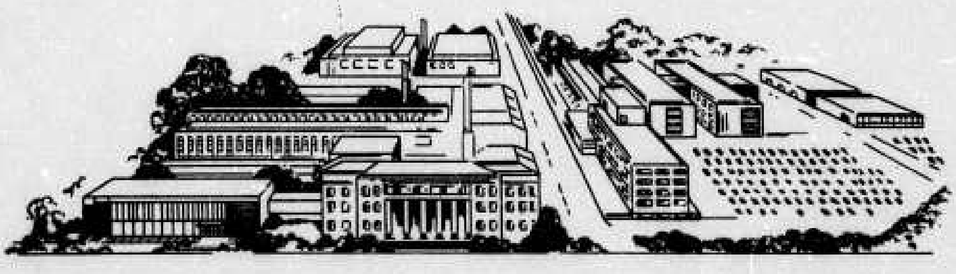
RESEARCH REPORT



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FINAL TECHNICAL REPORT

on

COMPACT HEAT-EXCHANGER STUDY

Contract No. DA-44-009-AMC-313(X)

to

U. S. ARMY ENGINEER REACTORS GROUP
ARMY NUCLEAR POWER PROGRAM
FORT BELVOIR, VIRGINIA

April 15, 1964

by

F. A. Creswick, S. G. Talbert, and J. W. Bloemer

THE VIEWS CONTAINED HEREIN REPRESENT ONLY THE VIEWS OF THE PREPARING AGENCY AND HAVE NOT BEEN APPROVED BY THE DEPARTMENT OF THE ARMY.

BATTELLE MEMORIAL INSTITUTE
505 King Avenue
Columbus, Ohio 43201

PREFACE

This report is a summary of the study conducted by Battelle Memorial Institute during the period from July 1, 1963, to March 31, 1964, for the U. S. Army Engineers Reactors Group at Fort Belvoir, Virginia, under USAERDL Contract No. DA-44-009-AMC-313(X). The project was administered by the Advanced Power Conversion Development Branch (APCDB), with Mr. E. D. Collins as project coordinator. The project requirements were outlined in PD10800-S20, dated April, 1963. The project staff at Battelle consisted of J. A. Eibling, Group Director; F. A. Creswick, Program Director; S. G. Talbert, Project Engineer; and J. W. Bloemer, Research Engineer.

Battelle Memorial Institute

5 0 5 KING AVENUE COLUMBUS, OHIO 43201

AREA CODE 614, TELEPHONE 299-3191

April 15, 1964

Property Officer
Warehouse 335
USAERDL
Fort Belvoir, Virginia

M/F: Contract No. DA-44-009-AMC-313(X)

Dear Sir:

Compact Heat-Exchanger Study

We are pleased to submit this final technical report on our study of compact heat exchangers. The objective of the program has been to establish design goals for a compact heat-exchanger development program.

The study has been more technical in nature than that implied in the purchase request. This was necessary because we found that a literature survey alone would not lead to the required information.

As a result of the additional technical effort, not only has the basic objective been achieved but also data and procedures are now available for estimating the size of heat exchangers without conducting a complete design analysis. These results should be extremely useful in conducting optimization studies in which heat-exchanger size and performance are trade-off parameters.

If there are any items in this report that need clarification, we will be pleased to discuss them.

Very truly yours,

James A. Eibling

James A. Eibling
Group Director
Thermal Systems Research

JAE/mln
Enc. (25)

cc: APCDB
Building 322
USAERDL
Fort Belvoir, Virginia
Attention Mr. E. D. Collins

AD

Battelle Memorial Institute, Columbus, Ohio, COMPACT HEAT-EXCHANGER STUDY, Final Technical Report, by F. A. Creswick, S. G. Talbert, and J. W. Bloemer. March 31, 1964. [64] p. incl. illus. and tables. [Contract No. DA-44-009-AMC-313(X)]. Unclassified report.

Heat-exchanger-performance data found in the literature are reduced to a form that allows compactness of core surfaces on the basis of geometrical configuration alone, by correcting for the influence of fluid properties and performance requirement parameters. Hydraulic radius of the core surface is shown to be the most important parameter affecting compactness.

(over)

A compactness parameter is derived with which the required weight and volume of a heat exchanger for a given application may be estimated. Data and examples are given.

The results of discussions with several heat-exchanger manufacturers to obtain state-of-the-art information on limitations in producing more compact surfaces are presented.

Design studies are conducted to show the influence of hydraulic radius on the size and shape of a recuperator and precooler for a mobile, closed-cycle gas-turbine power plant and to illustrate the usefulness and accuracy of the compactness parameter.

Recommendations for goals in a compact-heat-exchanger development program are presented.

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COMPACT HEAT-EXCHANGER STUDY

by

F. A. Creswick, S. G. Talbert, and J. W. Bloemer

SUMMARY

This study of compact heat exchangers included all types of fluids and surface configurations in order to determine realistic compactness goals for the recuperator and precooler heat exchangers to be used in mobile, nuclear-powered, closed-cycle, gas-turbine, electric-generating plants. The study was based mainly on a literature survey, supplemented by information obtained from heat-exchanger manufacturers and from a design study.

The data found in the literature were reduced to a form that permitted comparison of the effectiveness of core surfaces on the sole basis of geometrical configuration. In this way, the effects of fluid properties and operating conditions were eliminated. These reduced data can be used to predict easily the heat-exchanger size and performance for many combinations of core surface, heat-transfer fluid, and operating conditions. The hydraulic radius of the core surfaces was shown to be the most important factor affecting the compactness of a heat-exchanger core.

Discussions with eight heat-exchanger manufacturers produced information on present production capabilities, the state of the art of compact-heat-exchanger technology, and the difficulties likely to be encountered in attempting to produce more compact core surfaces than are now available.

Preliminary design studies were conducted for several recuperators and one precooler in order to compare the sizes and weights resulting from the use of different core surfaces that represent a wide range of compactness and manufacturing difficulty.

The various phases of this study led to recommendations concerning design goals for a compact-heat-exchanger development program.

INTRODUCTION

The purposes of this study were (1) to evaluate the present state of the art of compact-heat-exchanger design and performance and (2) to recommend realistic goals for the first step of a compact-heat-exchanger development program. In conducting the study, evaluations were made of heat exchangers now in use, or proposed for use, in the Army Gas-Cooled Reactor Systems Program, as well as of those in the literature representing the entire heat-exchanger field, including all gases and liquids, effects of change of phase and material conductivity, and surface configurations that result in more compact core designs.

Certain types of heat exchangers and some related side effects were excluded from consideration in this study, for example:

(1) Periodic-flow regenerators

(2) Natural-convection heat exchangers

(3) Transient performance

(4) Fouling and corrosion

(5) Dissociation or chemical change

(6) Rarefied gases

(7) Classified reports.

The degree of each heat exchanger's compactness was judged, whenever possible, by evaluating its specific heat transfer and heat-transfer density. These terms were developed by the Advanced Power Conversion Development Branch (APCDB) and are defined as follows:

(1) Specific heat transfer (SHT) is the heat transferred in Btu per hour per 100 degrees Fahrenheit logarithmic-mean-temperature difference per pound of heat-exchanger weight

(2) Heat-transfer density (HTD) is the heat transferred in Btu per hour per 100 degrees Fahrenheit logarithmic-mean-temperature difference per cubic foot of heat-exchanger volume.

Heat-exchanger weight and volume, as defined by APCDB, include the core, manifolds, pressure vessel, insulation, and any auxiliary equipment necessary for successful operation.

INVESTIGATION

The state of the art of compact-heat-exchanger technology was determined both from data gathered during the literature survey and from discussions with heat-exchanger manufacturers. From this start-of-the-art investigation, design and performance goals have been given for an APCDB recuperator and precooler.

Virtually no heat-exchanger data were found which would enable calculation of specific heat transfer (SHT) or heat-transfer density (HTD) as defined and requested by APCDB, because the majority of the technical literature presented investigations only on the performance of heat-exchanger core surfaces [References (112) through (129)*]. Also, it was suspected, and later verified, that over-all heat-exchanger performance data would depend so markedly on the particular fluids and operating conditions used that

* See Bibliography, page 44.

no meaningful comparisons could be made. Accordingly, it was decided to evaluate compactness on the basis of size and weight of the core surfaces, rather than on size and weight of over-all heat-exchanger designs. For this purpose, a correction procedure for analyzing performance data was required, and a compactness parameter (P_c) was derived to compare core surfaces on the basis of their ability to transfer heat because of their geometrical configuration (including such factors as fin design, friction factor, Colburn factor, etc.) rather than any particular fluid properties or operating conditions. These compactness-parameter data can be used also in conjunction with a particular set of design and performance requirements to calculate easily the expected size and performance for a heat exchanger. This application of the data is illustrated later by example problems. The terms heat-transfer density and specific heat transfer, were usually evaluated on the basis of core volumes and weights, respectively, to conform to the data gathered in the literature survey and to judge the relative merits of heat-transfer surface configurations and core designs found in the technical reports.

The approach used to arrive at values to satisfy the original APCDB definitions of HTD and SHT (i.e., including manifolds, insulation, etc.) was, first, to employ the correction procedure described above to calculate a practical minimum core volume and weight, and then to estimate the size and weight of typical manifolds, pressure vessel, insulation, etc.

The complete derivation of the compactness parameter (P_c) is presented in Appendix A, and whenever possible, the data were corrected and plotted using this procedure. The final expression, Equation (A-38) is

$$\mathrm{HTDC} = P_c\ [\text{Design Parameter}] \quad , \tag{A-38}$$

where

$$\mathrm{HTDC} = (\mathrm{HTD}) \left[(C/C_{min})(N_{tu_o}/\Delta p)^{1/3}\ (N_{Pr})^{8/9}\ (\mu\rho)^{-1/3}\ (c_p)^{-1} \right] \tag{A-39}$$

$$= (\mathrm{HTD})\ [\text{Correction Factor}] \tag{A-40}$$

$$P_c = \left[(g/2)^{1/3}\ c_j\ \eta_o^{4/3}\ r_h^{-4/3}\ (f/j)^{-1/3} \right] \tag{A-42}$$

$$\text{Design Parameter} = \left[(N_{tu_o}/N_{tu})^{4/3} \right] \quad . \tag{A-43}$$

The nomenclature is given on a fold-out page at the end of this report. HTDC, which stands for "corrected heat transfer density", and HTD in the above expression are evaluated on the basis of the void volume of one side of the core. P_c stands for "compactness parameter".

A simple substitution of the material volume and density in the above expression gives a correction factor for the specific heat transfer (SHT) and results in the following relationships:

$$\mathrm{SHT} = \mathrm{HTD}/(1-\sigma)d \quad , \tag{1}$$

$$\mathrm{SHTC} = \mathrm{HTDC}/(1-\sigma)d \quad . \tag{1a}$$

The compactness-parameter method of plotting data successfully separated fluid properties and design conditions and therefore allowed each core design to be compared primarily on the basis of surface-property factors. Another salient advantage was that only reduced data of the form f and j versus N_{Re} were needed, and a large percentage of heat-exchanger data are presented in this manner only. "Raw" data, including pressure drops, flow rates, temperatures, etc., were much more scarce.

An estimated HTD and void volume for a core can be calculated easily for a given set of design conditions and performance requirements by calculating a P_c for a chosen surface configuration, then finding the HTD for one side by multiplying the P_c by the design parameter and dividing by the correction factor calculated for the intended design conditions. [See Equation (A-37).] Then the void volume for that side can be found by multiplying this HTD by the log-mean-temperature difference and dividing by the heat-transfer rate. The total core volume is obtained by adding the void volume for each side and then dividing by σ. The above procedure is demonstrated later in this report.

Literature Survey Results

The literature survey covered the period from 1945 to the present, and the subject categories of Heat Exchangers and Heat Transfer were searched to include all references having pertinent information regarding compact heat exchangers. The sources searched included:

Applied Science and Technology Index

Engineering Index

Nuclear Science Abstracts

OTS Selective Bibliography

Scientific and Technical Aerospace Reports (NASA)

Technical Abstract Bulletin (DDC)

U. S. Government Research Reports (OTS).

Approximately 500 references were initially extracted. When it became apparent that time would not permit a detailed study of each, it was decided to concentrate on the reports containing heat-exchanger performance data. This reduced to approximately 200 the number of references to be reviewed. These are listed in the selected bibliography. During the latter stages of reviewing, the calculations were expedited by including only reports having data in the more compact range of hydraulic radius, since there were already ample data for surfaces having a hydraulic radius larger than about 0.0015 feet.

Figures 1 through 4 show the results, plotted both in "raw" form and in the more usable corrected form, of data gathered in the literature survey. As expected, the data plotted using the corrected heat-transfer density and compactness-parameter terms show much less scatter than do the uncorrected data, which contain the influence of fluid

