The first tests made were misleading, as it appeared that there might have been a sizable radiation factor. Later tests showed that the rise in temperatures attributed to the sun was actually a conduction heat transfer heating up stagnant air in the duct. In the early tests no forced draft was used. Later it was found that if the air was in turbulent flow past the duct and through the duct, this temperature rise largely disappeared. We were unable to determine exactly what the radiation correction factor amounted to, but it is known that under certain conditions such an error may be introduced. Since the correction factor appeared to be slight, it could never begin to account for the 50% discrepancy in heat loads. It was therefore concluded that the error was in the air velocity measurement.

3. Air Velocity Correction Factor.—The possibility of a large duct correction factor took on added importance since an air test apparatus for studying bond resistance of bimetal tubes was under investigation by the research group at the University. A major investigation of this factor was undertaken and the results indicated that a large correction factor is introduced when four- and six-in. ducts are used with a four-in. Taylor anemometer.

This duct investigation resulted in Report No. 37 entitled "Investigation of the Performance of Vane-Type Anemometers in a Four-Inch Duct." This investigation was extended to cover the six-in. duct used in the field tests. The investigation involved the use of a water-calibrated air rotameter and critical-flow orifices for measuring the flow of air through the anemometer. The test arrangement and test procedure used in conducting the investigation are covered in Report No. 37 and will not be reproduced here. The results of the tests using the anemometer in a six-in. duct resulted in Fig. 2. The slope of the calibration line in Fig. 2 is 0.636. The actual air velocity flowing through the six-in. duct is 63.6% of that indicated by the anemometer when placed in the duct. The measured air velocities are therefore 57% higher than the actual velocities. This type of anemometer reads correctly, according to the manufacturer, when used in a duct six to eight times the diameter of the anemometer (24- to 32-in. duct for the test anemometer). Using the instrument in a smaller duct results in a change in velocity profile of the air moving past the vanes of the instrument, which in turn changes the performance of the instrument. This is due to the fact that this is an inertial type of instrument.

The recomputation of the field test data is presented in Appendix D and is summarized in Section II-E of this report.

E. REVISED ANALYSIS OF THE FIELD TEST DATA

1. General Considerations.—The recalculations of the field test data, including the 0.636 duct correction factor on the air anemometer readings, are presented in Appendix D and are summarized below.

2. Thursday's Data.—

a. East bay.

Reference is made to Appendix A where the indicated standard air velocity is 508 ft/min. Introducing the duct correction factor of 0.636 reduces this velocity to 323 ft/min. This reduces the indicated air-side heat load of 13,000,000 Btu/hr to a corrected value of 8,290,000 Btu/hr. The corresponding water velocity computed from the air-side data is 5.95 ft/sec. The revised water coefficient becomes 1766. The corrected overall coefficient based on liner area is 77 as compared to the uncorrected value of 121. The revised air-side coefficient is 90.8, based on liner area, as compared with the earlier value of 154.5

b. West bay.

Reference is made to Appendix A where the indicated standard air velocity is 445 ft/min. Introducing the duct correction factor of 0.636

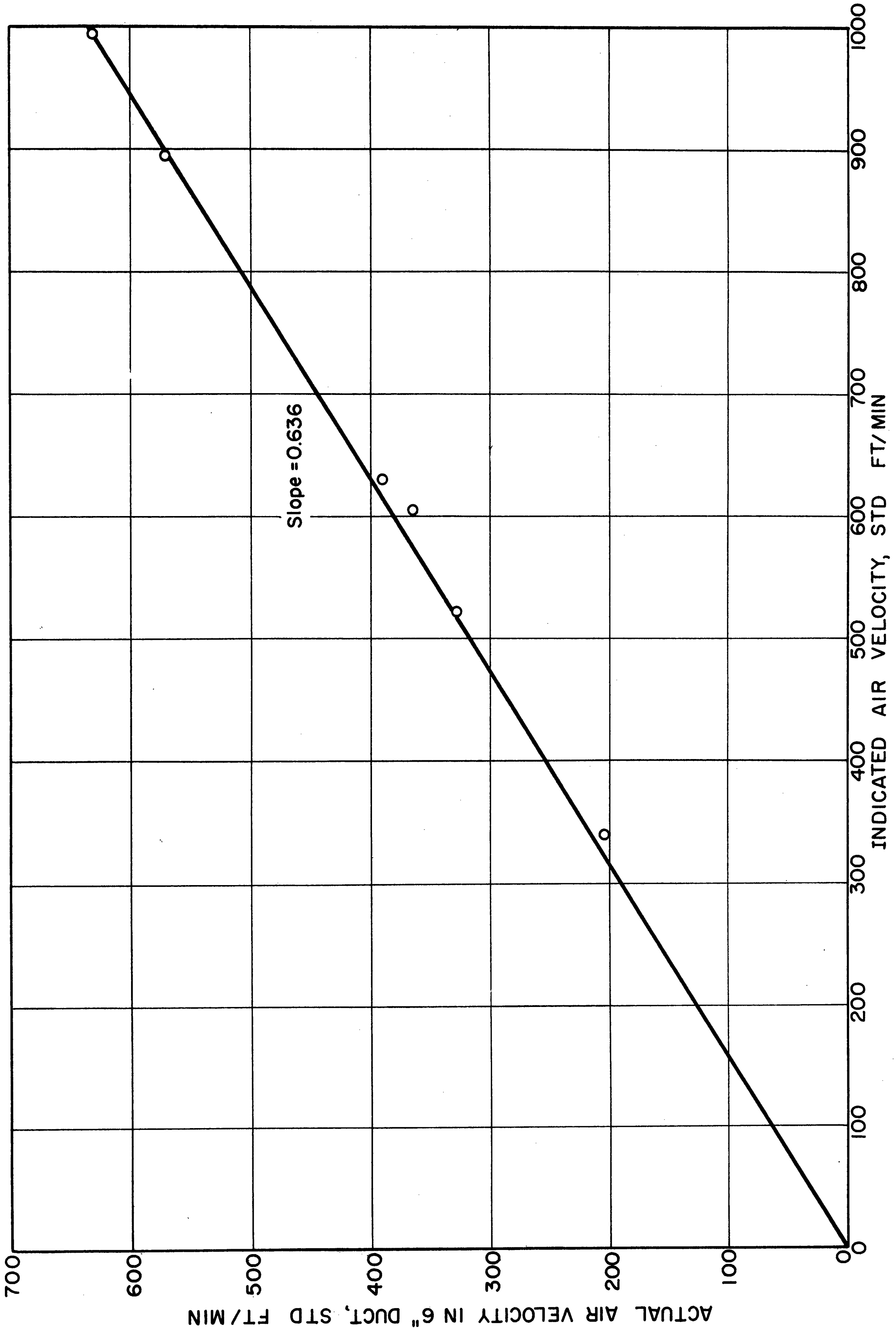

Fig 2. Calibration of a four-inch vane-type anemometer in a six-inch-diameter duct.

reduces this velocity to 283 ft/min. This reduces the indicated air-side heat load of 9,730,000 Btu/hr to 6,180,000 Btu/hr. The corresponding water velocity computed from the revised air velocity is 5.09 ft/sec. The revised water coefficient becomes 1552. The corrected overall coefficient based on liner area is 52.7 as compared to the earlier value of 82.8. The revised air-side coefficient is 59.1, based on liner area, as compared with the earlier value of 97.7.

c. Average of east bay and west bay.

The computed results given above are compared and averaged below.

	East	West	Avg
Face Velocity	323	283	303
Heat Duty	8,290,000	6,180,000	7,235,000
U_o	77.0	52.7	64.9
h_o	90.8	59.1	75

d. Discussion.

The average air-face velocity of 303 ft/min is about one-half the design-face velocity of 600 ft/min. This velocity is believed to be on the edge of the transition region between laminar and turbulent flow. Inspection of the data for this unit, given in Table I, indicates that large areas of the two bays have velocities significantly below the average values. It is possible therefore that the air flowing past the tubes in these regions of low velocity would not have a fully established turbulence pattern. The air-film coefficients for these areas would be much lower than that predicted by any correlation obtained from turbulent air-film data and extrapolated back to this velocity.

3. Monday's Data on Three-Rows-in-Line, Fourth-Row-Staggered Unit, Assuming a Water-Side Fouling Factor of 0.001.—Reference is made to Appendix B where the average standard air-face velocity was originally computed to be 747 ft/min. Introducing the duct correction factor of 0.636 gives 475 ft/min. This results in a new air-side heat duty of 8,840,000 Btu/hr (see Appendix D). The corrected water velocity turns out to be 5.58 ft/sec. The revised overall coefficient is 97, based on liner area, and the recomputed air-side coefficient is 118.5, based on the liner area.

A comparison may be made between the revised computed performance given above and the corresponding information computed from the water-side heat load. The water-side heat duty (see Appendix B) was computed to be 9,380,000 Btu/hr as compared with the revised value from the air-side data of 8,840,000 Btu/hr. The air-side heat duty is only 5.76% below the water-side heat duty. The check is reasonably close. In Appendix B, a calculation of the air-side coefficient is made, based on the water-side heat load and the air temperatures, assuming the air velocity was incorrect (later proved by laboratory test). The computed air velocity turned out to be 504 ft/min as compared to the revised air velocity of 475 ft/min. The computed U_o based on liner area is 103 as compared to the revised air velocity U_o value of 97. The corresponding h_o based on liner area was 126 as compared to the revised air-side h_o value of 118.5.

It is quite apparent that the introduction of the anemometer duct correction factor brings the air-side performance on this unit into reasonable agreement with the water-side predicted performance.

4. Monday's Data on the Triangular-Pitch Unit, Assuming a Water-Side Fouling Factor of 0.001.—Reference is made to Appendix D, Section B-2, where the air-side data are recomputed, using the duct correction factor of 0.636. The uncorrected air velocity (std) was 626 ft/min, whereas the corrected value becomes 398 ft/min. The original computed air-side heat load of 12,410,000 Btu/hr is reduced 7,900,000 Btu/hr. The uncorrected overall coefficient of 159 is reduced to 101.3. The uncorrected air-side coefficient, based on liner area as computed in Appendix C, was 216. The corrected value as given in Appendix D, Section B-2,is 125, based on liner area.

In Appendix C a calculation of the air-side performance, using the water-side heat load and inlet and exit air temperatures, is presented. The calculations indicate a computed air-side velocity of 428 std ft/min, an overall coefficient of 108, and an air-side coefficient of 135 based on liner area. These values compare favorably with the corresponding corrected air-side values of $V = 398$ ft/min, $U_o = 101.3$, and $h_o = 125$.

It is interesting to compare the revised air-side heat load of 7,900,000 Btu/hr with the water-side heat load of 8,450,000 Btu/hr. The air-side heat load is only 6.5% below the water-side heat load.

It is quite obvious that the introduction of the anemometer duct correction factor brings the air-side performance on this unit also into reasonable agreement with the water-side predicted performance. The discrepancy between the air-side and water-side heat loads for the three-in.-line, fourth-row-staggered unit amounted to 5.76% in the same direction, as compared to 6.5% for the triangular-pitch unit for an average discrepancy of only 6.1%.

5. Monday's Data Re-evaluated, Assuming No Fouling on the Water Side.—

a. Three-in-line, fourth-row-staggered unit.

The air-side coefficient recomputed from the air-side data, assuming no fouling on the water side, is 105, based on liner area, or 7.64, based on the outside area (see Appendix D). These values compare with 118.5 and 8.65, with an assumed water-side fouling factor of 0.001. The assumption of no water-side fouling results in a decrease of 12.8% in the computed air-side coefficients. This emphasizes the importance of water-side fouling in analyzing the data.

b. Triangular-pitch unit.

The air-side coefficient recomputed from the air-side data, assuming no fouling on the water side, is 110, based on the liner area, or 8.02, based on the outside area (see Appendix D). These values compare with 125 and 9.13, with an assumed fouling factor of 0.001 on the water side. The assumption of no water-side fouling results in a decrease of 13.6% in the computed air-side coefficients.

c. Discussion.

Examination of the above results indicates a significant difference in the computed performance of the units when the inside fouling factor is assumed to be zero or is assumed to be 0.001. Fouling rates vary exponentially as a function of time. Therefore, there probably is a stronger argument in favor of the case in which no fouling is assumed, since in all probability several months presumably would have to pass before a degree of fouling equal to 0.001 would be developed.

6. Summary.—For comparison purposes the final computed performance of the units are summarized below:

TABLE IV

TABULATED RESULTS

Thursday's Data

East bay		
V (air-face velocity)	=	323 std ft/min
Q (air side)	=	8,290,000 Btu/hr
U_o (liner area)	=	77.0
$(h_o)_L$ (liner area)	=	90.8
h_o (outside area)	=	6.63
$(\Delta t)_{LM}$	=	47.8°F
West bay		
V (air-face velocity)	=	283 std ft/min
Q (air side)	=	6,180,000 Btu/hr

U_o (liner area)	=	52.7
$(h_o)_L$ (liner area)	=	59.1
h_o (outside area)	=	4.32
$(\Delta t)_{LM}$	=	52.3°F

Average Performance of Unit on Thursday, April 28, 1955

V (air-face velocity)	=	303 std ft/min
Q (air side)	=	7,235,000 Btu/hr
U_o (liner area)	=	64.9
$(h_o)_L$ (liner area)	=	75.0
h_o (outside area)	=	5.48
$(\Delta t)_{LM}$	=	50.05°F

Monday's Data, Three-Rows-in-Line, Fourth Row Staggered Bay

V (face velocity) air side	=	475 std ft/min
V (face velocity) water side	=	504 std ft/min
V (face velocity) average	=	490 std ft/min
Q (air side)	=	8,840,000 Btu/hr
Q (water side)	=	9,380,000 Btu/hr
Q (average)	=	9,110,000 Btu/hr
U_o (air side, liner)	=	97.0
U_o (water side, liner)	=	103
U_o (average, liner)	=	100.0
$(\Delta t)_{LM}$	=	40.5°F

Assuming an inside fouling factor of zero:

$(h_o)_L$ (liner) air side	=	105
$(h_o)_L$ (liner) water side	=	110
$(h_o)_L$ (average)	=	107.5
h_o (outside) air side	=	7.65
h_o (outside) water side	=	8.05
h_o (outside) average	=	7.85

Assuming an inside fouling factor of 0.001:

$(h_o)_L$ (liner) air side	=	118.5
$(h_o)_L$ (liner) water side	=	126
$(h_o)_L$ (liner) average	=	122.3
h_o (outside) air side	=	8.65
h_o (outside) water side	=	9.18
h_o (outside) average	=	8.91

Monday's Data, Triangular-Pitch Unit

V (face velocity) air side	=	398 std ft/min
V (face velocity) water side	=	428 std ft/min

V (face velocity) average	=	413 std ft/min
Q (air side)	=	7,900,000 Btu/hr
Q (water side)	=	8,450,000 Btu/hr
Q (average)	=	8,175,000 Btu/hr
U_o (air side, liner)	=	101.3
U_o (water side, liner)	=	108
U_o (average, liner)	=	104.7
$(\Delta t)_{LM}$	=	34.7°F

Assuming an inside fouling factor of zero:

$(h_o)_L$ (liner) air side	=	109.8
$(h_o)_L$ (liner) water side	=	117
$(h_o)_L$ (liner) average	=	113.4
h_o (outside) air side	=	8.02
h_o (outside) water side	=	8.56
h_o (outside) average	=	8.29

Assuming an inside fouling factor of 0.001:

$(h_o)_L$ (liner) air side	=	125
$(h_o)_L$ (liner) water side	=	135
$(h_o)_L$ (liner) average	=	130
h_o (outside) air side	=	9.13
h_o (outside) water side	=	9.85
h_o (outside) average	=	9.49

III. HEAT TRANSFER CORRELATIONS FOR TUBE BANKS

A. CORRELATIONS FOR AIR-SIDE COEFFICIENTS FOR THREE-IN-LINE, FOURTH-ROW-STAGGERED UNITS

1. 651-A-B-C.—The Wolverine Tube Division has published a series of three brochures entitled "Data Sheet 651-A," published July 1, 1950, "Data Sheet 651-B," published July 1, 1950, and "Data Sheet 651-C," published July 1, 1950, and revised February 1, 1951. This series of data is not reproduced in this report. The data as published are presented in tabular form, giving the length of various-sized finned tubes having various numbers of fins per inch required to transfer 10,000 Btu/hr (condensing steam) as a function of the maximum air velocity and the log mean Δt driving force.

The tabulated data are of such a nature that one can readily compute the air-side coefficients used in making up the tables. A sample calculation is presented in Appendix E.

Figures 3, 4, and 5 give the h_o-vs-V_{max} curves for 651 A, B, and C, respectively. Figure 6 presents a single line representing all the three curves from Figs. 4, 5, and 6, which has the following equation:

$$h_o = 0.20(V_{max})^{0.6}$$

where

h_o = air-side coefficient based on outside area and
V_{max} = maximum air velocity at the minimum cross section, ft/min.

This equation can be converted to a corresponding relationship for a 9-fins-per inch tube, giving the air-side coefficient, h_o, based on the liner area, as a function of face velocity. The relationship is computed as follows:

$$(h_o)_{liner} = 13.7(h_o)_{outside\ area}$$

$$V_{max} = 2.428\ V_{face}.$$

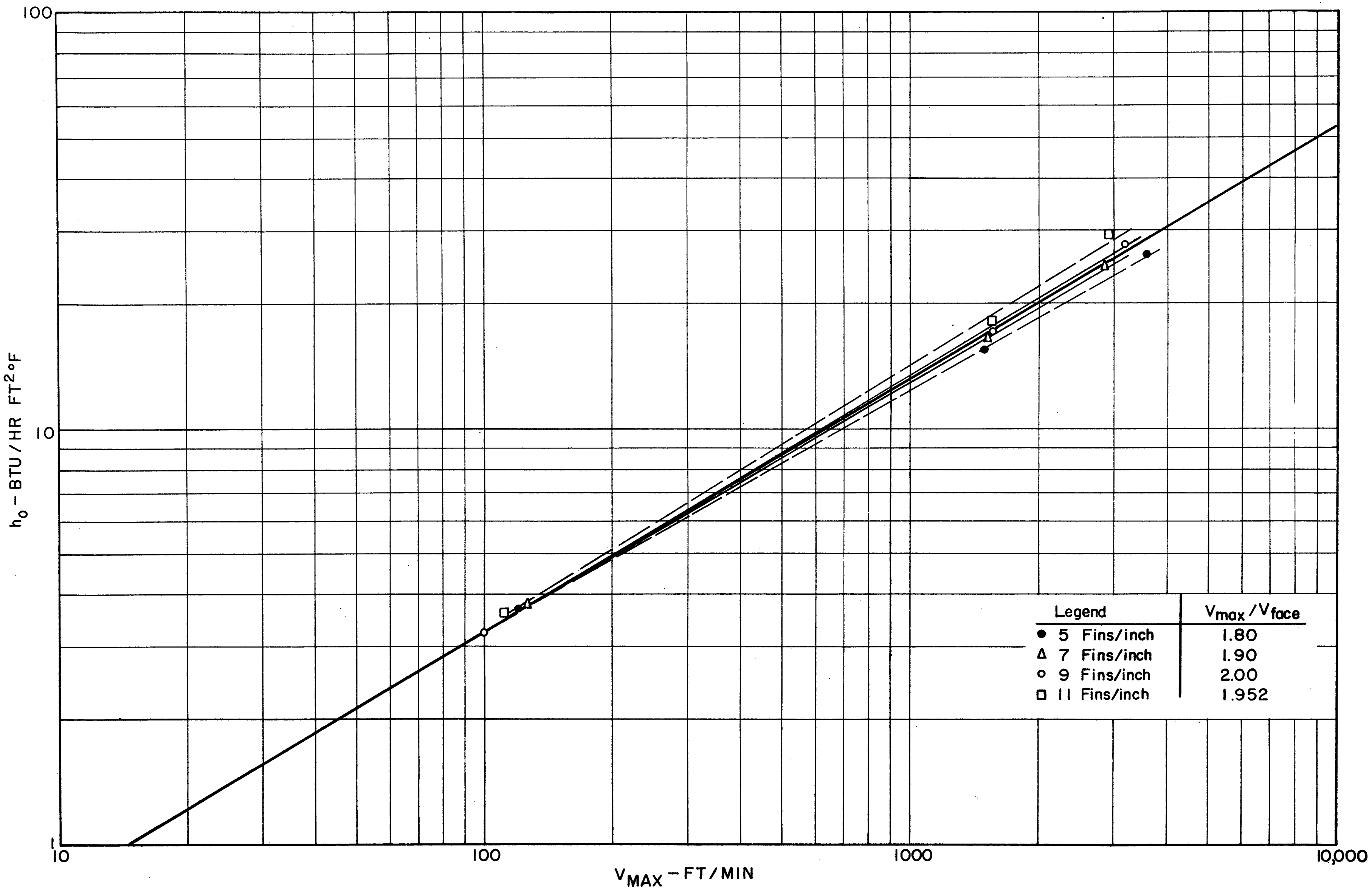

Fig. 3. Wolverine tube data sheet 651A, published July 1, 1950. Outside coefficient, calculated from data of tables 2 and 4, 6 and 8, 10 and 12, and 14 and 16.

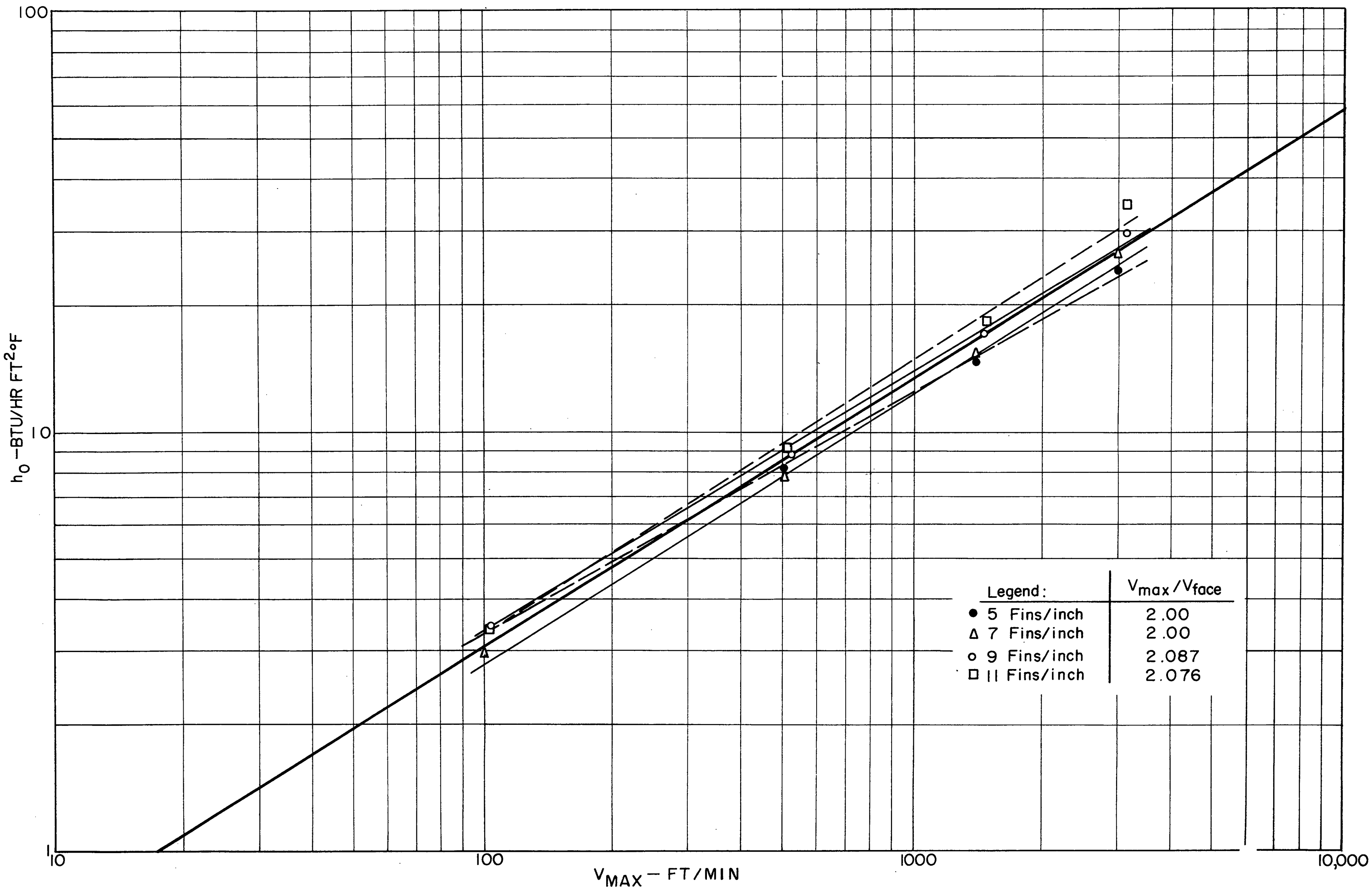

Fig. 4. Wolverine tube data sheet 651B, published July 1, 1950. Outside coefficient, calculated from data of tables 2 and 4, 6 and 8, 10 and 12, and 14 and 16.

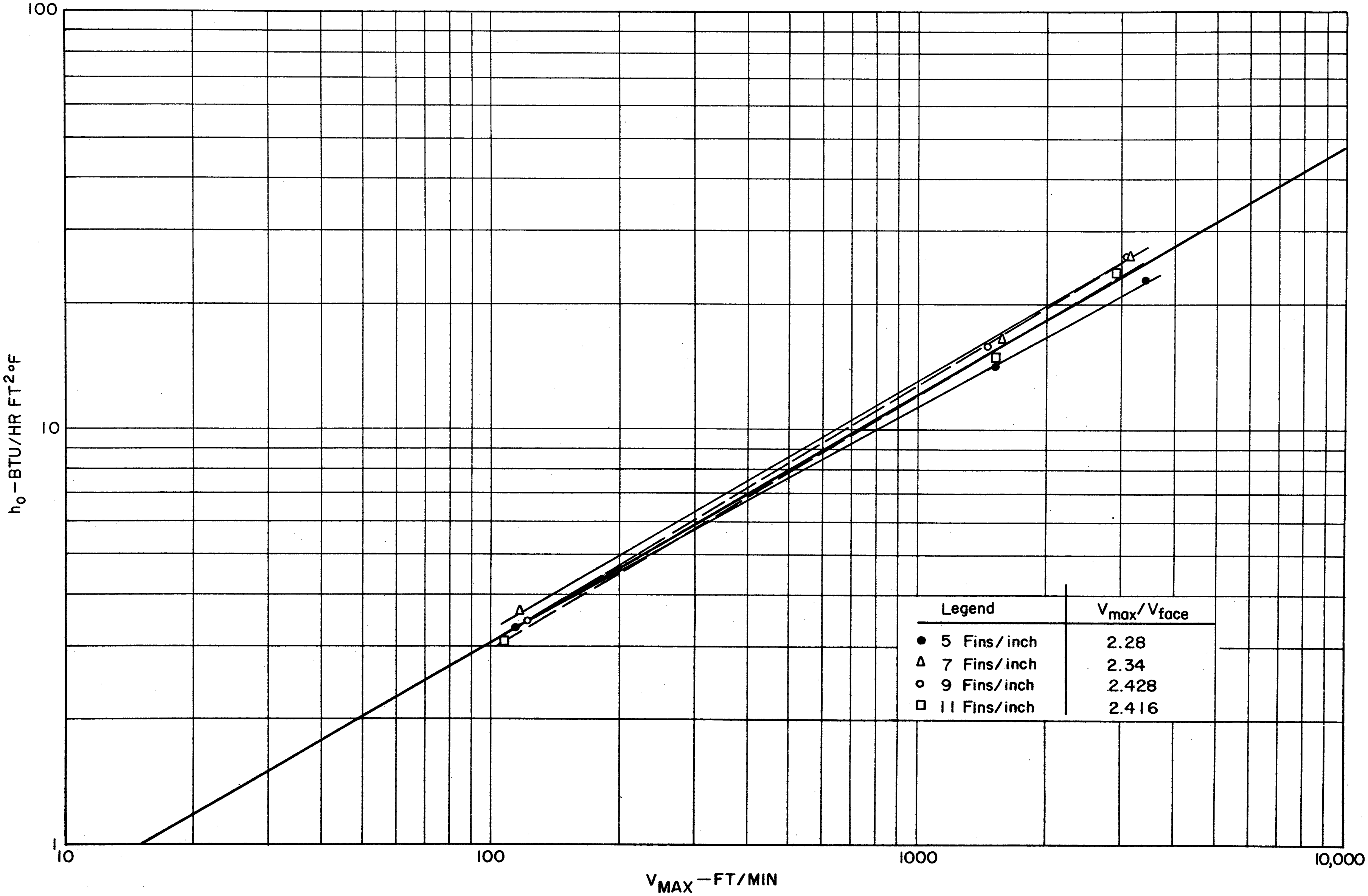

Fig. 5. Wolverine tube data sheet 651C, published July 1, 1950. Outside coefficient, calculated from data of tables 2 and 4, 6 and 8, 10 and 12, and 14 and 16.