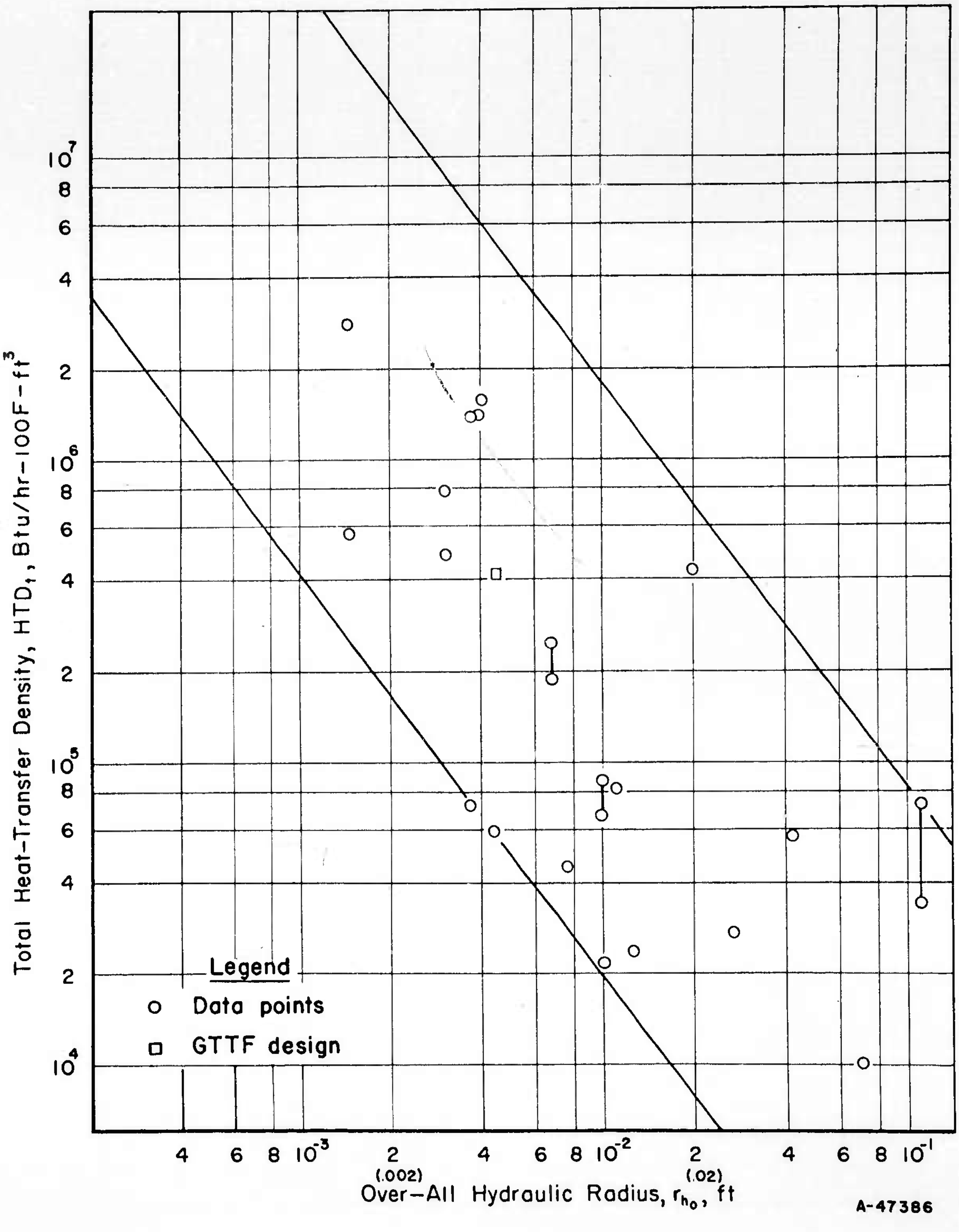

FIGURE 1. TOTAL HEAT-TRANSFER DENSITY VERSUS HYDRAULIC RADIUS

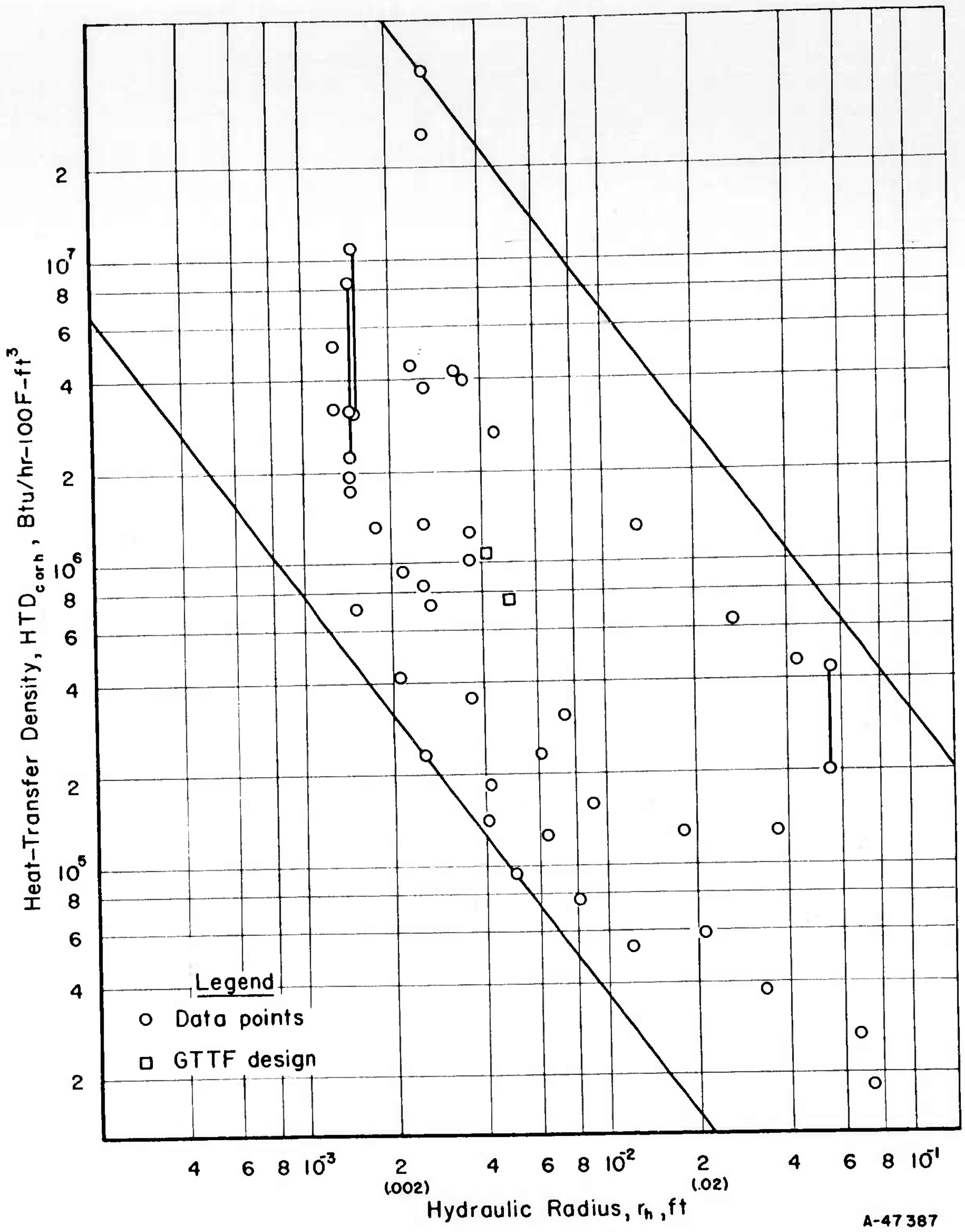

FIGURE 2. HEAT-TRANSFER DENSITY VERSUS HYDRAULIC RADIUS

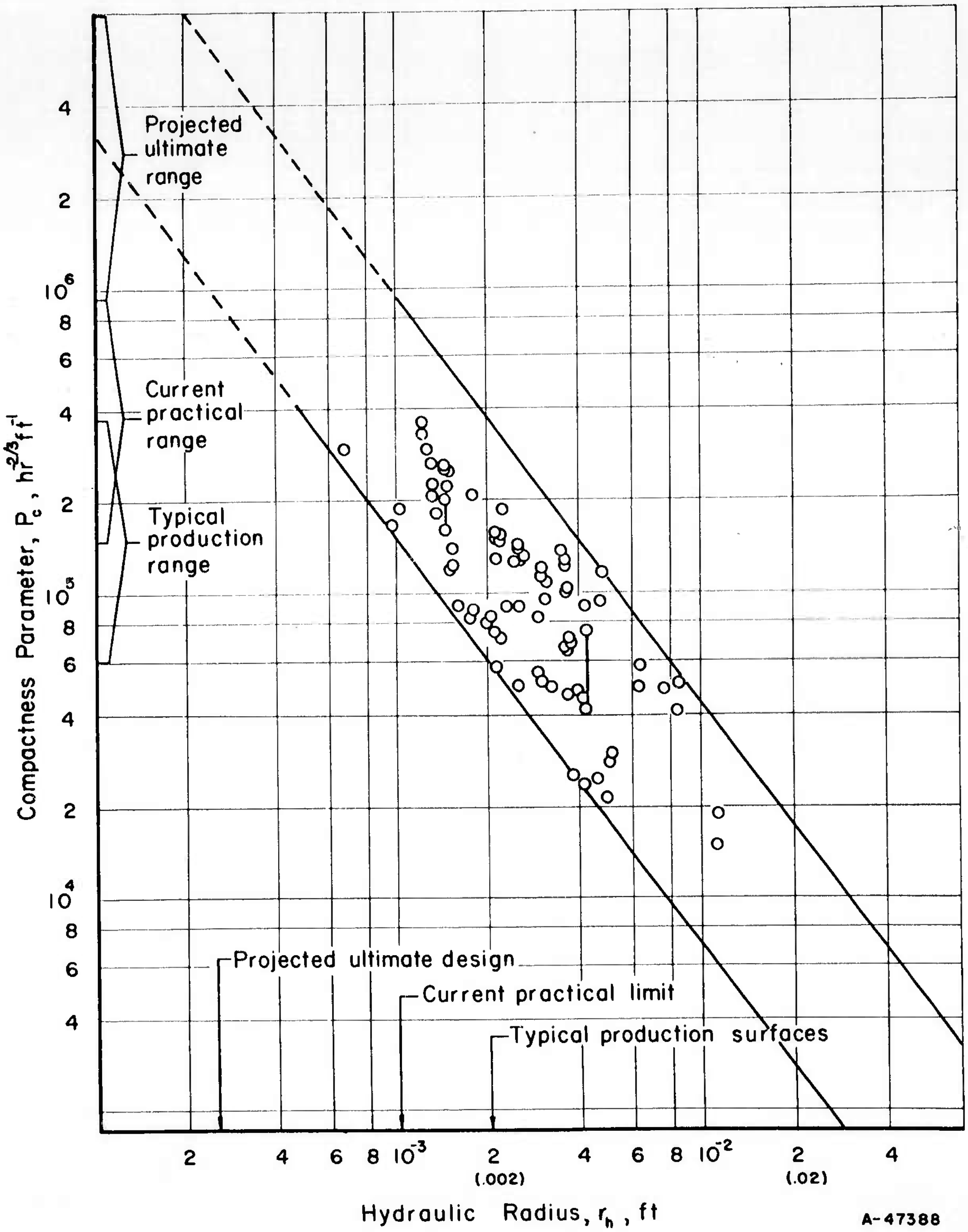

FIGURE 3. COMPACTNESS PARAMETER VERSUS HYDRAULIC RADIUS

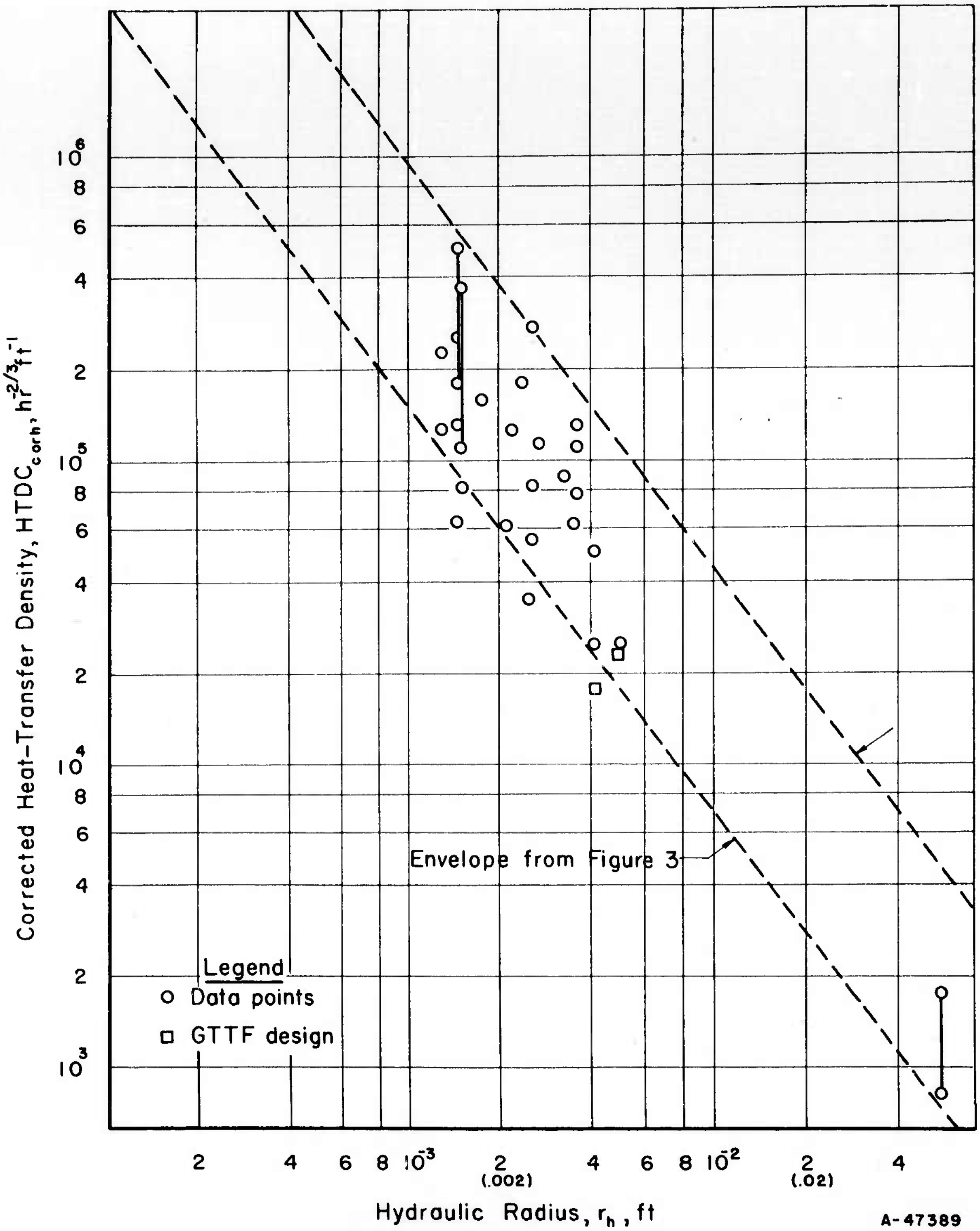

FIGURE 4. CORRECTED HEAT-TRANSFER DENSITY VERSUS HYDRAULIC RADIUS

properties and operating conditions. Indeed, no accurate or meaningful conclusions could have been reached concerning the state of the art for compact-heat-exchanger technology had not the effects of fluid properties and operating conditions been essentially eliminated or properly accounted for.

Figures 1 and 2 show HTD data points plotted without correction for the influence of fluid properties or operating conditions; therefore, the data were widespread by a factor of 150, on the average, for a given hydraulic radius. The HTD calculations are based on the void volume of the core for both sides in Figure 1, and for one side in Figure 2.

Figures 3 and 4 show the corrected HTDC data points obtained from raw data, and P_c data points calculated from reduced data using the j and f factors only. These "corrected" plots produced much better grouping of the data, the spread being only a factor of 6 for a given hydraulic radius; therefore, more accurate boundaries could be given, which lent confidence to extrapolation of the results to extreme compactness ranges. The state-of-the-art limits for r_h, P_c, and HTDC are also shown in Figure 3. Both the present and future technological capabilities are represented by boundary lines, and the average hydraulic radius for surfaces presently being produced is included for comparison.

The HTD, HTDC, and P_c terms are plotted versus the hydraulic radius because the equations revealed this was the most important single design variable affecting the compactness. The enclosing envelope lines are drawn with a -4/3 slope because the hydraulic radius is raised to this power in Equation (A-42) for the compactness parameter. The points designated "GTTF Design" in Figures 1, 2, and 4 are for the recuperator described in the "Engineering Manual for Ferrotherm Heat Exchanger for Closed-Cycle Gas Turbine Test Facility". This was the only recuperator previously designed or proposed for APCDB for which enough data was available to permit plotting on these graphs.

Tables 1 through 4 represent all the data points shown in Figures 1 through 4, respectively, and specify the heat exchangers' operating conditions, the parameters used in calculating each point, and the reference from which the data were taken.

Figures 5 and 6 are sketches of representative core surfaces for which the actual dimensions are included in Tables 1 through 4 for each data point.

Manufacturer Survey Results

Battelle engineers visited the following eight heat-exchanger manufacturers to inquire about the practical manufacturing limits of compactness and the difficulties expected in producing still smaller hydraulic radius surfaces.

Ferrotherm Company, Cleveland, Ohio

Harrison Radiator Division, Lockport, New York

Janitrol Aero Division, Columbus, Ohio

Modine Manufacturing Corporation, Racine, Wisconsin

TABLE 1. HEAT-TRANSFER DATA, TOTAL CORE

r_{h_o}, ft	Heat Exch. No.	HTD_t, Btu/hr-100 F-ft^3	SHT_t, Btu/hr-100 F-16	Type of Surface	Fluids: Hot	Fluids: Cold	Flow Geometry*	Temperatures, F: T_{h1}	T_{h2}	T_{c1}	T_{c2}	ΔT_{ln}	Heat Flux, Btu/hr	Maximum Pressure, psia: Hot	Cold	Surface Dimension, Hot Side: Fins/in. (1/a)*	Plate or Tube Spacing, in. (b)*	(c)	Surface Dimension, Cold Side: Fins/in. (1/a)*	Plate or Tube Spacing, in. (b)*	(c)	Volume, ft^3	Weight, (lbs)	Reference Number
0.00146	1	6.4×10^5	–	Strip fin	Air	Air	CF	213	136	65	145	63	6.75×10^3	15	15	?	?	?	?	?	?	0.0168	?	154
0.00308	2	7.9×10^5	5.0×10^4	Louvered fin	Air	Water	CF	200	74	60	91	37	6.1×10^5	15	?	11.1	0.025	0.375	11.1	0.25	–	2.08	33	120
0.00308	3	4.9×10^5	3.1×10^4	Plain fin	Air	Water	CF	200	74	60	91	37	6.1×10^5	15	?	11.1	0.25	–	11.1	0.25	–	3.37	54	120
0.00370	4	7.3×10^5	–	0.210" OD tube and shell	Steam	Air	NA	–	–	73	169	85.2	5.74×10^3	15	15	–	0.250	0.216	–	–	–	0.0921	?	154
0.00376	5	6.8×10^5	–	0.210" OD tube bank (staggered)	Helium	Hydrogen	CF	627	587	558	588	30.0	5.79×10^3	269	786	–	0.315	0.262	–	–	–	0.0284	?	212
0.00390	6	1.4×10^6	–	0.188" OD tube and shell	Sodium	Air	CF-CC	845	721	58	719	–	1.27×10^4	?	63	–	0.25	0.216	–	–	–	0.23	?	136
0.00403	7	1.6×10^6	1.1×10^5	0.036" O pin fin	Air	Water	CF	200	74	60	91	23	6.1×10^5	15	?	?	0.25	?	–	0.125	–	1.67	25	120
0.0045	8	4.20×10^5	5.7×10^3	Elliptical pin fin	Nitrogen	Nitrogen	CF-CC	848	420	329	761	89	7.75×10^6	85	220	10	0.440	0.110	12.5	0.320	?	19.66	1,530	GTTF design
0.00683	9	1.9×10^5	1.0×10^3	Triangular fin	Exhaust gas	Air	CF	1600	1305	60	353	655	3.0×10^5	15	15	?	0.505	–	?	0.313	–	0.52	45	197
0.00758	10	4.6×10^4	8.3×10^2	Wavy fin	Exhaust gas	Air	CF	1600	1328	50	503	1188	3.26×10^5	15	15	–	0.50	–	–	0.22	2.0	0.596	33	38
0.0099	11	2.18×10^4	4.1×10^2	Parallel plate (no fins)	Exhaust gas	Air	CF	1600	1253	50	628	1053	4.16×10^5	15	15	–	0.313	–	–	0.188	–	1.81	97	38
0.010	12	6.75×10^4	7.0×10^2	Parallel plate (no fins)	Exhaust gas	Air	CF	1600	1305	60	355	655	2.3×10^5	15	15	–	0.3125	–	–	0.1875	–	0.52	50	197
0.0112	13	8.33×10^4	1.5×10^3	0.750" OD internally finned tube bank (staggered)**	Exhaust gas	Air	CF	1025	896	70	212	820	3.15×10^5	15	15	–	1.375	0.812	45°	–	–	0.461	26	177
0.0125	14	2.4×10^4	–	0.375" OD tube bank (staggered)	Sodium	Air	CF	426	456	83	227	299	4.41×10^3	?	?	–	–	–	–	0.562	0.562	0.0625	?	49
0.0168	15	2.82×10^6	–	Strip fin	Steam	Air	NA	–	–	66	198	61	2.88×10^4	15	15	?	?	?	?	?	?	0.0168	?	154
0.0198	16	4.3×10^5	–	Spiral parallel plate (no fins)**	Water	Water	CC	156	149	83	100	60.5	2.55×10^4	?	?	–	0.625	–	–	0.25	–	0.0983	?	195
0.0268	17	2.31×10^4	5.8×10^2	Double tube with fins**	Exhaust gas	Air	PF	1600	1374	50	426	1230	2.71×10^5	15	15	–	~0.45	–	–	~0.45	–	0.956	38	38
0.0418	18	5.73×10^4	9.2×10^2	Flat tube bank (staggered)	Exhaust gas	Air	CF	1600	1278	50	586	1100	3.86×10^5	15	15	–	0.56	0.716	–	–	–	0.611	38	38
0.063	19	6.0×10^4	–	0.250" tube bank (in line)	Steam	Air	CF	228	228	77.6	169	95.2	1.31×10^4	20.0	15	–	–	–	–	0.375	0.312	0.23	?	121
0.0711	20	1.02×10^4	3.9×10^2	Tube with pin fins**	Exhaust gas	Air	CF	1600	1538	50	153	1445	7.40×10^4	15	15	30°	–	–	30°	–	–	0.504	13	38
0.11	21	7.4×10^4	–	Plate fin	Oil	Steam	CF	400	348	211	211	161	2.7×10^5	400	15	12	0.50	–	4	–	–	2.27	?	159

*See nomenclature in Appendix.
**See Figure 12 in Supplement.

TABLE 2. HEAT-TRANSFER DENSITY DATA FOR ONE SIDE OF CORE

r_h, ft	Heat Exch. No.	HTD_c or h', Btu/hr-100 F-ft^3	Type of Surface (Hot or Cold Side)	Fluids		Flow Geometry*	Temperature, F					Heat Flux, Btu/hr	Maximum Pressure, psia		Surface Dimensions			Volume, ft^3	Reference Number
				Hot	Cold		T_{h1}	T_{h2}	T_{c1}	T_{c2}	ΔT_{ln}		Hot	Cold	Fins/In. (1/a)*	Plate or Tube Spacing, in. (b)*	(c)*		
0.00129	22	5.23×10^6 (c)	0.156" OD finned tube bank	NaK	Air	CF-CC	859	586	499	772	–	2.18×10^6	102	?	24	0.452	0.397	0.448	71
0.00129	23	3.26×10^6 (h)	0.156" OD finned tube bank	Exhaust gas	NaK	CF-CC	566	859	1019	754	–	2.18×10^6	16	?	24	0.452	0.397	0.376	71
0.00146	1	3.19×10^6 (c)	Strip fin	Steam	Air	NA	–	–	66	198	62.1	1.23×10^4	15	15	?	?	?	0.0062	154
0.00146	1	2.21×10^6 (c)	Strip fin	Steam	Air	NA	–	–	69	211	37.3	5.1×10^3	15	15	?	?	?	0.0062	154
0.00146	1	1.92×10^6 (h)	Strip fin	Air	Air	CF	213	136	65	145	63	6.75×10^3	15	15	?	?	?	0.0056	154
0.00146	1	1.73×10^6 (c)	Strip fin	Air	Air	CF	213	136	65	145	63	6.75×10^3	15	15	?	?	?	0.0062	154
0.001495	24	7.11×10^5 (c)	Triangular fin	Steam	Air	NA	229	229	78	205	69.2	5.51×10^4	15	15	19.86	0.250	–	0.112	113
0.00152	25	1.1×10^7 (h)	Flat tube and shell	Oil	Boiling water	CC-CF	396	351	86	298	134.6	3.12×10^4	?	15	–	0.174	?	0.00185	159
0.00152	25	3.12×10^6 (h)	Flat tube and shell	Oil	Boiling water	CC-CF	400	360	83	218	204.5	1.34×10^4	?	15	–	0.174	?	0.00185	159
0.00174	26	1.31×10^6 (c)	Ruffled fin	Steam	Air	NA	229	229	79	222	46.7	5.78×10^4	15	15	17.8	0.413	0.375	0.0948	114
0.00212	27	4.20×10^5 (c)	Triangular fin	Steam	Air	NA	228	228	65	194	82.5	3.85×10^4	15	15	14.77	0.330	–	0.111	113
0.00218	28	9.39×10^5 (c)	Strip fin	Steam	Air	NA	229	229	85.3	219	51	4.64×10^4	15	15	15.2	0.414	0.125	0.0969	114
0.00236	5	2.2×10^6 (c)	0.190" ID tube bank (staggered)	Helium	Hydrogen	CF	627	587	558	588	30.0	5.79×10^6	269	786	–	–	–	0.00875	212
0.00253	2	1.35×10^6 (h)	Louvered fin	Air	Water	CF	200	74	60	91	37	6.1×10^5	15	?	11.1	0.25	0.375	1.22	120
0.00253	3	8.4×10^5 (h)	Plain fin	Air	Water	CF	200	74	60	91	37	6.1×10^5	15	?	11.1	0.25	–	1.97	120
0.00253	29	2.30×10^5 (c)	Plate fin	Steam	Air	NA	229	229	84	172	101	2.56×10^4	15	15	11.1	0.250	–	0.110	113
0.00256	7	3.8×10^6 (h)	0.036" D pin fin	Air	Water	CF	200	74	60	91	23	6.1×10^5	15	?	?	0.25	?	0.70	120
0.00262	22	4.1×10^7 (h)	0.125" ID finned tube bank	NaK	Air	CF-CC	859	586	499	772	–	2.18×10^6	102	?	–	–	–	0.057	71
0.00262	23	2.56×10^7 (c)	0.125" ID finned tube bank	Exhaust gas	NaK	CF-CC	586	859	1019	754	–	2.18×10^6	16	?	–	–	–	0.0479	71
0.00265	30	7.26×10^5 (c)	Ruffled fin	Steam	Air	NA	229	229	75	216	57	3.94×10^6	15	15	11.44	0.413	0.375	0.095	114
0.00325	6	4.3×10^6 (c)	0.156" ID tube and shell	Sodium	Air	CF-CC	845	721	58	719	–	1.27×10^4	?	63	–	–	–	0.0748	136

TABLE 2. (Continued)

r_h, ft	Heat Exch. No.	HTD_c or h, Btu/hr-100 F-ft^3	Type of Surface (Hot or Cold Side)	Fluids: Hot	Fluids: Cold	Flow Geometry*	Temperature, F: T_{h1}	T_{h2}	T_{c1}	T_{c2}	ΔT_{ln}	Heat Flux, Btu/hr	Maximum Pressure, psia: Hot	Cold	Surface Dimensions: Fins/In. (1/a)*	Plate or Tube Spacing, in.: (b)*	(c)*	Volume, ft^3	Reference Number
0.00348	5	2.0×10^6 (h)	0.210" OD tube bank (staggered)	Helium	Hydrogen	CF	627	587	558	588	30.0	5.79×10^6	269	786	–	0.315	0.262	0.00944	212
0.00357	31	1.27×10^6 (c)	Elliptical pin fin (in line)	Steam	Air	NA	228	228	81	149	113	1.26×10^5	15	15	10	0.440	0.110	0.0874	128
0.00357	31	1.01×10^6 (c)	Elliptical pin fin (in line)	Steam	Air	NA	228	228	76	161	109.5	9.65×10^4	15	15	10	0.440	0.110	0.0874	128
0.00357	31	3.60×10^5 (c)	Elliptical pin fin (in line)	Steam	Air	NA	228	228	86	193	76.6	2.41×10^4	15	15	10	0.440	0.110	0.0874	128
0.00410	8	1.073×10^6 (c)	Elliptical pin fin (in line)	Nitrogen	Nitrogen	CF-CC	848	420	329	760	89	7.75×10^6	85	220	12.5	0.320	?	8.11	GTTF design
0.0041	4	1.38×10^5 (c)	0.197" ID tube and shell	Steam	Air	NA	–	–	73	169	85.2	5.74×10^3	15	15	–	–	–	0.0487	154
0.00415	19	1.84×10^5 (c)	0.250" DD tube bank (in line)	Steam	Air	CF	228	228	78	169	95.2	1.31×10^4	20	15	–	0.375	0.312	0.075	121
0.00443	6	2.7×10^6 (h)	0.188" OD tube and shell	Sodium	Air	CF-CC	845	721	58	719	–	1.27×10^4	?	63	–	0.25	0.216	0.120	136
0.00486	8	7.54×10^5 (h)	Elliptical pin fin	Nitrogen	Nitrogen	CC-CF	848	420	329	760	89	7.75×10^6	85	220	≈10	0.440	0.110	11.55	GTTF design
0.00504	32	9.43×10^4 (c)	Triangular fin	Steam	Air	NA	228	228	79	153	108	1.205×10^4	15	15	5.3	0.470	–	0.114	113
0.0061	10	2.35×10^5 (c)	Wavy fin	Exhaust gas	Air	CF	1600	1328	50	503	1188	3.26×10^5	15	15	–	0.22	~2.0	0.117	38
0.00728	13	1.28×10^5 (h)	0.750" DD internally finned tube bank (staggered)**	Exhaust gas	Air	CF	1025	896	70	212	820	3.15×10^5	15	15	–	1.375	0.812	0.300	177
0.00807	11	7.65×10^4 (c)	Parallel plate (no fins)	Exhaust gas	Air	CF	1600	1253	50	628	1053	4.16×10^5	15	15	–	0.188	–	0.516	38
0.00906	10	1.58×10^5 (h)	Wavy fin	Exhaust gas	Air	CF	1600	1328	50	503	1188	3.26×10^5	15	15	–	0.50	–	0.174	38
0.0118	11	5.34×10^4 (h)	Parallel plate (no fins)	Exhaust gas	Air	CF	1600	1253	50	628	1053	4.16×10^5	15	15	–	0.313	–	0.741	38
0.0130	16	1.3×10^6 (c)	Spiral parallel plate (no fins)**	Water	Water	CC	156	149	83	100	60.5	2.54×10^4	?	?	–	0.25	–	0.0323	195
0.0176	13	3.12×10^5 (c)	0.694" ID internally finned tube bank (staggered)**	Exhaust gas	Air	CF	1025	896	70	212	820	3.15×10^5	15	15	45°	–	–	0.123	177
0.0208	17	5.91×10^4 (c)	Double tube with fins**	Exhaust gas	Air	PF	1600	1374	50	426	1230	2.71×10^5	15	15	1.6	~0.45	–	0.373	38
0.0266	16	6.4×10^5 (h)	Spiral parallel plate (no fins)**	Water	Water	CC	156	149	83	100	60.5	2.55×10^4	?	?	–	0.625	–	0.066	195
0.0327	17	3.77×10^4 (h)	Double tube with fins**	Exhaust gas	Air	PF	1600	1374	50	426	1230	2.71×10^5	15	15	1.6	~0.45	–	0.585	38
0.0373	18	1.275×10^5 (h)	Flat tube bank (staggered)	Exhaust gas	Air	CF	1600	1278	50	586	1100	3.86×10^5	15	15	–	0.56	0.716	0.268	38
0.0437	18	4.63×10^5 (c)	Flat tube bank (staggered)	Exhaust gas	Air	CF	1600	1278	50	586	1100	3.86×10^5	15	15	–	–	–	0.0757	38
0.0546	21	4.39×10^5 (h)	Plate fin	Oil	Steam	NA	400	348	211	211	161	2.7×10^5	400	15	12	0.5	–	0.383	159
0.0546	21	2.05×10^5 (h)	Plate fin	Oil	Steam	NA	300	266	211	211	70.5	5.52×10^4	400	15	12	0.5	–	0.383	159
0.0672	20	2.74×10^4 (c)	Tube with pin fins**	Exhaust gas	Air	CF	1600	1538	50	153	1445	7.40×10^4	15	115	30°	–	–	0.187	38
0.074	20	1.84×10^4 (h)	Tube with pin fins**	Exhaust gas	Air	CF	1600	1538	50	153	1445	7.40×10^4	15	15	30°	–	–	0.276	38

* See nomenclature in Appendix.
**See Figure 12 in Supplement.

TABLE 3. COMPACTNESS PARAMETER DATA

r_h, ft	Heat Exch. No.	P_c, $hr^{-2/3}$ ft^{-1}	Type of Surface	Fluids		Flow Geometry*	j/f	c_j	Surface Dimensions			Reference Number
				Hot	Cold				Fins/in. (1/a)*	Plate or Tube Spacing, in. (b)*	(c)	
0.00066	33	2.98×10^5	Triangular fin	Steam	Air	NA	0.267	0.048	46.45	0.100	–	46
0.00094	34	1.65×10^5	Triangular fin	Steam	Air	NA	0.29	0.045	25.79	0.101	–	129
0.00100	35	1.85×10^5	Triangular fin	Steam	Air	NA	0.28	0.055	30.33	0.170	–	129
0.00122	36	3.64×10^5	Offset rectangular fin	Steam	Air	NA	0.282	0.130	20.06	0.100	0.125	46
0.00122	36	3.35×10^5	Offset rectangular fin	Steam	Air	NA	0.265	0.122	20.06	0.100	0.125	46
0.00127	37	3.00×10^5	Strip fin – double sandwich	Steam	Air	NA	0.25	0.12	16.12	0.206	0.125	129
0.00127	38	2.62×10^5	Strip fin – double sandwich	Steam	Air	NA	0.24	0.113	16.12	0.206	0.125	125
0.00128	39	2.5×10^5	Strip fin – triple sandwich	Steam	Air	NA	0.21	0.112	16.12	0.314	0.125	125
0.00131	40	2.10×10^5	Slit fin	Steam	Air	NA	0.194	0.097	24	0.250	0.125	**
0.00132	41	2.27×10^5	Strip fin-double sandwich	Steam	Air	NA	0.27	0.087	15.4	0.206	0.250	125
0.00134	42	1.85×10^5	0.031" D pin fin (in line)	Steam	Air	NA	0.216	0.082	16.15	0.750	0.062	120
0.00146	1	2.66×10^5	Strip fin	Steam	Air	NA	0.158	0.142	?	?	?	154
0.00146	1	2.51×10^5	Strip fin	Air	Air	NA	0.145	0.138	?	?	?	154
0.00146	1	2.28×10^5	Strip fin	Air	Air	NA	0.131	0.129	?	?	?	154
0.00146	1	1.63×10^5	Strip fin	Steam	Air	NA	0.122	0.116	?	?	?	154
0.001495	24	1.17×10^5	Triangular fin	Steam	Air	NA	0.271	0.0555	19.86	0.250	–	113
0.00150	43	1.22×10^5	Triangular fin	Steam	Air	NA	0.285	0.057	19.86	0.250	–	116
0.00153	44	1.40×10^5	Herringbone	Steam	Air	NA	0.25	0.073	18	0.426	?	201
0.00159	45	9.12×10^4	Plain serpentine	Steam	Air	NA	0.204	0.055	22	0.320	?	**
0.00174	26	2.09×10^5	Wavy fin	Steam	Air	NA	0.224	0.13	17.8	0.413	0.375	123, 4
0.00174	26	8.26×10^4	Ruffled fin	Steam	Air	NA	0.209	0.128	17.8	0.413	0.375	114
0.00175	46	8.52×10^4	Serrated fin	Steam	Air	NA	0.25	0.14	15	0.375	0.125	201
0.00198	47	8.0×10^5	Straight fin	Steam	Air	NA	0.238	0.0515	14	0.200	–	**
0.00205	48	1.3×10^5	Perforated plate fin	Steam	Air	NA	0.38	0.09	13.95	0.200	–	126
0.00206	49	5.65×10^4	Plate fin	Steam	Air	NA	0.43	0.037	12.5	0.310	–	201
0.00211	50	1.60×10^5	Serrated strip fin	Steam	Air	NA	0.188	0.135	15	0.375	0.125	**
0.00212	27	7.45×10^4	Triangular fin	Steam	Air	NA	0.235	0.059	14.77	0.330	–	113
0.00217	51	1.54×10^5	Strip fin	Steam	Air	NA	0.188	0.135	15.2	0.414	0.125	123, 4
0.00218	23	1.48×10^5	Strip fin	Steam	Air	NA	0.187	0.135	15.2	0.414	0.125	114
0.00219	52	7.14×10^4	Plate fin	Steam	Air	NA	0.267	0.056	15.08	0.418	–	119
0.00220	53	1.87×10^5	Strip fin	Steam	Air	NA	0.16	0.19	13.95	0.375	0.125	126
0.00226	54	8.66×10^4	Herringbone	Steam	Air	NA	0.23	0.078	12	0.426	?	201
0.00236	5	1.54×10^5	0.190" ID tube bank (staggered)	Helium	Hydrogen	CF	0.217	0.134	–	–	–	212
0.00244	55	9.03×10^4	0.375" OD finned tube bank (staggered)	Steam	Air	CF	0.226	0.088	11.46	0.975	0.80	140
0.00248	56	1.46×10^5	Wavy fin	Steam	Air	NA	0.20	0.16	11.5	0.375	0.375	129
0.00248	57	1.43×10^5	Wavy fin	Steam	Air	NA	0.19	0.16	11.5	0.375	0.375	126
0.00253	58	1.27×10^5	Louvered fin	Steam	Air	NA	0.240	0.120	11.1	0.250	0.375	116
0.00253	59	1.26×10^5	Louvered fin	Steam	Air	NA	0.217	0.13	11.1	0.250	0.250	123, 4
0.00253	60	5.03×10^4	Triangular fin	Steam	Air	NA	0.239	0.0512	11.1	0.250	–	113

TABLE 3. (Continued)

r_h, ft	Heat Exch. No.	P_c, $hr^{-2/3}ft^{-1}$	Type of Surface	Fluids Hot	Fluids Cold	Flow Geometry*	j/f	c_j	Surface Dimensions Fins/in. (1/a)*	Plate or Tube Spacing, in. (b)*	(c)	Reference Number
0.00265	30	1.32×10^5	Ruffled fin	Steam	Air	NA	0.182	0.153	11.44	0.413	0.375	114
0.00288	61	8.43×10^4	Finned flat tube bank (staggered)	Steam	Air	CF	0.255	0.097	11.32	0.55	0.79	116
0.00288	62	5.50×10^4	Rectangular plate fin	Steam	Air	NA	0.30	0.060	11.11	0.250	–	140
0.00293	63	1.22×10^5	0.04" D pin fin (in line)	Steam	Air	NA	0.146	0.175	8.33	0.398	0.096	120
0.00294	64	1.13×10^5	Strip fin	Steam	Air	NA	0.184	0.151	12.22	0.485	0.094	116
0.00295	65	5.14×10^4	Finned flat tube bank (staggered)	Steam	Air	CF	0.273	0.060	9.68	0.436	1.06	123, 4
0.00298	66	9.67×10^4	0.402" OD finned tube bank (staggered)	Steam	Air	CF	0.35	0.105	8.0	1.0	0.866	123, 4
0.00309	67	1.09×10^5	0.375" OD tube bank (in line)***	Steam	Air	CF	0.302	0.13	–	0.469	0.469	121
0.00315	68	4.94×10^4	Triangular fin	Steam	Air	NA	0.25	0.07	10.27	0.544	–	46
0.00348	5	1.38×10^5	0.210" OD tube bank (staggered)	Helium	Hydrogen	CF	0.231	0.199	–	0.315	0.262	212
0.00357	31	1.28×10^5	Elliptical pin fin	Steam	Air	NA	0.207	0.215	10	0.440	0.110	128
0.00357	31	1.23×10^5	Elliptical pin fin	Steam	Air	NA	0.310	0.18	10	0.440	0.110	128
0.00357	31	1.01×10^5	Elliptical pin fin	Steam	Air	NA	0.181	0.194	10	0.440	0.110	128
0.00357	69	4.66×10^4	Flat tube bank (staggered)	Steam	Air	NA	0.296	0.068	–	0.444	0.344	123, 4
0.00361 1	70	1.05×10^5	0.04" D pin fin (in line)	Steam	Air	NA	0.227	0.170	8	0.240	0.125	120
0.00365	71	7.03×10^4	Louvered fin	Steam	Air	NA	0.163	0.13	6.06	0.250	0.375	123, 4
0.00365	72	6.85×10^4	Louvered fin	Steam	Air	NA	0.217	0.115	6.06	0.250	0.375	116
0.00365	73	6.68×10^4	Louvered fin	Steam	Air	NA	0.177	0.12	6.06	0.25	0.50	123, 4
0.00365	74	6.55×10^4	Louvered fin	Steam	Air	NA	0.216	0.11	6.06	0.25	0.50	123, 4
0.00375	75	2.46×10^4	Plate fin	Steam	Air	NA	0.267	0.040	4.0	0.18	–	119
0.00385	76	4.80×10^4	0.38" OD finned tube bank (staggered)	Steam	Air	CF	0.192	0.090	7.34	0.975	0.80	140
0.00397	77	4.51×10^4	0.774" OD finned tube bank (staggered)	Steam	Air	CF	0.208	0.086	9.05	2.725	0.80	123, 4
0.0041	4	2.4×10^4	0.197" ID tube and shell	Steam	Air	NA	0.24	0.0464	?	?	–	154
0.00415	78	9.17×10^4	0.250" OD tube bank (staggered)	Steam	Air	CF	0.222	0.018	–	0.375	0.312	123, 4
0.00415	79	7.56×10^4	0.250" OD tube bank (in line)	Steam	Air	CF	0.182	0.159	–	0.375	0.312	120
0.00415	79	3.98×10^4	0.250" OD tube bank (in line)	Steam	Air	CF	0.264	0.074	–	0.375	0.312	120
0.00455	80	2.46×10^4	Plate fin	Steam	Air	NA	0.294	0.050	6.2	0.405	–	119
0.00465	81	1.17×10^5	0.065" D pin fin (staggered)	Steam	Air	NA	0.158	0.30	5	0.502	0.125	123, 4
0.00465	81	9.42×10^4	0.065" D pin fin (staggered)	Steam	Air	NA	0.113	0.27	5	0.502	0.125	123, 4
0.00482	82	2.07×10^4	0.231" ID single tube	Steam	Air	NA	0.277	0.047	–	–	–	140
0.00504	83	2.98×10^4	Triangular fin	Steam	Air	NA	0.342	0.065	5.3	0.470	–	116
0.00504	32	2.76×10^4	Triangular fin	Steam	Air	N	0.236	0.0695	5.3	0.470	–	113
0.00620	84	5.85×10^4	0.375" OD tube bank (in line)	Steam	Air	CF	0.500	0.15	–	0.563	0.469	121
0.00620	84	4.84×10^4	0.375" OD tube bank (in line)	Steam	Air	CF	0.206	0.167	–	0.563	0.469	121
0.00742	85	4.78×10^4	Plate fin	Steam	Air	NA	0.193	0.214	4.2	0.510	–	123, 4
0.0082	86	4.96×10^4	0.375" OD tube bank (staggered)	Steam	Air	CF	0.180	0.26	–	0.750	0.375	123, 4
0.00835	87	4.03×10^4	0.125" D pin fin (in line)	Steam	Air	NA	0.217	0.206	2.67	0.875	0.250	120
0.0111	88	1.86×10^4	1.024" OD finned tube bank (staggered)	Steam	Air	CF	0.207	0.140	8.8	3.079	2.063	123, 4
0.0111	89	1.47×10^4	0.774" OD finned tube bank (staggered)	Steam	Air	CF	0.154	0.122	9.05	2.725	1.75	123, 4

* See nomenclature in Appendix.
** Private correspondence, source confidential.
*** Continuous sheet fins.

TABLE 4. CORRECTED HEAT-TRANSFER DENSITY DATA, ONE SIDE OF CORE

r_h, ft	Heat Exch. No.	HTOC$_c$ or h', $hr^{-2/3}$ ft^{-1}	Type of Surface	Fluids: Hot	Fluids: Cold	Flow Geometry*	Heat Flux, Btu/hr	C/Cmin	N_{tu_o}	Δp, lb/ft^2	N_{Pr}	c_p, Btu/lb-F	μ, lb/hr-ft	ρ, lb/ft^3	Correction Factor	HTO, Btu/hr-F-ft^3	Reference Number
0.00129	23	2.28×10^5 (h)	0.156" OO finned tube bank	Exhaust gas	Nak	CF	2.04×10^6	1.032	1.64	43.5	0.69	0.26	0.085	0.0303	7.00	3.26×10^4	71
0.00129	22	1.28×10^5 (c)	0.156" OO finned tube bank	Nak	Air	CF	2.04×10^6	1	3.126	264	0.69	0.252	0.0752	0.2507	2.445	5.23×10^4	71
0.00146	1	5.04×10^5 (c)	Strip fin	Steam	Air	NA	2.88×10^4	1	2.15	1,240	0.71	0.24	0.048	0.089	5.98	8.43×10^4	154
0.00146	1	2.54×10^5 (c)	Strip fin	Steam	Air	NA	1.23×10^4	1	1.89	32	0.71	0.24	0.046	0.071	7.95	3.19×10^4	154
0.00146	1	1.31×10^5 (c)	Strip fin	Air	Air	CF	6.75×10^3	1	1.15	20.8	0.71	0.24	0.046	0.071	7.56	1.73×10^4	154
0.00146	1	6.34×10^4 (h)	Strip fin	Air	Air	CF	6.75×10^3	1	1.15	271	0.71	0.24	0.046	0.071	3.30	1.92×10^4	154
0.001495	24	8.25×10^4 (c)	Triangular fin	Steam	Air	NA	5.51×10^4	1	1.83	9.83	0.71	0.24	0.046	0.071	11.6	7.11×10^3	113
0.00152	25	3.8×10^5 (h)	Flat tube and shell	Oil	Boiling water	CF-CC	3.12×10^4	1	1.568	99.2	110	0.521	15.0	49.5	3.46	1.1×10^5	159
0.00152	25	1.05×10^5 (h)	Flat tube and shell	Oil	Boiling water	CF-CC	1.34×10^4	1	0.660	42.5	104	0.524	14.5	49.1	3.27	3.12×10^4	159
0.00174	26	1.62×10^5 (c)	Ruffled fin	Steam	Air	NA	5.78×10^4	1	3.06	13.6	0.71	0.24	0.046	0.071	12.4	1.31×10^4	114
0.00212	27	6.18×10^4 (c)	Triangular fin	Steam	Air	NA	3.85×10^4	1	1.5[illegible]	4.25	0.71	0.24	0.046	0.071	14.7	4.20×10^3	113
0.00218	28	1.27×10^5 (c)	Strip fin	Steam	Air	NA	4.64×10^4	1	2.63	8.98	0.71	0.24	0.046	0.071	13.6	9.39×10^3	114
0.00236	5	6.85×10^4 (c)	0.190" ID tube bank (staggered)	Helium	Hydrogen	CF	5.79×10^6	1.43	1.40	0.60	0.7	3.4	0.0223	0.1	3.11	2.2×10^4	212
0.00253	2	8.4×10^4 (h)	Louvered fin	Air	Water	CF	6.1×10^5	1	3.3[illegible]	82	0.70	0.242	0.048	0.10	6.24	1.35×10^4	120
0.00253	3	5.5×10^4 (h)	Plain fin	Air	Water	CF	6.1×10^5	1	3.38	71.5	0.70	0.242	0.048	0.10	6.52	8.4×10^3	120
0.00253	29	3.50×10^4 (c)	Plate fin	Steam	Air	NA	2.56×10^4	1	0.923	2.24	0.71	0.24	9.046	0.071	15.2	2.30×10^3	113
0.00256	7	2.8×10^5 (h)	0.036" D pin fin	Air	Water	CF	6.1×10^5	1	5.5	78	0.70	0.242	0.048	0.10	7.35	3.8×10^4	120
0.00262**	22	9.58×10^2 (h)	0.125" ID finned tube bank	Nak	Air	CF	2.04×10^6	1	3.126	2,480	0.010	0.234	0.70	52	0.00234	4.1×10^5	71
0.00262**	23	4.01×10^2 (c)	0.125" ID finned tube bank	Exhaust gas	Nak	CF	2.04×10^6	1	1.64	2,900	0.070	0.23	0.45	49	0.00157	2.56×10^5	71
0.00265	30	1.15×10^5 (c)	Ruffled fin	Steam	Air	NA	3.94×10^4	1	2.46	5.2	0.71	0.24	0.046	0.071	15.9	7.26×10^3	114
0.00325	6	9.0×10^4 (c)	0.156" ID tube and shell	Sodium	Air	CF	7.60×10^5	1	2.72	2,528	0.68	0.245	0.062	0.076	2.09	4.3×10^4	136