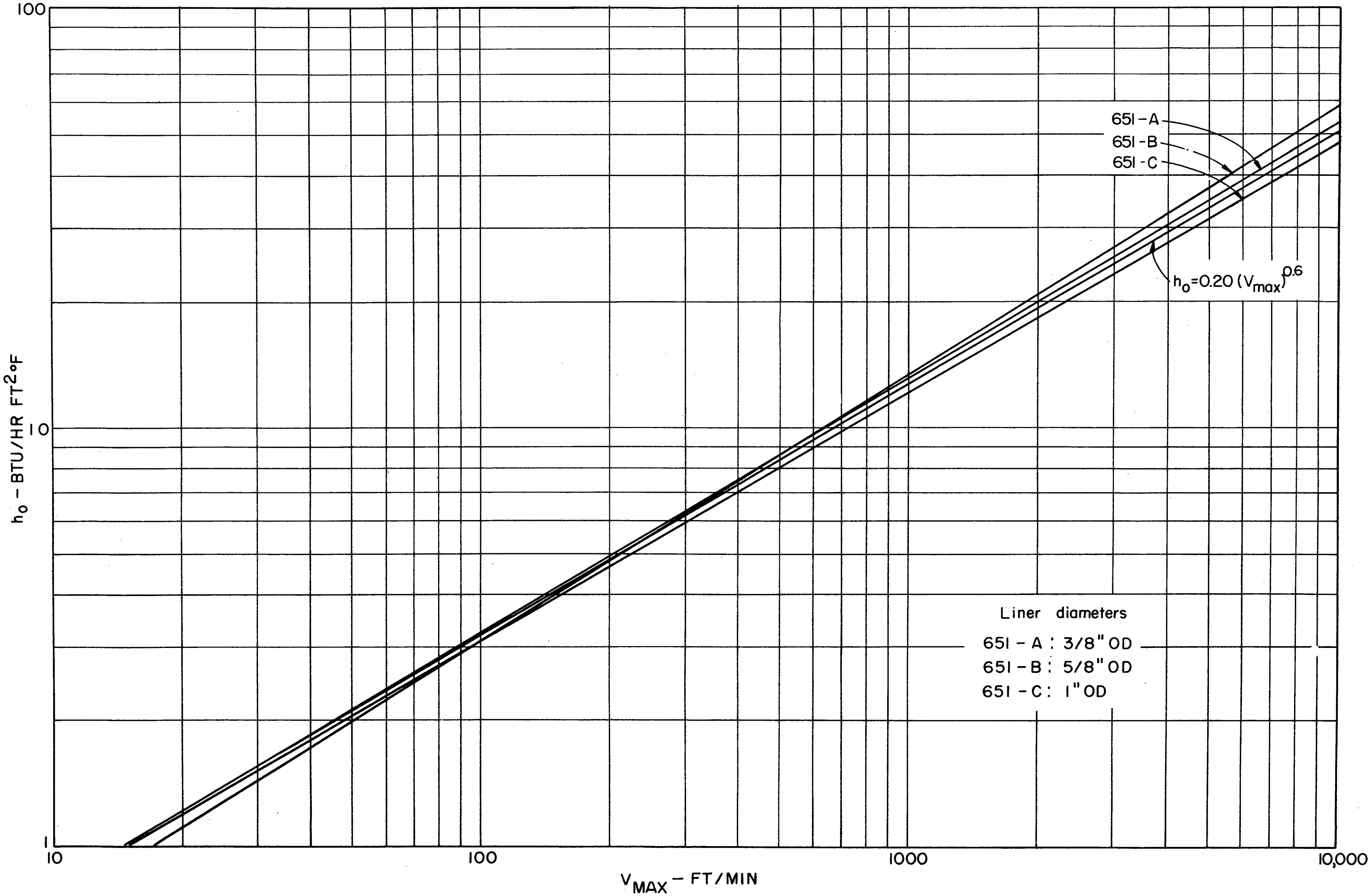

Fig. 6. Wolverine tube data sheets 651A, 651B, and 651C, published July 1, 1950. Outside coefficients, calculated from data of tables 2 and 4, 6 and 8, 10 and 12, and 14 and 16.

Substituting,

$$(h_o)_{liner} = (13.7)(0.20)(2.428\ V_{face})^{0.6}$$

$$\therefore \quad (h_o)_{liner} = 4.67(V_{face})^{0.6}$$

The air-side heat transfer coefficients were computed from the tables indicated by assuming a steam-condensing coefficient of 1000 with no allowance for fouling, metal, or bond resistance. The resistance of the metal and of the fouling factors for steam condensing to air are slight in comparison with the resistances of the air-side film coefficient. The effect of bond resistance is unknown.

Examination of Figs. 3, 4, 5, and 6 indicates that a single correlating line was probably used by someone in the preparation of the tables. The correlation report, Report No. 30, presents an equation (on page 6 of the report) which indicates the influence of the number of fins per inch on the air-side coefficient. This equation indicates that the coefficient varies inversely with the square root of the number of fins. No such factor was used in 651A, B, or C.

Figures 7, 8, and 9 graphically present the pressure-drop information for the air side of the tubes as given in 651A, B, and C. Figure 10 summarizes the curves presented in Figs. 7, 8, and 9. In Fig. 10 the air-side pressure-drop equation for a 2-in.-OD tube having 9 fins per inch and a 1-in. liner is

$$\Delta P = 1.84 \times 10^{-6} (V_{face})^{1.89} ,$$

where

ΔP = inches of water for four rows deep (three in. line, fourth row staggered) and

V_{face} = face velocity, ft/min.

a. Prediction of coefficient and pressure drop, using 651, Thursday's test data.

Reference is made to Table IV in which the average face velocity is reported as 303 ft/min with an h_o on liner area of 75.0. Substituting this face velocity into the 651A, B, and C equation gives

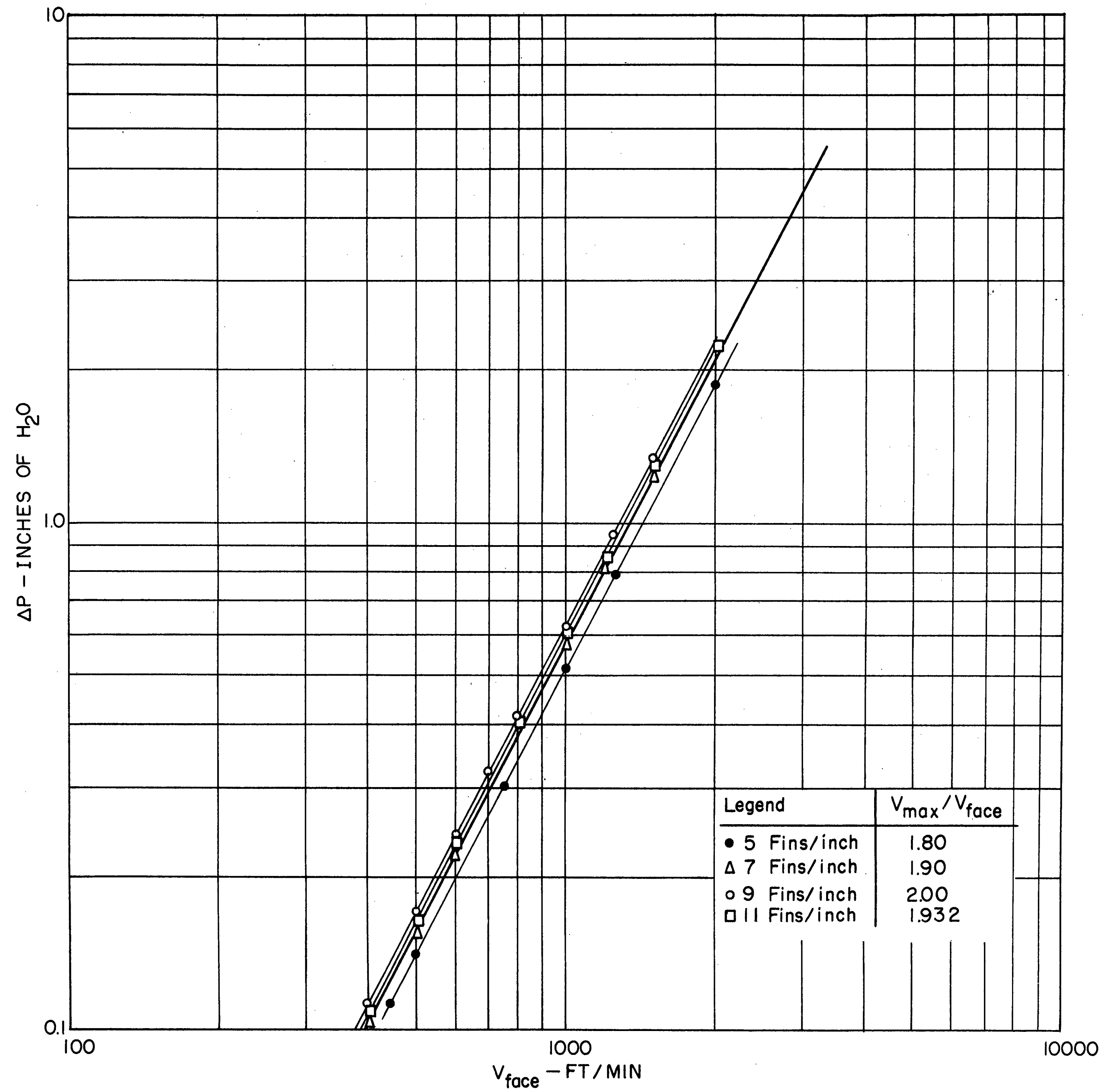

Fig. 7. Wolverine tube data sheet 651A, published July 1, 1950. Air-side pressure drop, calculated from data of tables 5, 9, 13, and 17.

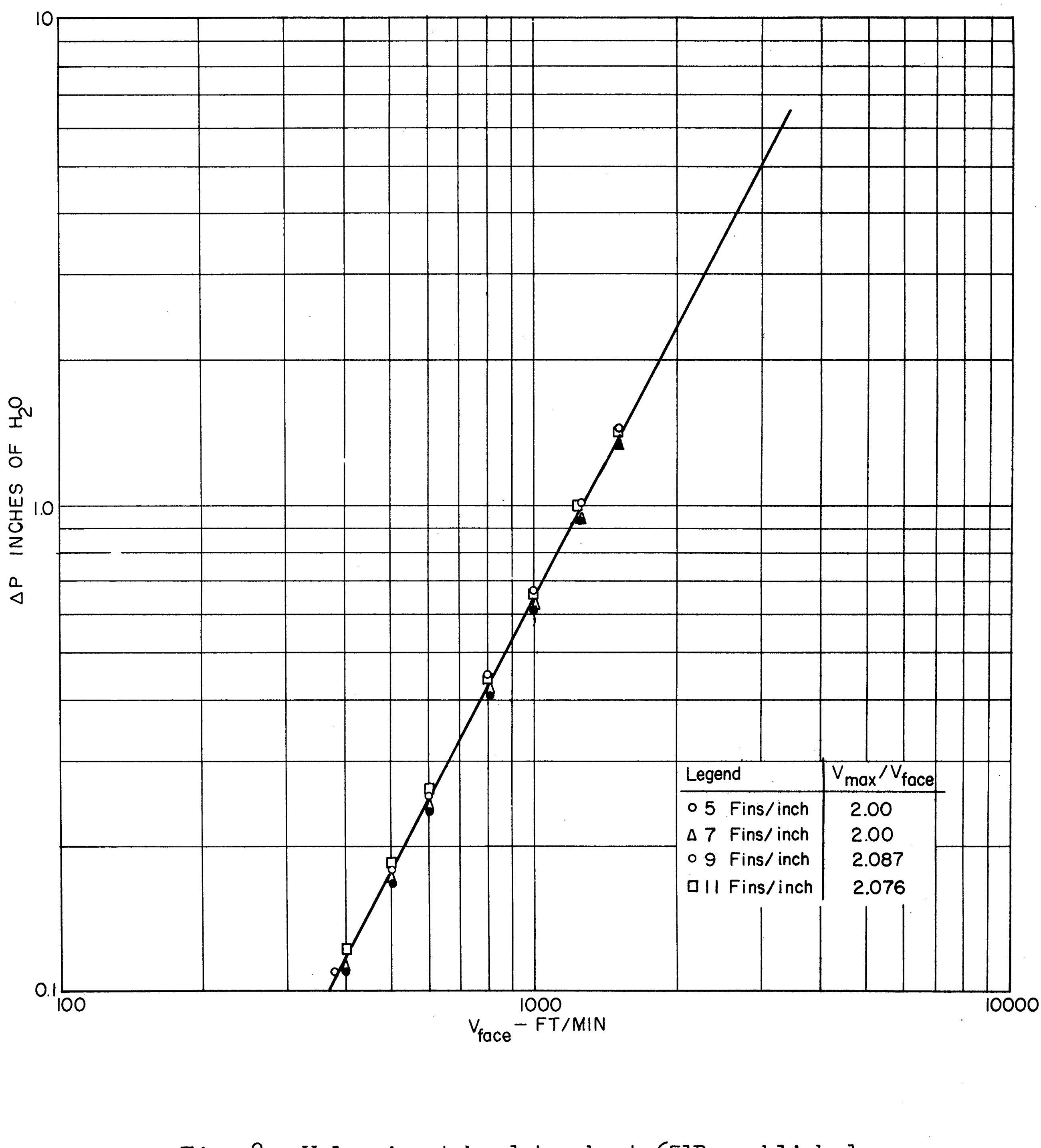

Fig. 8. Wolverine tube data sheet 651B, published July 1, 1950. Air-side pressure drop, calculated from data of tables 5, 9, 13, and 17.

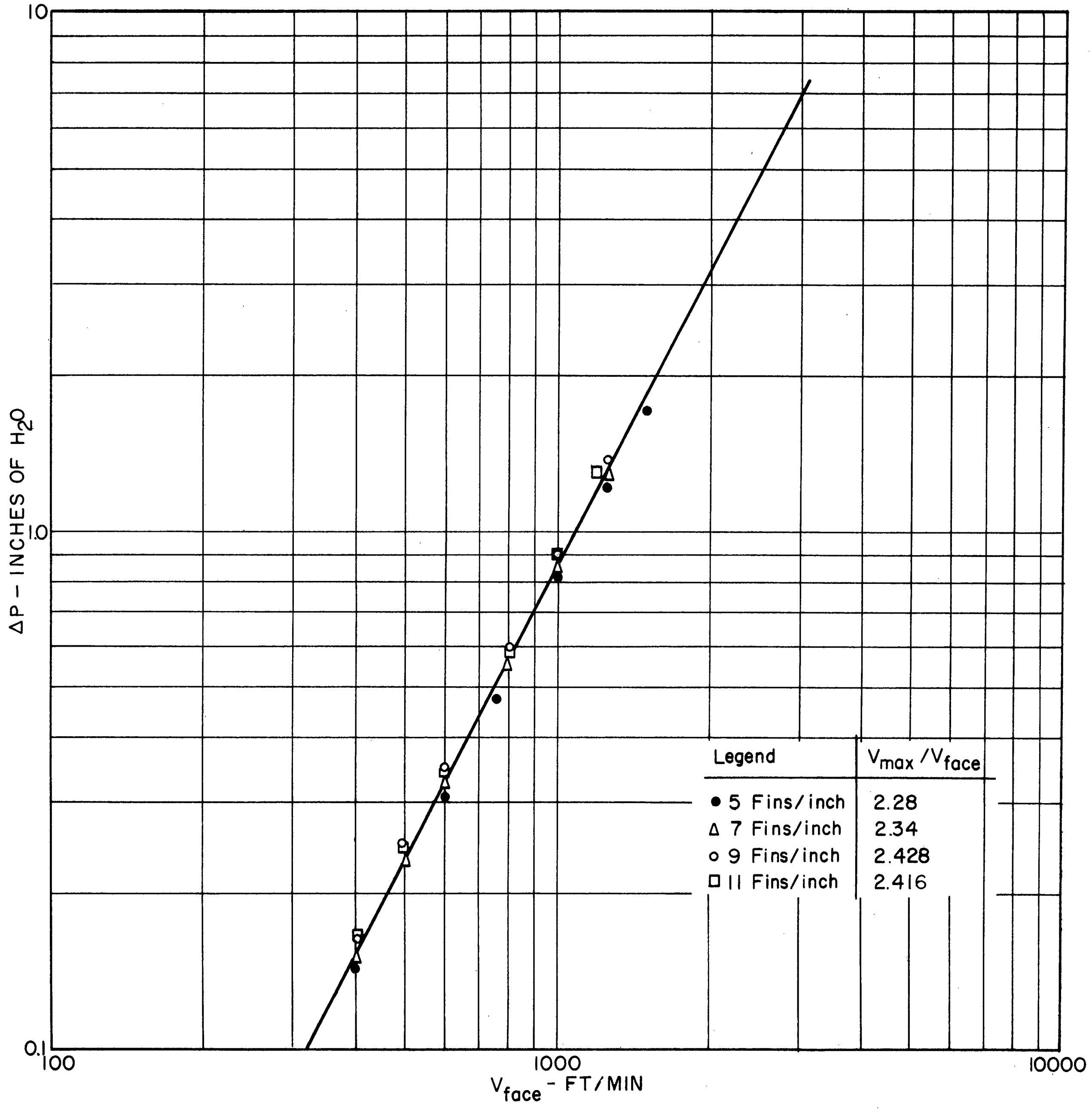

Fig. 9. Wolverine tube data sheet 651C, published July 1, 1950. Air-side pressure drop, calculated from data of tables 5, 9, 13, and 17.

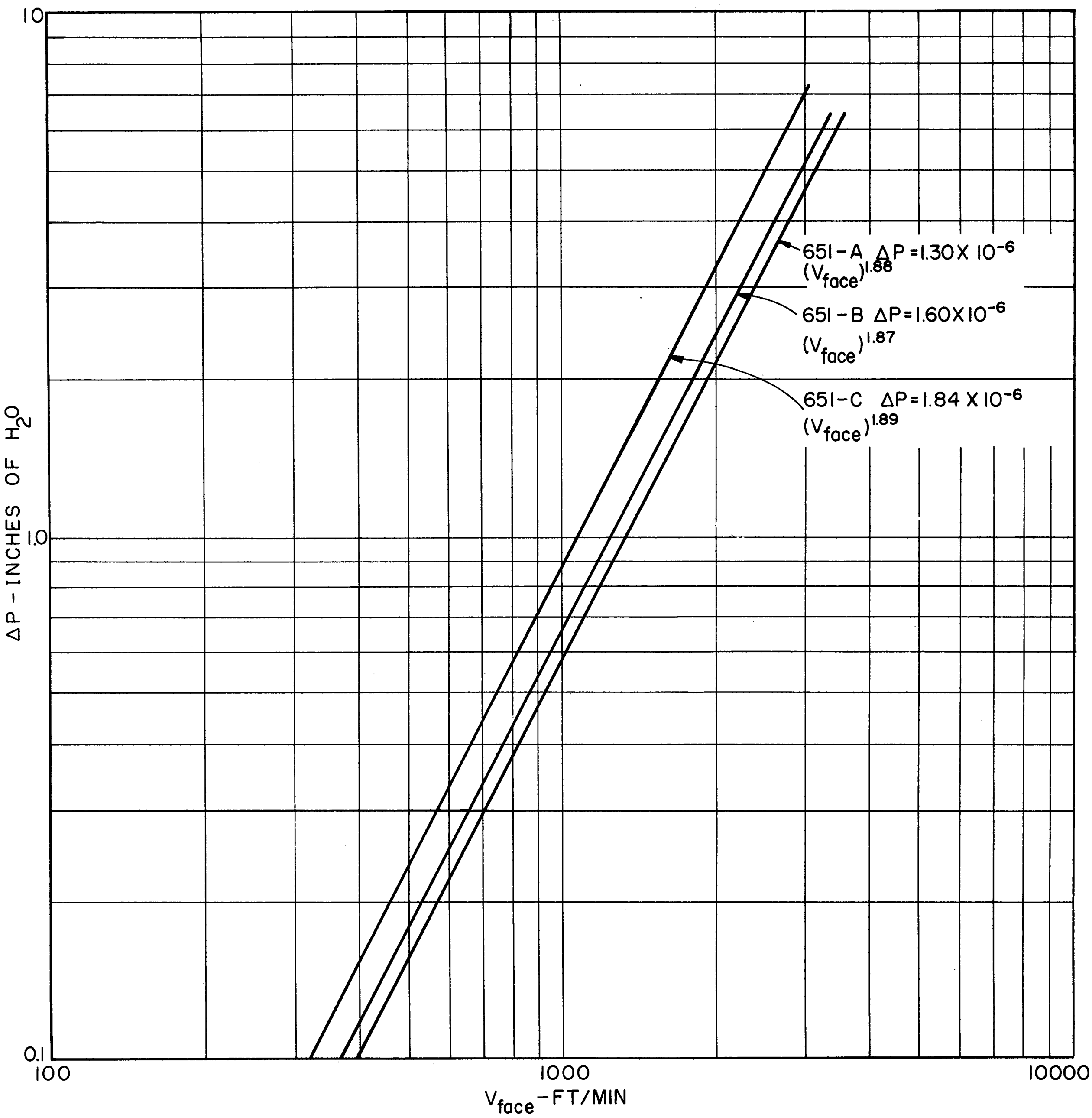

Fig. 10. Wolverine tube data sheets 651A, 651B, and 651C. Air-side pressure drop, calculated from data of tables 5, 9, 13, and 17.

$$(h_o)_{liner} = 4.67(V_{face})^{0.6}$$
$$= 4.67(303)^{0.6}$$
$$= 145$$

The 651 value of 145 is 93% greater than the average value obtained from the field test data.

The air-side pressure drop predicted by 651C is

$$\Delta P = 1.84 \times 10^{-6} (V_{face})^{1.89} \text{ in. of water}$$
$$= 1.84 \times 10^{-6} (303)^{1.89}$$
$$= 0.09 \text{ in. of water.}$$

This value will be discussed in Section VI.

b. Prediction of coefficient and pressure drop, using 651, Monday's test data for 3-in-line, fourth-row-staggered unit.

In Table IV the air-side face velocity is given as 475 ft/min with an h_o value of 105 ($r_i = 0$) based on the liner area. The 651 equation gives

$$(h_o)_{liner} = 4.67(475)^{0.6}$$
$$= 188.$$

The 651 equation gives a value that is 79% greater than the field-test-data value.

The air-side pressure drop predicted by 651C is

$$\Delta P = 1.84 \times 10^{-6} (475)^{1.89}$$
$$= 0.202 \text{ in. of water.}$$

This value will be discussed in Section VI of this report.

c. Prediction of coefficient and pressure for triangular-pitch unit, using 651, Monday's test data.

The 651A, B, and C air-side-coefficient equation and air-side pressure-drop equation are limited to three-in-line, fourth-row-staggered units and therefore are not applicable to this test data.

2. American Locomotive Co. Data.—The only test data in the University Research Project 1592 files bearing on the performance of three-in-line, fourth-row-staggered units were supplied by the American Locomotive Co. under the date of September 15, 1949. Figure 11 presents the ALCO data for 1-in.-liner, 2-in.-OD, 7-fins-per-inch Wolverine tubes. The figure presents h_o, based on the outside area, as a function of face velocity and the pressure drop in inches of water for the four-row-deep bank, also as a function of face velocity.

Unfortunately, the line which correlates the test data is for a 7-fins-per-inch tube. Equation 6, page 6, of the correlation report (Report No. 30) indicates that the air-side coefficient varies inversely with the square root of the number of fins per inch. Another line, corrected for the difference between 7 fins per inch and 9 fins per inch, using this equation, is presented in Fig. 11. Also indicated is the 651C correlating line. It is evident that the 651C line gives considerably higher values than indicated by the ALCO data. The equation of the ALCO test-data line is

$$h_o \quad = \quad 0.20\, V_{face}^{0.6} \text{ (for a 7-fins-per-inch tube),}$$

where h_o = air-side coefficient based on outside area. The corresponding equation in terms of liner area is

$$(h_o)_{liner} \quad = \quad 2.16\, V_{face}^{0.6} \text{ (for a 7-fins-per-inch tube).}$$

The equation of the ALCO line converted over to 9 fins per inch is

$$(h_o)_{liner} \quad = \quad 2.42\, V_{face}^{0.6} \text{ (for a 9-fins-per-inch tube).}$$

In Section II of this report the equation of the 651 coefficient line was given as

$$(h_o)_{liner} \quad = \quad 4.67\, V_{face}^{0.6} \ .$$

Taking the ratio of the ALCO equation for a 9-fins-per-inch tube to the equation of 651 gives

$$\frac{\text{ALCO}(h_o)_{liner}}{651(h_o)_{liner}} \quad = \quad \frac{2.42}{4.67} \quad = \quad 0.52 \ .$$

The ALCO line therefore gives a value which is 52% of that predicted by 651.

The pressure-drop curve of ALCO, as indicated in Fig. 11, gives excellent agreement with the pressure drop predicted by the 651C line on the same plot.

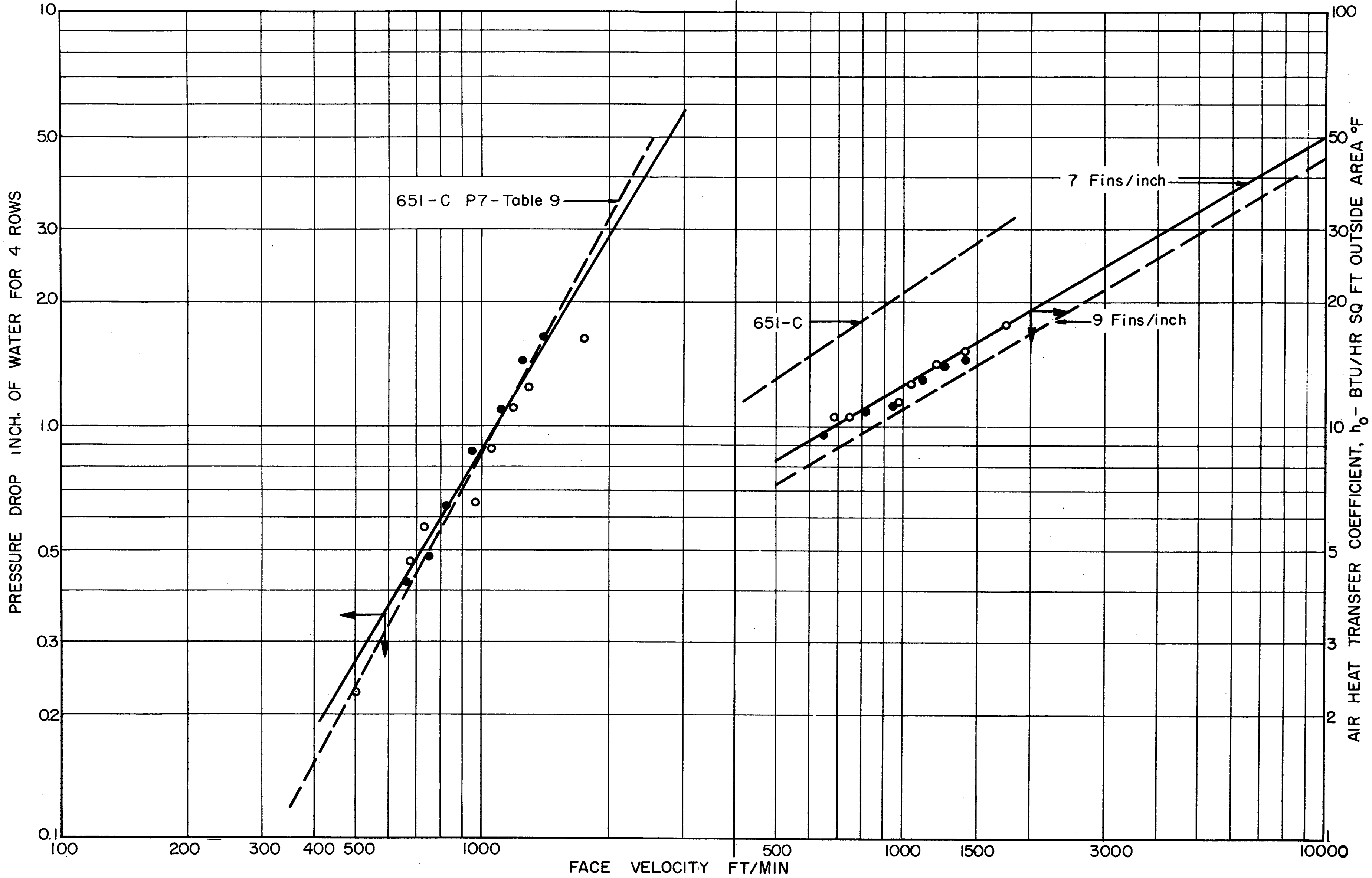

Fig. 11. ALCO Products Division, American Locomotive Company, heat transfer and pressure drop, Wolverine bi-metal x-fin tubes, 1" OD liner (7 fins) 3 in line, 4th staggered, September 15, 1949.

a. Thursday's test data, using ALCO equation.

Reference again is made to Table IV in which the average of Thursday's data gave a face velocity of 303 ft/min and an h_o of 75, based on the liner area. The ALCO line converted to 9 fins per inch gives the following h_o on liner area for this velocity:

$$
\begin{aligned}
(h_o)_{liner} &= 2.42\ (V_{face})^{0.6} \\
&= 2.42(303)^{0.6} \\
&= 75.
\end{aligned}
$$

The corresponding air-side pressure drop, as predicted by the ALCO curve of Fig. 11 for 7 fins per inch, is 0.115 in. of water as compared to 0.09 in. of water by 651C.

b. Monday's test data, using ALCO equation for three-in-line, fourth-row-staggered unit.

In Table IV the air-side-face velocity is given as 475 ft/min with an h_o value of 105 ($r_i = 0$), based on the liner area. The ALCO equation predicts

$$(h_o)_{liner} = 2.42\ (475)^{0.6} = 97.5.$$

The ALCO pressure-drop curve, Fig. 11, predicts a value of 0.248 in. of water as compared to 0.202 in. by 651C.

c. Monday's test data, using ALCO data for triangular-pitch unit.

The ALCO information is strictly limited to three-in-line, fourth-row-staggered units and is not applicable to this unit.

3. Happy Co. Rating Curve.—The Happy Co. has furnished a rating sheet which is given in Fig. 12. The figure presents the outside coefficient h_o, based on the liner area, as a function of the face velocity in std ft/min. The figure also presents the pressure drop in inches of water per row as a function of the same velocity. The data are presented for a 2-in.-OD tube having a 1-in. liner and 9 fins per inch.

a. Thursday's test data, using the Happy Co. rating curves.

Table IV gives an average face velocity of 303 ft/min and an average h_o of 75, based on the liner area. The Happy Co. heat transfer curve, Fig. 12, predicts an h_o of 130, based on the liner area, for this face velocity. The Happy Co. pressure-drop curve predicts a pressure drop of 0.024 in. of

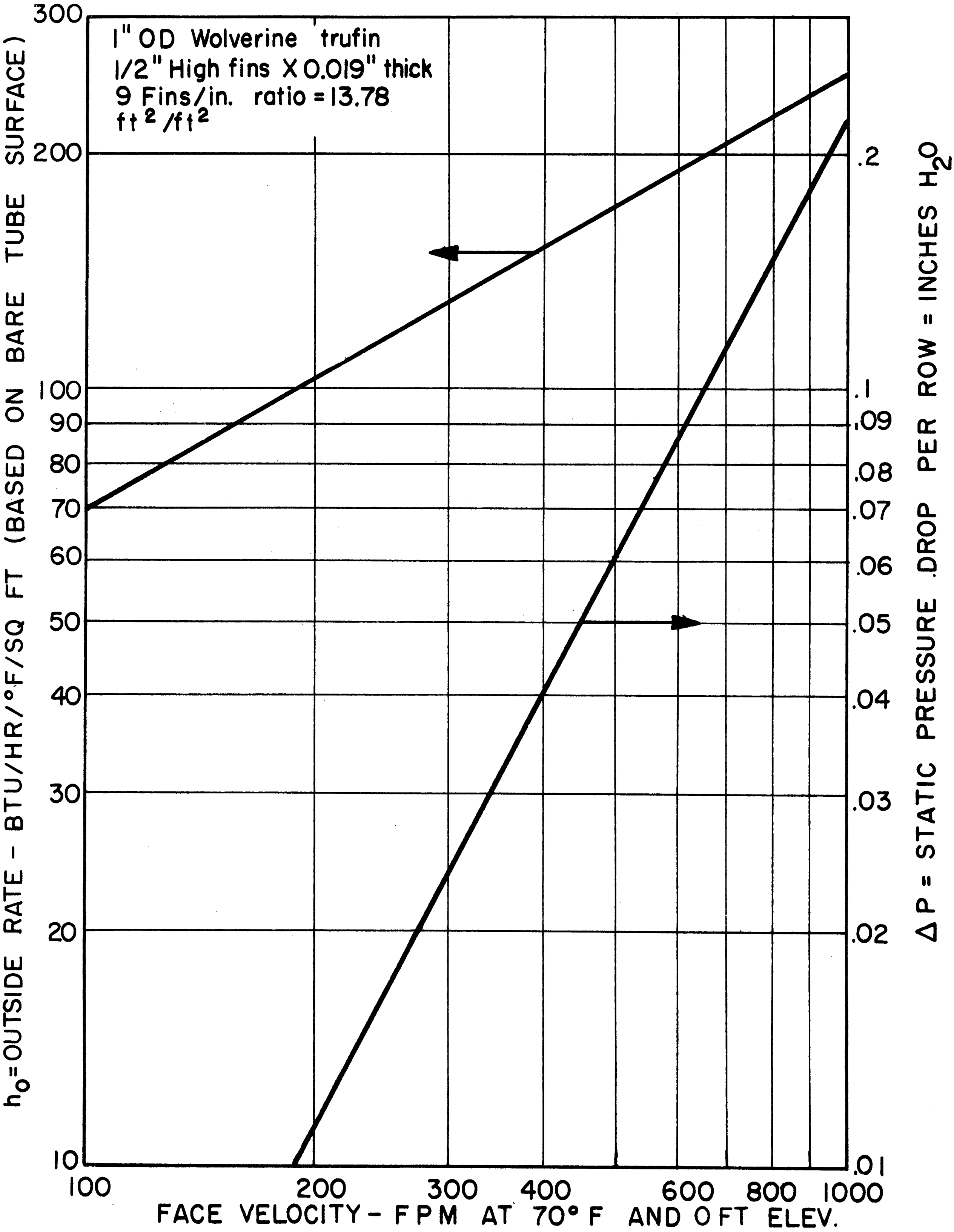

Fig. 12. The Happy Company heat transfer and pressure-drop rating curves.

water per row. For four rows deep the pressure drop would be 0.096 in. of water.

b. Monday's test data, using the Happy Co. rating curve.

In Table IV the air-side-face velocity is given as 475 ft/min with a corresponding h_o value of 105 (r_i = 0), based on the liner area for the three-in-line, fourth-row-staggered unit. The Happy Co. curve, Fig. 12, predicts a corresponding h_o of 167 for this face velocity. This figure also predicts an air-side pressure drop of 0.055 in. of water per row. For a four-row-deep tube bank this amounts to a pressure drop of 0.22 in. across the bank.

4. Summary.—The field-test-data results are compared with the predicted values from 651C, ALCO data, and Happy Co. rating curve in Table V.

TABLE V

COMPARISON OF PREDICTED COEFFICIENTS WITH TEST RESULTS FOR THREE-IN-LINE, FOURTH-ROW-STAGGERED UNIT

Item	$(h_o)_{liner}$	ΔP in. H_2O
Thursday's data computed (V_f = 303)	75	--
651C	145	0.09
ALCO (ΔP, 7 fins per inch)	75	0.115
Happy Co. rating curve	130	0.096
Monday's data computed (V_f = 475)	105	--
651C	188	0.202
ALCO (ΔP, 7 fins per inch)	97.5	0.248
Happy Co. rating curve	167	0.22

An examination of Table V indicates that the field-test coefficients agree reasonably well with the ALCO line converted for a 9-fins-per-inch tube and do not agree with the 651C predictions nor with the Happy Co. ratings.

B. TRIANGULAR-PITCH CORRELATIONS

1. Correlation Report.—The correlation report prepared by the University of Michigan Engineering Research Institute Project 1592, entitled "Correlation of Heat Transfer and Pressure Drop for Air Flowing Across Banks of Finned Tubes," Report No. 30, presents correlations that are restricted to triangular pitch arrangements.

Figure 13 presents the correlation for air coefficients as given on page 39 of Report No. 30. The correlation presents the correlating group as a function of the maximum air velocity in ft/min. The equation of the correlating line is

$$h_o = 1.9\left[\frac{D_r}{S(ND_o)^{0.5}}\right] V_{max}^{0.56} ,$$

where

h_o = air-side coefficient based on outside area,
D_r = root diameter, inches,
D_o = fin diameter, inches,
N = number of fins per inch,
S = tube pitch, inches, and
V_{max} = maximum air velocity, ft/min.

A generalized heat transfer correlation for any gas as a function of the Reynolds number and the physical properties of the gas is also presented in Report No. 30, page 41. This correlation is reproduced here as Fig. 14.

A third air-coefficient correlation is presented on page 45 of Report No. 30. This correlation gives the air coefficient as a function of the tube pitch, S, the number of fins per inch, N, and the root diameter, D_r and is reproduced here as Fig. 15.

The correlation report also presents several pressure-drop correlations for air flowing across staggered-pitch tube banks. A correlation presenting air-pressure drop as the product $(\Delta P)De$ against maximum air velocity in ft/min with a parameter of

$$K = \frac{ND_o}{D_r^{0.2}}$$

is given on page 42 of Report No. 30. The correlation is reproduced here as Fig. 16.

An empirical pressure-drop correlation for air is also given on page 46 of Report No. 30. The figure is reproduced here as Fig. 17. A generalized pressure-drop correlation is presented on page 47 of Report No. 30 and is reproduced here as Fig. 18.

The field test data taken on Monday on the triangular-pitch unit can be readily analyzed by the correlations of Report No. 30. Table IV gives the air-side-face velocity as 398 ft/min. This must be converted to a V_{max}

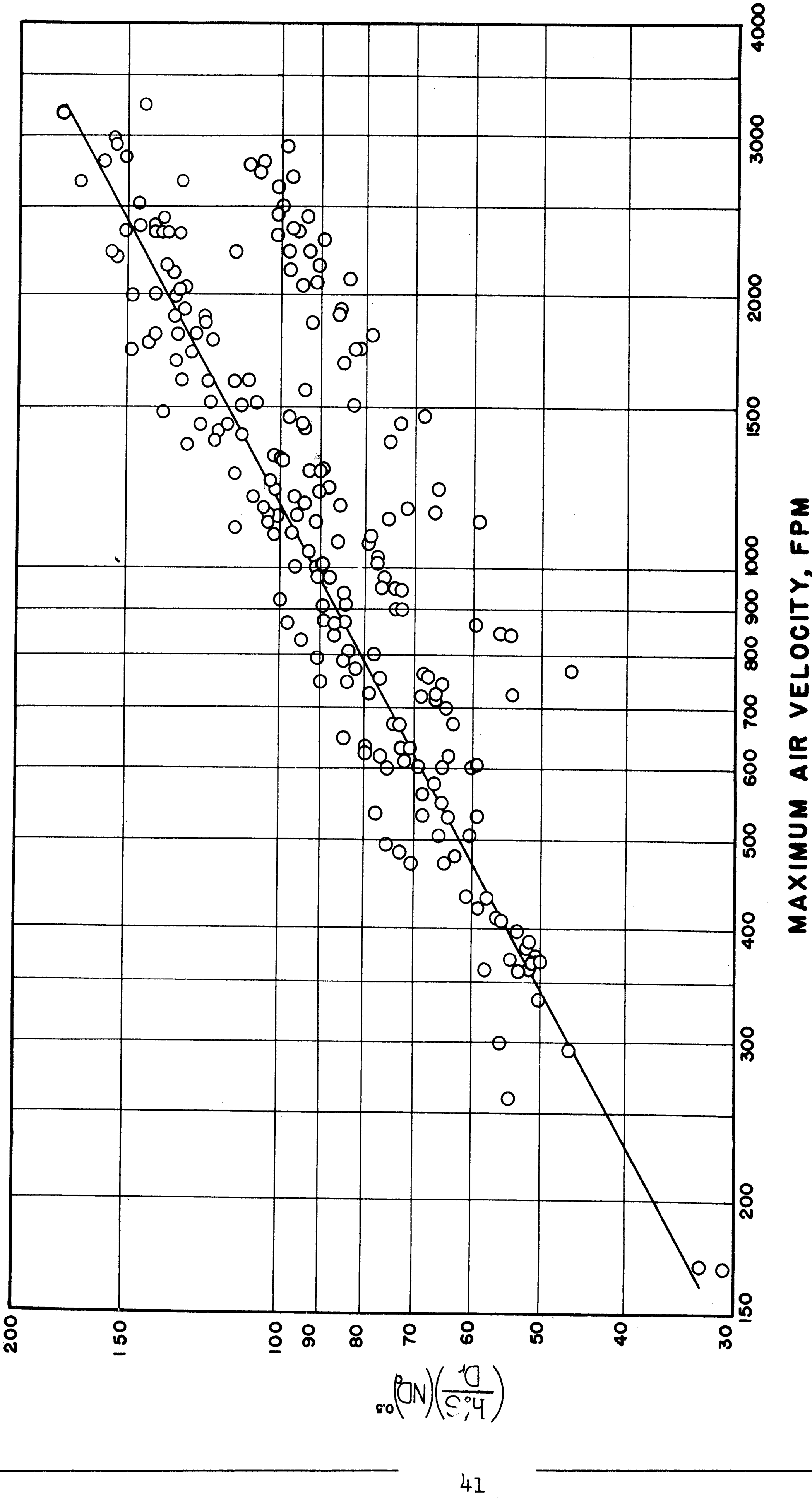

Fig. 13. Heat transfer correlation for air.

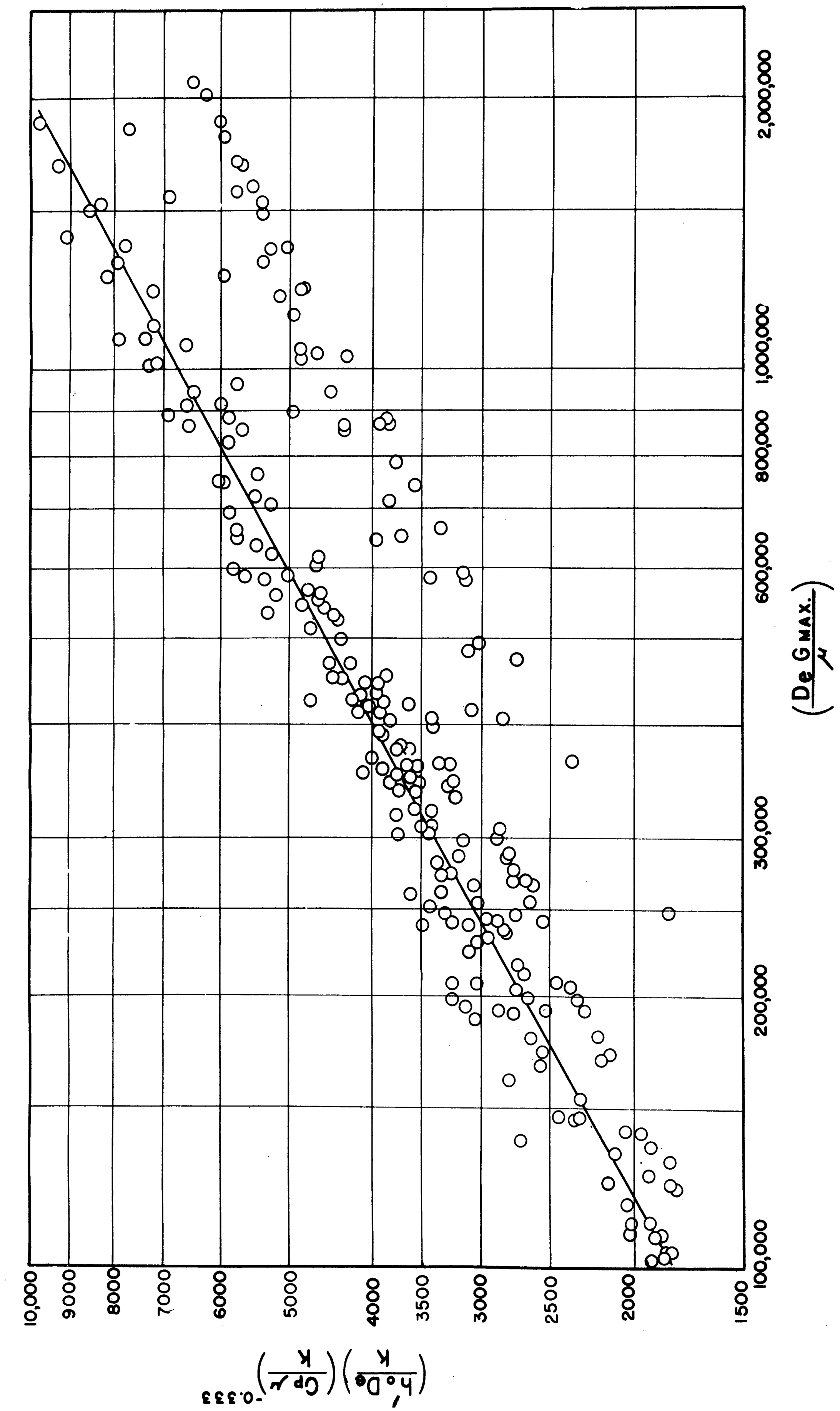

Fig 14. Generalized heat transfer correlation.

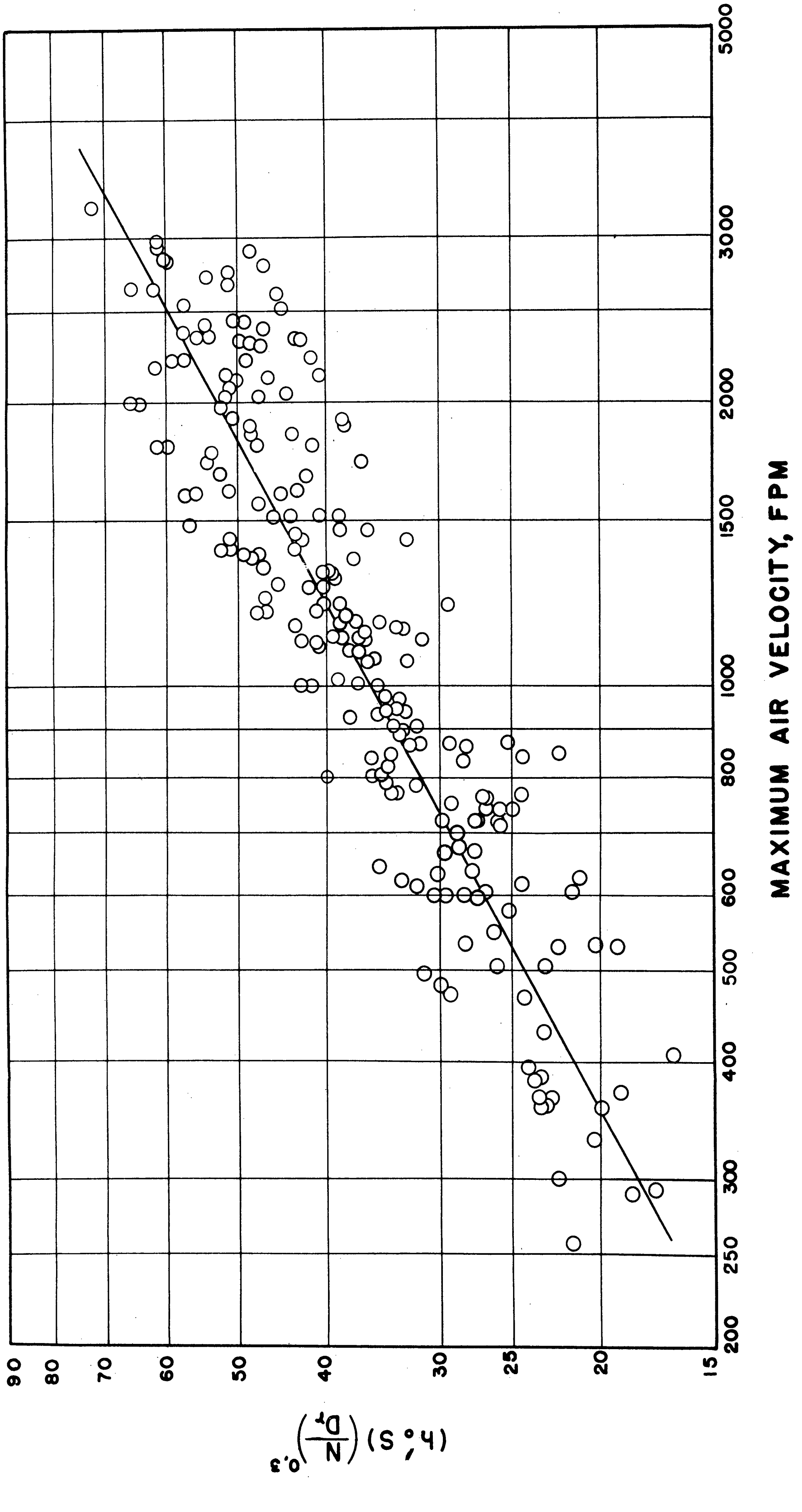

Fig. 15. Heat transfer correlation for air as a function of S, N, and D_r.

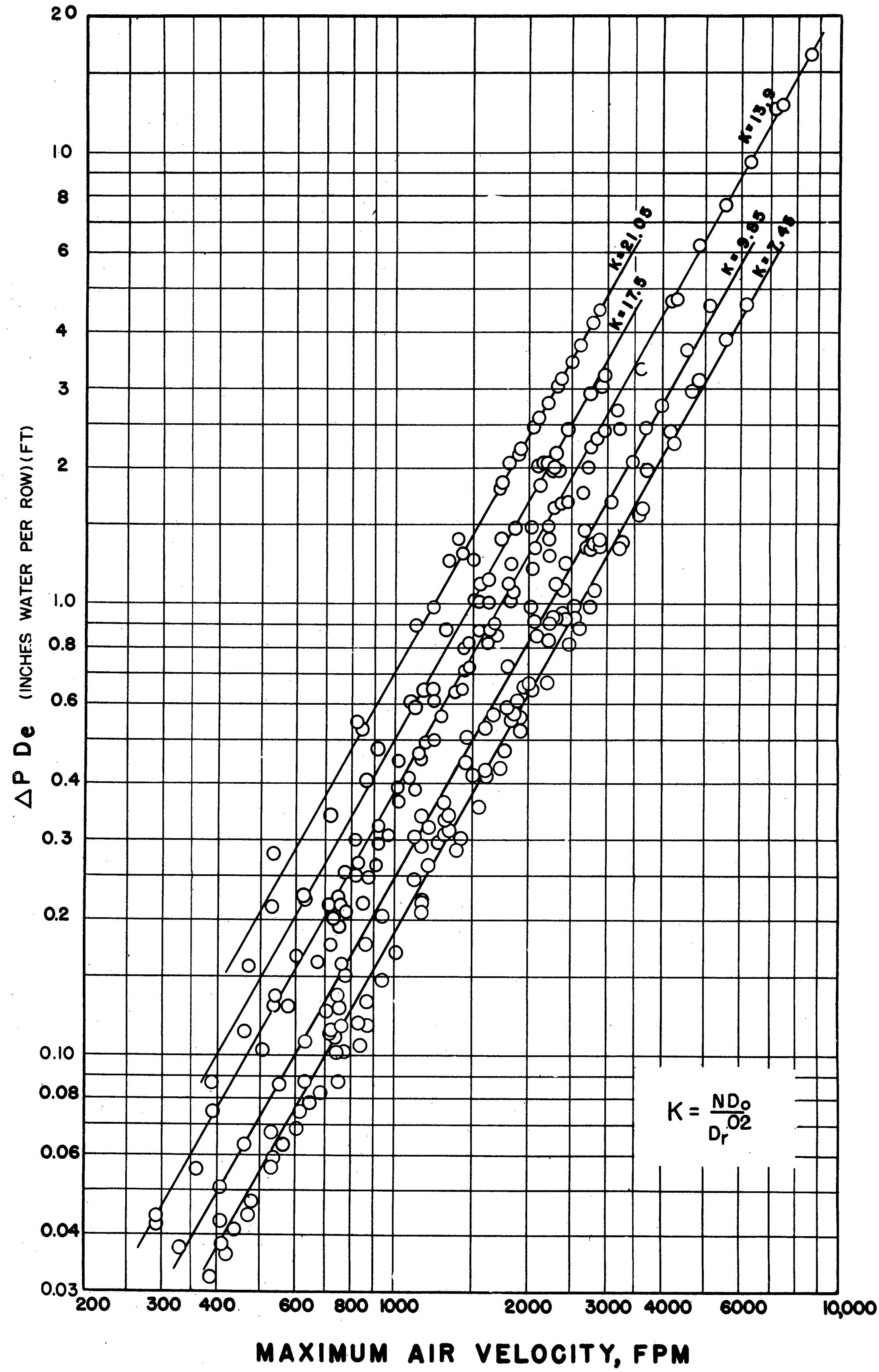

Fig. 16. Pressure-drop correlation for air.

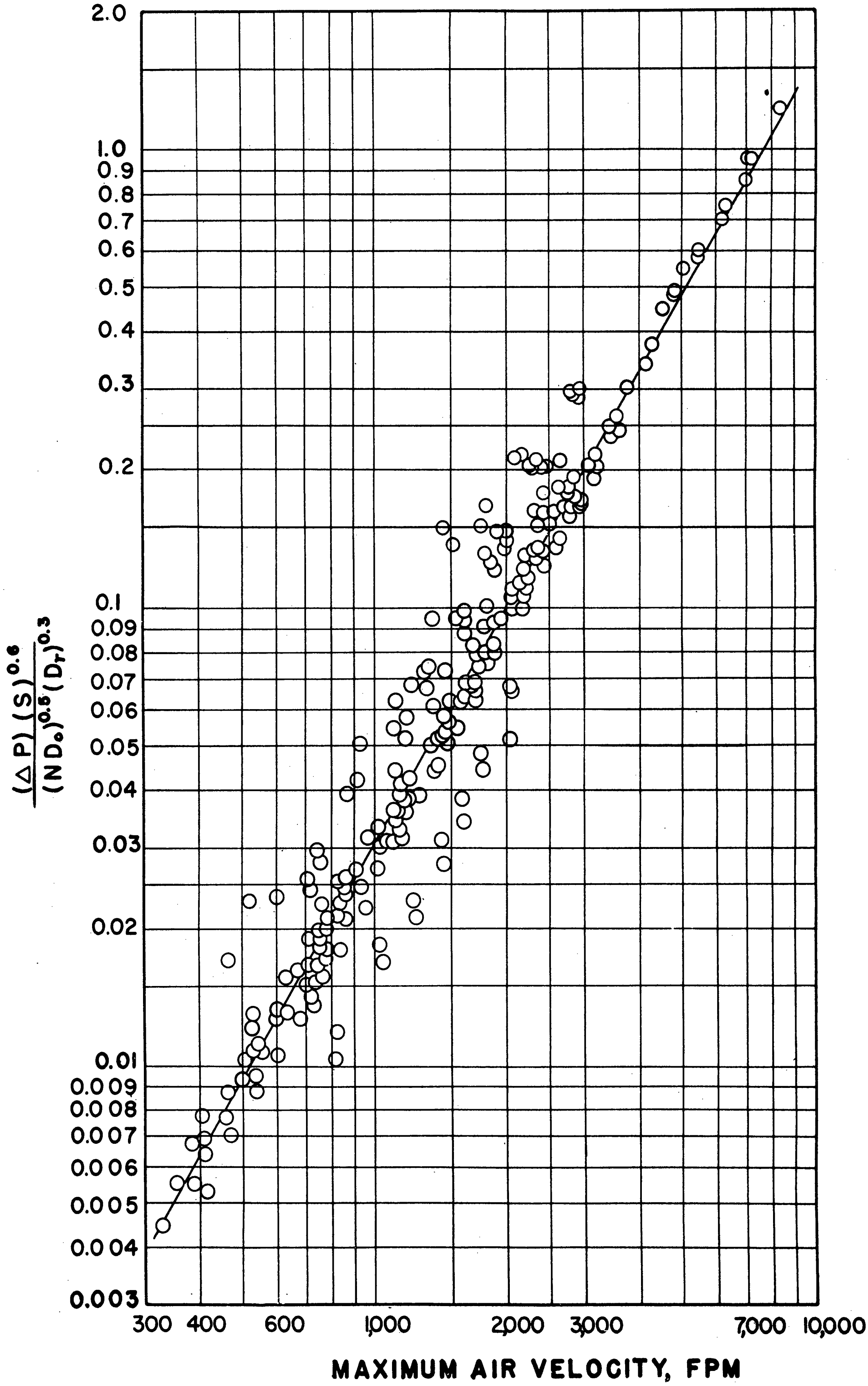

Fig. 17. Empirical pressure-drop correlation for air.

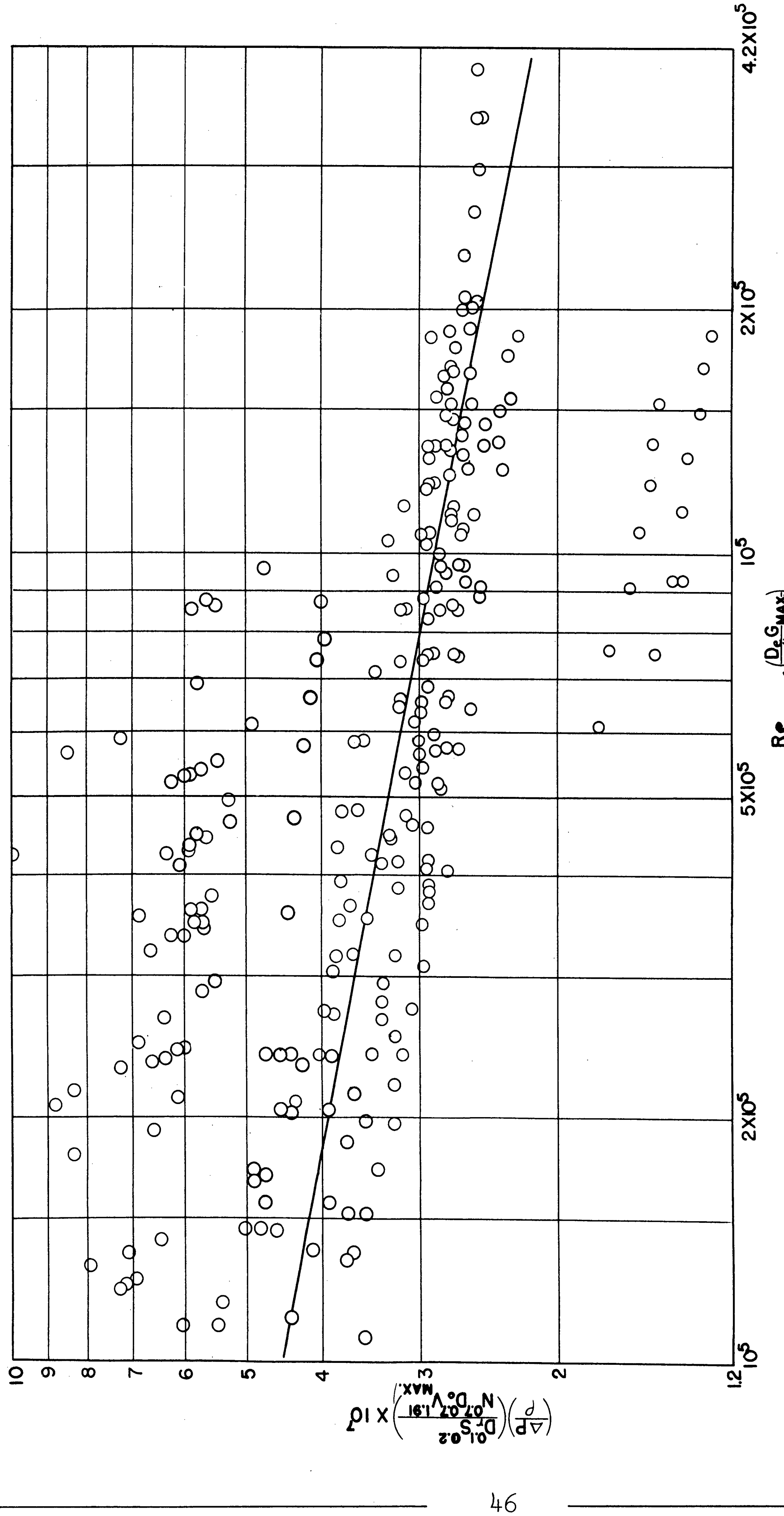

Fig. 18. Generalized pressure-drop correlation.

and is equal to

$$V_{max} = 2.428 \, V_{face} = (2.428)(398) = 968.$$

This value is used for computing the air-side coefficients and pressure drop by Figs. 13, 14, 15, 16, and 17, respectively. The calculations are presented in Appendix F and are summarized below. The h_o computed by use of Fig. 13 is 9.9, by Fig. 14 is 10.6 and by Fig. 15 is 8.5, all based on the outside area. The corresponding values based on liner area are 135.5, 145.0, and 116.4, respectively. The air-side pressure drop by Fig. 16 is computed to be 0.356 in. of water.