TABLE 4. (Continued)

r_h, ft	Heat Exch. No.	HTDC$_c$ or h', $hr^{-2/3} ft^{-1}$	Type of Surface	Fluids: Hot	Fluids: Cold	Flow Geometry*	Heat Flux, Btu/hr	C/C min	N_{tu_0}	Δp, lb/ft^2	N_{Pr}	c_p, Btu/lb-F	μ, lb/hr-ft	ρ, lb/ft^3	Correction Factor	HTD, $Btu/hr\text{-}F\text{-}ft^3$	Reference Number
0.00348	5	6.2×10^4 (h)	0.210" OD tube bank (staggered)	Helium	Hydrogen	CF	5.79×10^6	1	1.40	8.8	0.72	1.26	0.054	0.0209	3.10	2.0×10^4	212
0.00357	31	1.30×10^5 (c)	Elliptical pin fin	Steam	Air	NA	9.65×10^4	1	0.777	30.8	0.71	0.24	0.046	0.071	12.9	1.01×10^4	128
0.00357	31	1.11×10^5(c)	Elliptical pin fin	Steam	Air	NA	1.25×10^5	1	0.60	76.0	0.71	0.24	0.046	0.071	8.77	1.27×10^4	128
0.00357	31	7.91×10^4 (c)	Elliptical pin fin	Steam	Air	NA	2.41×10^4	1	1.46	1.16	0.71	0.24	0.046	0.071	22.0	3.60×10^3	128
0.0041	4	2.52×10^4(c)	0.197" ID tube and shell	Steam	Air	NA	5.74×10^3	1	1.13	1.56	0.71	0.24	0.046	0.071	18.3	1.38×10^3	154
0.00410	8	1.79×10^4(c)	Elliptical pin fin	Nitrogen	Nitrogen	CF-CC	7.75×10^6	1	11	507	0.70	0.254	0.086	0.57	1.67	1.073×10^4	GTTF Design
0.00415	19	5.0×10^4(c)	0.250" OD tube bank (in line)	Steam	Air	CF	1.31×10^4	1	0.975	0.437	0.70	0.242	0.047	0.068	27.0	1.84×10^3	121
0.00486	8	2.39×10^4(h)	Elliptical pin fin	Nitrogen	Nitrogen	CF-CC	7.75×10^6	1.012	4.86	204	0.71	0.26	0.092	0.20	3.17	7.54×10^3	GTTF Design
0.00504	32	2.47×10^4 (c)	Triangular fin	Steam	Air	NA	1.205×10^4	1	0.661	0.312	0.71	0.24	0.046	0.071	26.2	9.43×10^2	113
0.0546	21	1.78×10^3 (h)	Plate fin	Oil	Steam	NA	2.7×10^5	1	0.311	475	18	0.58	0.30	50	0.404	4.39×10^3	159
0.0546	21	8.27×10^2 (h)	Plate fin	Oil	Steam	NA	5.52×10^5	1	0.311	475	18	0.58	2.30	50	0.404	2.05×10^3	159

*See nomenclature in Appendix.
**These points not plotted because of low Prandtl Number.

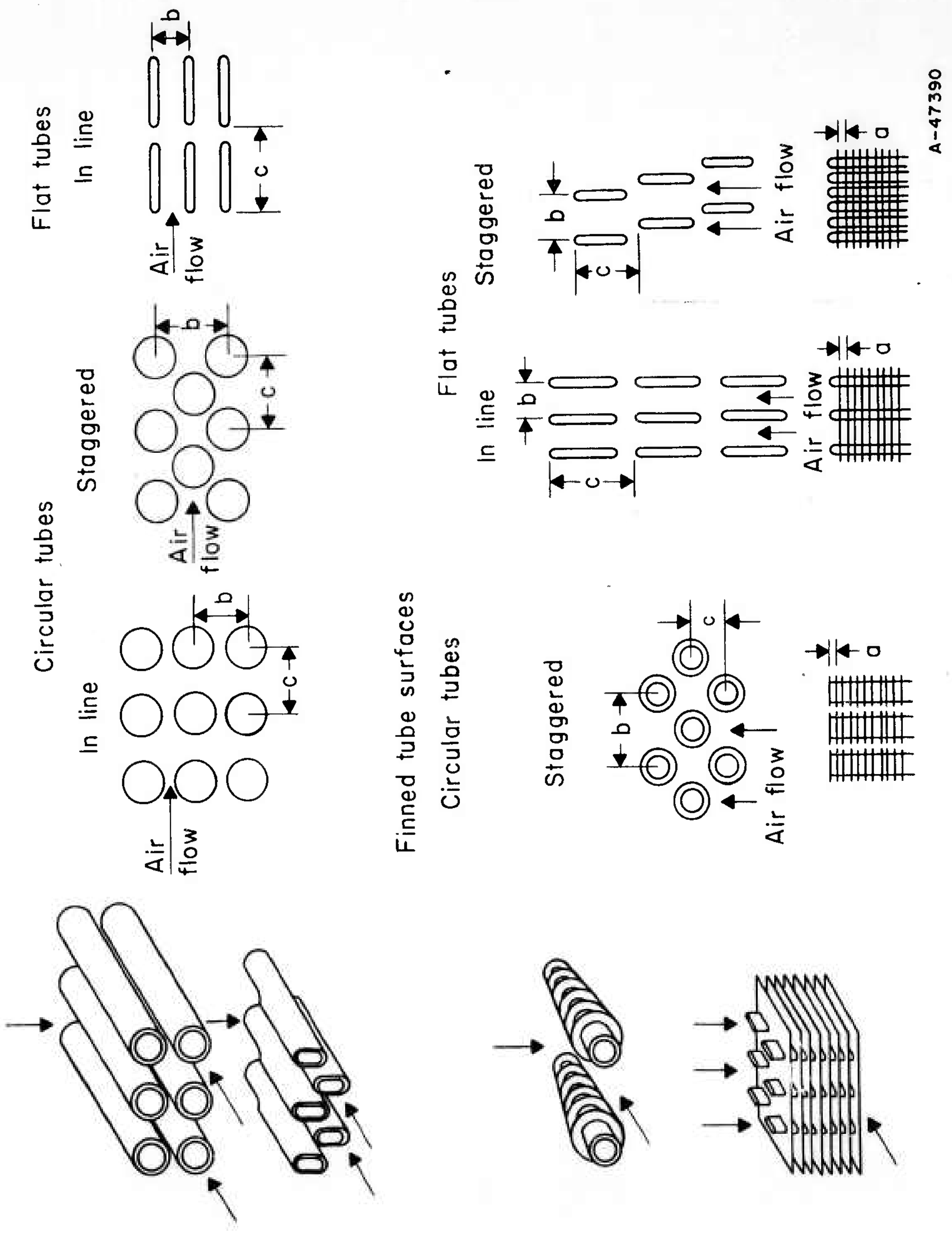

FIGURE 5. TUBULAR CORE SURFACES

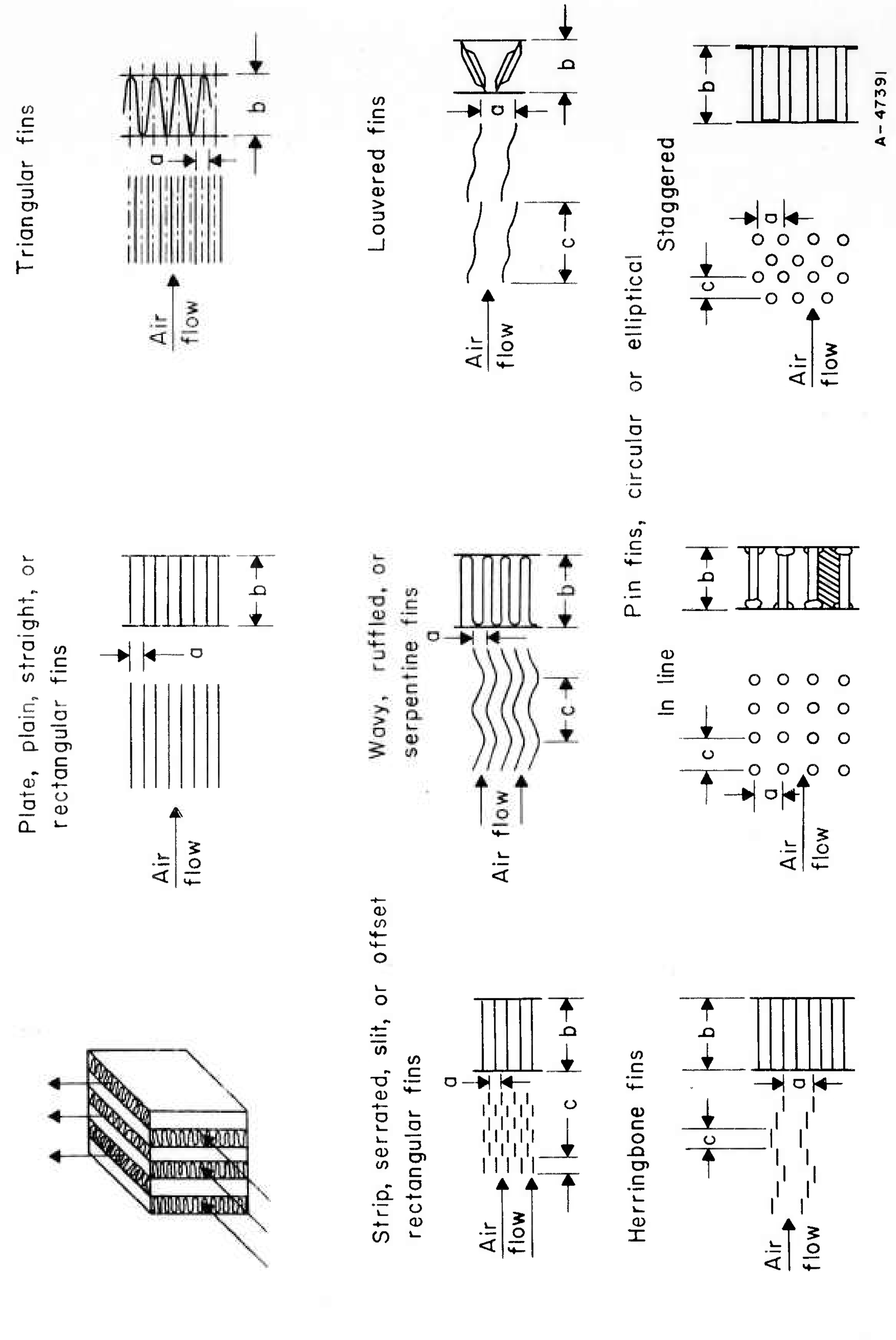

FIGURE 6. PLATE-FIN CORE SURFACES

Stewart-Warner Corporation, Indianapolis, Indiana

The Trane Company, LaCrosse, Wisconsin

United Aircraft Products, Inc., Dayton, Ohio

Young Radiator Company, Racine, Wisconsin

The companies were selected somewhat arbitrarily, although prominence in the manufacture of heat exchangers and convenience of location were considerations. It was not intended to interview all manufacturers experienced in this field but merely to include a sufficient number of them to assure a reasonably good representation of the industry.

For the most part, the manufacturers' comments were consistent and were helpful in developing an appreciation of manufacturing capabilities and limitations. The discussions revealed that manufacturing technology is capable of approaching the compactness limits determined by the data collected from the literature survey and presented in the preceding section in Figure 3. However, current production surfaces have typical hydraulic radii about double those of the present capability limit of approximately 0.001 ft.

The results of the discussions are summarized under the headings listed below.

Factors Limiting Compactness. The main factors limiting further compactness are: cost of manufacturing and materials, present machinery, brazing techniques, manifolding difficulty, dimensional tolerance factors, and fouling or cleaning. Customers often specify the number of fins per inch, but their chief criterion is cost per Btu transferred rather than size or weight, and neither of these demands gives the manufacturers much incentive toward further compactness.

Hydraulic Radius Limits. Typical production core surfaces have hydraulic radii around 0.002 ft, which correspond to a β of approximately 500 ft^2/ft^3, and a plate spacing of perhaps 0.20 inch with 15 fins per inch, but occasionally they approach 0.001 ft with a β of about 1,000 ft^2/ft^3, and a plate spacing of about 0.10 inch with 30 fins per inch. This factor, β, is the total heat-transfer surface area present in one cubic foot of core volume between plates. Under laboratory conditions, experimental test cores have been built having hydraulic radii in the range of 0.00025 to 0.0005 ft, which means that β is in the range of 4,000 to 2,000 ft^2/ft^3, respectively, and a surface having a hydraulic radius in this range may have a plate spacing of only 0.050 inch and require 100 fins per inch. Obviously, other combinations of plate spacing and number of fins per inch could be used to achieve the same hydraulic radius and compactness. The manufacturing limits given above will probably represent the state of the art for the next 5 to 10 years.

Materials. The two heat-exchanger materials for which heat-exchanger technology is most advanced are aluminum and stainless steel. Other steel alloys, having conductivities approaching that of carbon steel, are available for high-temperature uses and would be suitable for the APCDB recuperator, but their application technology is not as advanced and their cost is higher. Nickel may be a consideration for the recuperator design if even higher material conductivity is necessary, but otherwise a 300-series stainless steel would be satisfactory. Stainless-steel-clad copper fins have been used by several manufacturers, but at recuperator temperatures of 1400 F, the copper would alloy with the stainless steel. The cost of materials increases as thinner gages and more accurate tolerances are demanded. Thermal-expansion coefficients for all types of steels are similar enough so that fin material could differ from plate material to some extent and not cause problems.

Brazing. The primary difficulty arising in brazing extremely compact surfaces with many fins per inch is that of capillary action causing the brazing material to fill the flow passages between fins. This action may limit the fin spacing and also may cause the hydraulic radius to change from its design value. Vacuum brazing is the most successful technique for very compact surfaces. The current brazing technology is most advanced for aluminum and stainless steel, but if smaller hydraulic radii are required than those used in the experimental cores listed above in the section on hydraulic radius limits, new brazing techniques would be needed. The lack of a good brazing alloy for the 1400 F requirement may also be a durability barrier. Other materials would be even more difficult to braze. Spacer plates should not be made so thin as to allow the brazing alloy to diffuse completely through the material, thereby increasing the likelihood of leakage.

Manifolding. As core sizes of heat exchangers are designed more compactly, the ratio of manifold volume to core volume increases. Actually, the manifold volumes remain relatively constant because flow areas do not change appreciably for a given set of design conditions; only the flow length decreases appreciably as smaller hydraulic-radius surfaces are used. Therefore, a point of diminishing returns is reached for compactness when most of the heat exchanger volume becomes manifolding. A "rule-of-thumb" limit for counterflow heat exchangers might be a core length equal to only one-half the nonflow dimension. Manifolding becomes more difficult when the number of passages increases as they become more compact. Pressure loading may be a greater problem in the manifolds than in the core passages because the thin dividing sheets, although rigid enough in the core, may not be rigid enough in the manifold section. A floating-header type of manifold may be needed in the recuperator to prevent excessive thermal stressing during transient periods.

Leakage. The most common points of leakage occur from yielding of plate sheets by pressure loading, failure of the joints between side closures and spacers plates, and separation of the core and headers due to differential thermal expansion. Naturally, because the number of passages increases in a heat exchanger as the core surfaces become more compact, leakage problems will increase correspondingly. A successful design will be easier to achieve if APCDB will allow some measure of leakage, however small, rather than specifying no leakage whatever.

Durability. The 10,000-hour life requirement is probably not out of reach, but it is definitely beyond the state of the art and could not be guaranteed until many tests were performed. Thermal- and pressure-cycling severity would be the determining factors for longevity, and therefore may have to be specified for the manufacturers. Fatigue failures caused by differential thermal expansion are likely to occur at the junction of the core and manifold because, with highly compact surfaces, the thin sections of the core have high thermal response rates while thicker manifold materials have a slower response rate. Therefore, flexible headers, mentioned in the section under manifolding, may be necessary. At the higher recuperator temperatures, pressure loading will also be a design problem and may be a life-expectancy factor. High-temperature-brazing technology may also limit the life expectancy.

These discussions with manufacturers aided in projecting realistic goals and recommendations for compact-heat-exchanger design.

Design-Study Results

Using various core surfaces having a wide range of hydraulic radii, design studies were conducted of three recuperators and one precooler, in order to reveal the advantages and limitations of employing more compact heat-transfer surfaces and to help formulate practical compactness goals. During the design studies for the recuperator, pure counterflow was assumed and an attempt was made, analytically, to minimize both flow area and volume. (The procedure is presented in the calculations section.) However, the requirements for satisfying both of these design goals simultaneously were incompatible. Therefore, minimum volume was selected as the primary objective. To achieve the smallest core volume, the equations showed that the hydraulic radius should be equal on both fluid sides of the core, if a particular hydraulic radius is taken to be the minimum allowable for a particular design. In other words, if any other geometrically similar surface having a larger hydraulic radius were used for the other fluid side of the core, a larger total core volume would result, regardless of pressure-drop ratios, number of heat-transfer-unit ratios, etc.

Table 5 gives the design conditions specified by APCDB and used in sizing the three recuperators and one precooler.

Table 6 shows the average fluid properties at the mean mixed-bulk temperatures for the design conditions listed in Table 5, which were obtained from handbooks.

Figure 7 is a schematic diagram for a typical nuclear-powered, closed-cycle, gas-turbine, electric-generating plant.

Figure 8 shows the design used to estimate the over-all volumes and weights for the recuperators. Because only core data were available, it was necessary to assume some manifold configuration and to calculate a pressure-vessel thickness and an average insulation thickness to reduce the outside temperature to 150 F in order to predict an over-all HTD and SHT for APCDB. The flow area was equal for each fluid side because the hydraulic radius was the same on both sides to give minimum volume. The frontal area of the core was assumed to be square.

Figure 9 shows the design used for the precooler calculations. The precooler incorporated a crossflow design and would require ten 10-hp motors on the air side to produce the needed air flow.

Table 7 summarizes the numerical results of the design study and lists the more important parameters for the three recuperators and the one precooler. Example calculations and the procedure and assumptions used are given in the calculations section.

Figure 10 shows the relative sizes of the three recuperator-core designs resulting from the use of three quite different heat-transfer surfaces having hydraulic radii of 0.00253, 0.00122, and 0.00033 ft. These three particular surfaces were chosen because j and f data were available for them and they represented three stages of technology. As predicted, the flow area is not appreciably changed by reducing the hydraulic

TABLE 5. HEAT-EXCHANGER DESIGN CONDITIONS

	Precooler	Recuperator
Hot Fluid	Nitrogen	Nitrogen
Flow Rate, lb/hr	108,000	108,000
Inlet Tem, F	500	1400
Outlet Temp, F	110	500
Pressure, psia	175	175
Coolant	Ambient air	Nitrogen
Flow Rate, lb/hr	432,000*	180,000
Pressure, psia	14.7	500
Inlet Temp F	80	350
Outlet Temp, F	129*	1250
Heat Load, Btu/hr	12×10^6	17.5×10^6
Effectiveness, per cent	93	85.7
Pressure Drop	$(\Delta P/P)_h = 0.015$* $\Delta P_c = 4$ in. H_2O*	$(\Delta P/P)_c + (\Delta P/P)_h = 0.04$

*Conditions not specified by APCDB, but resulting from design study.