2. Schmidt's Correlation.—Mr. T. E. Schmidt published a correlation in an article entitled "Heat Transmission and Pressure Drop in Banks of Finned Tubes and in Laminated Coolers." The article appeared in the Institute of Mechanical Engineering and ASME Proceedings of the General Discussion on Heat Transfer, Section II, 186, London (1951).

T. E. Schmidt's heat transfer correlation applied to air is given on page 44 of Report No. 30 and is reproduced here as Fig. 19. The correlation can readily be used for predicting the air-side coefficient for staggered tube banks.

The calculation of the air-side coefficient, using this correlation, is presented in Appendix G for the air-face velocity of 398 ft/min obtained in Monday's test on the staggered-pitch unit. The computed air-side coefficient is 7.73, based on the outside area, or 106, based on liner area.

3. Summary.—The analysis of the field test data is compared with the available correlations in Table VI.

TABLE VI

SUMMARY OF TRIANGULAR-PITCH ANALYSIS FROM AIR-SIDE DATA

	$(h_o)_{liner}$	h_o	ΔP
Field test data (Table IV) (r_i = 0)	110	8.02	--
Field test data (Table IV) (r_i = 0.001)	125	9.13	--
Predicted by Fig. 13	136	9.9	--
Predicted by Fig. 14	145	10.6	--
Predicted by Fig. 15	116	8.5	--
Predicted by Fig. 19 (Schmidt)	106	7.73	--
Predicted by Fig. 16	---	---	0.356

An analysis of Table VI indicates that the field test data check the Schmidt correlation. The field test coefficient is 19% lower than that predicted by the correlation report curve for air (Fig. 13).

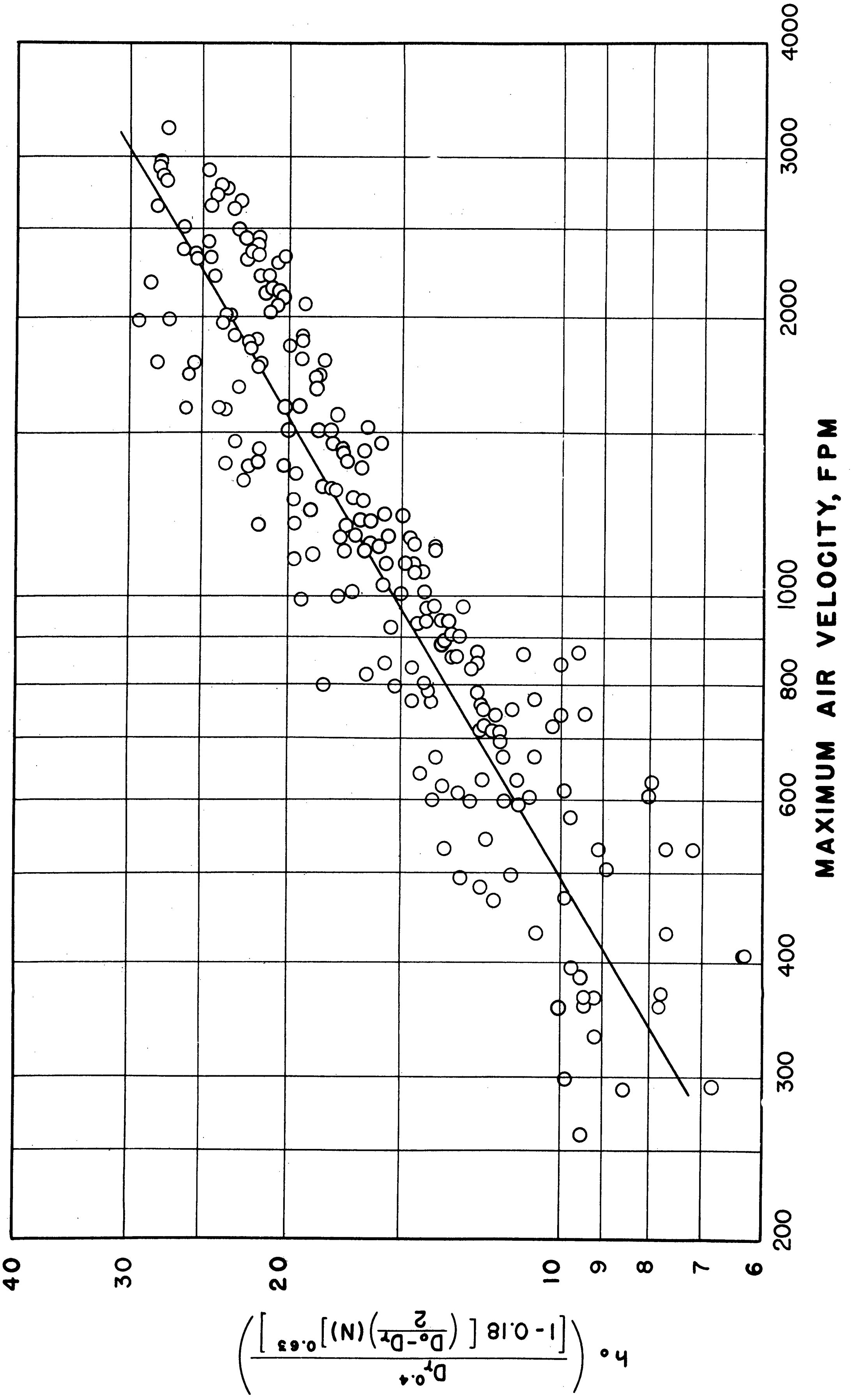

Fig. 19. Heat transfer correlation of T. E. Schmidt applied to air.

IV. PERFORMANCE OF THE THREE-IN-LINE, FOURTH-ROW-STAGGERED UNIT COMPARED WITH THE PERFORMANCE OF THE TRIANGULAR-PITCH UNIT

Table VII presents a comparison of the performance of the two units (Monday's test data).

TABLE VII

COMPARISON OF TRIANGULAR PITCH AND THREE IN LINE, FOURTH ROW STAGGERED

	Triangular-Pitch Unit			Three in Line, Fourth Row Staggered		
	Air Side	Water Side	Avg	Air Side	Water Side	Avg
V_{face}	398	428	413	475	504	490
V_{max}	969	1039	1004	1154	1222	1188
U_o	7.4	7.9	7.65	7.08	7.52	7.30
U_o(liner)	101.3	108	104.7	97	103	100
$Q_{million}$	7.9	8.45	8.175	8.84	9.38	9.11
$(\Delta t)_{mean}$	34.7	34.7	34.7	40.5	40.5	40.5
$h_o(r_i = 0)$	8.02	8.56	8.29	7.65	8.05	7.85
h_o(liner)$(r_i = 0)$	110	117	113.4	105	110	107.5
$h_o(r_i = .001)$	9.13	9.85	9.49	8.65	9.18	8.91
h_o(liner)$(r_i = .001)$	125	135	130	118.5	126	122.3

A number of comparisons can be made between the performances of the two units. A ratio of the performance variables is as follows:

1. $$\frac{V_{face\ 3\text{-}4}}{V_{face\ \Delta}} = \frac{V_{max\ 3\text{-}4}}{V_{max\ \Delta}} = \frac{490}{413} = 1.185 \text{ or } 18.5\% \text{ greater}$$

2. $$\frac{U_{o\ 3\text{-}4}}{U_{o\ \Delta}} = \frac{7.30}{7.65} = 0.954 \text{ or } 4.6\% \text{ less}$$

3. $\frac{Q\ 3\text{-}4}{Q\ \Delta} = \frac{9{,}110{,}000}{8{,}175{,}000} = 1.115$ or 11.5% greater

4. $\frac{(\Delta t)\ 3\text{-}4}{(\Delta t)\ \Delta} = \frac{40.5}{34.7} = 1.166$ or 16.6% greater

5. $\left(\frac{h_o\ 3\text{-}4}{h_o\ \Delta}\right)_{r_i = 0} = \frac{7.85}{8.29} = 0.948$ or 5.2% less

The 18.5% greater air velocity flowing through the 3-4 unit resulted in a 4.6% lower overall coefficient U_o. The greater throughput of air in the 3-4 unit permitted the air to leave at a lower exit air temperature, which in turn resulted in a 16.6% larger Δt driving force for the 3-4 unit. This greater Δt driving force more than compensates for the 4.6% lower overall coefficient. There is approximately 0.5% more area in the 3-4 unit.

It is apparent that if the triangular-pitch unit had the same air throughput as the 3-4 unit, the overall coefficient would have been considerably more than 4.6% better than the 3-4 unit coefficient. Also, the exit air temperature would have been lowered and the Δt driving force would have been increased correspondingly. This would have resulted in a considerable increase in the heat transferred by the triangular-pitch unit. Therefore, at the same face velocity, the triangular-pitch unit would have outperformed the 3-4 unit.

In effect, the triangular-pitch unit was seriously penalized by the lower air-face velocity since this resulted in a coefficient only 4.6% greater than the 3-4 unit. In addition to this, the reduced air velocity resulted in a penalizing Δt driving force which dragged down the effectiveness of the coefficient still further.

V. HORSEPOWER AND Δt CONSIDERATIONS

A comparison can be made between the staggered-pitch unit and the three-in-line, fourth-row-staggered unit on the basis of h_o vs theoretical horsepower, using the correlation report and the ALCO data. Figure 20 graphically presents the curves obtained. Sample calculations for preparing the curves are given in Appendix H. For reference purposes face velocity scales are superimposed on the figure. The upper face velocity scale is to be used only with the lines labeled "triangular." The lower face velocity scale is to be used only with the lower curve labeled "three-in-line, fourth-row-staggered, ALCO data." It should also be noted that Fig. 20 is limited to 2-in.-OD finned tubes having 7 fins per inch, since the ALCO data are for such a tube.

Figure 20 can be used for making two comparisons, one based on equal face velocities and the other based on equal horsepower.

Comparison 1 (same face velocity):

$$\text{assume } V_{face} = 800 \text{ ft/min},$$

$\therefore$

$$(h_o)_\Delta = 16.4, \quad (hp)_\Delta = 0.135$$

$$(h_o)_{3,4} = 11.0, \quad (hp)_{3,4} = 0.075$$

$\therefore$

$$\frac{(h_o)_\Delta}{(h_o)_{3,4}} = \frac{16.4}{11.0} = 1.493 \text{ or } 49.3\% \text{ greater } h_o$$

$$\frac{(hp)_\Delta}{(hp)_{3,4}} = \frac{0.135}{0.075} = 1.80 \text{ or } 80\% \text{ greater.}$$

Comparison 2 (same horsepower):

$$\text{assume hp} = 0.1$$

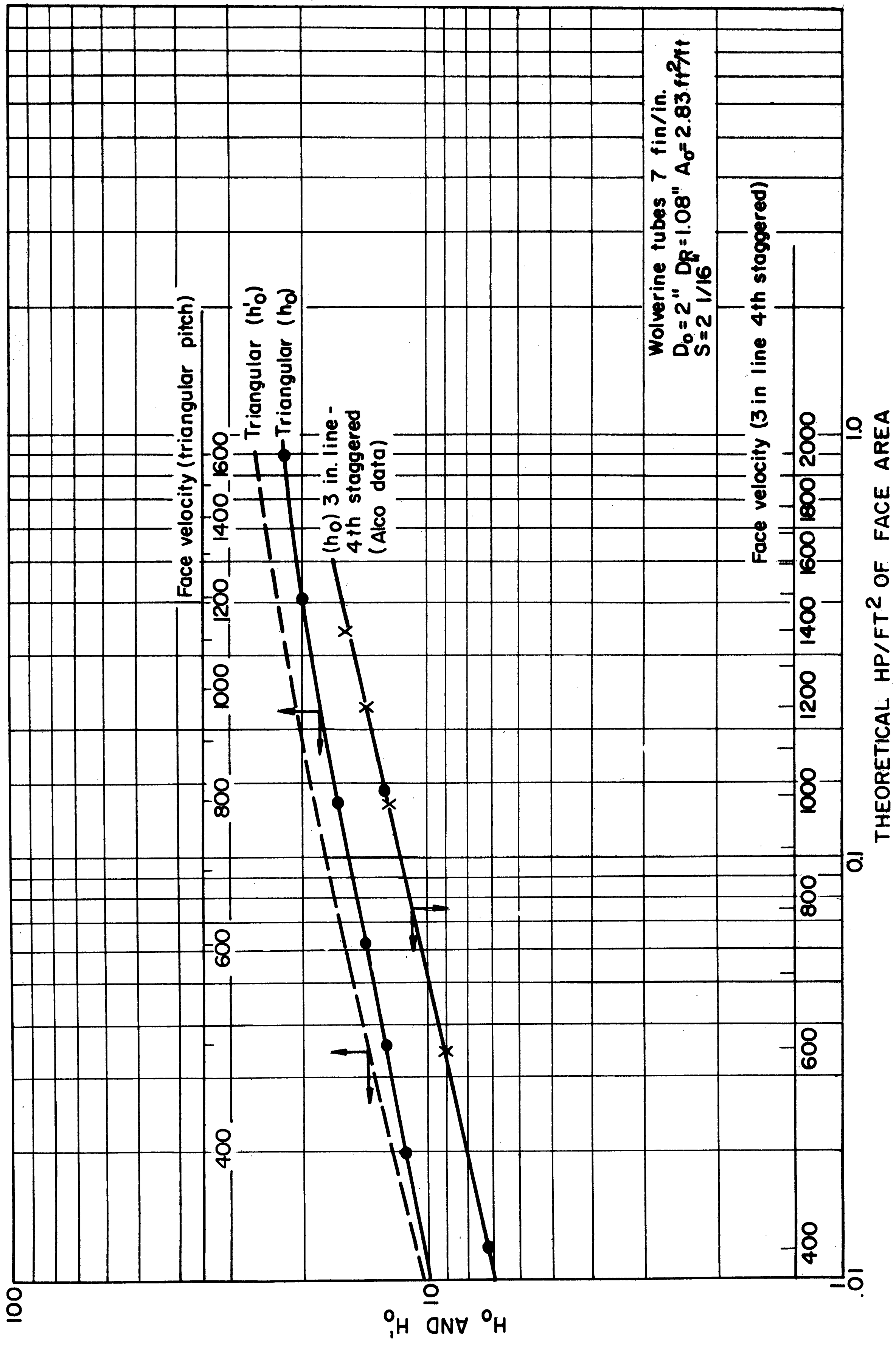

Fig. 20. Outside coefficients vs theoretical horsepower.

$$(h_o)_\Delta = 15.4, \quad (V_{face})_\Delta = 720$$

$$(h_o)_{3,4} = 11.6, \quad (V_{face})_{3,4} = 880$$

$\therefore$

$$\frac{(h_o)_\Delta}{(h_o)_{3,4}} = \frac{15.4}{11.6} = 1.327 \text{ or } 32.7\% \text{ greater for the same hp.}$$

The ratio of volume of air moved by blower:

$$\frac{(V)_\Delta}{(V)_{3,4}} = \frac{720}{880} = 0.818$$

and (1.000 - 0.818) = .182 or 18.2%, or, the triangular unit gives a 32.7% higher air-side coefficient but 18.2% less air is pumped with the same theoretical horsepower.

The above discussion concerned only the controlling coefficient, h_o, and the corresponding theoretical horsepower. The corresponding heat duties, Q, cannot be obtained directly from Fig. 20. The quantity of energy transferred is given by

$$Q = U_o A_o (\Delta t)_{LM}$$

and U_o is a function of h_o, h_i, r_o, r_i, and r_w. For discussion purposes, h_i, r_o, r_i, and r_w may be assumed to be constant. Taking

h_i = 1000
r_o = 0
r_i = 0.0125 based on outside area
r_w = 0.0005 based on outside area.

Referring to comparison 2, above (same hp):

$\therefore$

$$\left(\frac{1}{U_o}\right)_\Delta = \left(\frac{1}{15.4}\right) + 0.0125 + 0.0005 + \left(\frac{12.48}{1000}\right)$$

$$= 0.09$$

$$\therefore \quad (U_o)_\Delta = 11.1$$

$$\left(\frac{1}{U_o}\right)_{3,4} = \left(\frac{1}{11.6}\right) + 0.0125 + 0.0005 + \left(\frac{12.48}{1000}\right)$$

$$= 0.1112$$

$$\therefore \qquad (U_o)_{3,4} = 8.95$$

$$\therefore \qquad \frac{(U_o)_\Delta}{(U_o)_{3,4}} = \frac{11.1}{8.95} = 1.24$$

or 24% greater U_o (as compared with 32.7% on h_o basis).

Therefore,

$$\frac{Q_\Delta}{Q_{3,4}} = \frac{U_o}{U_{3,4}} \frac{A_\Delta}{A_{3,4}} \frac{(\Delta t)_\Delta}{(\Delta t)_{3,4}}$$

for one sq ft of face area, $A_\Delta = A_{3,4}$

$$\therefore \qquad \frac{Q_\Delta}{Q_{3,4}} = 1.24 \frac{(\Delta t)_\Delta}{(\Delta t)_{3,4}} .$$

The fact that the triangular-pitch unit is handling 18.2% less air with a 24% greater overall coefficient will result in heating the air passing through the triangular-pitch unit to a higher exit air temperature than the air leaving the three-in-line, fourth-row-staggered unit. This will result in a lower log mean temperature difference driving force for the triangular-pitch unit. This in turn will reduce the ratio of Q_Δ to $Q_{3,4}$ to less than 1.24. The triangular pitch arrangement gives a higher h_o and U_o than the three-in-line, fourth-row-staggered unit at the same theoretical horsepower input.

VI. PRESSURE-DROP COMPARISONS

The predicted air-side pressure drop in inches of water for the 3-4 unit (Monday's data) is 0.248, using the ALCO curve (see Table V). The predicted air-side pressure drop for the triangular-pitch unit (Monday's data) is 0.356 using Fig. 16 (see Table VI). The value taken from the ALCO data as given in Table V is used because it is based on experimental test data. The values are given for the test conditions and therefore are not based on any ideal comparison basis.

The Happy Co. measured 0.78 in. of water for the triangular-pitch unit and 0.54 in. of water for the 3-in-line, fourth-row-staggered unit (Monday's data, see Section X).

Tables VIII and IX (see Section X) present the data obtained by the Happy Co. on Thursday, April 28, and May 2, 1955, respectively.

The Happy Co. measurements included a static-pressure reading which was made between the fan and the tube bank. These measurements were made by inserting a plain copper tube underneath the tube bank, with one end open and the other end connected by means of rubber tubing to an inclined manometer.

The values obtained by the Happy Co. are given in Table X where they are compared with the values computed from the available correlations. The values tabulated for Monday's test are the test values for 2:25 p.m., as this time corresponds closest to the time during which the velocity profile was being made.

TABLE X

SUMMARY OF PRESSURE-DROP ANALYSIS

Three in line, fourth row staggered (see Table V)

	ΔP, Inches of Water
Thursday's test data (V_{face} = 303)	
(Happy Co.) East bay (avg)	0.325
West bay (avg)	0.245
average of bays	0.285
Predicted by the Happy Co. rating sheet	0.096

TABLE X (Concl.)

	ΔP, Inches of Water
Predicted by 651C	0.09
Predicted by ALCO data line (7 fins per inch)	0.115
Monday's test (V_{face} = 475)	
Field test data (Happy Co.) (avg)	0.54
Predicted by the Happy Co. rating sheet	0.22
Predicted by 651C	0.202
Predicted by ALCO data line (7 fins per inch)	0.248

The design-face velocity was 600 ft/min. The predicted pressure drops for this velocity are given in Table XI.

TABLE XI

COMPARISON OF PRESSURE DROPS WITH PREDICTED VALUES FOR 600-FT/MIN FACE VELOCITY

	ΔP, Inches of Water
Field test data (Thursday)	0.285
Field test data (Monday)	0.540
Predicted by Happy Co. rating curve	0.344
Predicted by 651C	0.315
Predicted by ALCO data (7 fins per inch)	0.370

Tables X and XI indicate that the Happy Co. pressure-drop measurements do not agree with any of the predicted values for design or test flowrate conditions.

VII. FAN PERFORMANCE

The fans used in the field-test units were manufactured by the Moore Co. of Kansas City, Mo. The fans are classified as series 48, 16-ft, 4-blade Moore Pressure Blowers, 7248-AM-16-4, 250 rpm. The Happy Co. furnished "anticipated-performance" curves for the fans, a copy of which is given in Fig. 21. The maximum fan pitch indicated on the figure is 15.6°. Figure 21 gives the anticipated static pressure in inches of water and corresponding brake horsepower input to the fan as a function of volume of air handled with fan pitch as a parameter.

The average face velocity measured on Thursday was 303 ft/min ($\rho = 0.074$) which corresponds to 345 ft/min ($\rho = 0.065$). The face area of 366 sq ft, when put together with the face velocity of 345, gives 126.2 thousands of CFM (see Fig. 21). The predicted static-pressure drop was 0.096 in. by Happy Co. and 0.09 in. by 651C for an average of 0.093 in. of water. These values, when plotted on Fig. 21, do not fall near the curves.

The average face velocity for the three-in-line, fourth-row-staggered unit (Monday's test data) was 475 ft/min ($\rho = 0.074$), which corresponds to 541 ft/min ($\rho = 0.065$). This corresponds to 198 thousands of CFM. The predicted static-pressure was 0.22 in. of water by Happy Co. and 0.20 in. by 651C for an average of 0.21 in. These values, when plotted on Fig. 21, do not fall near the curves.

The average face velocity for the triangular-pitch unit (Monday's data) was 398 ft/min ($\rho = 0.074$), which corresponds to 453 ft/min ($\rho = 0.065$). This in turn corresponds to 166 thousands of CFM. The predicted static pressure by the correlation report was 0.356 in. of water. These values, when plotted on Fig. 21, do not fall on the curves.

Blade-efficiency curves can be prepared, using the anticipated-performance curves of Fig. 21. Sample computations are presented in Appendix L and resulting curves are presented in Fig. 22. Superimposed on Fig. 22 are vertical lines corresponding to the predicted pressure drops for the units, based on Monday's test data. Also indicated on Fig. 22 are two points corresponding to the Happy Co. field-test static-pressure measurements. The figure indicates that the predicted static pressures fall within the curves in the range of 32 to 45% blade efficiency. The peak of the curve reaches a maximum of 55% efficiency. It is believed that the existing fans are incapable of ever providing 600 ft/min face velocity air to the units tested.

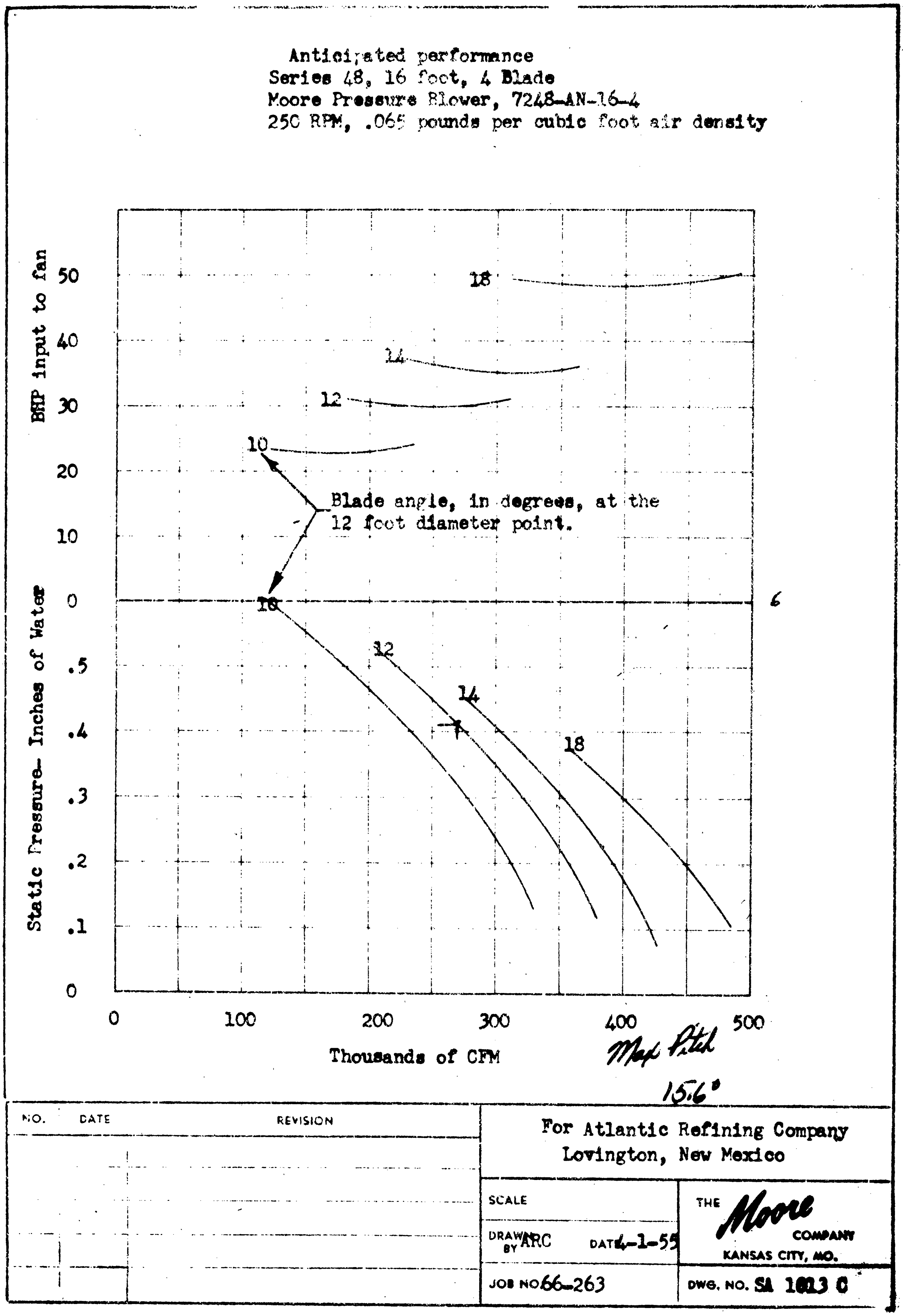

Fig. 21. Jacket-water-cooler anticipated fan performance curves.

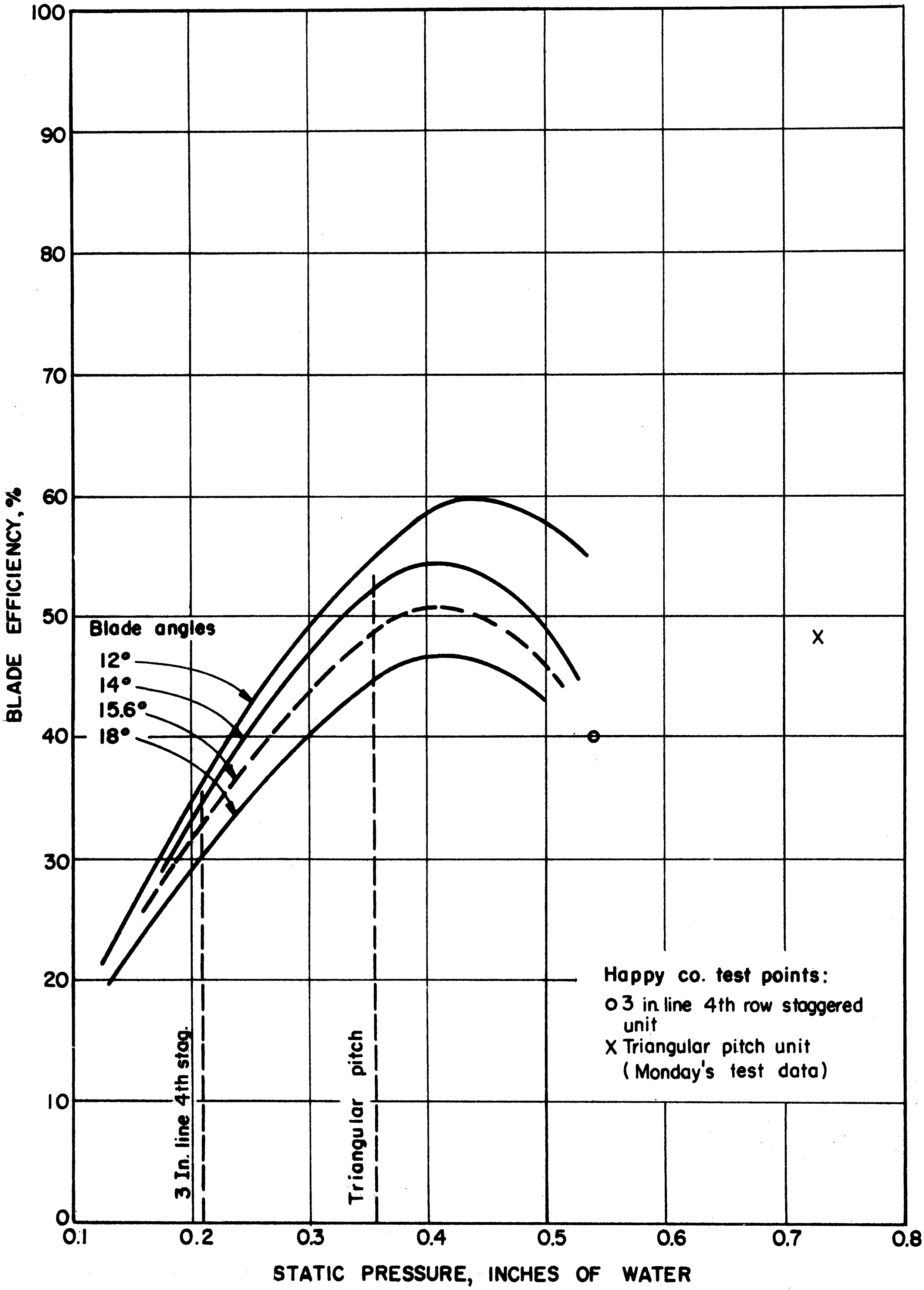

Fig. 22. Predicted blade efficency of jacket-water-cooler fans.

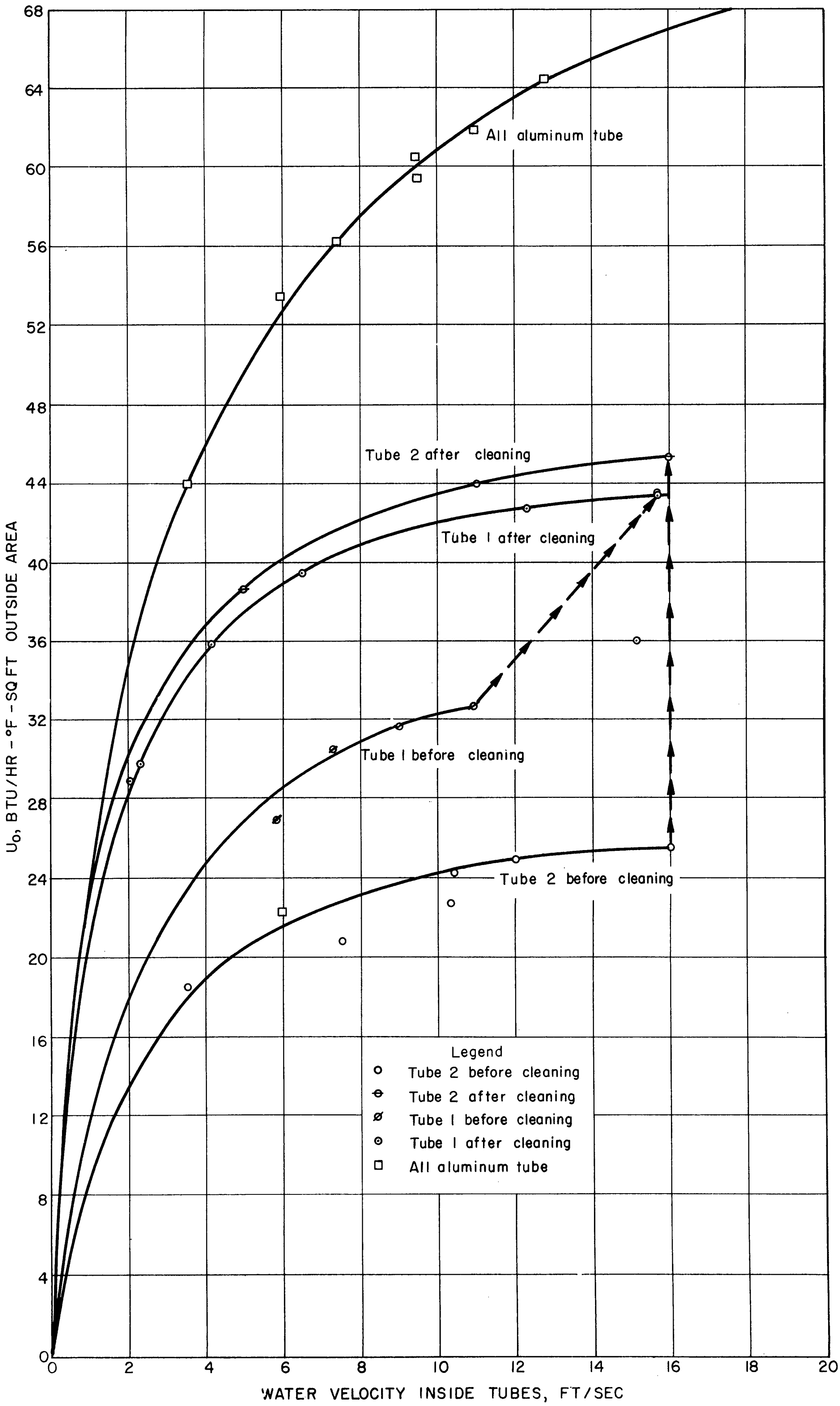

Fig. 23. Results of tests conducted on tubes cut from the jacket water cooler.

Difference of resistances $= \frac{1}{25.0} - \frac{1}{44.4} = 0.0175$

$\therefore$

$r_i = 0.0175$, based on outside area,

or $r_i = \frac{0.0175}{13.7} = 0.00128$, based on liner area.

(Note: Any bond resistance present would cancel out.)

Also: $r_i = \frac{0.0175}{15.2} = 0.00115$, based on inside area.

Also given in Fig. 23 is a test curve for an all-aluminum tube which was tested at the same time. This curve is useful for predicting the amount of bond resistance presumably existing in the tubes. To indicate how this curve can be used, Fig. 24 was prepared. Figure 24 is based on the all-aluminum test curve of Fig. 23. and gives the decrease in heat transfer as indicated by the percentages when the outside coefficient is 10.

Figure 24 has a superimposed dashed line which represents the average performance of tubes No. 1 and No. 2 after cleaning, as given in Fig. 23. It is apparent that the bond resistance present amounts to approximately 5% of the overall resistance. This degree of resistance cannot account for more than 5% of the field-performance discrepancy as compared to the specifications for the units tested.

One of the tubes tested was shipped to Wolverine tube for expanding of the liner. After the liner had been expanded, the tube was retested under identical test conditions. The results of the expanding are shown in Fig. 24 by a dash-dot line indicating reduced bond resistance from 5% to 4%. The tubes used in the fabrication of the units that were tested on Monday had expanded liners.

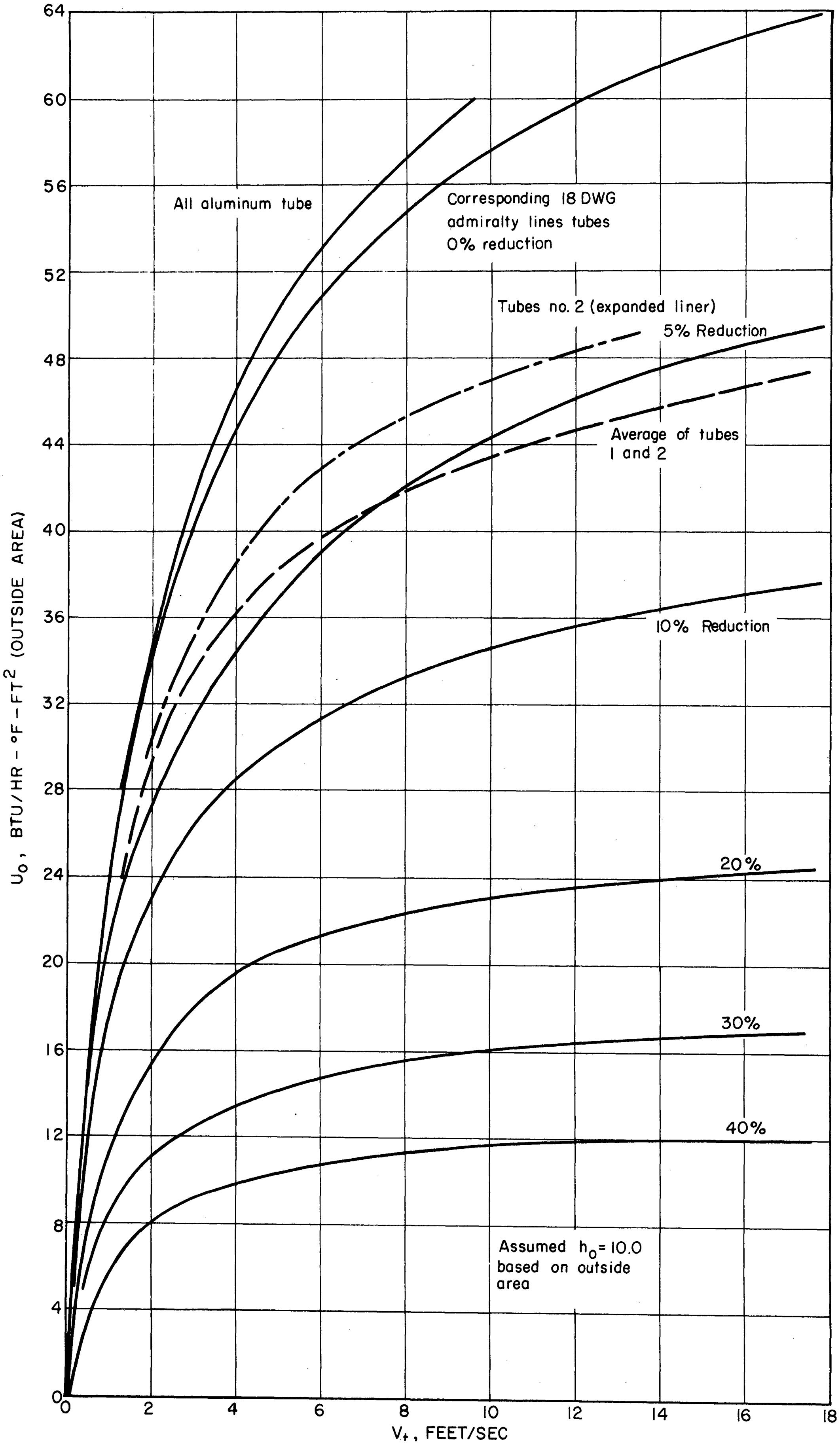

Fig. 24. Predicted performance of tubes tested in field application.

IX. JACKET WATER TEST-ANALYSIS CONCLUSIONS

The conclusions drawn from the analysis given in Sections II through VIII of this report are:

1. The units are operating at air-face velocities that are well below the design-face velocity of 600 ft/min.

2. The design air-side coefficients for three-in-line, fourth-row-staggered units are greatly overrated. The air-side coefficient controls the performance of the unit.

3. A combination of the above two factors explains the poor performance obtained from the units.

X. HAPPY CO. TEST DATA

The representatives of the Happy Co. took test data of their own on Thursday and on Monday, April 28 and May 2, respectively. Their Thursday test data are given in Table VIII and their Monday test data are given in Table IX.

TABLE VIII

HAPPY CO. TEST DATA OF THURSDAY, APRIL 28, 1955

Air temperatures, °F:

West Fan	East Fan	
(3:43 p.m.)	(3:43 p.m.)	(4:40 p.m.)
77.0	75.0	75.0
76.0	77.5	77.5
76.5	75.0	75.0
76.5	75.0	74.0
76.0	74.0	75.0

Water temperatures, °F:

	Coil 1	Coil 2	Coil 3	Coil 4
Inlet	164.2	164	164.2	164
Inlet	164.4	164	164.0	
Outlet	157.3	156.2	155.9	156
Outlet	157.6	157.1	155.0	
Outlet		157.4		
Outlet		156.6		

Water-main inlet temperature = 164°F
Water-main outlet temperature = 156°F

Static pressures:

West Fan	East Fan
0.26	0.33
0.23	0.32

Motor amps: West Fan = 18.5
East Fan = 19.0

Pump suction: 4.5 psi
Pump discharge: 25.3 psi

Water-main Pitot tube: 3.5 to 8 in. of meriman oil, specific gravity = 2.9. (readings fluctuated widely)

Air pressure to fans at 4:40 p.m. (controller): 9.0 psi.

TABLE IX

HAPPY CO. TEST DATA OF MONDAY, MAY 2, 1955

	Time 1:45 p.m.	2:25 p.m.
Water In, T_0, °F	154.7	149.7
Water Out, T_1, °F	147.2	142.1
Water Out, T_2, °F	145.5	140.9
Water Out, T_3, °F	145.8	141.0
Water Out, T_4, °F	144.9	140.3
Water Out, T_2, °F (Overall)	146.0	141.3
Flow Meter, in H_2O	65 avg	66 avg
Pump Suction, psig	2.8	2.75
Pump Disc, psig	22.0	22.0
Air Wet Bulb, °F	67	
Air Dry Bulb, °F	74	
Barometer		
Wind Direction	WSW	WSW
Wind Velocity, Mph, est.	12	12
Fan Position, (Air Pressure)	0(15.6°)	0(15.6°)
Static Pressure, West Unit	0.75-0.85	0.78
Static Pressure, East Unit	0.48-0.56	0.53-.55
Air Temp., °F, West, in	81	79
Air Temp., °F, East, in	81.5	79.5
Air Temp., out, Avg, °F, W.		
Air Temp., out, Avg, °F, E.		
Air Velocity, Avg, FPM, W.		
Air Velocity, Avg, FPM, E.		
Fan Load, amps, West	40	42
Fan Load, amps, East	40	42
Fan Volts	470 at Gen.	470 at Gen.
Speed, rpm. West	250	
Speed, rpm. East	250	
No. Compressor on.	5	5
Discharge Press, psig	880	885
Suction Press, psig	6	6
Flowmeter, Diff, "H_2O	35	35
Temp. Gas In, °F	69	69
No. 1 Generator Load, KW	390	350
No. 2 Generator Load, KW	390	320

XI. HAPPY CO. SPECIFICATION SHEETS

The specification sheets for the five Happy Co. units at the Atlantic gasoline plant are reproduced for reference purposes on the following pages.

The HAPPY COMPANY
TULSA, OKLAHOMA

SPECIFICATIONS FOR

HAPPY FIN-TYPE COOLERS

Customer	Dresser Engineering Company
Quotation No.	9117
Address	P. O. Box 2518 Tulsa, Oklahoma
Date	March 6, 1953
Plant	Atlantic Refining, Lovington, N.M.
Reference	Item 2-1, Alternate
Cooler Model	HFR-360-2
Service	Jacket Water

	INSIDE TUBES	AIR DATA	
Fluid		Elevation, Feet	3700
Sp. Gr. @ 60°F.		Inlet Air Temp., °F.	100
Sp. Ht. @ Avg. Temp.		Total SCFM	445,000
Density @ Avg. Temp.		SCFM/Fan	222,500
Viscosity		Temp. Rise, °F.	44.8
~~xxxx~~ M.T.D.	Based on bare tube 30°	**MECHANICAL EQUIPMENT**	
	U based on bare tube 159.5	Fans: Type	Moore °48 Adj. Pitch
	Inside fouling factor 0.001	No.	2
Qty.—Gas		Dia.	16'
H. C. Vapor, #/Hr.		No. Blades	4
Steam, #/Hr.		RPM	250
H. C. Liquid, #/Hr.		Hp./Fan	29.9
Water GPM	3886	Total Hp.	59.8
		Gears: Make	Cleveland
Inlet Temp., °F.	161.06	Model	CU-400
Outlet Temp., °F.	150	Rating AGMA	47
Operating Press., PSI	12	Ratio	7:1
Pressure Drop, PSI	4.45	~~V-Belt~~ Drive:	Tripod
Design Pressure, PSIG	50	~~Fan Sheave~~	Supported
Test Pressure, PSIG	100	~~Drive Sheave~~	Fan
Design Temperature, °F.	175	~~V-Belts~~	
Heat Exchanged, BTU/Hr.	21,492,000		
Surface, Sq. Ft.	Total bare 4500; total external 62,000	Motors: Make	Louis Allis
Surface per Coil, Sq. Ft.	Total bare 1125; total external 15,502	Type	Explosion proof
		Number	2
	MATERIALS & CONSTRUCTION	Hp.	40
Units Reqd.	1	R.P.M.	1760
Coils per Unit	4	Volts	440
Coil Width	7.5'	Phase	3
Tubes per Coil	179	Cycle	60
Tube Length	24'	Gas Eng.: Make	
Tubes Wolverine	1" x 18 BWG Inhibited Admiralty Liner	Model	
Fins Extruded	Aluminum 9/inch, 1/2" high .019"	Number	
Headers Box	Fabricated Steel, Brass Plug	Hp.	
Connections	8" - 150# R.F.	R.P.M.	
No. Passes	2		

Cooler made up of 2 bays, 1 fan per bay, each bay approximately 16' x 24' x 12'. Both bays combined in common structure.

The HAPPY COMPANY

TULSA, OKLAHOMA

SPECIFICATIONS FOR

HAPPY FIN-TYPE COOLERS

Page 2- Alternate

Customer	Dresser Engineering Company	Quotation No.	9117
Address	Tulsa, Oklahoma	Date	March 11, 1953
Plant	Atlantic Refining Co., Lovington, N.M.	Reference	Item 2-2 Alternate
Cooler Model	HFR-72		

Service	Evaporator Condenser

	INSIDE TUBES	AIR DATA	
Fluid		Elevation, Feet	3700
Sp. Gr. @ 60°F.		Inlet Air Temp., °F.	100
Sp. Ht. @ Avg. Temp.		Total SCFM	46,200
Density @ Avg. Temp.		SCFM/Fan	23100
Viscosity		Temp. Rise, °F.	79.2
M.T.D.	Based on bare tube surface 78	MECHANICAL EQUIPMENT	
U	Based on bare tube 172	Fans: Type	Koppers Adj. Pitch
	Inside fouling factor .001	No.	2
Qty.—Gas		Dia.	5'
H. C. Vapor, #/Hr.		No. Blades	4
Steam, #/Hr.	Saturated 4,000	RPM	870
H. C. Liquid, #/Hr.		Hp./Fan	3.3
Water		Total Hp.	6.6
		Gears: Make	
Inlet Temp, °F.	240	Model	
Outlet Temp., °F.	200	Rating AGMA	
Operating Press., PSI	25	Ratio	
Pressure Drop, PSI	2	Fan direct	
Design Pressure, PSIG	50	connected to	
Test Pressure, PSIG	100	motor	
Design Temperature, °F.	300	V-Belts	
Heat Exchanged, BTU/Hr.	3,970,000		
Surface, Sq. Ft. Bare	298.5 Total External 4120	Motors: Make	Louis Allis
Surface per Coil, Sq. Ft.	298.5 " " 4120	Type	TEFC
		Number	2
MATERIALS & CONSTRUCTION		Hp.	5
Units Reqd.	1	R.P.M.	870
Coils per Unit	1	Volts	440
Coil Width	6'	Phase	3
Tubes per Coil	95 - 3 rows	Cycle	60
Tube Length	12'	Gas Eng.: Make	
Tubes Wolverine	1" x 18 BWG Inhibited Admiralty Liner	Model	
Fins Extruded Aluminum	9/inch, 1/2" high .019"	Number	
Headers Box	Fabricated Steel Brass Plug	Hp.	
Connections	6" - 150# R.F. In: 2" -150# R.F. Out	R.P.M.	
No. Passes	2 - reducing 66-2/3% - 33-1/3%		

The HAPPY COMPANY

TULSA, OKLAHOMA

SPECIFICATIONS FOR

HAPPY FIN-TYPE COOLERS

Customer	Dresser Engineering Company	Quotation No.	9117
Address	Tulsa, Oklahoma	Date	March 10, 1953
Plant	Atlantic Refining Co., Lovington, N.M.	Reference	Item 2-3 Alternate
Cooler Model	HFR-430		
Service	Still Overhead Partial Condenser		

	INSIDE TUBES	AIR DATA	
Fluid	Gasoline Vapor	Elevation, Feet	3700
Sp. Gr. @ 60°F.		Inlet Air Temp., °F.	100
Sp. Ht. @ Avg. Temp.		Total SCFM	258,000
Density @ Avg. Temp.		SCFM/Fan	129,000
Viscosity		Temp. Rise, °F.	27.7
MTD bare tube	46.0	**MECHANICAL EQUIPMENT**	
U based on bare tube	68.2	Fans: Type	Moore 48
Inside fouling factor	.001	No.	2
Qty.—Gas		Dia.	11'
H. C. Vapor, #/Hr.	In: 74,249 M.W. 58.4	No. Blades	3
Steam, #/Hr.	In: 2,072	RPM	318
H. C. Liquid, #/Hr.	Out: 27,900 M.W. 69.3	Hp./Fan	19.9
Water	Out: 2,072	Total Hp.	39.8
	Assume straight line condensation	Gears: Make	Cleveland
Inlet Temp., °F.	180	Model	CU-200
Outlet Temp., °F.	140	Rating AGMA	24
Operating Press., PSI	87	Ratio	5-4/5:1
Pressure Drop, PSI	2	V-Belt Drive:	Gear & Motor
Design Pressure, PSIG	150	Fan Sheave	on concrete
Test Pressure, PSIG	225	Drive Sheave	pedestal
Design Temperature, °F.	200	V-Belts	
Heat Exchanged, BTU/Hr.	7,747,000		
Surface, Sq. Ft.	Bare 2472 Total External 34,050	Motors: Make	Louis-Allis
Surface per Coil, Sq. Ft.	824 11,350	Type	Expl. Proof
		Number	2
MATERIALS & CONSTRUCTION		Hp.	25/6.25
Units Reqd.	1	R.P.M.	1800/900
Coils per Unit	3 - - Parallel	Volts	440
Coil Width	6'	Phase	3
Tubes per Coil	131 - 4 rows	Cycle	60
Tube Length	24'	Gas Eng.: Make	
Tubes	Wolverine 1" x 18 BWG Inhibited Admiralty Liner	Model	
Fins	Extruded Aluminum, 9/inch, 1/2" x .019"	Number	
Headers	Box Fabricated Steel, Beass Plug	Hp.	
Connections	In: 8"-150# R.F. Out: 6"-150# R.F.	R.P.M.	
No. Passes	2 reducing 75% - 25%		

Bay Size Approx. 20' x 24' x 11'

The HAPPY COMPANY

TULSA, OKLAHOMA

SPECIFICATIONS FOR

HAPPY FIN-TYPE COOLERS

Customer	Dresser Engineering Company	Quotation No.	9117
Address	Tulsa, Oklahoma	Date	
Plant	Atlantic Refining Co. LOVINGTON, N.M.	Reference	Item 2-4 Alternate
Cooler Model	HFR-180		
Service	Debutanizer Overhead Condenser		

	INSIDE TUBES	AIR DATA	
Fluid	Butane Vapor	Elevation, Feet	3700
Sp. Gr. @ 60°F.		Inlet Air Temp., °F.	100
Sp. Ht. @ Avg. Temp.		Total SCFM	125,000
Density @ Avg. Temp.		SCFM/Fan	62,500
Viscosity		Temp. Rise, °F.	35.6
Mol. Wt.	58	MECHANICAL EQUIPMENT	
Fouling Factor	0.001	Fans: Type	Series 24 Moore Adj. Pitch
U based on bare tube surface	104.9	No.	2
Qty.—Gas		Dia.	7'
H. C. Vapor, #/Hr.	IN: 33,859	No. Blades	5
Steam, #/Hr.		RPM	557
H. C. Liquid, #/Hr.	OUT: 33,859	Hp./Fan	12.8
Water		Total Hp.	25.6
		Gears: Make	Cleveland Worm
Inlet Temp., °F.	174	Model	CU-100
Outlet Temp., °F.	140	Rating AGMA	15.7 HP
Operating Press., PSI	158	Ratio	3-1/7 to 1
Pressure Drop, PSI	2	XXXXXX:	Gear and motor on
Design Pressure, PSIG	175	XXXXXXX	concrete
Test Pressure, PSIG	265	~~Drive Sheave~~	pedestal
Design Temperature, °F.	200	V-Belts	
Heat Exchanged, BTU/Hr.	4,800,000		
Surface, Sq. Ft. Bare	1125 Total External 15,500	Motors: Make	Louis Allis
Surface per Coil, Sq. Ft.	1125 Total External 15,500	Type	Expl. proof, 2-speed
		Number	2
MATERIALS & CONSTRUCTION		Hp.	15/3.75
Units Reqd.	1	R.P.M.	1750/875
Coils per Unit	1	Volts	440
Coil Width	7.5'	Phase	3
Tubes per Coil	179 - 4 rows	Cycle	60
Tube Length	24'	Gas Eng.: Make	
Tubes	Wolverine 1" OD x 18 BWG Inhibited Admiralty	Model	
Fins	Extruded Smooth Aluminum, 9/inch, 1/2" x 0.019"	Number	
Headers	Fabricated Steel, Box Brass Plug	Hp.	
Connections	In 8" 150# R.F. Out 6" - 150# R.F.	R.P.M.	
No. Passes	2 Reducing - 75% - 25%		

MTD = 40.8 based on straight line heat release. Composition indicates a 169° bubble point so this should be a very conservative selection.

The HAPPY COMPANY
TULSA, OKLAHOMA
SPECIFICATIONS FOR

HAPPY FIN-TYPE COOLERS

Page 5

Customer	Dresser Engineering Company	Quotation No.	9117
Address	P. O. Box 2518, Tulsa, Oklahoma	Date	March 4 , 1953
Plant	AtlanticRefining Company, Livington, New Mexico	Reference	Item 2-5
Cooler Model	HFR-60		
Service	Lean Oil Cooler		
	INSIDE TUBES	AIR DATA	
Fluid	Lean Oil	Elevation, Feet	3700
Sp. Gr. @ 60°F.	.825	Inlet Air Temp., °F.	100
Sp. Ht. @ Avg. Temp.		Total SCFM	46,500
Density @ Avg. Temp.		SCFM/Fan	46,500
Viscosity @ 156.5°F	1.49	Temp. Rise, °F.	30.5
Mol. Wt.	200	MECHANICAL EQUIPMENT	
Fouling factor	0.002	Fans: Type	Moore 24, Axial Flow
	U based on bare tube surface 98.4	No.	1
Qty.—Gas		Dia.	7'
H. C. Vapor, #/Hr.		No. Blades	5
Steam, #/Hr.		RPM	557
H. C. Liquid, #/Hr.	90,659	Hp./Fan	10.5
Water		Total Hp.	10.5
MTD	41.2	Gears: Make	Cleveland Worm
Inlet Temp, °F.	173	Model	CU-100
Outlet Temp., °F.	140	Rating AGMA	15.7 HP
Operating Press., PSI	46	Ratio	3-1/7 to 1
Pressure Drop, PSI	10	V-Belt Drive:	
Design Pressure, PSIG	125	Fan Sheave	
Test Pressure, PSIG	190	Drive Sheave	
Design Temperature, °F.	250	V-Belts	
Heat Exchanged, BTU/Hr.	1,538,000		
Surface, Sq. Ft.	Bare tube 376. Total external 5180	Motors: Make	Louis Allis
Surface per Coil, Sq. Ft.	Bare tube 376. Total external 5180	Type	Expl. proof - 1-speed
		Number	1
	MATERIALS & CONSTRUCTION	Hp.	15
Units Reqd.	1	R.P.M.	1750
Coils per Unit	1	Volts	440
Coil Width	7.5'	Phase	3
Tubes per Coil	179 - 4 rows	Cycle	60
Tube Length	8'	Gas Eng.: Make	
Tubes	Wolverine 1" x 18 BWG Inhibited Admiralty	Model	
Fins	Extruded, Smooth, 9/inch, 1/2" x .019" Aluminum	Number	
Headers	Fabricated Steel, Brass Plug	Hp.	
Connections	6" - 150# R.F.	R.P.M.	
No. Passes	10		

XII. ANALYSIS OF UNITS NOT TESTED

A. LEAN OIL COOLER

The specifications for this unit are given in Section XI. This unit had been field tested during 1954 by the Happy Co. and the Dresser Engineering Co. The results of their tests indicated that this unit was performing at about the same level as the jacket water cooler. The jacket-water-cooler performance, as indicated in Sections II and III of this report, was due to the air-side coefficient being considerably lower than that given by the Happy Co. rating sheet. Therefore, the lean-oil-cooler design was evaluated in terms of the ALCO data converted to 9 fins per inch. This analysis is given in Appendix I and is summarized below.

The design air flow rate of 46,500 std cu ft/min as given in the specifications was assumed to be that delivered by the fans. The corresponding face velocity is computed as 760 std ft/min. The design heat duty as given in the specification sheets for the unit is 1,538,000 Btu/hr.

TABLE XII

PREDICTED PERFORMANCE OF LEAN OIL COOLER

(Based on ALCO data converted to 9 fins per inch, assuming fan produces 760 ft/min face velocity)

V_{face}, std ft/min	760
Q, Btu/hr	1,210,000
Q/Q_{design}	79%
U_o (based on outside area)	5.49
$(U_o)_L$ (based on liner area)	66.9
$U_o/U_{o(specified)}$	71%
$t_{outlet\ oil}$, °F	147.5
$(h_o)_L$ (liner area)	133
h_o (outside area)	9.7
h_i (based on inside area)	216
$r_{fouling\ inside}$	0.002
r_{metal}	0.00062
ΔT_M, °F	48.1

For comparison purposes, similar calculations were made, using the outside air film coefficient obtained from 651C. The analysis is given in Appendix I and is summarized in Table XIII.

TABLE XIII

LEAN-OIL-COOLER ANALYSIS BASED ON 651C AND V_{face} OF 760 STD FT/MIN

V_{face}, std ft/min	760
Q, Btu/hr	1,430,000
Q/Q_{design}	93%
U_o (based on outside area)	6.4
$(U_o)_L$ (based on liner area)	87.8
$U_o/U_{o(specified)}$	89%
$t_{outlet\ oil}$, °F	142.5
$(h_o)_L$ (liner area)	252
h_o (outside area)	18.4
h_i (based on inside area)	216
$r_{fouling\ inside}$	0.002
r_{metal}	0.00062
ΔT_M, °F	43.2

For comparison purposes, similar calculations were made, using the outside air film coefficient obtained from the Happy Co. rating curve. The analysis is given in Appendix I and is summarized in Table XIV

TABLE XIV

LEAN-OIL-COOLER ANALYSIS BASED ON HAPPY CO. RATING CURVE AND V_{face} OF 760 STD FT/MIN

V_{face}, std ft/min	760
Q, Btu/hr	1,390,000
Q/Q_{design}	90.5%
U_o (based on outside area)	6.08
U_o (based on liner area)	83.5
$U_o/U_{o(specified)}$	85%
$t_{outlet\ oil}$, °F	143.5
h_o(liner)	220
h_i(inside area)	216
r_i	.002
r_{metal}	.00062
ΔT_M, °F	44.2

The specified heat duty of 1,538,000 Btu/hr cannot be realized by use of the unit specified. If the face velocity of 760 ft/min is obtained from the fan specified, the unit could be expected to perform at about 71% of design demand. On the basis of the Happy Co. rating curve, the unit would be 9.5% low on heat duty and 15% low on overall coefficient if the rating curve was correct.