TABLE 6. AVERAGE FLUID PROPERTIES FOR HEAT-EXCHANGER DESIGN CONDITIONS

	Recuperator		Precooler	
Property	Hot Side	Cold Side	Hot Side	Cold Side
c_p, Btu/lb_m-F	0.267	0.262	0.250	0.240
N_{Pr}, dimensionless	0.70	0.69	0.68	0.70
μ, lb_m/hr-ft	0.083	0.078	0.056	0.048
ρ, lb_m/ft^3	0.324	1.037	0.597	0.0676

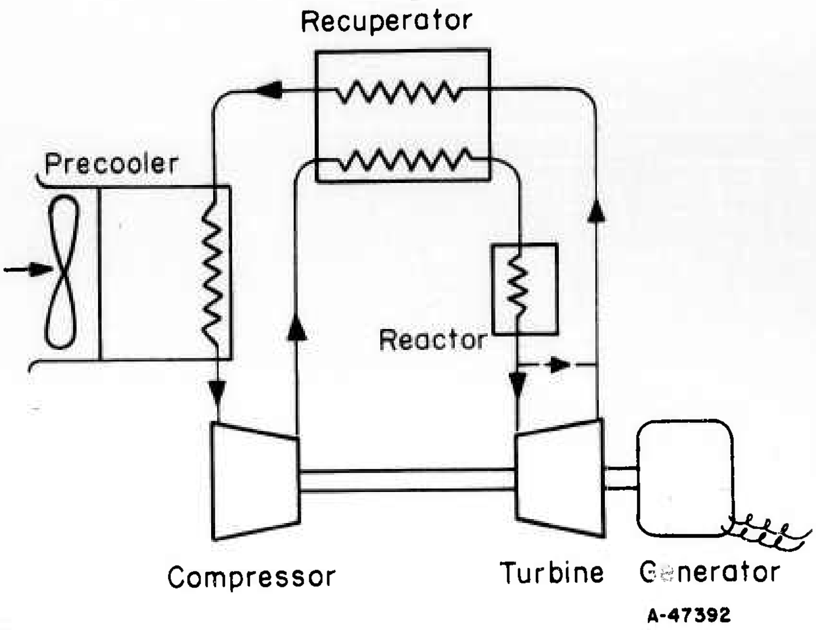

FIGURE 7. SCHEMATIC DIAGRAM OF POWER PLANT

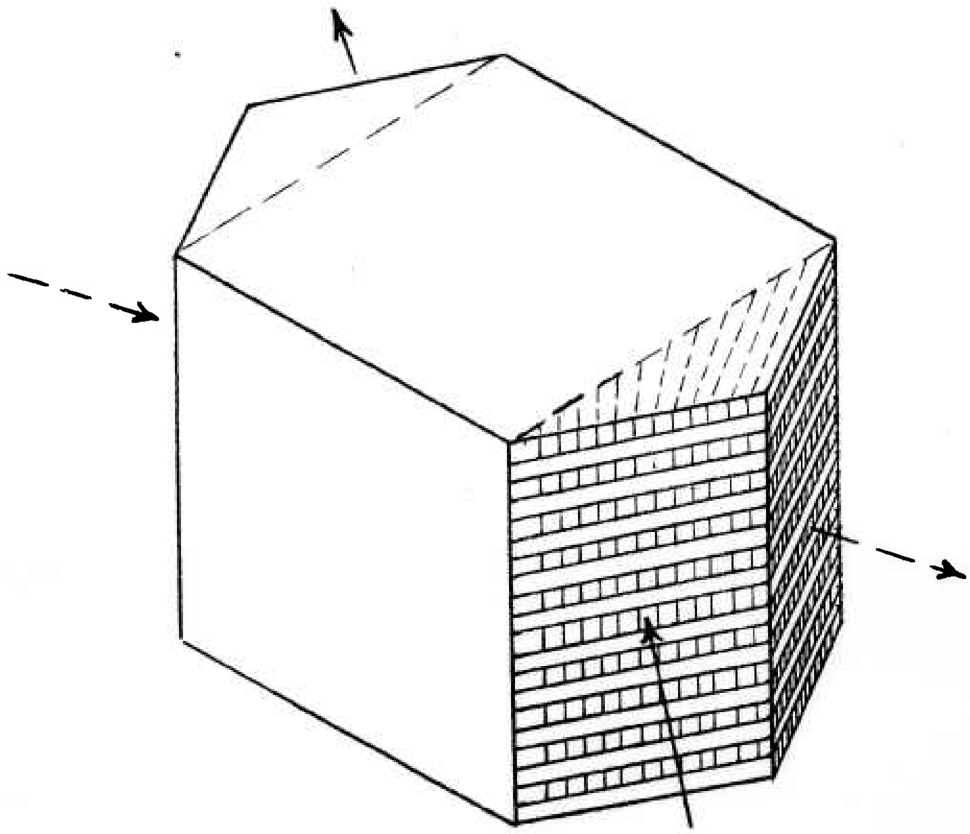

FIGURE 8. COUNTERFLOW RECUPERATOR DESIGN, CORE AND MANIFOLD CONFIGURATION

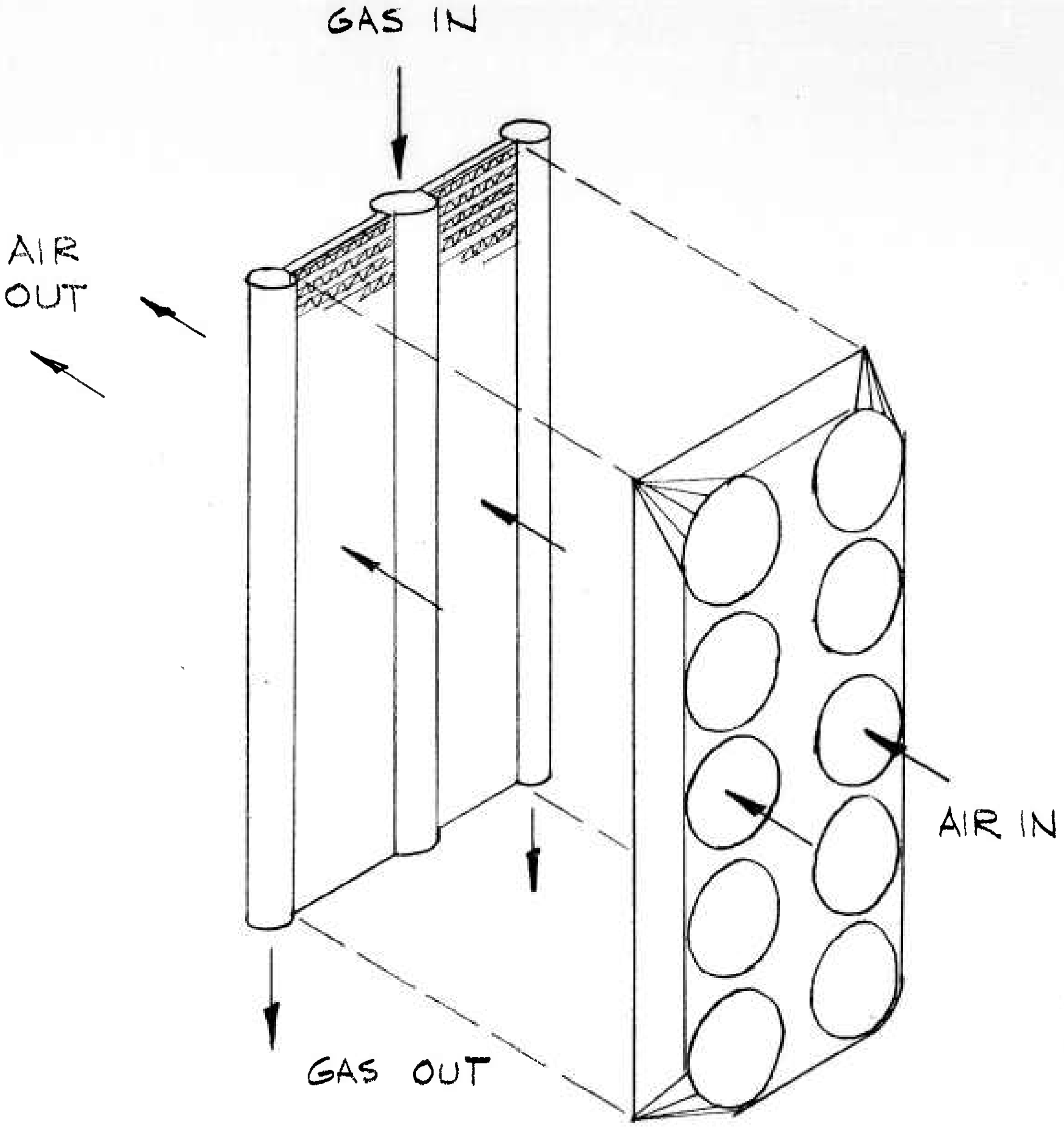

FIGURE 9. CROSSFLOW PRECOOLER DESIGN, CORE AND MANIFOLD CONFIGURATION

TABLE 7. DESIGN-STUDY RESULTS

	Recuperator			Precooler	
	1	2	3	1	
Surface					
Type of Fin	Louvered-triangular	Rectangular-strip	Plain-triangular	Rectangular-strip	
r_h, ft	0.00253	0.00122	0.00033	0.00153	
β, ft^2/ft^3	367	698	2,665	550	
Plate Spacing, in.	0.25	0.10	0.05	0.255	
Plate Thickness, in.	0.006	0.004	0.001	0.022	
Fin Thickness, in.	0.006	0.004	0.001	0.006	
Number Fins Per Inch	11.1	20	93	16	
σ, Ratio V_x/V_t	0.91	0.85	0.89	0.72	
				Hot Side	Cold Side
f, Friction factor	0.030	0.0275	0.014	0.026	0.038
j, Colburn number	0.0065	0.0071	0.0035	0.0072	0.0105
h, $Btu/hr\text{-}ft^2\text{-}F$	154	213	108	136	53.3
η_f, Fin effectiveness	0.49	0.71	0.83	0.76	0.88
η_s, Surface effectiveness	0.61	0.76	0.86	0.80	0.90
P_c, $hr^{-2/3}ft^{-1}$	0.722×10^5	2.44×10^5	6.13×10^5	2.17×10^5	2.56×10^5
Core					
A_s, ft^2	4,530	2,560	4,480	5,520	
A_x, ft^2	3.02	2.48	2.38	1.76	28.5
A_t, ft^2	3.33	2.92	2.66	2.44	39.6
L_n, ft	1.82	1.71	1.63	2.85	
L_x, ft	4.07	1.48	0.71	2.85	0.176
V_x, ft^3	12.32	3.67	1.68	5.02	5.02
V_t, ft^3	13.56	4.32	1.89	14.0	
Weight, lb	585	310	100	680	
N_{tu_h}/N_{tu_c}	1.007	0.992	0.992	10.1	
$\Delta p_h/\Delta p_c$	3.20	3.20	3.20	18.2	
N_{Re}, Hot side	8,700	5,130	1,450	6,480	--
N_{Re}, Cold side	9,260	5,470	1,540	--	2,190
HTD_t, Btu/hr-100 F-ft^3	8.61×10^5	27.0×10^5	61.7×10^5	7.62×10^5	
SHT, Btu/hr-100 F-lb	1.99×10^4	3.76×10^4	11.7×10^4	1.57×10^4	
Manifolds					
Volume, ft^3	1.75	1.44	1.25	144*	
Weight, lb	75	70	65	573*	
Pressure Vessel					
Weight, lb	1,020	325	145	Incl. with manifold	
Insulation					
Thickness, in.	2.25	2.25	2.25	None required	
Volume, ft^3	6.0	2.2	1.0	None required	
Weight, lb	120	45	20	None required	
Total Heat Exchanger					
Volume, ft^3	21.3	8.0	4.1	128*	
Weight, lb	1,800	750	330	1,253*	
HTD_o, Btu/hr-100 F-ft^3	5.48×10^5	14.6×10^5	28.4×10^5	0.83×10^5	
SHT_o, Btu/hr-100 F-lb	6.48×10^3	15.5×10^3	35.4×10^3	8.5×10^3	

*Excluding fans and motors.

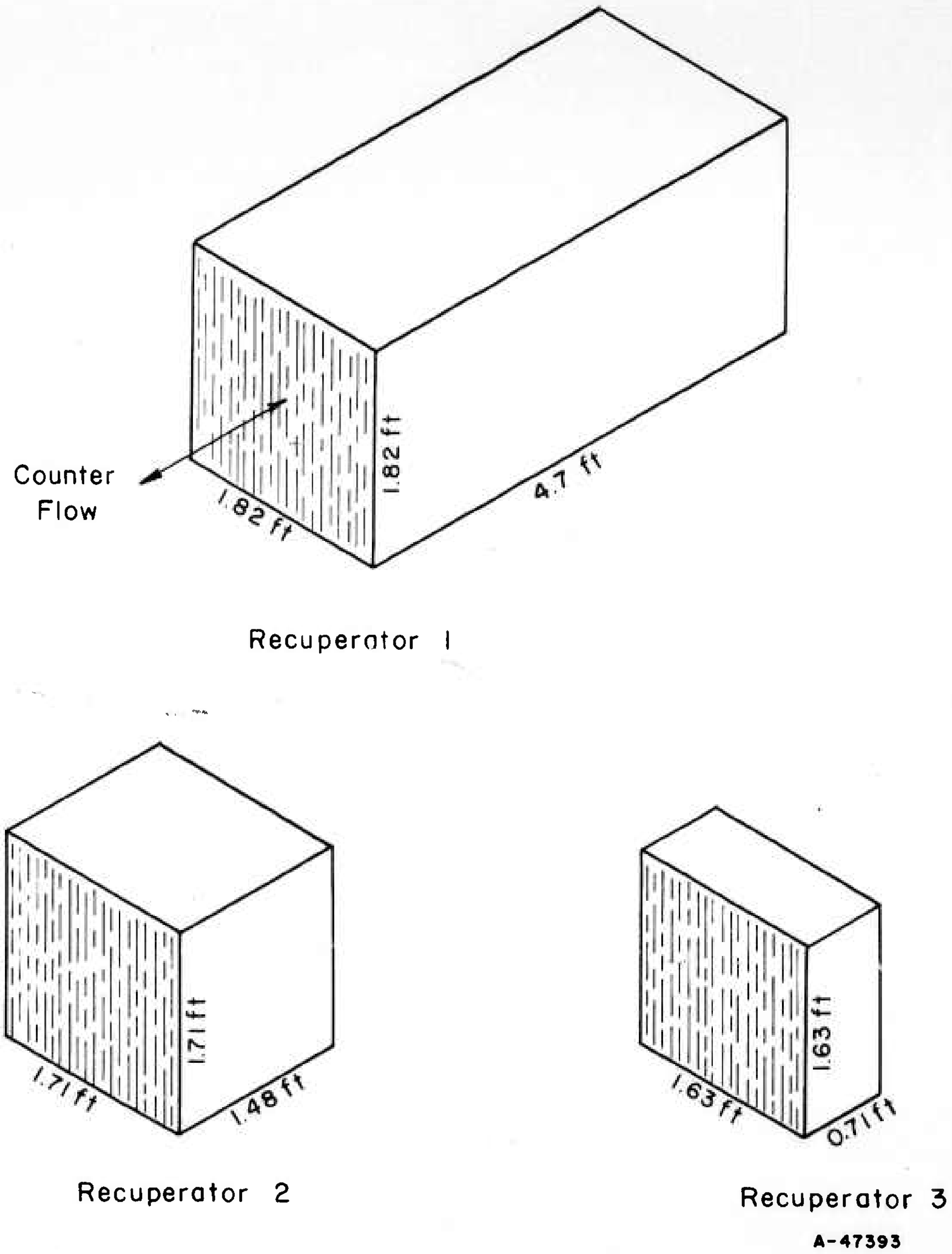

FIGURE 10. RECUPERATOR CORE SIZES

radius, but the flow length is markedly affected. However, because the flow areas change very little, the manifold volumes likewise do not change much and thus they become a larger percentage of the total volume.

Figure 11 reveals that very little advantage would be gained by trying to achieve still more compact surfaces than used in the Recuperator 3 design.

Recuperator 1 incorporated a typical heat-exchanger surface now being produced by manufacturers in general and represents about an average hydraulic radius. This design might be considered an "off-the-shelf" possibility. The surface used was designated as "1/4 – 11.1" in Reference (124).

Recuperator 2 is a design which could be realized by utilizing manufacturing capabilities, but most manufacturers are not tooled for such a compact surface configuration. This design might represent a "practical" state-of-the-art limit. This surface was designated as "20.06R-.100/.098-1/8(O)-.004(AL)" in Reference (46).

Recuperator 3 is a design considered to be an ultimate goal, representing experimental test cores of some manufacturers, but which could be realized only with extreme difficulty and extensive development of manufacturing techniques. This core design might represent an "ultimate" state-of-the-art limit, approachable in the next 5 to 10 years. It should also be noticed that in this instance, the core design approaches the practical limit, mentioned previously, of having a flow length about half that of the nonflow dimensions; thus the manifold volume is almost equal to the core volume itself. Perhaps the over-all volume of Recuperator 3, including manifolds and insulation, might have been reduced slightly if the flow area had been optimized by finding the optimum Δp and N_{tu} ratios to accomplish this, but it is not too likely. The core volume would increase because the length would increase, but the manifold volume would decrease because the flow area would decrease. However, it is not known whether or not the decrease in the manifold volume would exceed the increase in insulation and core volume, because this design was not investigated. The surface used for the Recuperator 3 design was a one-half scale version of the surface designated as "46.45T-.100/.100-2.63(P)-.002(S.S.)" in Reference (46).

CALCULATIONS

Minimum-Core-Volume and Flow-Area Equations

The objective sought in deriving expressions for minimum core volume and flow area was to optimize the pressure ratio, the N_{tu} ratio, and the hydraulic radius ratio between the two fluid sides of the core.

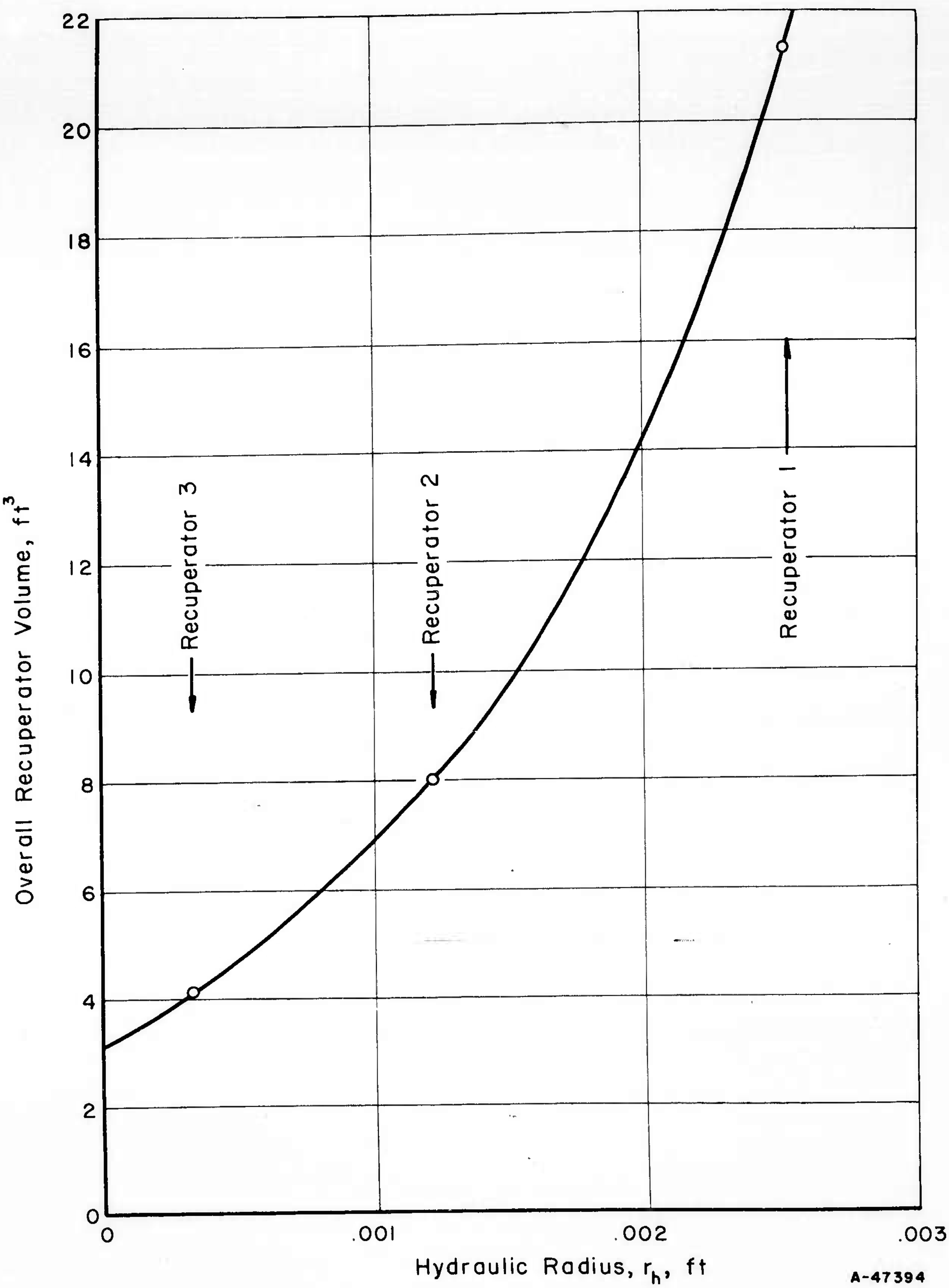

FIGURE 11. OVER-ALL RECUPERATOR VOLUME VERSUS HYDRAULIC RADIUS

Minimum Volume Requirements

The equation for the core volume on one side was derived in the appendix, Equation (A-33), and is*

$$V_v = (2/g)^{1/3}[w(N_{tu})^{4/3}(\Delta p)^{-1/3}][N_{Pr})^{8/9}(\mu\rho)^{-1/3}][(r_h)^{4/3}(\eta_s)^{-4/3}(f/j)^{1/3}(c_j)^{-1}] \quad . \quad \text{(A-33)}$$

Now, letting V_t represent the total void volume of both sides of the core, we can write

$$V_t = K_c(N_{tu_c})^{4/3}(\Delta p_c)^{-1/3}(r_{h_c})^{4/3} + K_h(N_{tu_h})^{4/3}(\Delta p_h)^{-1/3}(r_{h_h})^{4/3} \quad , \tag{2}$$

where

$$K = (2/g)^{1/3}(w)(N_{Pr})^{8/9}(\mu\rho)^{-1/3}(\eta_s)^{-4/3}(f/j)^{1/3}(c_j)^{-1} \quad , \tag{3}$$

and the subscripts c and h refer to the cold and hot sides, respectively. For a given set of heat-exchanger-design conditions, K is essentially constant. Now, letting N, P, and R represent the N_{tu} ratio, (N_{tu_h}/N_{tu_c}), the Δp ratio, $(\Delta p_h/\Delta p_c)$, and the r_h ratio, (r_{h_h}/r_{h_c}), respectively, we have an equation of the form

$$V_t = V_t(N,P,R) \quad , \tag{4}$$

in which V_t is a function of the three variables N, P, and R. Therefore, we can determine N, P, and R ratios that will yield a minimum volume by setting

$$\frac{\partial V_t}{\partial N} = 0 \tag{5}$$

$$\frac{\partial V_t}{\partial P} = 0 \tag{6}$$

$$\frac{\partial V_t}{\partial R} = 0 \tag{7}$$

and solving these equations simultaneously. However, the hydraulic radius ratio, R, is such a predominating influence that the other ratios are of relatively minor importance and probably cannot be achieved without going to a larger volume. It must also be remembered that these equations refer to the total void volume of the core only.

After differentiating Equation (2) with respect to N, P, and R and setting the resulting expressions equal to zero, as indicated in Equations (5), (6), and (7), we obtain the following equations:

$$N = N_{tu_h}/N_{tu_c} = (\Delta p_h/\Delta p_c)^{1/2}(w_c/w_h)^{6/7}(c_{p_c}/c_{p_h})^{3/7}(N_{Pr_c}/N_{Pr_h})^{8/21}(\mu_h\rho_h/\mu_c\rho_c)^{1/7}$$

$$(r_{h_c}/r_{h_h})^{4/7}(c_{j_h}/c_{j_c})^{3/7}(\eta_{s_h}/\eta_{s_c})^{4/7}[(f/j)_c/(f/j)_h]^{1/7} \tag{8}$$

*Terms are defined on a fold-out nomenclature page in the appendix to this report.

$$P = \Delta p_h/\Delta p_c = (N_{tu_h}/N_{tu_c})(w_h/w_c)^{3/4}(p_h/p_c)^{3/4}(N_{Pr_h}/N_{Pr_c})^{2/3}(\mu_c\rho_c/\mu_h\rho_h)^{1/4}$$

$$(r_{h_h}/r_{h_c})(c_{j_c}/c_{j_h})^{3/4}(\eta_{s_c}/\eta_{s_h})[(f/j)_h/(f/j)_c]^{1/4} \qquad (9)$$

$$R = r_{h_h}/r_{h_c} = 0. \qquad (10)$$

This last equation indicates that r_{h_h} should be made as small as possible to achieve minimum volume, but obviously r_{h_h} cannot equal zero. Therefore, if r_{h_c} is fixed and assumed to be the minimum practical hydraulic radius that can be used in a given heat-exchanger-core design, then we can conclude that r_{h_h} should equal r_{h_c}, or R = 1 for minimum core volume.

Solving Equations (8) and (9) simultaneously, we find that we can obtain minimum core volume when

$$N = N_{tu_h}/N_{tu_c} = (w_c/w_h)^{7/8}(p_h/p_c)^{1/8}(c_{p_c}/c_{p_h})^{1/2}(N_{Pr_c}/N_{Pr_h})^{1/3}(\mu_h\rho_h/\mu_c\rho_c)^{1/8}$$

$$(r_{h_c}/r_{h_h})^{1/2}(c_{j_h}/c_{j_c})^{3/8}(\eta_{s_h}/\eta_{s_c})^{1/2}[(f/j)_c \quad (f/j)_h]^{1/8} \qquad (11)$$

$$P = \Delta p_h/\Delta p_c = (w_c/w_h)^{1/8}(p_h/p_c)^{7/8}(c_{p_c}/c_{p_h})^{1/2}(N_{Pr_h}/N_{Pr_c})^{1/3}(\mu_c\rho_c/\mu_h\rho_h)^{1/8}$$

$$(r_{h_h}/r_{h_c})^{1/2}(c_{j_c}/c_{j_h})^{3/8}(\eta_{s_c}/\eta_{s_h})^{1/2}[(f/j)_h/(f/j)_c]^{1/8} \quad , \qquad (12)$$

and we see that

$$\Delta p_h/\Delta p_c = (N_{tu_c}/N_{tu_h})[p_h/p_c)(c_{p_c}/c_{p_h})(w_c/w_h)] \quad , \qquad (13)$$

or

$$N_{tu_h}/N_{tu_c} = (\Delta p_c/\Delta p_h)[(p_h/p_c)(c_{p_c}/c_{p_h})(w_c/w_h)] \quad . \qquad (14)$$

However, as pointed out previously, $r_{h_h} = r_{h_c}$ for minimum volume, and for this design requirement, the above equations, (11) and (12), become incompatible with the requirement of R = 1, which fixes the Δp and N_{tu} ratios. Therefore, Equations (11) and (12) are not useful, because simply making the hydraulic radius equal for both sides and as small as practical will result in the minimum core volume for that particular surface.

Minimum-Flow-Area Requirements

The equation for the required flow area on one side of the core was derived in the appendix and is

$$A_x = w[(1/2g)(N_{tu}/\Delta p)(N_{Pr}{}^{2/3}/\rho)(f/j)(1/\eta_s)]^{1/2} \quad , \tag{A-29}$$

and for the total flow area, A_t, we can write

$$A_t = Y_c\ (N_{tu_c}/\Delta p_c)^{1/2} + Y_h\ (N_{tu_h}/\Delta p_h)^{1/2} \quad , \tag{15}$$

where

$$Y = w\ [(1/2g)(N_{Pr}{}^{2/3}/\rho)(f/j)(1/\eta_s)]^{1/2} \quad . \tag{16}$$

We now have an equation of the form

$$A_t = A_t\ (N,P) \quad , \tag{17}$$

in which A_t is a function of the two variables, N and P. It will be noticed that flow area is not directly a function of the hydraulic radius.

Following the same procedure as outlined for obtaining minimum volume, we can also obtain the optimum N_{tu} and Δp ratios that will result in minimum flow area. The results only are given below.

For minimum flow area:

$$N = N_{tu_h}/N_{tu_c} = (w_c/w_h)^{5/4}(p_h/p_c)^{1/4}(c_{p_c}/c_{p_h})^{3/4}(N_{Pr_c}/N_{Pr_h})^{3/4}(\rho_h/\rho_c)^{1/4}$$

$$(\eta_{s_h}/\eta_{s_c})^{1/4}[(f/j)_c/(f/j)_h]^{1/4} \tag{18}$$

$$P = \Delta p_h/\Delta p_c = (w_h/w_c)^{1/4}(p_h/p_c)^{3/4}(c_{p_c}/c_{p_h})^{3/4}(N_{Pr_h}/N_{Pr_c})^{3/4}(\rho_c/\rho_h)^{1/4}$$

$$(\eta_{s_c}/\eta_{s_h})^{1/4}[(f/j)_h/(f/j)_c]^{1/4} \quad , \tag{19}$$

and again Equations (13) and (14) express the relationship between the ratios of N_{tu} and Δp. By comparing exponents in Equations (18) and (19) with those in (11) and (12), we see that different ratios are required to achieve either minimum volume or minimum flow area. Again it should be pointed out that the hydraulic radius should be equal on each side of the core to produce a minimum volume design, and this will cause considerably different N_{tu} and Δp ratios than are calculated from Equations (11), (12), (18), or (19).

The minimum-area equations would be used only for a situation requiring minimum frontal, or flow, area. All the minimum-volume equations are not needed, because simply by making the hydraulic radius as small as possible on both sides, the minimum core volume will result.

Estimating Core Volume and HTD With P_c Data

An estimated core volume and HTD can be obtained from the P_c data plot, Figure 3, together with Equation (A-37), which is

$$\text{HTD}\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} = P_c\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} , \qquad \text{(A-37)}$$

where

$$\text{HTD} = \frac{Q}{\Delta T_{ln} V} \qquad \text{(A-34)}$$

$$\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} = \left[(C/C_{min})(N_{tu_o}/\Delta p)^{1/3}(N_{Pr})^{8/9}(\mu\rho)^{-1/3}(c_p)^{-1}\right] \qquad \text{(A-41)}$$

$$P_c = \left[(g/2)^{1/3}(c_j)(\eta_s)^{4/3}(r_h)^{-4/3}(f/j)^{-1/3}\right] \qquad \text{(A-42)}$$

$$\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} = (N_{tu_o}/N_{tu})^{4/3} \quad . \qquad \text{(A-43)}$$

We can solve Equation (A-37) for V, which is the void volume on one side of the heat-exchanger core, and we obtain

$$V = (Q/\Delta T_{ln})\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} \Big/ P_c \begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} \quad . \qquad (20)$$

If we assume both sides of a counterflow heat-exchanger core have equal hydraulic-radius surfaces, and equal areas, then

$$V_t = 2V \quad . \qquad (21)$$

The procedure for estimating core volume and HTD for a given heat-exchanger application is outlined below:

(1) Calculate the correction factor for one side of the heat-exchanger core using the design conditions specified.

(2) Choose a particular surface having the desired hydraulic radius and find from Figure 3 the average P_c expected. The P_c could be calculated if the j, f, and η_s are known.

(3) Estimate the design parameter. This value will equal 1/2 for a balanced (C_R = 1) heat exchanger, with equal flow areas on both sides.

(4) Solve Equation (20) above for the core volume on one side and double this to obtain the over-all void volume of the core. Then Equation (A-34) can be used to calculate the total HTD for both sides by substituting in this total volume, or Equation (A-37) can be solved for HTD.

An example problem is given below to illustrate the technique and compare the results with the detailed recuperator design given previously. The design conditions listed in Table 5 and the fluid properties given in Table 6 are used.

Step 1. Use the cold-side parameters to calculate the correction factor of Equation (A-41):

$$C_c = (w\,c_p)_c = (108{,}000\ \mathrm{lb/hr})(0.262\ \mathrm{Btu/lb\ F}) = 28{,}300\ \mathrm{Btu/hr\ F}$$

$$C_c = C_{min} \quad ,$$

so

$$C/C_{min} = 1 \quad .$$

$$N_{tu_o} = Q/\Delta T_{ln}\, C_{min} = (17.5 \times 10^6\ \mathrm{Btu/hr}/150\ \mathrm{F})\ (28{,}300\ \mathrm{Btu/hr\ F})$$

$$N_{tu_o} = 4.12 \quad .$$

For the assumptions of $A_{x_c} = A_{x_h}$, $L_c = L_h$, $r_{h_c} = r_{h_h}$, and $f_c = f_h$, Equation (A-25) gives the ratio

$$\Delta p_h/\Delta p_c = \rho_c/\rho_h \quad . \qquad (22)$$

We will now let

$$\xi = (\Delta p_h/\Delta p_c)(p_c/p_h) = (\Delta p/p)_h/(\Delta p/p)_c \quad . \qquad (23)$$

Combining (23) with the specification

$$(\Delta p/p)_h + (\Delta p/p)_c = 0.04 \quad , \qquad (24)$$

we obtain

$$\Delta p_c = p_c\ 0.04/(\xi + 1) \quad . \qquad (25)$$

Combining (22) and (23), we have

$$\xi = \rho_c P_c/\rho_h P_h$$

$$\xi = (1.037\ \mathrm{lb/ft^3})(500\ \mathrm{psia})/(0.324\ \mathrm{lb/ft^3})(175\ \mathrm{psia})$$

$$\xi = 9.14 \quad .$$

So, using Equation (25),

$$\Delta p_c = (500\ \mathrm{psia})(144)(0.04)/10.14$$

$$\Delta p_c = 284\ \mathrm{lb/ft^2} \quad .$$

The average fluid property values were either obtained from handbooks or calculated, and are

$$N_{Pr_c} = 0.69$$

$$\mu_c = 0.078 \text{ lb/hr-ft}$$

$$\rho_c = 1.037 \text{ lb/ft}^3$$

$$c_{p_c} = 0.262 \text{ Btu/lb F} \quad .$$

Inserting all these values into Equation (A-41) gives

$$\begin{bmatrix} \text{Correction} \\ \text{Factor} \end{bmatrix} = 1.55 \text{ hr}^{1/3} \text{ F ft}^2/\text{Btu} \quad . \tag{26}$$

Step 2. The same surface used for the design of Recuperator 2 will be assumed for both sides of the core. From the graph of P_c versus r_h, Figure 3, this surface had a

$$P_c = 3.65 \times 10^5/\text{hr}^{2/3}\text{ft} \quad .$$

The design studies discussed previously had a calculated P_c value for the Recuperator 2 design of $2.44 \times 10^5/\text{hr}^{2/3}$ft. The difference between the graph and the calculated value arises because the actual η_s was 0.76 and the Reynolds number was around 5,000 for each side, whereas the P_c data in Figure 3 is based on a η_s = 0.95 and usually a Reynolds number near 1,000. Therefore, we must correct any P_c data taken from the graph, especially to account for expected differences in η_s. It will be realized that estimated volumes will deviate from actual design values to some extent, depending on how much the shape of the j and f curves changes over the range of Reynolds numbers, how accurately the η_s is predicted, etc. If we correct this P_c of 3.65×10^5 to account for a surface effectiveness of 0.76 instead of 0.95, we will have

$$P_c = 3.65 \times 10^5 \, (0.76/0.95)^{4/3} = 2.61 \times 10^5/\text{hr}^{2/3} \text{ ft} \quad . \tag{27}$$

This value now agrees much more closely with the value found in the actual design. It is realized that the accuracy of this estimating method is very dependent on assuming η_s accurately.

Step 3. The design parameter will be taken to be

$$(N_{tu_o}/N_{tu_c})^{4/3} = (1/2)^{4/3} = 0.396 \quad , \tag{28}$$

because the recuperator is very nearly a balanced heat exchanger, having a $C_R = 0.98$.

Step 4. The previously determined values in Equations (26), (27), and (28) will now be substituted into Equation (20) to obtain a predicted core volume for the APCDB specifications using a rectangular strip-fin surface having a hydraulic radius of only 0.00122 ft:

$$V = (17.5 \times 10^6 \text{ Btu/hr/150 F})(1.55 \text{ hr}^{1/3} \text{ F ft}^2/\text{Btu})/(2.61 \times 10^5/\text{hr}^{2/3} \text{ ft})(0.396)$$

$$V = 1.75 \text{ ft}^3 \quad . \tag{29}$$

The total estimated void volume of the core will be twice this amount, or

$$V_{t_{est.}} = 3.50 \text{ ft}^3 \quad . \tag{30}$$

This value compares very favorably with the actual core volume in the design study of Recuperator 2 (Table 6), which had a total void core volume of

$$V_{t_{act.}} = 3.67 \text{ ft}^3 \quad . \tag{31}$$

Equation (A-34) can be used to estimate the HTD_t for the core, using the total void volume found in Equation (30):

$$HTD_t = (17.5 \times 10^6 \text{ Btu/hr})/(150 \text{ F})(3.50 \text{ ft}^3)$$

$$HTD_{t_{est.}} = 3.34 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \quad . \tag{32}$$

This value compares favorably with the actual design value of

$$HTD_{t_{act.}} = 3.18 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \quad . \tag{33}$$

The above example problem illustrates the usefulness of the P_c data-presentation method. Once the correction factor has been calculated for a particular application, estimated core volumes can be obtained very quickly for a wide range of heat-transfer surfaces by inserting various P_c values; thus the relative advantages of using more compact surfaces can be ascertained. Also, this method yields answers much more quickly than do actual design calculations, which are described below step by step.

Design of Three Recuperators and One Precooler

The results of the design studies of three recuperators and one precooler were presented in a previous section, where Tables 5 and 6 outlined the design conditions and fluid properties used in these calculations and Figures 8, 9, and 10 showed the final heat-exchanger configurations. The calculation methods will be presented in this section.

Recuperators

The assumptions used in designing the recuperators were:

(1) Counterflow arrangement

(2) Heat transferred only in the core section

(3) Single-sandwich construction, having the same hydraulic radius on both sides

(4) Material conductivity, k = 17 Btu/hr ft F, and material density, d = 480 lb/ft^3

(5) Hot and cold fluid-flow lengths equal because of Assumption (1)

(6) Hot and cold fluid-flow areas equal

(7) Frontal area square in shape

(8) Manifold configuration as shown in Figure 8 with a 30-degree angle

(9) Manifold weight estimated using the core density

(10) Pressure vessel calculated as a cylinder enclosing the core envelope only and withstanding 500 psia with a tensile stress of 10,000 psi, but the volume of the pressure vessel considered negligible because it was assumed to be included in the insulation thickness

(11) Average insulation thickness calculated to reduce the average core-surface temperature of 875 to 150 F with a thermal conductivity, k = 0.018 Btu/hr ft F

(12) Insulation weight estimated on the basis of a density of 20 lb/ft^3.

When Equation (A-29) is combined with Assumption (6), an expression is obtained for N_{tu} and Δp ratios, and with Equation (22) written in the form,

$$\Delta p_h/\Delta p_c = f_h \rho_c / f_c \rho_h \quad , \tag{34}$$

and substituted in for the Δp ratio, we arrive at the following expression:

$$N_{tu_h}/N_{tu_c} = (N_{Pr_c}/N_{Pr_h})^{2/3} (j_h/j_c)(\eta_{s_h}/\eta_{s_c}) \quad . \tag{35}$$

Equation (A-8) can be rearranged to give

$$N_{tu_c} = N_{tu_o} \left[C_{min}/C_h \, (N_{tu_h}/N_{tu_c}) + C_{min}/C_c \right] \quad . \tag{36}$$

A similar expression for Δp_h, as Equation (25) gives for Δp_c, can be derived and is

$$\Delta p_h = p_h \, \xi (0.04)/(\xi + 1) \quad . \tag{37}$$

The procedure for designing a recuperator core which satisfies the above assumptions is enumerated below:

<u>Step 1.</u> Choose a heat-transfer surface having the desired hydraulic radius and use this same configuration for both sides.

<u>Step 2.</u> Estimate η_s and N_{Re}.

<u>Step 3.</u> Calculate f/j.

<u>Step 4.</u> Calculate N_{tu_h}/N_{tu_c} ratio, using Equation (35).

<u>Step 5.</u> Calculate N_{tu_c} from Equation (36) and N_{tu_h} from the ratio of N_{tu_h}/N_{tu_c}.

<u>Step 6.</u> Calculate $\Delta p_h/\Delta p_c$ ratio, using Equation (34).

<u>Step 7.</u> Calculate Δp_c from Equation (25) and Δp_h with Equation (37).

<u>Step 8.</u> Calculate flow area for one side, A_x, from Equation (A-29). [Flow area is the same for both sides, from Assumption (5).]

<u>Step 9.</u> Calculate flow length, L_x, using either Equation (A-25) or (A-31).