B. DEBUTANIZER OVERHEAD CONDENSER

The specification sheet for this unit given in Section XI calls for a heat duty of 4,800,000 Btu/hr with an inlet hydrocarbon temperature of 174°F and an exit hydrocarbon temperature of 140°F. The design inlet air temperature is specified as 100°F, with an air temperature rise of 35.6°F.

The computed bubble point of the hydrocarbon mixture as given in Appendix J is 172°F and the dew point is 174°F. Therefore, the inlet mixture is saturated vapor which condenses over a 2°F range. The condensate must subcool 32°F, from 172° to 140°F. The computed condensing load on this basis is 4,070,000 Btu.hr and the subcooling load is 724,000 Btu/hr for a total of 4,794,000 Btu/hr. This checks the specified heat load.

A major consideration in this design is one of insuring that the subcooling load will be obtained. The proposed design does not provide any means for controlling the condensate subcooling film coefficient. It should be emphasized that a partially flooded tube will have (a) a low condensate subcooling film coefficient resulting in high film resistance to heat transfer and (b) a tube-wall temperature that will approach the condensing-vapor temperature. In order to insure that the subcooling will be realized the bottom row of tubes should be completely flooded.

The proposed design had two passes on the tube side with the top three rows as the first pass and the bottom row as the second pass. The condensing coefficient was computed to be 300, based on the inside surface area. The air-side coefficient as determined by the ALCO data converted to 9 fins per inch was 9.28 for a face velocity of 700 std ft/min. This results in an overall condensing coefficient of 5.73, based on the outside area. The mean temperature difference for the condensing zone is 49.7°F. Using the condensing heat duty of 4,070,000 Btu/hr and the above data, a condensing surface area of 13,100 sq ft (outside area) is required. The top three rows of the proposed design provide 11,300 sq ft or a deficiency of 1800 sq ft, which amounts to 13.7%.

Considering the subcooling zone to be the bottom row of tubes completely flooded, a condensate cooling film coefficient of 86.4, based on the inside area, is computed. If the air-side coefficient is taken as 9.28, the same as above, and is combined with the inside coefficient of 86.4, an overall coefficient of 3.11 is obtained, based on the outside area. The temperature-difference driving force for this zone is 52.5°F. The required subcooling heat transfer area, based on the subcooling load of 724,000 Btu/hr, is 4380 sq ft of outside area. The bottom row actually provides only 3800 sq ft. The deficiency of 580 sq ft amounts to 13.3%.

Since no provisions were made for flooding the bottom row of tubes, the bottom row will be only partially flooded. This will result in the exposing of additional condensing surface area to the condensing vapors. Therefore, the 13.7% deficiency of condensing area will be eliminated, resulting in the condensation of the required condensing load. The required degree of subcooling will not be obtained under this arrangement.

The analysis given above, based on ALCO data, indicates that the proposed design will not provide the required area for effecting the specified heat transfer.

A similar set of calculations based on 651C can be made for comparison with the above analysis. Only the air-side film coefficient and Δt will be greatly affected. The air-side film coefficient was determined to be 16.6, based on outside area. The overall coefficient for the condensing zone was computed to be 7.86 with a Δt of 49.7°F. This results in a required condensing area of 10,520 sq ft (outside area). The top three rows provide 11,300 sq ft, or an excess of 7.6%.

The overall coefficient for the subcooling zone, assuming the bottom row is flooded, was computed to be 3.95, based on the outside area. The Δt was computed to be 52.2°F. The corresponding required external surface area is 3800 sq ft. The bottom row provides 3800 sq ft.

The analysis given above, based on 651C, indicates that if the bottom row were flooded, the unit would handle the specified heat duty. This assumes that the face velocity of 700 SFM is provided by the fan.

The air-side coefficient as given by the Happy Co. rating curve at 700 std ft/min is 220, based on the liner area. This corresponds to 16.05, based on the outside area. This compares to 16.6, based on 651C. The condensing-zone overall coefficient becomes 7.73, based on outside area. This is 1.6% lower than the 651C value of 7.86. Therefore, the required condensing area is 10,720 sq ft. This compares with 11,300 sq ft provided by the top three rows for an excess area of 5.4%.

The overall coefficient for the subcooling zone, based on the Happy rating line, is 3.91, based on outside area. The required subcooling area becomes 3840 sq ft as compared to the 3800 sq ft provided by the bottom row.

The field tests conducted on the jacket water cooler essentially substantiate the converted ALCO data for predicting the air-side coefficient for three-rows-in-line, fourth-row-staggered arrangement. The most reliable prediction of performance of the debutanizer condenser is that predicted by the converted ALCO data. The Happy Co. rating curve and 651C both overrate the air-side coefficient. It is therefore concluded that the proposed design is not capable of providing both the required condensing load and the required subcooling load.

C. EVAPORATOR CONDENSER

The specification sheet for this unit given in Section XI calls for a heat duty of 3,970,000 Btu/hr with an inlet steam temperature of 240°F and an outlet condensate temperature of 200°F. The design inlet air temperature is specified as 100°F with an air temperature rise of 79.2°F.

The required heat duty consists of two parts: (a) that of condensing 4000 lb of saturated steam per hour and (b) that of subcooling the resulting condensate 40°F. These heat loads are 3,808,000 and 160,000 Btu/hr, respectively.

The proposed design provides a total external area of 4120 sq ft for accomplishing both the required condensing and the subcooling load.

The specification sheet calls for an MTD of 78°F. If the inlet air temperature is 100°F, the exit air temperature is 179.2°F, the steam inlet temperature is 240°F, and the condensate outlet temperature is 200°F, a LMTD based on these terminal temperatures of 78.7°F is obtained. There is an implication that the use of this conservative MTD, when employed in conjunction with some overall coefficient, will give the proper heat transfer area. The problem here is one of determining the applicable overall coefficient. The overall coefficient given in the specification sheet is 172, based on liner area, or 12.55, based on the outside area.

It can be readily shown that the overall coefficient of 12.55 is considerably larger than the applicable overall coefficient. The heat exchanger consists esentially of two zones, a condensing zone and a subcooling zone. Each zone has its effective overall coefficient and corresponding LMTD driving force.

An overall coefficient for the condensing zone can be computed from the data given in 651C, Table 12, page 9. This overall coefficient per foot of tube is 50.5; this assumes no fouling. This value corresponds to 14.05, based on outside area. Allowing for inside fouling of 0.001 reduces this value to 11.6 for the condensing zone. This value is 8.6% lower than the Happy Co. value of 12.55 for the entire unit.

The subcooling-zone overall coefficient will be considerably lower than 11.6. It must be emphasized that it is necessary to insure complete flooding of some tubes in order to obtain the degree of subcooling specified for this unit. If the tubes in the subcooling section are partially filled with condensing steam, the tube-wall temperatures will approach those computed in Appendix K. It is shown in this appendix that if steam is present in the tube, it is impossible to attain the specified outlet steam condensate temperature of 200°F. If the bottom row of the evaporator condenser is completely flooded, the overall subcooling coefficient for this zone can be shown to be 4.53, based upon the air film coefficient given by 651C.

The overall subcooling-section coefficient of 4.53 can be combined with the condensing-section overall coefficient of 11.6 to give an effective overall coefficient of 11.0 for the entire unit. This mean overall coefficient is obtained by combining the overall coefficient for each section with its respective Δt and area. The mean overall coefficient of 11.0 predicted by 651C is 12.3% lower than the Happy Co. value of 12.55.

Since the jacket-water-cooler field test results indicate that the air film coefficient predicted by 651C is too high, it is believed that the unit is not capable of producing both the required condensing load and the required subcooling load.

The ALCO data converted to 9 fins per inch are limited to the three-rows-in-line, fourth-row-staggered arrangement. These data do not apply to a two-rows-in-line, third-row-staggered arrangement. Therefore, no analysis was made, using these data.

D. STILL OVERHEAD PARTIAL CONDENSER

The specification sheet for this unit calls for a heat duty of 7,747,000 Btu/hr with an inlet hydrocarbon and steam temperature of 180°F. The exit vapor-liquid mixture is to be 140°F. The molecular weight of hydrocarbons entering is 58.4 and the molecular weight of condensing material is 69.3. The air to the unit is 100°F with a temperature rise of 27.7°F.

In order to evaluate the proposed design for this application, the feed-stream composition used as the basis for the design must be available. The only composition available was that provided by the Happy Co. as the result of a field test made on August 11, 1954. Whether or not this composition checks with the design composition is not known. A series of calculations based on this available stream composition was made. First, a check of the molecular weight of the feed stream was made and was found to be 57.45, as compared to the 58.4 value given above. The computed molecular weight of the condensed material was calculated as 68.0, as compared to the 69.3 value given above. These computed values appear to check fairly well with the specification-sheet values. The computed heat duty, using the specification sheet flow rates and the field test analysis, is 5,729,000 Btu/hr, as compared to 7,747,000 Btu/hr given in the specifications. Calculations indicate that all the steam does not condense in this unit. This accounts for part of the reduced computed heat duty.

The feed stream contains eight components, C_2, C_3, iC_4, nC_4, iC_5, C_6, C_7^+, and steam. In flowing through the heat exchanger, about 29% of the material condenses (based on field analysis). This compares with the specification sheet's computed condensing value of 37.5%.

The mechanisms involved in partial condensing are complex. The condensing material must diffuse to the condensing surface by means of molecular diffusion or eddy diffusion. The remaining gas stream must cool by convective heat transfer. The complex mechanisms involved when eight components are partially condensed simultaneously at varying rates throughout the exchanger are such that exact rate analysis can be made only by use of electronic digital computers. Therefore, recourse must be made to the use of empirical short-cut methods. In the case at hand, the condensing film coefficients will be very large in comparison with the hydrocarbon-gas film cooling coefficient. The air-side coefficient will be very low and the combination of hydrocarbon-gas phase cooling resistance with the air-side film resistance will control the performance of the unit. An evaluation of the unit can be made on this basis by proper distributions of coefficients. The computed gas film cooling resistance is frequently reduced by multiplying by the ratio of the gas cooling sensible heat duty to the total heat transfer heat duty. An analysis of the proposed unit was made on this basis.

The unit designed contained two passes on the tube side. The inlet top pass contained three rows of tubes and the bottom pass contained one row of tubes. Inside gas film coefficients were computed for both the upper and the lower passes at the average temperature. The values of h_i obtained were 62.5 and 92 for the upper and lower pass, respectively. As can be seen from the numerical values of these coefficients, the effect of the decreased flow area in the lower pass as compared to the upper pass is to increase the inside coefficient for this region, even though the vapor-mass flow rate has been reduced through condensation of part of the vapor.

An analysis using the inside gas film coefficients given above, combined with an air-side coefficient computed from 651C, was made. This analysis resulted in a predicted required area for the design conditions and the field-test feed-stream composition of 23,100 sq ft of external area. The area provided in this unit is 34,050 sq ft for an excess area of 48% over that required.

A similar analysis, using an air-side coefficient given by the converted ALCO data, predicts a required area of 33,200 sq ft (external) as compared to the provided area of 34,050 sq ft. This amounts to 2.5% excess area over that provided.

In both the above analyses the condensing film coefficient has been ignored. The diffusion-rate phenomenon also has been neglected. The assumption has been made that the gas film coefficient which is controlling can be increased due to the fact that the higher condensing coefficient results in a greater overall effective coefficient. In the case under consideration, the condensing load is about 75% of the total heat duty. This would result in a much larger effective overall coefficient than the hydrocarbon-gas film coefficient alone would indicate.

On the basis of the above analysis, the proposed design would be expected to give the specified outlet hydrocarbon temperature when the specified mass of material of field-test composition is fed to the unit.

Results of tests made by the Dresser Engineering Co. and the Happy Co. on August 8, 1954, indicated that this unit had a performance index of 65%. This discrepancy with the calculated values, using the converted ALCO data, could be due to one or more of the following reasons:

1. The field-test composition and pressure of the inlet vapor mixture for this test indicates a dew point of approximately 182°F as compared to the inlet feed temperature given as 192.8°F. This indicates that an appreciable amount of superheat was present in the feed gas mixture. The removal of the superheat prior to the start of condensation was not provided for in the design and would tend to reduce the performance index of this unit.

2. It was determined that the heat duty specified was based on the assumption that all the steam entering the unit would condense. Calculations indicate that approximately one-fourth of the steam entering the unit would not be condensed. This would cause the heat load of the unit to be less than the value given in the specifications when the composition flow rates, and temperatures were those specified. The net effect of this discrepancy would be to reduce the performance index of this unit.

3. The fan-blade angle for this test was less than the maximum attainable, due to mechanical difficulties. This reduction of blade angle would decrease the air velocity past the tubes. The reduced flow rate of air would not only result in a lowering of the outside coefficient, but also cause a decrease in the mean temperature difference due to the greater temperature rise of the air. Since the performance index given as 65% was computed while assuming design flow rate and a heat balance, the real performance index should be appreciably higher than that indicated.

E. CONCLUSIONS

1. Lean Oil Cooler.—This unit cannot be expected to perform in accordance with the specifications.

2. Debutanizer Overhead Condenser.—This unit cannot be expected to perform in accordance with the specifications.

3. Evaporator Condenser.—This unit cannot be expected to perform in accordance with the specifications.

4. Still Overhead Partial Condenser.—This unit can reasonably be expected to perform in accordance with the specifications.

APPENDICES

APPENDIX A

CALCULATIONS ON DATA TAKEN IN LOVINGTON, NEW MEXICO

CALCULATIONS ON THE PERFORMANCE OF THE JACKET WATER COOLER FROM DATA TAKEN THURSDAY, APRIL 28, 1955

A. East Bay.—

1. Data.

Velocity profile (see Table I)
Average anemometer velocity = 539.9 ft/min
Inlet air temperature (avg) = 72.1°F
Outlet air temperature (avg):

Ref. EMF	Position	$(MV)_{read}$	$(MV)_{32°F}$	Temp °F
1.36 MV	3CR	1.875	3.235	143.1
	3CL	1.93	3.29	145.0
	3DR	1.77	3.13	139.7
	3DL	1.755	3.115	139.1
	3EL	1.82	3.18	141.4
	3ER	1.715	3.075	137.9
			avg =	141.0°F
	4CR	1.46	2.90	131.7
1.44	4CL	1.625	3.065	136.4
	4DR	1.50	2.94	133.0
	4DL	1.715	3.15	140.4
	4EL	1.50	2.94	133.0
	4ER	1.49	2.93	132.7
			avg =	134.5°F

Average outlet temperature:

$$\frac{141.0 + 134.5}{2} = 137.8°F \text{ average temperature}$$

Inlet water temperature (avg) = 162.5°F
Outlet water temperature (avg) = 154.5°F

2. Calculation of heat duty.

$$Q = W\,C_p\,\Delta t$$

The face area of the tube bay is (allowing four inches for rolling in and end effects):

$$A_{bay} = (23.67) \times \left(\frac{92.5}{12}\right) \times 2 = 366 \text{ ft}^2$$

Average indicated air velocity = 539.9 ft/min
Average exit air temperature = 137.8°F

The corrected face velocity (correction for density) =

$$539.9\sqrt{\frac{530}{597.8}} = 508 \text{ std ft/min}$$

Standard density = 0.074 lb/cu ft
Face area of the bay = 366 ft^2
Lb/hr of air (indicated) = (60)(366)(0.074)(508) = 825,000 lb/hr
Air-side heat duty:

$$Q = W\,C_p\,\Delta t$$

$$\Delta t_{air} = 137.8 - 72.1 = 65.7°\text{F}$$

$$C_{p(air)} = 0.24 \text{ Btu/lb - °F}$$

$$Q = (825{,}000)(0.24)(65.7) = 13{,}000{,}000 \text{ Btu/hr}$$

3. Calculation of ΔT_{LM}.

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ell n \dfrac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_1 = 154.5 - 72.10 = 82.4°\text{F}$$

$$\Delta T_2 = 162.5 - 137.8 = 24.7°\text{F}$$

$$\Delta T_{LM} = \frac{82.4 - 24.7}{\ln \frac{82.4}{24.7}} = \frac{57.7}{1.204} = 47.8°F$$

ΔT_{LM} correction (Kern) is negligible

4. Calculation of overall coefficient.

The overall coefficient is obtained by:

$$U = \frac{Q}{A \Delta T_{LM}}$$

From the Happy Co. specification sheets:

$$A_L = 2250 \text{ ft}^2 \text{ of liner area}$$

$$U_L = \frac{13,000,000}{(2250)(47.8)} = 121 \text{ Btu/hr-°F-ft}^2 \text{ (liner area)}$$

$$U_o = \frac{121}{13.7} = 8.84 \text{ Btu/hr-°F-ft}^2 \text{ (outside area)}$$

5. Calculation of corresponding water flow rate.

Assuming an air-side—water-side heat balance, the water-side flow rate is:

$$W = \frac{Q}{C_p \, \Delta t_{water}} = \frac{13,000,000}{(1)(162.5 - 154.5)} = 1,625,000 \text{ lb/hr}$$

The water-flow area is:

$$A_{flow} = N \, A_{flow/tube}$$

$$A_{flow/tube} = \frac{\pi (0.902)^2}{(4)(144)} = 0.00443 \text{ ft}^2\text{/tube}$$

$$A_{flow} = (179)(0.00443) = 0.794 \text{ ft}^2 \text{ per pass per bay}$$

The water velocity inside the tubes is:

$$V = \frac{1,625,000}{(0.794)(3600)(61)} = 9.33 \text{ ft/sec}$$

This is too high a velocity.

6. Water flow rate from the Happy Co. specification sheet.

$$GPM = 3886 \text{ from specification sheet}$$

$$W = \frac{(3886)}{(2)7.48(60)} = 4.34 \text{ cu ft/sec/bay}$$

$$V = \frac{4.34}{0.794} = 5.47 \text{ ft/sec}$$

The velocity from air-side data as computed in the previous section was 9.33 ft/sec. This value is 71% higher than specified.

7. Calculation of the outside coefficient.

The outside coefficient will be computed from the relationship:

$$\frac{1}{U_o} = \frac{1}{h_o} + r_o + r_m + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i h_i}$$

where

U_o = overall coefficient of heat transfer,
h_o = outside film coefficient,
r_o = outside fouling film resistance,
r_m = metal resistance,
r_i = inside fouling film resistance, and
h_i = inside water film coefficient.

a. Calculation of inside film coefficient.

The inside water film coefficient will be determined by

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

$$(t_w)_{avg} = \frac{162.5 + 154.5}{2} = 158.5°F$$

$$d_i^{0.2} = (0.902)^{0.2} = 0.979 \cong 0.98$$

Using the water velocity computed from the air-side data,

$$V_t^{0.8} = (9.33)^{0.8} = 6.0$$

Substituting

$$h_i = \frac{150[1 + 0.011(158.5)]6.0}{0.98} = 2520$$

$$\frac{A_o}{A_i} = \frac{3.59 \times 12}{(\pi)(0.902} = 15.2$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{2520} = 0.00604 \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}} .$$

b. Outside fouling.

The outside fouling will be assumed equal to zero:

$$r_o = 0.$$

c. Metal resistance.

The computed outside coefficients will be compared with other corresponding coefficients which include in them a copper liner resistance and a root wall resistance; therefore, the metal resistance to be used here will be the difference between an admiralty and a copper wall resistance. The metal resistance will be given by

$$r_m = \frac{XA_o}{A_m K_1} - \frac{XA_o}{A_m K_2}$$

$$= \frac{XA_o}{A_m}\left[\frac{K_2 - K_1}{K_1 K_2}\right] ,$$

where

X = liner metal thickness, ft,
A_o = outside heat transfer area, ft²/ft,
A_m = mean metal heat transfer area, ft²/ft,
K_1 = thermal conductivity of admiralty, Btu/hr-°F-ft,
K_2 = thermal conductivity of copper, Btu/hr-°F-ft.

$$X = \frac{0.049}{12} = 0.00408 \text{ ft},$$

$$A_o = 3.59 \text{ ft}^2/\text{ft},$$

$$A_m = 0.249 \text{ ft}^2/\text{ft (liner)},$$

$$K_2 = 220 \text{ Btu/hr-°F-ft (copper), and}$$

$$K_1 = 64 \text{ Btu/hr-°F-ft (admiralty).}$$

Substituting

$$r_m = \frac{(0.00408)(3.59)}{0.249}\left[\frac{220 - 64}{(220)(64)}\right]$$

$$= 0.000652 \; \frac{\text{hr-°F-ft}^2 \text{ (outside area)}}{\text{Btu}} .$$

d. Inside fouling.

Reference is made to Section VIII of this report, in which the inside fouling was determined to be

$$r_i\left(\frac{A_o}{A_i}\right) = 0.0175 \text{ (based on the outside area).}$$

e. Calculation of outside coefficient.

$$\frac{1}{h_o} = \frac{1}{U_o} - \left[\frac{A_o}{A_i h_i} + \frac{A_o}{A_i} r_i + r_m\right]$$

Substituting from Sections a, b, c, and d:

$$\frac{1}{h_o} = \frac{1}{8.84} - [0.00604 + 0.000652 + 0.0175]$$

$$\frac{1}{h_o} = 0.1131 - 0.02419 = 0.0889$$

$$h_o = 11.26 \text{ Btu/hr-°F-ft}^2 \text{ (outside area)}$$

$$h_{o(\text{liner})} = 11.26 \times 13.7 = 154.5 \text{ (based on liner area)} .$$

(The above values of h_o are based on the field test data without the application of the duct correction factor.)

B. West Bay.—

1. Data.

Velocity profile (see Table I)

average anemometer velocity = 470.1 ft/min.

Inlet air temperature (avg) = 75.7°F
Outlet air temperature (avg):
(Reference temperature = 27.8°C = 82.0°F = 1.44 MV)

Position	MV reading	0°C, MV	Temperature °F
1CR	1.43	2.87	130.6
1CL	1.60	3.04	136.4
1DR	1.465	2.905	131.7
1DL	1.485	2.925	132.5
1EL	1.49	2.93	132.7
1ER	1.47	2.91	132.0
		Avg	132.65°F

Position	MV reading	32°F, MV	Temperature °F
2CR	1.375	2.735	126.2
2CL	1.60	2.96	133.7
2DR	1.46	2.82	129.0
2DL	1.445	2.805	128.5
2EL	1.545	2.905	131.9
2ER	1.60	2.96	133.7
		Avg	131.0 °F

Average outlet temperature:

$$\frac{132.65 + 131.0}{2} = 131.8°F$$

Inlet water temperature (avg) = 163.5°F
Outlet water temperature (avg) = 156.0°F

2. Calculation of heat duty.

$$Q = W \ C_p \ \Delta t$$

The face area of the tube bay is (see Section A) 366 sq ft.

Average indicated air velocity = 470.1 ft/min
Average exit air temperature = 131.8°F
The corrected face velocity (correction for density) =

$$470.1\sqrt{\frac{530}{591.8}} = 445 \text{ ft/min.}$$

Std density = 0.074 lb/cu ft
lb/hr of air (indicated) =

$$(60)(366)(0.074)(445) = 723{,}000 \text{ lb/hr}$$

$$C_p = 0.24 \text{ Btu/lb-°F}$$

$$\Delta t_{air} = 131.8 - 75.7 = 56.1\text{°F}$$

$$Q = (0.24)(56.1)(723{,}000) = 9{,}730{,}000 \text{ Btu/hr}$$

3. Calculation of ΔT_{LM}.

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_1 = 156.0 - 75.7 = 80.3\text{°F}$$

$$\Delta T_2 = 163.5 - 131.8 = 31.7\text{°F}$$

$$\Delta T_{LM} = \frac{80.3 - 31.7}{\ln \frac{80.3}{31.7}} = \frac{48.6}{0.93} = 52.3\text{°F}$$

ΔT_{LM} correction (TEMA) is negligible.

4. Calculation of the overall coefficient.

$$U = \frac{Q}{A\Delta T_{LM}}$$

From Section A-4

$$A_L = 2250 \text{ sq ft of liner area.}$$

Substituting:

$$U_L = \frac{9{,}730{,}000}{(2250)(52.3)} = 82.8 \frac{\text{Btu}}{\text{hr-°F-ft}^2\text{(liner area)}}$$

$$U_o = \frac{82.8}{13.7} = 6.05 \frac{\text{Btu}}{\text{hr-°F-ft}^2} \text{ (outside area).}$$

5. Calculation of corresponding water flow rate.

Assuming a water-side, air-side heat balance, the water-side flow rate is

$$W = \frac{Q}{C_p \, \Delta t_{water}} = \frac{9,730,000}{(1)(163.5 - 156)} = 1,390,000 \text{ lb/hr.}$$

From Section A-5, the water flow area is

$$A_{flow} = 0.794 \text{ sq ft per pass per bay.}$$

The water velocity inside the tubes is

$$V = \frac{1,390,000}{(0.794)(3600)(61)} = 8.0 \text{ ft/sec.}$$

This is too high a velocity since the velocity computed from the specification sheet is 5.47 ft/sec (section A-6). The velocity of 8.0 is 46% higher than that specified.

6. Calculation of the outside coefficient.

The outside coefficient will be computed in the same manner as Section A.

a. Calculation of inside film coefficient.

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

$$d_i^{0.2} = 0.98$$

$$(t_w)_{avg} \cong 160°F$$

$$V_t^{0.8} = 8.0^{0.8} = 5.28.$$

Substituting:

$$h_i = \frac{150[1 + 0.011(160)]5.28}{0.98} = 2230$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{2230} = 0.00681 \text{ (based on outside area).}$$

b. Outside fouling.

$$r_o \text{ (assumed)} = 0.$$

c. Metal resistance (see Section A-7c).

$$r_m = 0.000652 \text{ (based on outside area).}$$

d. Inside fouling (see Section A-7d).

$$r_i\left(\frac{A_o}{A_i}\right) = 0.0175 \text{ (based on outside area).}$$

e. Calculation of outside coefficient.

$$\frac{1}{h_o} = \frac{1}{6.05} - [0.00681 + 0.000652 + 0.0175]$$

$$= 0.1655 - 0.02496 = 0.1405$$

$$h_o = \frac{1}{0.1405} = 7.13 \text{ (based on outside area)}$$

$$h_o\text{(liner)} = (7.13)(13.7) = 97.7 \text{ (based on liner area).}$$

(The above values of h_o are based on the field test data without the application of the duct correction factor.)

C. Summary.—

	East Bay	West Bay	Total
Q, Btu/hr	13,000,000	9,730,000	22,730,000
ΔT_{LM}, °F	47.8	52.3	---
U_o (based on outside area)	8.84	6.05	---
U_L (based on liner area)	121	82.8	---
V_t, ft/sec	9.33	8.0	---
h_o	11.26	7.13	---
$(h_o)_L$	154.5	97.7	---
V_{face}, std ft/min	508	445	---
Water flow rate, lb/hr	1,625,000	1,390,000	3,015,000
$h_{water\ flow}$	2520	2230	---

APPENDIX B

DATA OF MONDAY, MAY 2, 1955

THREE-IN-LINE, FOURTH-ROW-STAGGERED BAY

1. Air-Side (Indicated) Heat Duty.—

Average indicated air velocity = 785.6 ft/min
Average exit air mev = 1.317
Average cold junction temp = 29.1°C = 84.4°F
84.4°F = 1.50 mev
Total EMF = 1.50 + 1.317 = 2.817
Corresponding outlet temperature = 128.5°F
Temperature rise of air = 128.5 - 80.8 = 47.7°F
Lb per hour of air = 785.6 ft/min at 128.5°F and 29.87 in. Hg
Corrected face velocity = $785.6\sqrt{530/588.5}$ = 747 std ft/min
Std density = 0.074 lb/cu ft
Face area = 366 sq ft (see Appendix I, Section A)
Lb of air per hour (indicated)

$$= (366)(747)(0.074)(60)$$
$$= 1{,}215{,}000 \text{ lb/hr}$$

$$Q = (1{,}215{,}000)(0.24)(47.7)$$
$$= 13{,}900{,}000 \text{ Btu/hr.}$$

2. Heat Transferred From Water-Side Data.—For comparison purposes, the heat duties computed from the water-side measurements are as follows:

Inlet water temperature = 152.7°F
Exit water temperature = 143.6°F
∴ water temperature drop = 9.1°F
average water temp = 148°F
average water density = 61.3 lb/cu ft

The water velocity was determined by an orifice-plate pressure-drop measurement. The orifice diameter was 11.25 in. The water pipe was a 16-in. schedule-20 pipe (ID = 15.375 in. = 1.286 sq ft). The manometer reading was 66 in. of water (5.5 ft). The relationship for determining the velocity of flow through the orifice is given by*

$$V_o = \frac{q}{S_o} = \frac{C\sqrt{2g\ Hv}}{\sqrt{1 - \left(\frac{S_o^2}{S_i^2}\right)}} ,$$

where

V_o = orifice flow velocity, ft/sec,
S_o = orifice flow area, sq ft,
C = orifice coefficient, dimensionless
g = acceleration of gravity, 32.2 ft/sec^2
H_v = pressure drop across the orifice, feet of flowing fluid, and
S_i = cross-sectional flow area of the pipe, sq ft.

By a trial-and-error procedure the orifice coefficient is determined from a Reynolds number plot.** The Reynolds number is over two million, therefore the orifice coefficient is taken equal to 0.61. The value of H_v is 5.5 ft. Therefore,

$$V_o = \frac{0.61\sqrt{2(32.2)(5.5)}}{\sqrt{1 - (0.535)^2}}$$

$$= \frac{0.61(18.8)}{0.845} = 13.58 \text{ ft/sec.}$$

The pipe velocity is

$$V = 13.58(0.535) = 7.26 \text{ ft/sec.}$$

The lb per hour of water flowing through the heat exchanger is 7.26(1.286)(3600)(61.3) = 2,060,000. It is assumed that since this water was split between two units, one-half of the water flowed through the unit under consideration. Therefore, the total heat transferred is

$$Q = (w\ C_p\ \Delta t)_{water}$$

$$= \frac{2,060,000}{2}(1)(9.1)$$

$$= 9,380,000 \text{ Btu/hr.}$$

---

*G. G. Brown et al., Unit Operations. New York: John Wiley and Sons, 1950, p. 158.

**Loc. cit.

3. Comparison of Heat Duties.—

Air side = 13,900,000 Btu/hr
Water side = 9,380,000 Btu/hr
Discrepancy = 4,520,000 Btu/hr

$$\% \text{ greater than water side} = \left(\frac{4{,}520{,}000}{9{,}380{,}000}\right)100 = 48\%$$

4. Calculation of Overall Coefficient From Air-Side Data.—

$$U = \frac{Q}{A \Delta T_{LM}}$$

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ell n \dfrac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_1 = 143.6 - 80.8 = 62.8°F$$

$$\Delta T_2 = 152.7 - 128.5 = 24.2°F$$

$$\Delta T_{LM} = \frac{62.8 - 24.2}{\ell n \dfrac{62.8}{24.2}} = \frac{38.6}{0.954} = 40.5°F$$

$$A_{liner} = 2250 \text{ sq ft}$$

Substituting:

$$U_L = \frac{13{,}900{,}000}{(2250)(40.5)} = 152.8 \text{ (based on liner area)}$$

$$U_O = \frac{152.8}{13.7} = 11.15 \text{ (based on outside area).}$$

5. Calculation of Corresponding Water-Side Flow Rate.—

$$W = \frac{Q}{C_p \Delta t} = \frac{13{,}900{,}000}{(1)(152.7 - 143.6)} = 1{,}528{,}000 \text{ lb/hr}$$

Water flow area:

$$A_{flow} = 0.794 \text{ sq ft per pass per bay.}$$

The water velocity inside the tubes is

$$V_t = \frac{1,528,000}{(0.794)(61)(3600)} = 8.76 \text{ ft/sec.}$$

6. Calculation of Air-Side Coefficient From Air-Side Data.—The procedure is the same as in Appendix A.

a. Calculation of water film coefficient.

$$h_i = \frac{150[1 + 0.011(147.7)](8.76)^{0.8}}{0.98}$$

$$= 2270$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{2270} = 0.0067 .$$

b. Outside fouling.

$$r_o = 0 \text{ (assumed).}$$

c. Metal resistance (see Section I-A7c).

$$r_m = 0.000652 \text{ (based on outside area).}$$

d. Calculation of outside coefficient, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{11.15} - [0.0067 + 0.000652]$$

$$= 0.0896 - 0.00735 = 0.08225$$

$$h_o = \frac{1}{0.08225} = 12.18 \text{ (based on outside area)}$$

$$(h_o)_L = 12.18 \times 13.7 = 166.5 \text{ (based on liner area).}$$

e. Calculation of outside coefficient, assuming inside fouling of 0.001 (based on inside area).

$$\frac{1}{h_o} = 0.0896 - [0.00735 + 0.0152] = 0.06705$$

$$h_o = \frac{1}{0.06705} = 14.95 \text{ (based on outside area)}$$

$$(h_o)_L = (14.95)(13.7) = 205 \text{ (based on liner area).}$$

7. Calculation of Overall Coefficient From Water-Side Data and Measured Air Temperatures.

$$U = \frac{Q}{A\ \Delta T_{LM}}$$

where Q = the water side heat duty of 9,380,000 Btu/hr,
A = 2250 sq ft of liner area, and
ΔT_{LM} = 40.5°F (see Section II-A4).

Substituting:

$$U_L = \frac{9{,}380{,}000}{(2250)(40.5)} = 103.0 \text{ (based on liner area)}$$

$$U_o = \frac{103.0}{13.7} = 7.52 \text{ (based on outside area).}$$

8. Calculation of Corresponding Air-Side Flow Rate.

$$W = \frac{Q}{C_p\ \Delta t} = \frac{9{,}380{,}000}{(0.24)(128.5 - 80.8)} = 820{,}000 \text{ lb/hr.}$$

The corresponding face velocity is

$$V_{face} = \frac{820{,}000}{(366)(60)(0.074)} = 504 \text{ std ft/min.}$$

9. Calculation of Air-Side Coefficient From Water-Side Data.—

a. Calculation of water film coefficient.

$$h_i = 2270 \left(\frac{9{,}380{,}000}{13{,}900{,}000}\right)^{0.8}$$

$$= 1655 \text{ (based on inside area)}$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{1655} = 0.0092 \text{ (based on outside area).}$$

b. Outside fouling assumed zero.

c. Metal resistance (see Section I-A7c).

$$r_m = 0.000652 \text{ (based on outside area).}$$

d. Calculation of outside coefficient, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{7.52} - [0.0092 + 0.000652] = 0.1241$$

$$h_o = \frac{1}{0.1241} = 8.05 \text{ (based on outside area)}$$

$$(h_o)_L = (8.05)(13.7) = 110 \text{ (based on liner area).}$$

e. Calculation of outside coefficient, assuming an inside fouling of 0.001 (based on the inside area).

$$\frac{1}{h_o} = \frac{1}{7.52} - [0.0092 + 0.000652 + 0.0152] = 0.1089$$

$$h_o = \frac{1}{0.1089} = 9.18 \text{ (based on outside area)}$$

$$(h_o)_L = 9.18 \times 13.7 = 126 \text{ (based on liner area).}$$

APPENDIX C

TRIANGULAR-PITCH BAY

1. Air Side (Indicated Heat Duty).—

Average indicated air velocity = 661.8 ft/min
Average exit air mev = 1.0335
Average cold-junction temperature = 35.03°C = 95.1°F
95.1°F = 1.8228 mv
Total EMF = 1.8228 + 1.0335 = 2.8563 mv
Corresponding outlet air temperature = 130.2°F
Average air inlet temperature = 79.5°F
Temperature rise of air = 130.2 - 79.5
= 50.7°F
Lb per hour of air:
661.8 ft/min at 130.2°F and 29.87 in. Hg
Corrected face velocity = $661.8\sqrt{530/590.2}$ = 626 std ft/min
Std density = 0.074 lb/cu ft
Face area = 366 sq ft
Lb of air per hour (indicated) =
(366)(626)(0.074)(60) = 1,019,000 lb/hr

$Q = W\,C_p\,\Delta t = (1{,}019{,}000)(0.24)(50.7)$
$= 12{,}410{,}000$ Btu/hr.

2. Heat Transferred From Water Side.—

Lb of water per hour (from previous section) = 1,030,000
Temperature of inlet water = 148.7°F
Temperature of outlet water = 140.5°F
Δt of water = 8.2°F

$Q = (1{,}030{,}000)(1)(8.2) = 8{,}450{,}000$ Btu/hr.

3. Comparison of Heat Duties.—

Air side = 12,410,000 Btu/hr
Water side = 8,450,000 Btu/hr
Discrepancy = 3,960,000 Btu/hr

Percent greater than water side $= \dfrac{3{,}960{,}000}{8{,}450{,}000} = 46.9\%$.

4. Calculation of Overall Coefficient From Air-Side Data.—

$$U = \frac{Q}{A\Delta T_{LM}}$$

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}}$$

$$\Delta T_1 = 140.5 - 79.5 = 61.0°F$$

$$\Delta T_2 = 148.7 - 130.2 = 18.5°F.$$

Substituting:

$$\Delta T_{LM} = \frac{61.0 - 18.5}{\ln \frac{61.0}{18.5}} = \frac{41.5}{1.195} = 34.7°F$$

$$U_L = \frac{12,410,000}{(2250)(34.7)} = 159 \text{ (based on liner area)}$$

$$U_o = \frac{159}{13.7} = 11.62 \text{ (based on outside area).}$$

5. Calculation of Corresponding Water-Side Flow Rate.—

$$W = \frac{12,410,000}{(1.0)(148.7 - 140.5)} = 1,517,000 \text{ lb/hr}$$

$$V_t = 8.76 \left(\frac{1,517,000}{1,528,000}\right) = 8.84 \text{ ft/sec.}$$

6. Calculation of Air-Side Coefficient From Air-Side Data.—

a. Calculation of water film coefficient.

$$h_i = \frac{(150)[1 + 0.011\ (145.1)](8.84)^{0.8}}{0.98}$$

$$h_i = 2280$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{2280} = 0.0067.$$

b. Outside fouling assumed zero.

c. Metal resistance (see Section I-A7c).

$$r_m = 0.00062 \text{ (based on outside area).}$$

d. Calculation of outside coefficient, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{11.62} - [0.0067 + 0.000652] = 0.0786$$

$$h_o = \frac{1}{0.0786} = 12.7 \text{ (based on outside area)}$$

$$(h_o)_L = 12.7 \times 13.7 = 174 \text{ (based on liner area).}$$

e. Calculation of outside coefficient, assuming inside fouling of 0.001 (based on inside area).

$$\frac{1}{h_o} = 0.0860 - [0.007352 + 0.0152] = 0.06345$$

$$h_o = \frac{1}{0.06345} = 15.8 \text{ (based on outside area)}$$

$$(h_o)_L = 15.8(13.7) = 216 \text{ (based on liner area).}$$

7. <u>Calculation of Overall Coefficient From Water-Side Data.</u>—

$$U = \frac{Q}{A\Delta T_{LM}}$$

ΔT_{LM} = 34.7°F
A = 2250 sq ft of liner area
Q = 8,450,000 Btu/hr (based on water-side data).

Substituting:

$$U_L = \frac{8{,}450{,}000}{(2250)(34.7)} = 108 \text{ (based on liner area)}$$

$$U_o = \frac{108}{13.7} = 7.90 \text{ (based on outside area).}$$

8. Calculation of Corresponding Air-Side Flow Rate.—

$$W = \frac{Q}{C_p \, \Delta t_{air}} = \frac{8{,}450{,}000}{(0.24)(130.2 - 79.5)} = 695{,}000 \text{ lb/hr.}$$

The corresponding face velocity is

$$V_{face} = \frac{695{,}000}{(366)(60)(0.074)} = 428 \text{ std ft/min.}$$

9. Calculation of Air-Side Coefficient From Air-Side Data.—

a. Calculation of water film coefficient.

$$h_i = 2280 \left(\frac{8{,}450{,}000}{12{,}410{,}000}\right)^{0.8}$$

$$= 1673 \text{ (based on inside area)}$$

$$\frac{A_o}{A_i h_i} = \frac{15.2}{1673} = 0.00908 \text{ (based on outside area).}$$

b. Outside fouling assumed zero.

c. Metal resistance (see Appendix A,I-7c)

$$r_m = 0.000652 \text{ (based on outside area).}$$

d. Calculation of outside coefficient, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{7.90} - [0.00908 + 0.000652]$$

$$= 0.1266 - 0.009732 = 0.11687$$

$$h_o = \frac{1}{0.11687} = 8.56 \text{ (based on outside area)}$$

$$(h_o)_L = 8.56 \times 13.7 = 117 \text{ (based on liner area).}$$

e. Calculation of outside coefficient, assuming inside fouling of 0.001 (based on inside area).

$$\frac{1}{h_o} = 0.11687 - 0.0152 = 0.1015$$

$$h_o = \frac{1}{0.1015} = 9.85 \text{ (based on outside area)}$$

$$(h_o)_L = (9.85)(13.7) = 135 \text{ (based on liner area).}$$

APPENDIX D

CALCULATION OF THE PERFORMANCE OF THE JACKET WATER COOLER, USING AN ANEMOMETER DUCT FACTOR OF 0.636

A. DATA OF THURSDAY, APRIL 28, 1955

1. East Bay.—

a. Heat duty.

The indicated face velocity (see Appendix A) of 508 std ft/min is to be multiplied by 0.636 (see Fig. 2 of this report).

$$V_{face} = 508(0.636) = 323 \text{ std ft/min}$$

$$A_{face} = 366 \text{ sq ft of face area}$$

$$\rho_{std} = 0.074 \text{ lb/cu ft}$$

$$W = V\rho A = (323)(366)(0.074)(60)$$

$$= 525{,}000 \text{ lb air/hr}$$

$$Q = W\ C_p\ \Delta t$$

$$\Delta t = 137.8 - 72.1 = 65.7°\text{F (see Appendix A)}$$

$$C_p = 0.24 \text{ Btu/lb-°F}$$

$$Q = 525{,}000(65.7)(0.24) = 8{,}290{,}000 \text{ Btu/hr.}$$

b. Log-mean temperature difference.

$$\Delta T_{LM} = 47.8°\text{F (see Appendix A)}$$

c. Calculation of overall coefficient.

The overall coefficient is given by:

$$U = \frac{Q}{A \, \Delta T_{LM}}$$

$$A_L = 2250 \text{ sq ft of liner area}$$

$$U_L = \frac{8,290,000}{(2250)(47.8)} = 77.0 \text{ (based on liner area)}$$

$$U_o = \frac{77}{13.7} = 5.61 \text{ (based on outside area).}$$

d. Calculation of corresponding water flow rate.

From Appendix A:

$$V_T = 9.33 \times 0.636 = 5.95 \text{ ft/sec.}$$

e. Calculation of the outside coefficient.

(1) Inside (water) film coefficient.

$$h_i = 2520 \left(\frac{5.95}{9.33}\right)^{0.8} = 1766$$

$$\frac{A_o}{A_i \, h_i} = \frac{15.2}{1766} = 0.00862.$$

(2) Outside fouling assumed zero.

(3) Metal resistance (see Appendix A).

$$r_m = 0.000652 \text{ (based on outside area).}$$

(4) Inside fouling (see Appendix A).

$$\frac{A_o}{A_i} r_i = 0.0175 \text{ (based on outside area).}$$

(5) Outside air film coefficient.

$$\frac{1}{h_o} = \frac{1}{U_o} - \left[\frac{A_o}{A_i \, h_i} + \frac{A_o}{A_i} r_i + r_m + r_o\right].$$

Substituting from (1), (2), (3), and (4):

$$\frac{1}{h_o} = \frac{1}{5.61} - [0.00862 + 0.0175 + 0.000652]$$

$$= 0.178 - 0.02677 = 0.1512$$

$$h_o = \frac{1}{0.1512} = 6.63 \text{ (based on outside area)}$$

$$(h_o)_L = (6.63)(13.7) = 90.8 \text{ (based on liner area).}$$

2. West Bay.—

a. Heat duty.

Using the duct factor of 0.636:

$$V_{face} = 445 \times 0.636 = 283 \text{ std ft/min}$$

$$W = 723{,}000 \times 0.636 = 460{,}000 \text{ lb/hr}$$

lb of air per hour (see Appendix A)

$$Q = 9{,}730{,}000 \times 0.636 = 6{,}180{,}000 \text{ Btu/hr.}$$

b. Log-mean temperature difference (see Appendix A).

$$\Delta T_{LM} = 52.3°F.$$

c. Calculation of the overall coefficient.

$$U_L = \frac{Q}{A\ \Delta T_{LM}} = \frac{6{,}180{,}000}{(2250)(52.3)} = 52.7 \text{ (based on liner area)}$$

$$U_o = \frac{52.7}{13.7} = 3.85 \text{ (based on outside area).}$$

d. Calculation of the corresponding water flow rate (from Appendix A).

$$V_t = (0.636)(8.0) = 5.09 \text{ ft/sec.}$$

e. Calculation of the outside coefficient.

(1) Inside water film coefficient.

$$h_i = 2230 \left(\frac{5.09}{8.0}\right)^{0.8} = 1552 \text{ (based on inside area)}$$

$$\frac{A_o}{A_i\ h_i} = \frac{15.2}{1552} = 0.0098 \text{ (based on outside area).}$$

(2) The outside fouling will be assumed zero.

(3) Metal resistance.

$$r_m = 0.000652 \text{ (based on outside area).}$$

(4) Inside fouling (see Appendix A).

$$\frac{A_o}{A_i} r_i = 0.0175 \text{ (based on outside area).}$$

(5) Outside air film coefficient.

$$\frac{1}{h_o} = \frac{1}{U_o} - \left[\frac{A_o}{A_i\ h_i} + \frac{A_o}{A_i} r_i + r_m + r_o\right].$$

Substituting:

$$\frac{1}{h_o} = \frac{1}{3.85} - [0.0098 + 0.0175 + 0.000652]$$

$$\frac{1}{h_o} = 0.260 - 0.02795 = 0.2315$$

$$h_o = \frac{1}{0.232} = 4.32 \text{ (based on outside area)}$$

$$(h_o)_L = (4.32)(13.7) = 59.1 \text{ (based on liner area).}$$

B. DATA OF MONDAY, MAY 2, 1955

1. Three-in-Line, Fourth-Row-Staggered-Pitch Bay.—

a. Heat duty.

The heat duty calculated from the air-side data is directly proportional to the air velocity. Therefore,

$$Q = (Q) \text{ indicated} \times 0.636 = 13{,}900{,}000 \times 0.636$$

$$= 8{,}840{,}000 \text{ Btu/hr.}$$

The discrepancy between the computed air-side heat duty and the computed water-side heat duty then is 9,380,000 - 8,840,000 = 540,000 Btu/hr, which is 5.76% of the water-side heat duty.

The face velocity is

$$V_{face} = (747)(0.636) = 475 \text{ std ft/min.}$$

b. Log-mean temperature difference (see Appendix A).

$$\Delta T_{LM} = 40.5°F.$$

c. Calculation of the overall coefficient from the air-side performance data.

$$U = \frac{Q}{A\ \Delta T_{LM}}.$$

Substituting from a and b, with a liner area of 2250 sq ft:

$$U_L = \frac{8{,}840{,}000}{(2250)(40.5)} = 97.0 \text{ (based on liner area)}$$

$$U_o = \frac{97.0}{13.7} = 7.08 \text{ (based on outside area).}$$

d. Calculation of the corresponding water flow rate.

$$V_t = (8.76)(0.636) = 5.58 \text{ ft/sec.}$$

e. Calculation of the outside coefficient.

(1) Inside water film coefficient (see Appendix B-6).

$$h_i = 2270 \left(\frac{5.58}{8.76}\right)^{0.8} = 1580 \text{ (based on inside area)}$$

$$\frac{A_o}{A_i\ h_i} = \frac{15.2}{1580} = 0.0096.$$

(2) The outside fouling will be assumed zero.

(3) Metal resistance.

$$r_m = 0.000652 \text{ (based on outside area).}$$

(4) Calculation of h_o, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{7.08} - [0.0096 + 0.000652]$$

$$= 0.1411 - 0.01025 = 0.13085$$

$$h_o = \frac{1}{0.13085} = 7.64 \text{ (based on outside area)}$$

$$(h_o)_L = (7.64)(13.7) = 105 \text{ (based on liner area).}$$

(5) Calculation of h_o assuming an inside fouling factor of 0.001 (based on the inside area).

$$\frac{1}{h_o} = 0.13085 - 0.0152 = 0.11565$$

$$h_o = \frac{1}{0.11565} = 8.65 \text{ (based on the outside area)}$$

$$(h_o)_L = (8.65)(13.7) = 118.5 \text{ (based on the liner area).}$$

2. Triangular-Pitch Bay.—

a. Heat duty (see Appendix C).

$$Q \text{ indicated (air side)} = 12{,}410{,}000 \text{ Btu/hr}$$

$$Q = Q \text{ indicated} \times 0.636 = 12{,}410{,}000 \times 0.636$$

$$= 7{,}900{,}000 \text{ Btu/hr.}$$

The discrepancy between the computed air-side heat duty and the computed water-side heat duty is 8,450,000 - 7,900,000, or 550,000 Btu/hr, which is 6.50% of the water-side heat duty. The corrected face velocity is

$$V_f = (626)(0.636) = 398 \text{ std ft/min.}$$

b. Log-mean temperature difference (see Appendix C).

$$\Delta T_{LM} = 34.7°F.$$

c. Calculation of the overall coefficient from the air-side performance data.

$$U = \frac{Q}{A\ \Delta T_{LM}}$$

$$U_L = \frac{7,900,000}{(2250)(34.7)} = 101.3 \text{ (based on liner area)}$$

$$U_o = \frac{101.3}{13.7} = 7.40 \text{ (based on outside area).}$$

d. Calculation of the corresponding water-side flow rate.

$$V_t = (8.84)(0.636) = 5.63 \text{ ft/sec.}$$

e. Calculation of the outside coefficient.

(1) Inside water film coefficient (see Appendix C).

$$h_i = 2280 \left(\frac{5.63}{8.84}\right)^{0.8}$$

$$h_i = 1590 \text{ (based on inside area)}$$

$$\frac{A_o}{A_i\ h_i} = \frac{15.2}{1590} = 0.00956.$$

(2) The outside fouling is assumed zero.

(3) Metal resistance.

$$r_m = 0.000652 \text{ (based on outside area).}$$

(4) Calculation of outside coefficient, assuming no fouling.

$$\frac{1}{h_o} = \frac{1}{7.4} - [0.00956 + 0.000652] = 0.135 - 0.0102$$

$$h_o = \frac{1}{0.1248} = 8.02 \text{ (based on outside area)}$$

$$(h_o)_L = (8.02)(13.7) = 109.8 \text{ (based on liner area).}$$

(5) Calculation of outside coefficient, assuming an inside fouling factor of 0.001 (based on inside area).

$$\frac{1}{h_o} = 0.1248 - 0.0152 = 0.1096$$

$$h_o = 9.13 \text{ (based on outside area)}$$

$$(h_o)_L = (9.13)(13.7) = 125 \text{ (based on liner area).}$$

APPENDIX E

EVALUATION OF AIR-SIDE COEFFICIENT BY 651C FOR THREE-IN-LINE, FOURTH-ROW-STAGGERED UNIT

A calculation will be presented for a one-in.-OD-liner tube containing 9 fins per inch, having a nominal two-in. diameter over the fin.

Reference is made to 651C, pages 8 and 9, Tables 10, 11, 12, and 13.

According to Table 12, the tube length required to transfer 10,000 Btu/hr, using this tube under a log-mean temperature difference driving force of 20°F with a maximum air velocity of 1457 ft/min, is 10.9 ft.

According to Table 11, 3000 cu ft/min of air passing through an air-duct cross section of five sq ft results in a maximum air velocity of 1457 ft/min. The corresponding face velocity is therefore

$$V_{face} = \frac{3000}{5} = 600 \text{ ft/min.}$$

The overall heat transfer coefficient is given by

$$U_o = \frac{Q}{A_o \, \Delta T} \; .$$

According to Table 10, this tube has an A_o of 3.61 sq ft/ft and an A_o/A_i of 14.8.

Substituting:

$$U_o = \frac{10{,}000}{(10.9)(3.61)(20)} = 12.7 \; .$$

Assuming an industrial steam coefficient of 1000, with no fouling, and neglecting the metal resistance, the overall coefficient is given by

$$U_o = \frac{1}{\frac{1}{h_o} + \frac{A_o}{A_i h_i}}$$

or

$$h_o = \frac{1}{\frac{1}{U_o} - \frac{A_o}{A_i h_i}} = \frac{1}{\frac{1}{12.7} - \frac{14.8}{1000}} = \frac{1}{0.0639} = 15.7 \quad .$$

The ratio of $\frac{V_{max}}{V_{face}}$ is equal to $\frac{1457}{600} = 2.428$.

The h_o, if expressed on liner area, becomes

$$(h_o)_{liner} = 15.7 \times \frac{3.61}{0.262} = 216.5 \quad .$$

APPENDIX F

CALCULATION OF TRIANGULAR AIR-SIDE COEFFICIENTS AND PRESSURE DROP FROM THE CORRELATION REPORT, USING MONDAY'S TEST DATA

From Appendix C the face velocity measured was V_{face} = 398 std ft/min.

The corresponding maximum air velocity is

$$V_{max} = 2.428 \times V_{face} = 2.428 \times 398 = 968 \text{ ft/min.}$$

The dimensions of the tube tested, as given in the Wolverine Trufin Catalog "C," are:

Catalog Number 62-0916049-01, 1-in.-OD liner 18 B.W.G. tube.

OD fins:	1.938-2.063 in.
mean fin thickness:	0.019 in.
outside area:	3.59 sq ft/ft
ratio of outside to inside area A_o/A_i:	15.28.

1. Calculation of Air-Side Coefficient, Using Fig. 13.—From Fig. 13, at a maximum air velocity of 968 ft/min,

$$\left(\frac{h_o'S}{D_r}\right)(ND_o)^{0.5} = 89,$$

where the terms included are defined in Report No. 30, page 51.

S = average tube pitch from blue print $\left(\frac{2\text{-}1/16 + 2\text{-}1/8}{2}\right)$
= 2.094 in.,

D_r = 1.08 in. (assuming a 0.04-in. root wall metal thickness).,

N = 9 fins per inch, and

D_o = 2.00 in.

Substituting:

$$\frac{h_o' \cdot 2.094}{1.08} [(9)(2)]^{0.5} = 89$$

h_o' = 10.85 based on the equivalent outside area.

The outside coefficient given is this report has to be corrected for the fin efficiency, using the relationship

$$h_o' \, Aeq = [h_o]_{outside} \, A_o,$$

where

$Aeq = A_r + E_f A_f$,

A_r = area of root, sq ft/ft,

A_f = area of fin, sq ft/ft, and

E_f = fin efficiency.

The root area of this tube is

$$A_r = \frac{\pi}{12} D_r (1 - NY),$$

where Y is the fin thickness. Substituting:

$$A_r = \frac{\pi}{12} (1.08)[1 - (9)(0.019)]$$

$$= 0.232 \cong 0.23 \text{ sq ft/ft}$$

$\therefore$

$$A_f = 3.59 - 0.23 = 3.36 \text{ sq ft/ft.}$$

Determination of fin efficiency: From the attached plot on the next page, the abscissa is

$$\text{abscissa} = H\sqrt{\frac{2}{\left(\frac{1}{h_o'} + r_o\right)K_m Y}} = H\sqrt{\frac{2h_o'}{K_m Y}},$$

since the fouling on the outside of the tube is assumed to be zero.

Substituting the values of

$$h_o' = 10.85,$$

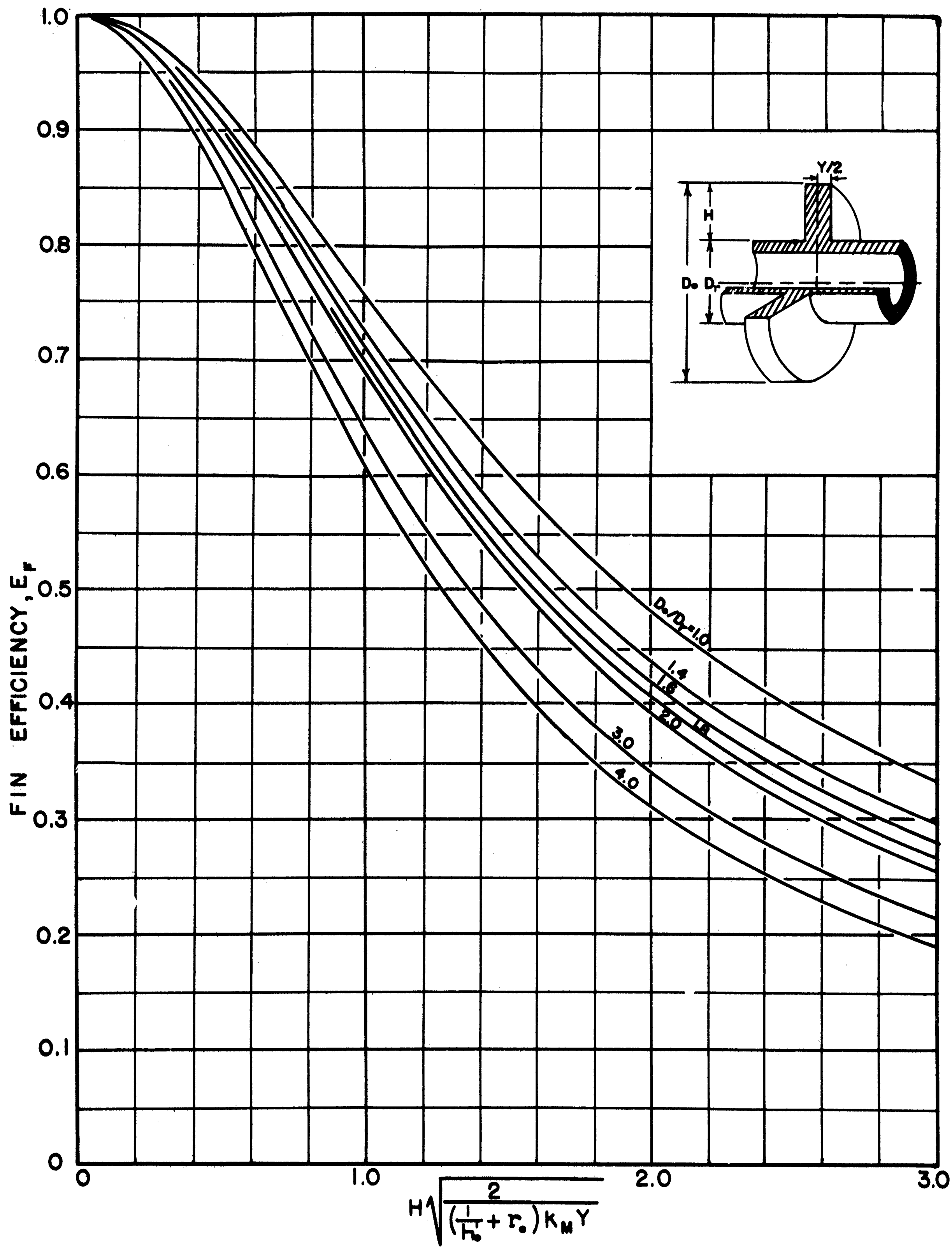

Fig. 25. Gardner's fin efficiency.

$$H = \frac{2.0 - 1.08}{(12)(2)} = 0.0384 \text{ ft},$$

$$K_m = 119 \text{ Btu/hr-°F-ft}$$

$$Y = 0.019/12 = 0.00158 \text{ ft},$$

$$\text{abscissa} = 0.0384\sqrt{\frac{(2)(10.85)}{(119)(0.00158)}}$$

$$= 0.413.$$

The parameter, D_o/D_r, $= \dfrac{2.00}{1.08} = 1.85.$

From the figure, $E_f = 0.905 = 90.5\%$.