<u>Step 10.</u> Check N_{Re} assumption, using

$$N_{Re} = 4\, r_h\, G/\mu = 4\, r_h\, w/A_x\, \mu \quad . \tag{38}$$

<u>Step 11.</u> Calculate heat-transfer coefficient, h, using the definition

$j \equiv N_{St}\, N_{Pr}{}^{2/3}$, giving

$$h = j\, G\, c_p/N_{Pr}{}^{2/3} \quad . \tag{39}$$

<u>Step 12.</u> Calculate the fin effectiveness, η_f, and surface effectiveness, η_s, using the formulas given in Reference (124):

$$\eta_f = (\tanh mL)/mL \quad , \tag{40}$$

where

$$m = (2\, h/k\, \delta)^{1/2} \quad , \tag{41}$$

and

$$\eta_s = 1 - (A_f/A_s)(1 - \eta_f) \quad . \tag{42}$$

<u>Step 13.</u> Repeat Steps (3) through (12), using these new values, as many times as necessary to reach satisfactory agreement.

<u>Step 14.</u> Calculate the total void volume of core and find the over-all core volume by

$$V_t = A_x L_x \tag{43}$$

$$V_o = V_t/\sigma \quad . \tag{44}$$

<u>Step 15.</u> Calculate core weight by using material volume, $V_o\,(1 - \sigma)$, and material density, d, which give

$$Wt_{core} = V_o\,(1 - \sigma)\, d \quad , \tag{45}$$

<u>Step 16.</u> Calculate the manifold volume, using Assumptions (7) and (8), which leads to

$$V_m = L_n{}^3(0.1442) \quad , \tag{46}$$

where

$$L_n = \text{nonflow lengths} \quad ,$$

and both these lengths were assumed equal in Assumption (7).

Step 17. Calculate manifold weight, using assumption (9), so

$$Wt_m = (V_m/V_o)\ Wt_{core} \quad . \qquad (47)$$

Step 18. Calculate pressure-vessel weight, using Assumption (10), which yields

$$Wt_{\substack{pressure\\vessel}} = 0.157\ L_n^{\ 2} L_x\ d \quad . \qquad (48)$$

Step 19. Calculate insulation thickness, using Assumption (11) and assuming an ambient air temperature of 80 F, which produces an average heat-transfer coefficient of 1.0 Btu/hr ft^2 F, using the equation [Reference (105)],

$$h = 0.3\ (\Delta T)^{0.25} \quad . \qquad (49)$$

Step 20. Calculate the outside surface area of the heat exchanger, excluding only the flow passages, from

$$A_i = A_{core} + A_m$$

$$A_i = 4\ L_n L_x + 0.577\ L_n^{\ 2} \quad . \qquad (50)$$

Step 21. Calculate volume and weight of the insulation, using Assumption (12).

Step 22. Calculate over-all HTD and SHT from Equations (A-35) and (1), respectively, for comparison with other APCDB Recuperator designs using the over-all sum of volumes and weights, respectively.

The above steps are straightforward; since Table 7 gives the results calculated for each successive step of the three recuperator designs, the calculations are not included in this report. Usually about three trials were needed to reach satisfactory agreement of Steps (3) through (12). It will also be noticed that of the 4 per cent total pressure drop allowed, 3.61 per cent is on the hot-gas, or low-pressure, side and only 0.39 per cent occurs on the cold-gas, or high-pressure, side when equal hydraulic-radius surfaces are used on each side.

As pointed out earlier, these design calculations produced a core for Recuperator 2 having a total void volume of 3.67 ft^3 (Table 6), whereas P_c data and a calculation correction factor for the given fluid and design conditions gave an estimated core void volume of 3.50 ft^3 when the P_c was corrected to the proper η_s.

Precooler

Initially, a counterflow configuration was also tried for a precooler core, but the dimensions of the core became absurd, being something like 9 ft square in the nonflow dimensions and having only a 3-inch flow length. A four-pass cross-counterflow design also produced a very odd configuration. Therefore, a pure crossflow design appeared to give the most realistic shape.

The assumptions used for designing a precooler were:

(1) Crossflow arrangement, fluids unmixed

(2) Double-sandwich construction, having the same hydraulic radius on both sides

(3) Aluminum materials, with k = 95 Btu/hr-ft-F

(4) Pressure drop on nitrogen side of 1.5 per cent

(5) Pressure drop on air side of 4 in. of water

(6) Capacity-rate ratio, $C_R = 0.25$

(7) Area ratio, $A_{x_h}/A_{x_c} = L_{x_c}/L_{x_h}$

(8) Manifold configuration as shown in Figure 9

(9) Manifold headers designed for gas velocities less than 100 ft/sec and as pressure vessels, using a maximum tensile stress of 5,000 psi at 100 F or 1,000 psi at 500 F

(10) No insulation required for ambient air-cooled heat exchangers, as mentioned in the APCDB requirements.

When Equation (A-29) is combined with Assumption (7), the following expression is obtained for the N_{tu} ratio in terms of the Δp ratio:

$$(N_{tu_h}/N_{tu_c})^{4/3}[(f/j)_h/(f/j)_c]^{1/3}(\eta_{s_c}/\eta_{s_h})^{2/3}(c_{j_c}/c_{j_h}) = 9.32\ (\Delta p_h/\Delta p_c)^{1/3} \quad . \tag{51}$$

The Δp ratio can be calculated from Assumptions (4) and (5) to give the N_{tu} ratio, and then the individual N_{tu} of each side can be calculated using Equation (36). These values are then substituted into Equation (A-29) to give the flow area for each side and into either Equation (A-25) or (A-31) to give the flow length required for each side. The procedure to be followed to arrive at a precooler core design is again an iterative process and follows very closely the procedure outlined above in the section on recuperators, except for substituting the new equations and assumptions just given for the crossflow design. The values calculated during the precooler design are listed in Table 7, so detailed calculations will not be outlined. The particular manifold design chosen requires ten 10-hp axial fans to move 432,000 lb of air/hr through the core with a pressure drop of 4 in. of water.

A comparison was also made between HTD values calculated using the P_c data method (Equation A-37) and the design study results for this precooler design. The correction factors, Equation (A-41), were 1.93 and 47.0 $hr^{1/3}\ ft^2$ F/Btu for the hot and cold sides, respectively, and the design parameters, Equation (A-43), were 0.17 and 4.19 for the hot and cold sides, respectively. The P_c calculated for the surfaces, assuming both $\eta_s = 0.95$, was $2.68 \times 10^5/hr^{2/3}$ ft. From Equation (A-37), the following equation can be written:

$$HTD = \begin{bmatrix}\text{Compactness}\\ \text{Parameter}\end{bmatrix}\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix}\Big/\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} \quad . \tag{52}$$

The HTD for each side can now be calculated:

$$HTD_h = 2.36 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \tag{53}$$

$$HTD_c = 2.39 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \quad . \tag{54}$$

The HTD_t for the total core void volume can be calculated from the equation

$$HTD_t = Q/\Delta T_{ln} (V_c + V_H) \quad , \tag{55}$$

which can be written as

$$1/HTD_t = (1/HTD_h) + (1/HTD_c) \quad . \tag{56}$$

Equations (53) and (54) then combine to give

$$HTD_t = 1.19 \times 10^6 \text{ Btu/hr 100 F ft}^3 \quad . \tag{57}$$

This value is an optimistic estimate because $\eta_s = 0.95$ was assumed. However, had the correct values of $\eta_{s_h} = 0.80$ and $\eta_{s_c} = 0.90$ been assumed by prior knowledge, a more realistic estimate would have been obtained. Equation (52) can be corrected in the following manner:

$$1/HTD_t = [1/HTD_h(\eta_{s_h}/0.95)^{4/3}] + [1/HTD_c(\eta_{s_c}/0.95)^{4/3}] \quad . \tag{58}$$

This would then give the following estimate:

$$HTD_{t_{est.}} = 1.02 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \quad , \tag{59}$$

which compares to an actual design value, based on the total void volume of the core, of

$$HTD_{t_{act.}} = 1.07 \times 10^6 \text{ Btu/hr-100 F-ft}^3 \quad . \tag{60}$$

The final precooler design was shown in Figure 9 with the dimensions and other parameters enumerated in Table 7.

This precooler design represents a present state-of-the-art capability; using a more compact surface would not change the flow areas to any extent and would only make the core thinner, but it is already about as thin as is practical.

Calculations and technical results on this program are recorded in Battelle Laboratory Record Books Numbers 20507, 20525, 20947, 20969, and 21134.

DISCUSSION

The initial investigation was planned to include some additional theoretical work regarding other compact-heat-exchanger problems, such as fluids with extreme Prandtl numbers, change of phase, effects of material and fluid conductivity, and effects of radiation, noise, leakage, thermal shock, etc. However, because the number of reports to be reviewed increased beyond preliminary estimates and because it was agreed with the sponsoring agency that theoretical studies were to be minimized, only analytical work on the compactness-parameter derivation and on the optimization study to minimize core volume and flow areas was pursued to completion. This work was believed to be essential to make the data meaningful and more generally applicable as well as to aid in understanding the way in which the several parameters affect compactness. The derivation of the compactness parameter is presented in the appendix. The optimization equations are given in the Calculations section.

Other phases of this study, which were begun but not completed, led to the following conclusions:

(1) The compactness-parameter data are useful only for fluids having a Prandtl number in the range of approximately 0.7 to 10.

(2) Heat conducted through the material longitudinally, in the flow direction, will not noticeably decrease the effectiveness of the Recuperator 3 design given herein, even with a spacer-plate thickness as large as 0.12 in.

(3) The net effect of fluid conductivity is to increase the total heat exchange [Reference (183)].

(4) Fatigue failures, due to acoustically induced vibrations, can be a serious problem but can be reduced by proper design of flow passages, fin spacings, divider plates, ductwork, etc.

CONCLUSIONS

The following conclusions resulted from this study of compact heat exchangers:

(1) The hydraulic radius of the heat-transfer surface is the most important factor affecting the core volume; the smaller the hydraulic radius, the more compact will be the core.

(2) Plate spacings of about 0.10 in. and hydraulic radii around 0.001 ft appear to be minimum practical values at the present time.

(3) Surfaces approaching a hydraulic radius of 0.00025 ft have been built under experimental laboratory conditions, but at present these could be incorporated into a complete recuperator design only with extreme difficulty.

(4) The main limitations in manufacturing surfaces more compact than those in production today are cost of manufacturing and materials, available machinery, brazing techniques, manifolding difficulty, dimensional tolerance factors, and the related problems of fouling and cleaning.

(5) The requirement of 10,000-hr life will necessitate a development program, but this objective should be attainable.

(6) The main durability problems are stresses induced by thermal cycling and pressure fluctuations, high temperatures in recuperator, different thermal-response rates of core and manifolds, high-temperature brazing-alloy technology, and perhaps acoustically induced vibrations.

(7) In general, for any given hydraulic radius surface, the more highly turbulated the fin design, the higher will be the compactness parameter within the range shown in Figure 3.

(8) The compactness parameter is accurate only for fluids in the range of normal Prandtl numbers – about 0.7 to 10.

(9) Relatively few reports were found on heat exchangers using liquid metals or oils, or of the evaporative or condensing type. Most of the literature reported tests using air to steam, with only the heat-transfer surface on the air side analyzed.

(10) None of the papers reviewed contained heat-exchanger designs of sufficient detail to permit calculation of HTD and SHT as originally defined by APCDB, which required that the over-all volume and weight include manifolds, pressure vessel, insulation, supporting structures, etc. Practically all of the studies reported in the literature have been conducted only on the cores themselves and included only such manifolding necessary for the laboratory setup, which of necessity was bulky to contain mixing chambers and well-designed diverging sections to produce uniform flow distribution. Therefore, all the data reported are based on the core surfaces only, except for the design studies, which estimated volumes and weights for manifolds, pressure vessel, insulation, etc.

(11) Data correlation was much more accurate when fluid properties and operating conditions were accounted for by using the compactness-parameter and corrected heat-transfer density terms derived for this study.

RECOMMENDATIONS

The state-of-the-art limits for compact heat-transfer surfaces are shown in Figure 3 in terms of hydraulic radius and compactness parameter for three categories: typical current production, current practical feasibility, and ultimate design. These limits, summarized below, should serve as a guide to the degree of core-surface compactness attainable in the next few years.

Typical Production Surfaces

$r_h \geq 0.002$ ft

$6.0 \times 10^4 \leq P_c \leq 3.8 \times 10^5 \text{ hr}^{-2/3} \text{ ft}^{-1}$.

Current Practical Limit

$r_h \geq 0.001$ ft

$1.5 \times 10^5 \leq P_c \leq 9.4 \times 10^5 \text{ hr}^{-2/3} \text{ ft}^{-1}$.

Projected Ultimate Design

$r_h \geq 0.00025$ ft

$9.6 \times 10^5 \leq P_c \leq 6.0 \times 10^6 \text{ hr}^{-2/3} \text{ ft}^{-1}$.

The design study of the three recuperators presented in this report was based on the above three state-of-the-art limits. The study led to predicted core sizes, over-all volumes and weights, and the over-all heat-transfer density and specific heat transfer expected for heat exchangers approaching the design limits. These designs, together with the literature and manufacturer surveys, led to recommended design goals which could be met or exceeded for the first step of a compact-heat-exchanger development for the Army Gas-Cooled Reactor Systems Program. The goals are listed below in the APCDB definitions of HTD and SHT and are based on the Recuperator 2 design-study results given previously in Table 7.

Recuperator

$HTD_o \geq 1.5 \times 10^6$ Btu/hr-100 F-ft^3

$SHT_o \geq 1.6 \times 10^4$ Btu/hr-100 F-lb.

Precooler

$HTD_o \geq 8.3 \times 10^4$ Btu/hr-100 F-ft^3

$SHT_o \geq 8.5 \times 10^3$ Btu/hr-100 F-lb.

Values for the precooler do not include the size and weight of fans, fan housings, or motors, which would depend on the voltage and frequency of the electric power.

It is also recommended that a system optimization study for the entire mobile, closed-cycle, gas-turbine facility be conducted, using the data and relationships for required heat-exchanger volume presented in this report.

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APPENDIX

DERIVATION OF COMPACTNESS PARAMETER

APPENDIX

DERIVATION OF COMPACTNESS PARAMETER

The parameter describing the number of heat-transfer units, N_{tu}, is convenient to use in heat-exchanger design because, for most well-defined flow arrangements (i.e., counterflow, crossflow, etc.), equations have been derived (Reference 124, Chapter II, and Appendix III) relating effectiveness (ϵ) to N_{tu} and capacity rate ratio (C_R), or

$$\epsilon = \epsilon \; (N_{tu_o}, C_R) \quad . \tag{A-1}$$

Capacity-Rate Ratio

The capacity rate is defined as

$$C = w \; c_p \quad . \tag{A-2}$$

This capacity rate may not be the same on both sides of the heat-exchanger core and, therefore, there may be a C_{min} and a C_{max}, so the capacity-rate ratio is taken to be

$$C_R = C_{min}/C_{max} \quad . \tag{A-3}$$

Effectiveness

The fundamental definition of effectiveness is

$$\epsilon = \text{actual heat-transfer rate/max heat-transfer rate theoretically possible} \quad . \tag{A-4}$$

The actual heat transferred is

$$Q = C_h \; (T_{h_1} - T_{h_2}) = C_c \; (T_{c_1} - T_{c_2}) \quad , \tag{A-5}$$

and the maximum possible rate is

$$Q_{max} = C_{min} \; (T_{h_1} - T_{c_1}) \quad . \tag{A-6}$$

Substituting (A-5) and (A-6) into (A-4) gives

$$\begin{aligned} \epsilon = Q/C_{min} \; (T_{h_1} - T_{c_1}) &= C_h \; (T_{h_1} - T_{h_2})/C_{min} \; (T_{h_1} - T_{c_1}) \\ &= C_c \; (T_{c_1} - T_{c_2})/C_{min} \; (T_{h_1} - T_{c_1}) \quad . \end{aligned} \tag{A-7}$$

It may be observed that either C_h/C_{min} or C_c/C_{min} will be unity, or if $C_R = 1$, both will equal unity.

N_{tu} and ΔT_{ln} Relationships

The over-all N_{tu} parameter, N_{tu_o}, is related to the individual N_{tu} parameters for each side of the core by

$$1/N_{tu_o} = 1/N_{tu_h} (C_h/C_{min}) + 1/N_{tu_c} (C_c/C_{min}) \quad , \tag{A-8}$$

where

$$N_{tu_h} = (\eta_s\ h\ A_s)_h/C_h \quad , \tag{A-9}$$

$$N_{tu_c} = (\eta_s\ h\ A_s)_c/C_c \quad . \tag{A-10}$$

The over-all heat-transfer coefficient can be defined by

$$N_{tu_o} = (UA)_{avg}/C_{min} \quad . \tag{A-11}$$

Then, since

$$Q = (UA)_{avg}\ \Delta T_{ln} \quad , \tag{A-12}$$

$$N_{tu_o} = Q/\Delta T_{ln}\ C_{min} \quad . \tag{A-13}$$

Equation (A-13) illustrates the relationship between the N_{tu} approach and the log-mean-temperature-difference approach to heat-exchanger design.

Laminar Flow

For fully established laminar flow in a constant area duct of simple geometry (i. e., a circular tube), theory [Reference (68)] predicts

$$N_{Nu} = \text{Constant} \tag{A-14}$$

and

$$f = \text{Constant}\,(N_{Re})^{-1} \quad . \tag{A-15}$$

Using the relationship

$$N_{St} = N_{Nu}/N_{Re}\ N_{Pr} \quad , \tag{A-16}$$

Equation (A-14) may be transformed to

$$N_{St}\ N_{Pr} = \text{Constant}\ (N_{Re})^{-1} \quad . \tag{A-17}$$

From (A-15) and (A-17) it would be expected that

$$N_{St}\ N_{Pr}/f = \text{Constant} \quad . \tag{A-18}$$

Turbulent Flow

For turbulent flow in circular tubes, experimental data [Reference (102)] can be correlated by

$$N_{Nu} \cong \text{Constant } (N_{Re})^{0.8} (N_{Pr})^{1/3} \quad \text{(A-19)}$$

for N_{Pr} between 0.7 and 120. Again, using Equation (A-16), Equation (A-19) becomes

$$N_{St} N_{Pr}^{2/3} \cong \text{Constant } (N_{Re})^{-0.2} \quad . \quad \text{(A-20)}$$

Experiment also shows that friction factors for smooth pipes in turbulent flow follow the relationship

$$f \cong \text{Constant } (N_{Re})^{-0.2} \quad . \quad \text{(A-21)}$$

Therefore, in turbulent flow, for circular tubes

$$N_{St} N_{Pr}^{2/3} f \cong \text{Constant} \quad . \quad \text{(A-22)}$$

General Equation

For fluids having a Prandtl number near unity

$$N_{St} N_{Pr}^{2/3} \cong N_{St} N_{Pr} \quad , \quad \text{(A-23)}$$

and comparing Equations (A-17) and (A-20) resulted in the correlation

$$j \equiv N_{St} N_{Pr}^{2/3} \cong c_j (N_{Re})^{-m} \quad , \quad \text{(A-24)}$$

which approximates the conditions for both laminar- and turbulent-flow regimes and can be used with reasonable accuracy for the majority of heat-transfer surfaces. However, for flow in smooth ducts, a different "m" may have to be used for the separate regimes of laminar and turbulent flow if a high degree of accuracy is desired. "j" is the Colburn number and "m" is the slope of the line for j vs. N_{Re}. For heat-transfer surfaces such as tube banks, louvered fins, etc., that produce highly separated flow, it is usually possible to use a single value of m to represent approximately all the surfaces in the range of normally encountered Reynolds numbers, and this value was found usually to be between 0.3 and 0.4. Notice that the one limiting assumption necessary to use Equation (A-24) is that the Prandtl number should be near unity, as is the case for fluids such as air and water. The friction factor, f, and the Colburn factor, j, curves are usually nearly parallel to log-log coordinates; and since two curves separated by a constant distance on log-log coordinates have a constant ratio, the ratio f/j is nearly constant for most heat-exchanger surfaces.

Required Core Volume

The usual heat-exchanger-design situation is that of having specified fluid-capacity rates, initial temperatures, a desired effectiveness, and allowable fluid-pressure drops. With the desired effectiveness and the capacity-rate ratio, C_R, known, the required over-all N_{tu} can be calculated directly by the $\epsilon = \epsilon(N_{tu_o}, C_R)$ relationships. For a balanced heat exchanger (i.e., $C_R = 1$), the required N_{tu} on each side will be $2\ N_{tu_o}$; however, if one film resistance is much lower than the other, the N_{tu} of the limiting side may approach N_{tu_o}. Since the design process is a trial-and-error procedure, the necessary N_{tu} on each side to produce the required over-all N_{tu} will become apparent.

Frontal Area. The required flow area on each fluid side can be calculated directly from the following equations.