The equivalent area is

$$Aeq = A_r + E_f A_f = (0.23) + (0.905)(3.36)$$

$$= 3.27 \text{ sq ft/ft}.$$

The outside coefficient then becomes

$$[h_o]_{outside} = h_o'\left(\frac{Aeq}{A_o}\right) = 10.85\left(\frac{3.27}{3.59}\right) = 9.89,$$

or $(h_o)_L$ based on liner area $= 135.5$.

2. Calculation of Air-Side Coefficient, Using Fig. 14.—The Reynolds number for finned tubes is defined as

$$R_e = \frac{D_e\, G_{max}}{\mu},$$

where

$$D_e = \left(\frac{S}{D_r}\right)^2 \left(\frac{ND_o}{12}\right), \text{ ft},$$

G_{max} = maximum mass rate of flow through the tube bank, lb/hr-sq ft, and

μ = average viscosity of fluid, lb/ft-hr.

Using the dimensions given in the previous section,

$$D_e = \left(\frac{S}{D_r}\right)^2 \left(\frac{ND_o}{12}\right) = \left(\frac{2.094}{1.08}\right)^2 \left(\frac{9 \times 2}{12}\right) = 5.62 \text{ ft}$$

$$G_{max} = V_{max}\,\rho = (60)(968) \times 0.074 = 4300 \text{ lb/hr-sq ft}$$

$$\mu^* = 0.046 \text{ lb/ft-hr at } 100°F.$$

Substituting,

$$Re = \frac{De\ G_{max}}{\mu} = \frac{(5.62)(4300)}{0.046} = 525{,}000.$$

From Fig. 14, at Re = 525,000,

$$\frac{h_o'\ De}{K}\left(\frac{C_p\ \mu}{K}\right)^{-0.33} = 4700$$

$$\left(\frac{C_p\ \mu}{K}\right)^{+0.33} = (0.71)^{+0.33} = 0.892$$

$$K = 0.0157 \text{ Btu/hr-°F-ft.}$$

Substituting these values,

$$h_o' = \frac{(4700)(0.0157)(0.892)}{5.62}$$

$$= 11.70, \text{ based on the equivalent area.}$$

Following the same procedure as given in the previous section, the abscissa of the fin-efficiency plot is

$$\text{abscissa} = 0.0384\sqrt{\frac{(2)(11.70)}{(119)(0.00158)}}$$

$$= 0.428.$$

From the fin-efficiency figure, $E_f = 0.90$.

Substituting,

$$Aeq = 0.23 + (0.90)(3.36) = 3.25 \text{ sq ft/ft,}$$

the outside coefficient, corrected for fin efficiency is

*The physical properties of air are obtained from: W. H. McAdams, Heat Transmission, third edition. New York: McGraw-Hill Book Co., 1954, p. 483.

$$(h_o)_{outside} = 11.7\left(\frac{3.25}{3.59}\right) = 10.6 \text{ (based on outside area)}$$

$$(h_o)_{liner} = 13.7(10.6) = 145.$$

3. Calculation of Air-Side Coefficient, Using Fig. 15.—From this figure at a maximum air velocity of 968 ft/min,

$$(h_o{}'S)\left(\frac{N}{D_r}\right)^{0.3} = 36$$

$$\left(\frac{N}{D_r}\right)^{0.3} = \left(\frac{9}{1.08}\right)^{0.3} = 1.89$$

$$S = 2.094 \text{ in. (avg)}.$$

Substituting:

$$h_o{}' = \frac{36}{(2.094)(1.89)} = 9.11 \text{ (based on equivalent area)}.$$

The abscissa of the fin-efficiency plot is

$$\text{abscissa} = 0.0384\sqrt{\frac{(2)(9.11)}{(119)(0.00158)}} = 0.378.$$

From the fin-efficiency figure,

$$E_f = 0.93,$$

$$Aeq = 0.23 + (0.93)(3.36)$$

$$= 3.35 \text{ sq ft/ft}.$$

The outside coefficient corrected for fin efficiency is

$$h_o = 9.11\left(\frac{3.35}{3.59}\right) = 8.5 \text{ (based on outside area)}$$

$$(h_o)_{liner} = (13.7)(8.5) = 116.4.$$

4. Pressure Drop, Using Fig. 16.—The parameter given on this figure for this tube is

$$K = \frac{N\,D_o}{D_r^{\,0.2}} = \frac{9 \times 2.0}{(1.08)^{0.2}} = 17.75.$$

For a maximum air velocity of 968 ft/min, the pressure drop times the tube equivalent diameter is

$$-\Delta P \cdot De = 0.50 \text{ ft (inches of water per row)}$$

$$De = 5.62 \text{ ft}$$

$$-\Delta P = 0.089 \text{ in. of water per row.}$$

The pressure drop for a bank four rows drop is $(4)(.089) = 0.356$ in. of water.

APPENDIX G

SCHMIDT'S CORRELATION

Reference is made to Fig. 19 in which the Schmidt group is plotted as a function of maximum air velocity. The maximum air velocity of 968 ft/min gives a value of 15 for the Schmidt group.

$$h_o' \left[\frac{D_r^{0.4}}{\left[1 - 0.18\left[\left(\frac{D_o - D_r}{2}\right)N\right]^{0.63}\right]} \right] = 15.$$

The dimensions of the tube (see Appendix F) are

$$D_o = 2.00 \text{ in.,}$$

$$D_r = 1.08 \text{ in., and}$$

$$N = 9 \text{ fins/per inch.}$$

Therefore,

$$D_r^{0.4} = (1.08)^{0.4} = 1.032$$

$$\frac{D_o - D_r}{2} = \frac{2.00 - 1.08}{2} = 0.46 \text{ in.}$$

$$\left[N \cdot \left(\frac{D_o - D_r}{2}\right)\right]^{0.63} = [9 \times 0.46]^{0.63} = 2.445.$$

Therefore,

$$\frac{D_r^{0.4}}{\left[1 - 0.18\left[\left(\frac{D_o - D_r}{2}\right)N\right]^{0.63}\right]} = \frac{1.032}{1 - 0.18(2.445)} = \frac{1.032}{0.56}.$$

Therefore,

$$h_o' = \frac{(15)(0.56)}{1.032} = 8.14$$

The abscissa of the fin-efficiency plot is

$$\text{abscissa} = H\sqrt{\frac{2h_o'}{K_m Y}} = 0.0384\sqrt{\frac{(2)(8.14)}{(119)(0.00158)}} = 0.357$$

$$\frac{D_o}{D_r} = \frac{2.0}{1.08} = 1.85.$$

From the figure, $E_f = 0.945$.

Using the area values given in Appendix F,

$$A_{eq} = A_r + E_f A_f = 0.23 + 0.945(3.36) = 3.41 \text{ sq ft/ft.}$$

The outside coefficient corrected for fin efficiency is

$$h_o = h_o' \left(\frac{A_{eq}}{A_o}\right) = 8.14\left(\frac{3.41}{3.59}\right) = 7.73 \text{ (based on outside area)}$$

$$(h_o)_L = (7.73)(13.7) = 106 \text{ (based on liner area).}$$

APPENDIX H

SAMPLE CALCULATIONS FOR FIG. 20, "h_o vs HP/ft^2"

A. STAGGERED PITCH

Catalog tube No. 62-0716065-01
Distance from center to center: $S = 2\text{-}1/16" = 2.0625$ in.
No. fins per inch: $N = 7$ $A_o = 2.83$ sq ft/ft $D_o = 2.0$ in.
$D_r = 1.08$ in. $V_{max}/V_{face} = 2.34$
Face area: 366 sq ft
Assume $V_{face} = 600$ std ft/min $\therefore V_{max} = 1404$ std ft/min

1. Calculation of h_o.—From Fig. 13: (entering with V_{max} of 1404)

$$\left(\frac{h_o{}'S}{D_r}\right)(ND_o)^{.5} = 110$$

$$\frac{S}{D_r}(ND_o)^{.5} = \frac{2.0625}{1.08}(7 \times 2)^{.5} = 7.15$$

$$\therefore h_o{}' = \frac{110}{7.15}$$

$$h_o{}' = 15.4 \frac{\text{Btu}}{\text{hr-°F-sq ft (outside equivalent area)}} .$$

This value must be referred to actual outside area.

$$h_o{}' A_oeq = h_oA_o, \quad \text{or} \quad h_o = h_o{}' \frac{A_oeq}{A_o}$$

$$A_oeq = A_r + \phi A_f$$

$$A_r = \frac{3.14 \times 1.08}{12} = .283 \text{ sq ft/ft}$$

$$A_f = 2.83 - 0.283 = 2.547 \text{ sq ft/ft.}$$

Fin efficiency is determined, using Gardner's plot:

$$\text{abscissa:} \quad H\sqrt{\frac{2}{\left(\frac{1}{h_o'} + r_o\right) K_m Y}}$$

$$H = \frac{0.92}{24} = 0.0383 \text{ ft}$$

$$K_m = 120 \qquad Y = \frac{.019}{12} \qquad r_o \cong 0$$

$$\text{abscissa} = 0.0383\sqrt{\frac{15.4 \times 2 \times 12}{120 \times 0.019}} = .487.$$

To this abscissa corresponds a fin efficiency of

$$\phi = .895$$

$$A_{o_{eq}} = 0.283 + (0.895)(2.547) = 2.563$$

$$h_o = h_o' \frac{A_{oeq}}{A_o} = 15.4 \frac{2.563}{2.830} = 13.95 \frac{\text{Btu}}{\text{hr-°F-sq ft (outside area).}}$$

2. Calculation of HP/sq ft Face Area.—

a. Pressure drop

From Fig. 16:

$$K = \frac{ND_o}{D_r^{0.2}} = \frac{7 \times 2}{(1.08)^{0.2}} = 13.8$$

$$D_e = \left(\frac{S}{D_r}\right)^2 \left(\frac{ND_o}{12}\right) = \left(\frac{2.0625}{1.080}\right)^2 \left(\frac{7 \times 2}{12}\right) = 4.28 \text{ ft.}$$

For a value of V_{max} = 1404 std ft/min, Fig. 16 gives

$$\Delta P\ D_e = 0.7 \text{ (in. water per row)ft}$$

$$\Delta P_t = \frac{0.7 \times 4 \times 62.4}{12 \times 4.28} = 3.4 \text{ lb/sq ft}$$

b. Theoretical power input per sq ft face.

$$HP = \left(\frac{\text{ft lb}}{\text{min}}\right)(3.03 \times 10^{-5})$$

$$HP/ft^2 = (V_f) \times (\Delta P_t)(3.03 \times 10^{-5})$$

$$= \frac{600 \times 3.4 \times 3.03}{10^5} = .0618 \frac{HP}{ft^2 \text{(face area)}}.$$

Figure 20 is obtained by plotting for each assumed V_{face}, h_o vs HP/ft^2 (face area).

B. THREE-IN-LINE, FOURTH-ROW-STAGGERED ARRANGEMENT

This curve has been constructed by using the ALCO data given in Fig. 11.

1. <u>For an Assumed Value of V_{face}, Read the Corresponding Value of h_o From ALCO Line.</u>—

$$V_{face} = 600 \text{ std ft/min} \qquad h_o = 9.2$$

2. <u>From the Same Plot, ΔP = .37 in. of Water for Four Rows.</u>—

$$\Delta P = \frac{.370 \times 62.4}{12} = 1.92 \text{ lb/sq ft}$$

Theoretical

$$\frac{HP}{ft^2} = \frac{1.92 \times 600 \times 3.03}{10^5} = 0.0349 \frac{HP}{ft^2 \text{ (face area)}} .$$

The corresponding graph can be obtained by plotting h_o vs HP/ft^2 for each assumed value of V_{face}.

APPENDIX I

CALCULATIONS FOR LEAN OIL COOLER (ITEM 2-5)

A. DESIGN CONDITIONS FROM SPECIFICATIONS SHEET (See Section XI of this report).

1. Air Side.—

Inlet air temperature: 100°F
Air flow rate: 46,500 st cu ft/min
Temperature rise of air: 30.5°F.

2. Tube Side.—

Inlet oil temperature: 173°F
Outlet oil temperature: 140°F
Average molecular weight: 200 lb/mole
Oil flow rate: 90,659 lb/hr
Mean temperature difference: 41.2°F
Number of passes: 10
Heat exchanged: 1,538,000 Btu/hr.

3. Heat Transfer Surface.—

Bare tube: 376 sq ft
External area: A_o = 5180 sq ft
Tube specifications: 1 in. OD x 18 B.W.G. Inhibited Admiralty
Tube inside diameter: 0.902 in.
Fin height: 1/2 in. nominal, 9 fins per inch
Fin thickness: 0.019 in., A_o = 3.59 sq ft/ft, A_o/A_i = 15.28.

B. PHYSICAL PROPERTIES OF LEAN OIL

Specific heat: 0.520 Btu/lb-°F (Maxwell's Data Book on Hydrocarbons, p. 93).
Specific gravity at 60°F: 0.825 (Given in specification sheet)
Viscosity at 156.5°F: 1.48 cps (Maxwell's Data Book on Hydrocarbons, p. 161)

Thermal conductivity at 156.5°F: 0.0780 (Btu/ft)/sq ft/hr-°F
(Kern's Process Heat Transfer)
API° = 39°

C. SPECIFIED HEAT LOAD

$$Q_{oil} = W\ C_p\ \Delta t_{oil}$$

$$C_p = 0.52 \text{ Btu/lb-°F}$$

$$\Delta t_{oil} = 173 - 140 = 33\text{°F.}$$

Substituting:

$$Q_{design} = (90{,}659)(0.52)(33) = 1{,}552{,}000 \text{ Btu/hr.}$$

This value compares with the heat duty given in the specification sheet as 1,538,000 Btu/hr.

D. CALCULATION OF EXIT TEMPERATURE OF THE EXISTING UNIT, USING THE ALCO DATA CONVERTED TO 9 FINS PER INCH

1. Air-Face Velocity.—From the blue print supplied by The Happy Co., the width of unit is 93-5/8 in. The tube length is given in the specifications as 8 ft. Allowing one-in. stripped ends for rolling in, the face area is

$$A_{face} = \left(\frac{93.625}{12}\right)(7.832) = 61.1 \text{ sq ft.}$$

The corresponding specified air velocity (face) is then

$$V_{face} = \frac{46{,}500}{61.1} = 760 \text{ std ft/min.}$$

2. Outside Film Coefficient.—From Fig. 11, using the dotted line corresponding to 9 fins per inch at a V_{face} of 760 std ft/min,

$$h_o = 9.7 \text{ (based on outside area)}$$

$$(h_o)_L = (9.7)(13.7) = 133 \text{ (based on liner area).}$$

3. Metal Resistance.—The metal resistance is obtained from Section I of Appendix A as

$$r_m = 0.00062 \text{ (based on outside area).}$$

4. Inside Film Coefficient.—The inside coefficient will be obtained, using the following relationship given by McAdams*:

$$\frac{h}{C_p\ G}\left(\frac{C_p\ \mu}{K}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14} = \frac{0.023}{\left(\frac{DG}{\mu}\right)^{0.2}} ,$$

where the symbols are defined by the author.

The area of flow for the oil is

$$A_{flow} = (18 \text{ tubes/pass})\ \frac{(0.902)^2\pi}{(4)(144)} = 0.0798 \text{ sq ft}$$

$$G = \frac{W}{A_{flow}} = \frac{90,659}{0.0798} = 1,136,000 \text{ lb/hr-sq ft}$$

$$\mu = 1.48 \text{ centipoise} \times 2.42 = 3.58 \text{ lb/ft-hr}$$

$$\left(\frac{DG}{\mu}\right)^{0.2} = \left[\frac{(0.902)(1,136,000)}{(12)(3.58)}\right]^{0.2} = [23,800]^{0.2} = 7.51$$

$$\left(\frac{C_p\ \mu}{K}\right)^{2/3} = \left[\frac{(0.52)(3.58)}{0.078}\right]^{2/3} = (23.9)^{2/3} = 8.3.$$

Substituting:

$$h_i = \frac{(0.023)(0.52)(1,136,000)}{(7.51)(8.3)}\left(\frac{\mu}{\mu_w}\right)^{0.14} = 218\left(\frac{\mu}{\mu_w}\right)^{0.14}$$

Assume that $(\mu/\mu_w)^{0.14}$ is 0.99 (to be checked later):

$$h_i = (218)(0.99) = 216 \text{ (based on inside area).}$$

5. The Inside Fouling is Obtained From TEMA as 0.002.—**

*Op. cit., eq. 9-10c, p. 219.

**Standards of Tubular Exchanger Manufacturer's Association, third edition, 1952, p. 87, Table T-2.

6. The Overall Heat Transfer Coefficient is Given by.—

$$\frac{1}{U_o} = \frac{1}{h_o} + r_m + \frac{A_o}{A_i} r_i + \frac{A_o}{A_i\, h_i} .$$

Substituting:

$$\frac{1}{U_o} = \frac{1}{9.7} + 0.00062 + (15.28)(0.002) + \frac{15.28}{216}$$

$$= 0.103 + 0.00062 + 0.03056 + 0.0708 = 0.20498$$

$$U_o = \frac{1}{0.205} = 4.88 \text{ (based on outside area)}$$

$$(U_o)_L = (4.88)(13.7) = 66.9 \text{ (based on liner area).}$$

7. Calculation of Heat Duty.—Trial and error for the outlet temperature:

$$\text{assume } t_{oil} \text{ out} = 147.5\ °F$$

$$Q = (90{,}659)(0.52)(173 - 147.5) = 1{,}210{,}000 \text{ Btu/hr.}$$

The temperature rise of the air is

$$\Delta t_{air} = \frac{Q}{W\, C_p} = \frac{1{,}200{,}000}{(46{,}500)(60)(0.074)(0.24)} = 24.2°F.$$

The outlet air temperature is then

$$t_{air} \text{ out} = 100 + 24.2 = 124.2°F.$$

The mean temperature difference driving force therefore can be obtained as

$$\Delta T_1 = (173 - 124.2) = 48.8°F$$

$$\Delta T_2 = (147.5 - 100) = 47.5°F$$

$$\Delta T_m = \Delta T_{average} = 48.1°F.$$

Check on heat duty:

$$Q = U_o\, A_o\, \Delta T_m = (4.88)(5180)(48.1) = 1{,}215{,}000 \text{ Btu/hr,}$$

which is reasonably close to the value of 1,200,000 Btu/hr obtained by assuming an outlet oil temperature of 147.5°F.

8. Check of Inside Coefficient.—The temperature drop across the oil film is obtained from

$$\frac{\Delta t_{oil}}{\Delta T_m} = \frac{U_o}{\left(\frac{A_i\ h_i}{A_o}\right)} .$$

Substituting:

$$\Delta t_{oil} = \frac{(4.88)(15.28)}{(216)}(48.1) = 16.6°F$$

$$t(\text{oil at wall}) = \frac{173 + 147.5}{2} - 16.6 = 143.6°F$$

$$\mu_w = 1.58 \text{ centipoise} \quad (\text{at } 143.6°F)^*$$

$$\therefore \quad \left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{1.48}{1.58}\right)^{0.14} = 0.991,$$

as compared to the assumed value of 0.99.

9. Comparisons with Design Values.—

a. Comparison of computed U_o with that given in the Happy Co. specifications.

The computed U_o = 4.88 (based on outside area)

= 66.9 (based on liner area).

The overall coefficient specified is

$$(U_o)_{specified} = 98.4 \text{ (based on liner area)}$$

$$\frac{(U_o)_{computed}}{(U_o)_{specified}} = \frac{66.9}{98.4} = 0.71.$$

*Maxwell, J. B. Data Book On Hydrocarbons. New York: D. Van Nostrand Co., Inc., 1950, p.161.

b. Comparison of heat duties.

$$\frac{Q}{Q_{design}} = \frac{1,210,000}{1,538,000} = 0.788.$$

E. CALCULATION OF EXIT OIL TEMPERATURE, USING 651C

1. Air Velocity (see Section D-1).—

$$V_{face} = 760 \text{ std ft/min}$$

$$V_{max} = (760)(2.428) = 1845 \text{ std ft/min.}$$

2. Outside Film Coefficient.—From Fig. 8,

$$h_o = 18.4 \text{ (based on outside area)}$$

$$(h_o)_L = (18.4)(13.7) = 252 \text{ (based on liner area)}$$

at a V_{max} of 1845.

3. Metal Resistance.—

$$r_m = 0.00062 \text{ (based on outside area).}$$

4. Inside Film Coefficient.—

$$h_i = 218\ (\mu/\mu_w)$$

$$\text{assume } (\mu/\mu_w) = 0.99 \text{ (to be checked later)}$$

$$h_i = 216 \text{ (based on inside area).}$$

5. The Inside Fouling is Assumed to be 0.002, Based on the Inside Area (See Section D-5).—

6. Overall Heat Transfer Coefficient.—

$$\frac{1}{U_o} = \frac{1}{18.4} + 0.00062 + (15.28)(0.002) + \frac{15.28}{216}$$

$$= 0.1564$$

$$U_o = \frac{1}{0.1564} = 6.4 \text{ (based on outside area)}$$

$$(U_o)_L = (6.4)(13.7) = 87.8 \text{ (based on liner area).}$$

7. Calculation of Heat Duty.—Trial and error for the outlet temperature:

$$\text{assume } t_{oil} \text{ (outlet)} = 142.5°F$$

$$Q = W\,C_p\,\Delta t = (90{,}659)(0.52)(173 - 142.5)$$

$$= 1{,}435{,}000 \text{ Btu/hr.}$$

The corresponding temperature rise of the air is

$$\Delta t_{air} = \frac{1{,}435{,}000}{(46{,}500)(60)(0.074)(0.24)} = 29.0°F$$

$$t_{air} \text{ outlet} = 100 + 29.0 = 129.0°F.$$

The mean temperature difference is

$$\Delta T_m = \frac{(173 - 129.0) + (142.5 - 100)}{2} = 43.2°F.$$

Check of heat duty:

$$Q = U_o\,A_o\,\Delta T_m = (6.4)(5180)(43.2) = 1{,}430{,}000 \text{ Btu/hr,}$$

which checks closely the assumed value of 1,435,000 Btu/hr.

8. Check of Inside Coefficient (See Section D-8).—

$$\Delta t_{oil} = \frac{(6.4)(15.28)}{216}(43.2) = 19.5°F$$

$$t_{oil} \text{ at wall} = \frac{173 + 142.5}{2} - 19.5 = 138.2°F$$

$$\mu_w = 1.60 \text{ centipoise}$$

$$\left(\frac{\mu}{\mu_w}\right)^{0.14} = \left(\frac{1.48}{1.60}\right)^{0.14} = 0.989,$$

as compared to the assumed value of 0.99.

9. Comparison with Design Values.—

a. Comparison of computed U_o with that given in the Happy Co. specifications.

$$\text{computed } U_o = 87.8 \text{ (based on liner area)}$$
$$\text{specified } U_o = 98.4 \text{ (based on liner area)}$$

$$\frac{(U_o)_{computed}}{(U_o)_{specified}} = \frac{87.8}{98.4} = 0.89.$$

b. Comparison of heat duties.

$$\frac{Q}{Q_{design}} = \frac{1,430,000}{1,538,000} = 0.93.$$

F. CALCULATION OF EXIT OIL TEMPERATURE, USING THE HAPPY CO. RATING CURVE

1. Air Velocity (See Section D-1).—

$$V_{face} = 760 \text{ std ft/min.}$$

2. Outside Film Coefficients.—From Fig. 8,

$$(h_o)_L = 220 \text{ (based on liner area)}$$

$$h_o = \frac{220}{13.7} = 16.05 \text{ (based on outside area).}$$

3. Metal Resistance.—

$$r_m = 0.00062 \text{ (based on outside area).}$$

4. Inside Film Coefficient.—

$$h_i = 216 \text{ (based on inside area).}$$

5. The Inside Fouling is 0.002, Based on the Inside Area.—

6. Overall Heat Transfer Coefficient.—

$$\frac{1}{U_o} = \frac{1}{16.05} + 0.00062 + 0.03056 + \frac{15.28}{216}$$

$$= 0.1643 \text{ (based on outsided area)}$$

$$U_o = \frac{1}{0.1643} = 6.08 \text{ (based on outside area)}$$

$$(U_o)_L = (13.7)(6.08) = 83.5 \text{ (based on liner area).}$$

7. Calculation of Heat Duty.—Trial and error for the outlet temperature:

$$\text{assume } t_{oil\ out} = 143.5°F$$

$$Q = (90{,}659)(0.52)(173 - 143.5) = 1{,}390{,}000 \text{ Btu/hr.}$$

The corresponding temperature rise of the air is

$$\Delta t_{air} = \frac{1{,}390{,}000}{(46{,}500)(60)(0.074)(0.24)} = 28.1°F$$

$$t_{air} \text{ outlet} = 100 + 28.1°F = 128.1°F.$$

The mean temperature difference is

$$\Delta T_m = \frac{(173 - 128.1) + (143.5 - 100)}{2} = 44.2°F.$$

Check of heat duty:

$$Q = (6.08)(5180)(44.2) = 1{,}395{,}000 \text{ Btu/hr,}$$

which checks closely with the assumed value of 1,390,000 Btu/hr.

8. Comparison with Design Values.—

a. Overall coefficients.

$$\frac{(U_o)_{computed}}{(U_o)_{specified}} = \frac{83.5}{98.4} = 0.848 \text{ or } 85\%.$$

b. Heat duty.

$$\frac{Q}{Q_{design}} = \frac{1{,}390{,}000}{1{,}538{,}000} = 0.905 = 90.5\%.$$

APPENDIX J

CALCULATION OF DEW POINT AND BUBBLE POINT OF FEED TO THE DEBUTANIZER CONDENSER

Inlet Vapor Composition (From Mr. W. F. Roberts of The Happy Co.).

Component	Mole %
C_3 (propane)	1.40
i-C_4 (isobutane)	21.80
n-C_4 (butane)	75.20
i-C_5 (isopentane)	1.60

DETERMINATION OF PHYSICAL PROPERTIES OF THE MIXTURE

1. Bubble Point (NGSMA, pp. 168-171).—

Assume t_b = 172°F:

Component	x	K	y=Kx
C_3	1.40	2.16	3.02
i-C_4	21.80	1.15	25.05
n-C_4	75.2	.94	72.00
i-C_5	1.60	.42	.75
		Σy =	100.82

Therefore, the assumed t_b = 172°F is correct.

2. Dew Point.—

Assume t_D = 174°F:

Component	y	K	x=y/K
C_3	1.40	2.2	0.637
i-C_4	21.80	1.18	18.500
n-C_4	75.2	.96	78.300
i-C_5	1.60	.48	3.330
		Σx =	100.767

Therefore, the assumed t_D = 174°F is correct, and the feed vapor is saturated.

APPENDIX K

CALCULATION OF TUBE WALL TEMPERATURE FOR PARTIALLY FILLED EVAPORATOR CONDENSER TUBE

The relationship used to calculate the tube wall temperature is

$$t_{wall} = t_{sv} - \Delta t_i,$$

where

t_{wall} = the wall temperature, °F,

t_{sv} = temperature of saturated vapors, °F,

= 240°F for this design, and

Δt_i = temperature drop across condensate and fouling film.

The temperature drop, Δt_i, is related to the overall temperature difference by

$$\Delta t_i = \left(\frac{1}{h_i} + r_i\right)_{liner} (U_o)_L \, \Delta T_{oa}$$

where

h_i = inside coefficient, based on liner area,

r_i = inside fouling film resistance, based on liner area,

$(U_o)_L$ = overall coefficient, based on liner area, and

ΔT_{oa} = overall temperature difference = $T_{sv} - T_{air}$.

Using:

t_{air} = 100°F (inlet air from specifications)*

*The actual air temperature on the outside of the tube will be somewhat higher than the assumed value. The effect of increasing this temperature would be to increase the computed wall temperature of the tube.

$(h_o)_L$ = 210 (based on liner area) from the Happy Co. rating curve

r_i = 0.0011 (based on liner area) from TEMA.

The relationships given can be reduced to

$$t_{wall} = 240 - 140 \left(\frac{0.0011\ h_i + 1}{0.00586\ h_i + 1}\right).$$

The following table can then be computed.

TABLE I (APPENDIX K)

COMPUTED WALL TEMPERATURES IN THE SUBCOOLING ZONE
OF THE EVAPORATOR CONDENSER FOR VARIOUS INSIDE COEFFICIENTS

h_i	t_{wall}, °F
1000	194
1200	200
1500	202
2000	205

APPENDIX L

CALCULATION OF FAN-BLADE EFFICIENCY

Reference is made to Fig. 21 of page 58.

For a 12° blade angle, with a pressure drop of 0.4 in. of water, the anticipated flow rate is obtained as 275,000 CFM (ρ = 0.065 lb/cu ft). The corresponding brake horsepower delivered to the blades is obtained as 30 BHP.

Computation of theoretical horsepower required:

$$HP = \left(275{,}000\ \frac{\text{cu ft}}{\text{min}} \times 60\ \frac{\text{min}}{\text{hr}}\right)\left(\frac{0.4\ \text{in. } H_2O}{12\ \text{in./ft}} \times 62.4\ \frac{\text{lb}}{\text{cu ft } H_2O}\right)\left(5.05 \times 10^{-7}\ \frac{\text{HP-hr}}{\text{ft-lb}}\right)^*$$

$$= 17.35\ \text{HP}$$

$$\text{Blade efficiency} = \frac{\text{Theoretical HP required}}{\text{Actual HP required}} = \frac{17.35}{30} = 58\%$$

*Maxwell, J. B., op. cit., p. 250.