Pressure drop is calculated by

$$\Delta p = f\ G^2\ L/2g\ \rho\ r_h \quad , \tag{A-25}$$

and by using the two equations

$$N_{tu} = \eta_s\ N_{St}\ L/r_h \quad , \tag{A-26}$$

$$j \equiv N_{St}\ N_{Pr}^{\ 2/3} \quad , \tag{A-27}$$

we arrive at

$$G = w\ A_x = \left[2g\ (\Delta p/N_{tu})(\rho/N_{Pr}^{\ 2/3})(j/f)\ \eta_s\right]^{1/2} \quad , \tag{A-28}$$

and solving for flow area, we have

$$A_x = w\left[(1/2g)(N_{tu}/\Delta p)(N_{Pr}^{\ 2/3}/\rho)(f/j)(1/\eta_s)\right]^{1/2} \quad . \tag{A-29}$$

This determines the required flow area on one side of the core for a given weight-flow rate in terms of the performance requirements N_{tu} and Δp, the fluid properties N_{Pr} and ρ, and the surface properties f/j and η_s. Note that the required flow area is not directly dependent upon the hydraulic radius, although f/j and η_s will be somewhat affected by hydraulic radius.

Flow Length. The required flow length may be calculated by combining Equations (A-24), (A-26), and (A-28), together with

$$N_{Re} \equiv 4r_h\ G/\mu \quad , \tag{A-30}$$

to give the following equation for flow length

$$L_x = \left[4^m(2g)^{m/2}\right]\left[(N_{tu})^{1-m/2}(\Delta p)^{m/2}\right]\left[(N_{Pr}^{\ 2/3})^{1-m/2}(\rho)^{m/2}(\mu)^{-m}\right]$$
$$\left[(r_h)^{1+m}(\eta_s)^{-1+m/2}(f/j)^{-m/2}(c_j)^{-1}\right] \quad . \tag{A-31}$$

<u>Core Volume for One Side</u>. The void volume is equal to the flow area times the flow length, so by multiplying Equation (A-49) by (A-31), we arrive at

$$V_v = \left[4^m (2g)^{-1+m/2}\right] \left[(w)(N_{tu})^{3-m/2}(\Delta p)^{-1+m/2}\right] \left[(N_{Pr}{}^{2/3})^{3-m/2}(\rho)^{-1+m/2}(\mu)^{-m}\right]$$

$$\left[(r_h)^{1+m}(\eta_s)^{-3+m/2}(f/j)^{1-m/2}(c_j)^{-1}\right] \quad . \qquad \text{(A-32)}$$

Now, let m = 1/3, which approximates the majority of curves of j versus N_{Re} for heat-exchanger surfaces over a wide range of Reynolds numbers. Then Equation (A-32) becomes

$$V_v = (2/g)^{1/3}\left[w(N_{tu})^{4/3}(\Delta p)^{-1/3}\right]\left[(N_{Pr})^{8/9}(\mu\rho)^{-1/3}\right]$$

$$\left[(r_h)^{4/3}(\eta_s)^{-4/3}(f/j)^{1/3}(c_j)^{-1}\right] \quad . \qquad \text{(A-33)}$$

Or we could think of this being

$$V_v = (\text{Constant}) \begin{bmatrix}\text{Performance}\\ \text{Parameter}\end{bmatrix} \begin{bmatrix}\text{Fluid-Property}\\ \text{Parameter}\end{bmatrix} \begin{bmatrix}\text{Surface-Property}\\ \text{Parameter}\end{bmatrix} .$$

From the above equation, it is seen that the core volume can be made to approach zero if the hydraulic radius is made to approach zero. However, the flow area will not change appreciably, because the hydraulic radius does not appear in Equation (A-29) for flow area. The volume can also be reduced somewhat by using a surface possessing a lower f/j ratio, and higher η_s and c_j factors.

Considering core volume only, it can be stated that the hydraulic radius is the most important single parameter affecting heat-exchanger compactness.

<u>Compactness Parameter and Corrected Heat Transfer Density</u>. The definition for heat-transfer density is

$$HTD = Q/\Delta T_{ln} V \quad , \qquad \text{(A-34)}$$

and it is apparent that this parameter can be based on the void volume on each side of the core, the total core volume, or the over-all heat-exchanger volume, including insulation, etc.

Using Equation (A-13) we obtain

$$HTD = N_{tu_o}\, C_{min}/V \quad . \qquad \text{(A-35)}$$

If V is defined as being the core void volume on one side, V_v, Equation (A-33) can be substituted into the above equation, and we arrive at the following expression:

$$\left(\mathrm{HTD}\right)\left[(C/C_{min})(N_{tu_o}/\Delta p)^{1/3}(N_{Pr})^{8/9}(\mu\rho)^{-1/3}(c_p)^{-1}\right]$$

$$=\left[(g/2)^{1/3}(c_j)(\eta_s)^{4/3}(r_h)^{-4/3}(f/j)^{-1/3}\right]\left[(N_{tu_o}/N_{tu})^{4/3}\right] \quad . \tag{A-36}$$

Or we can say

$$\left(\mathrm{HTD}\right)\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} = \begin{bmatrix}\text{Compactness}\\ \text{Parameter}\end{bmatrix}\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} \quad , \tag{A-37}$$

or

$$\mathrm{HTDC} = P_c\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} \quad , \tag{A-38}$$

where

$$\mathrm{HTDC} = \mathrm{HTD}\left[(C/C_{min})(N_{tu_o}/\Delta p)^{1/3}(N_{Pr})^{8/9}(\mu\rho)^{-1/3}(c_p)^{-1}\right] \tag{A-39}$$

$$= \mathrm{HTD}\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} \tag{A-40}$$

$$\begin{bmatrix}\text{Correction}\\ \text{Factor}\end{bmatrix} = \left[(C/C_{min})(N_{tu_o}/\Delta p)^{1/3}(N_{Pr})^{8/9}(\mu\rho)^{-1/3}(c_p)^{-1}\right] \tag{A-41}$$

$$P_c = \left[(g/2)^{1/3}(c_j)(\eta_s)^{4/3}(r_h)^{-4/3}(f/j)^{-1/3}\right] \tag{A-42}$$

$$\begin{bmatrix}\text{Design}\\ \text{Parameter}\end{bmatrix} = (N_{tu_o}/N_{tu})^{4/3} \quad . \tag{A-43}$$

HTDC stands for "corrected heat transfer density" and P_c for "compactness parameter". It must also be remembered that the above HTDC and HTD are based on the void volume for one side of the core.

SUPPLEMENT
to the
FINAL TECHNICAL REPORT

on

COMPACT HEAT-EXCHANGER STUDY

Contract No. DA-44-009-AMC-313(X)

to

U. S. ARMY ENGINEER REACTORS GROUP
ARMY NUCLEAR POWER PROGRAM
FORT BELVOIR, VIRGINIA

June 15, 1964

by

F. A. Creswick, S. G. Talbert, and J. W. Bloemer

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June 15, 1964

Property Officer
Warehouse 335
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Fort Belvoir, Virginia

M/F: Contract No. DA-44-009-AMC-313(X)

Dear Sir:

Compact Heat-Exchanger Study

We are submitting this supplement to be incorporated into the final technical report of April 15, 1964. The additional information contained herein was requested by APCDB and shows that certain items requested in the applicable work statement could not be obtained. Some additions and corrections are also included.

The attached Tables 1 through 4 and Figures 5 and 6 are modified and should replace those in the final technical report as initially submitted.

If there are any items in this supplement or in the final report that need further clarification, please let us know.

Very truly yours,

James A. Eibling

James A. Eibling
Group Director
Thermal Systems Research

JAE:jvd
Enc. (25)

cc: APCDB
Building 322
USAERDL
Fort Belvoir, Virginia
Attention Mr. E. D. Collins

SUPPLEMENT

This supplement has been written to incorporate certain pertinent discussions and information omitted from the final technical report as initially submitted.

Availability of Desired Data

Thermodynamic Data

It has been noted that, of more than 200 references examined, data are presented from only 25 of them. However, nearly 90 different core surfaces were analyzed, which yielded about 180 data points. The primary reasons that more data were not found useful were either that the thermodynamic performance data or descriptions of the heat-exchanger surface geometry were incomplete. Comprehensive data on both are necessary in order to compare performance data or relate the performance of a heat exchanger under one set of conditions to predicted performance under another set of conditions. The data presented in Tables 1 through 4 tabulate all the data presented in Figures 1 through 4 and represent all the meaningful information that was found that contributed directly to the objectives of this study. However, there are many good discussions and much interesting background information on the general subject of heat exchangers that are not directly pertinent to this study; this is the main reason for including all the references examined in the Bibliography.

Life and Location Information

Data on the mechanical performance of heat exchangers, e.g., operating-life and thermal- and pressure-cycling information, were, in general, not available for the heat-exchanger surfaces reported. This statement must be qualified by the fact that all the references reviewed were not read cover-to-cover; therefore, some life data could have been overlooked. However, this type of information is in an area of sensitivity to most manufacturers; therefore, we would expect considerable reluctance to put this type of information in the literature. Further, it is apparent that the scientific and technical community considers this type of information too specific to be of general interest and significance.

Information on the location of heat exchangers is generally not pertinent. As we have shown, most of the useful information available is in the form of reduced data obtained from tests on core samples. Of the complete heat-exchanger units on which data are available, most are either designed for mobile or vehicular applications or not for any specific location.

General reference to some specific heat-exchanger installations is made in Reference 98, but no performance data are given.

Manufacturers

The following table lists the manufacturers of the heat-exchanger equipment listed in Tables 1 through 4:

TABLE 8. HEAT-EXCHANGER MANUFACTURERS

Reference No.	Manufacturer Identification
38	Not given
46	AiResearch
49	Not given
71	Curtiss-Wright
113	G. M. Harrison
114	Trane
116	G. M. Harrison, Modine, or AiResearch
119	Not given
120	Air Preheater, or G. M. Harrison
121	Not given
123	Not given
124	Not given
125	AiResearch
126	Not given
128	Ferrotherm
129	Not given
136	Not given
140	Wolverine Tube, Modine, or AiResearch
154	AiResearch
159	AiResearch
177	Brown Fintube
195	Not given
197	Not given
201	Trane
212	AiResearch

In a few cases in which several heat-exchanger surfaces are presented and more than one manufacturer is listed, the respective manufacturer is not called out for each surface.

Discussion of Tables 1 Through 4

An identification number for each surface has been included in Tables 1 through 4. This is necessary because of the convention that was chosen for describing the surface geometry in Tables 2, 3, and 4. For these tables, data are given for only the "hot-fluid" side or the "cold-fluid" side of the heat exchanger, and the description of the surface and the hydraulic radius are given primarily to identify the respective side of the surface.

The "sandwich" designation in Table 3 refers to multiple layers of a similar core surface used for each hot or cold fluid passage. Two and three consecutive layers for one fluid passage are termed as double and triple sandwich surfaces, respectively. Such configurations allow design flexibility when choosing optimum flow areas and fin configurations for the two fluid sides of a heat exchanger.

The hydraulic radius is based on the surface area and void volume on the respective sides of the exchanger in Tables 2, 3, and 4. In Table 1, which gives over-all data on the heat-exchanger core, r_{h_o} is calculated on the basis of the total surface area and the total void volume of both fluid sides of the core.

Data on the ML-1 precooler or other heat exchangers used in the Army Gas-Cooled Reactor Systems Program were not included in the tables because there was insufficient information on the core dimensions, surface geometry, and operating conditions to enable a meaningful comparison to be made.

Figure 12 shows four core surfaces listed in Table 1, but not described by Figures 5 or 6.

Additions and Corrections to Final Report

Table of Contents
Add "Supplement" . . . Page S-1.

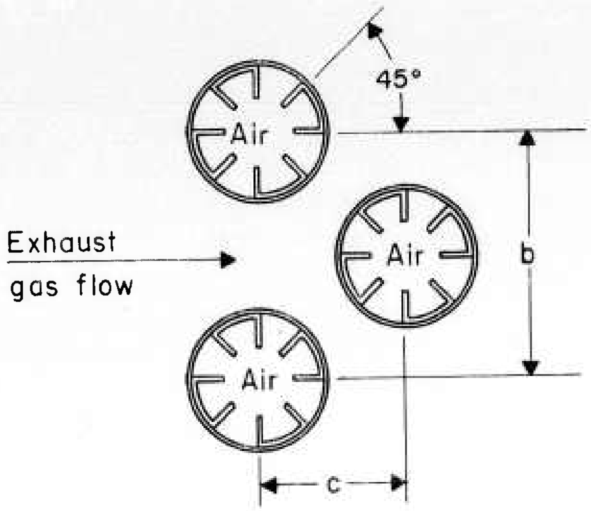
Table of Contents, List of Figures
Add "Figure 12. Additional Heat-Exchanger Surfaces in Table 1" . . . Page S-4.

Table of Contents, List of Tables
Add "Table 8. Heat-Exchanger Manufacturers" . . . Page S-2.

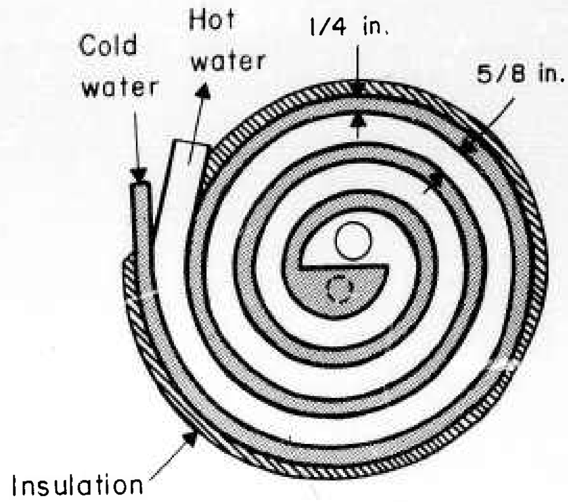
Selected Bibliography, Reference 195
Change "Trans. AIChE" to "Trans. Instn. Chem. Engrs."

Nomenclature, English-Letter Symbols
Add "c - Plate or tube spacing, inches"

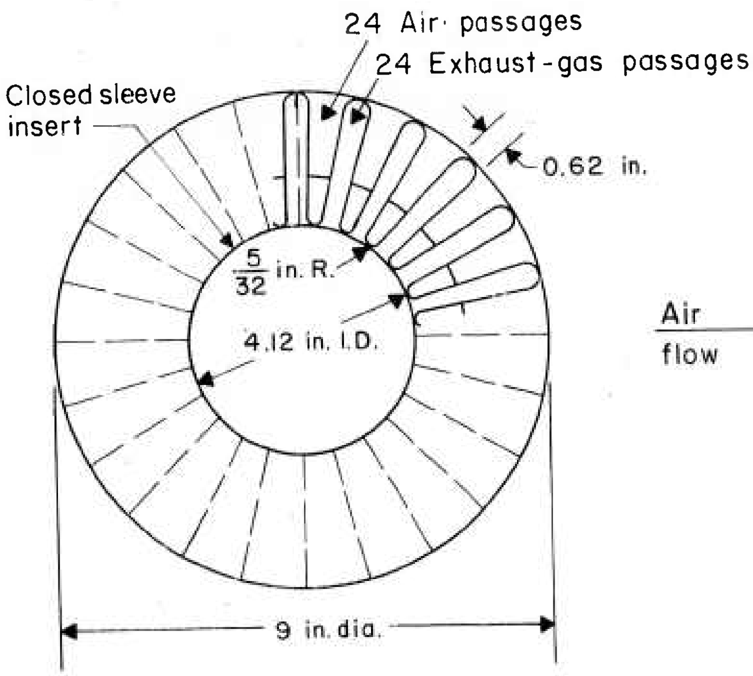
Nomenclature, Grouped-Letter Symbols
Add "PF - Parallel flow heat exchanger"



Heat Exchanger No. 13
(fins 0.188 x 0.018 in.)



Heat Exchanger No. 16



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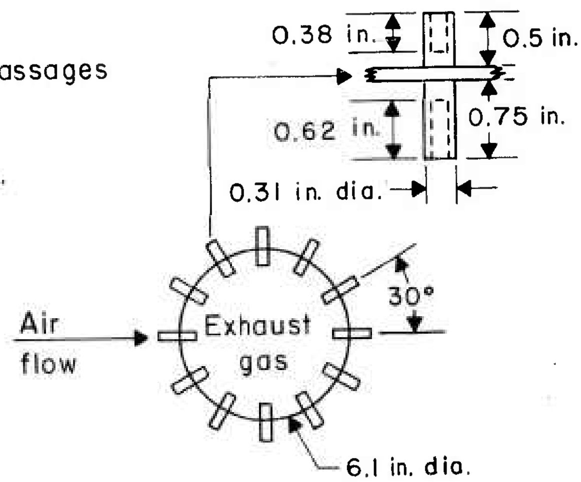

Heat Exchanger No. 20
(12 pins per row, 16 rows)

A-47733

FIGURE 12. ADDITIONAL HEAT-EXCHANGER SURFACES IN TABLE 1

NOMENCLATURE

English-Letter Symbols

A - Area, ft^2

a - Fin spacing, inches

b - Plate or tube spacing, inches

C - Capacity rate, $(w c_p)$, Btu/hr-F

c_j - Colburn-number correlation constant, dimensionless

c_p - Specific heat, Btu/lb-F

d - Material density, lb/ft^3

f - Fanning friction factor, dimensionless

G - Mass velocity, (w/A_x), $lb/hr\text{-}ft^2$

g - Gravitational conversion factor, 4.18×10^8 $lb_m\text{-}ft/lb_f\text{-}hr^2$

h - Convective heat-transfer coefficient, $Btu/hr\text{-}ft^2\text{-}F$

j - Colburn number, $N_{St} N_{Pr}{}^{2/3}$, dimensionless

K - Constant, $lb^{1/3}$-ft [defined by Equation (3)]

k - Thermal conductivity, Btu/hr-ft-F

L - Length, ft

m - Fin constant $\left(\sqrt{2h/k\delta}\right)$ ft^{-1}

N - N_{tu} ratio (N_{tu_h}/N_{tu_c}), dimensionless

P - Pressure-drop ratio $(\Delta p_h/\Delta p_c)$, dimensionless

p - Pressure, lb/ft^2

Q - Heat-transfer rate, Btu/hr

R - Hydraulic-radius ratio (r_{h_h}/r_{h_c}), dimensionless

r_h - Hydraulic radius $(A_x L_x/A_s)$, ft, ($4 r_h$ = hydraulic diameter)

T - Temperature, F

V - Volume, ft^3

w - Flow rate, lb/hr

Y - Constant, $lb^{1/2}$-ft [defined by Equation (16)]

Greek-Letter Symbols

β - Ratio of surface heat-transfer area to volume between spacer plates, ft^2/ft^3

Δ - Difference of temperatures, pressures, etc.

δ - Fin thickness, ft

ϵ - Effectiveness of heat exchanger, dimensionless

η - Effectiveness of fin or total surface, dimensionless

μ - Dynamic viscosity, lb/ft-hr

ξ - Ratio of pressure differences and pressures $(\Delta p_h p_c/\Delta p_c p_h)$, dimensionless

ρ - Fluid density, lb/ft^3

σ - Ratio of void volume to total volume of core, dimensionless

Grouped-Letter Symbols

CC - Counterflow heat exchanger

CF - Crossflow heat exchanger

HTD - Heat transfer density, $Btu/hr\text{-}100\ F\text{-}ft^3$

HTDC - Corrected heat transfer density, $hr^{-2/3}\ ft^{-1}$

NA - Not applicable

P_c - Compactness parameter, $hr^{-2/3}\ ft^{-1}$

SHT - Specific heat transfer, Btu/hr-100 F-lb

Dimensionless Numbers

ϵ - Effectiveness = Q_{act}/Q_{max}

f - Fanning friction factor = $2g\rho\Delta p r_h/G^2 L_x$

j - Colburn number = $N_{St} N_{Pr}{}^{2/3}$

N_{Nu} - Nusselt number = $4r_h h/k$

N_{Pr} - Prandtl number = $\mu c_p/k$

N_{Re} - Reynolds number = $4r_h G/\mu$

N_{St} - Stanton number = $h/Gc_p = N_{Nu}/N_{Re}N_{Pr}$

N_{tu} - Number of heat-transfer units = $\eta_s h A_s/wc_p$

Subscripts

1 - Inlet

2 - Outlet

act - Actual

avg - Average

c - Cold side, or compactness parameter (P_c)

est - Estimated

f - Fin, or force

h - Hot side

i - Insulation

ln - Log-mean difference

m - Manifold, or mass

max - Maximum

min - Minimum

n - Nonflow dimension

o - Over-all

R - Ratio

s - Surface

t - Total

v - Void

x - Flow length, area, or volume