THE HEAT TRANSFER AND FLOW FRICTION PERFORMANCE OF
THREE COMPACT PLATE-FIN HEAT EXCHANGER SURFACES

TECHNICAL REPORT NO. 7

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ABSTRACT

This report contains the results of basic heat transfer and flow friction tests of three compact high performance plate-fin-type heat transfer surfaces. One surface is of strip or serrated fin construction while the other two have wavy or ruffled fins. The heat transfer and flow friction characteristics, respectively, are given in both tabular and graphical form using the conventional dimensionless correlations employing the Stanton (h/Gc_p), Prandtl ($\mu c_p/k$), Fanning friction factor (f), and Reynolds ($Gr_h G/\mu_f$) numbers in the form, $(h/Gc_p)Pr^{2/3}$ and f versus $(Gr_h G/\mu_f)$. The heat transfer--friction power characteristics, evaluated at specified standard gas properties (air at 1 atm. and 500 °F) are presented on both a unit area and a unit volume basis.

The test results show that the wavy or ruffled fin surface is competitive with the completely interrupted fin surface from the heat transfer--friction power point of view and therefore may be an attractive surface for applications where high performance is desired but fouling is a major consideration. However, this opinion requires verification by tests at actual or simulated service conditions.

The strip fin surface exhibits the characteristics of short flat plates with laminar boundary layer, and the ruffled fin surfaces exhibit similar characteristics indicating the build-up of a laminar boundary layer on the fins between changes in fin direction.

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NOMENCLATURE

English Letter Symbols

- A - transfer area, ft^2
A_c - free-flow area, ft^2
A_{fr} - frontal area, ft^2
a - plate thickness between surfaces, ft
b - plate spacing, in, ft
c_p - specific heat at constant pressure, Btu/lb^oF
E_{std} - friction power evaluated at certain standard gas properties,
hp/(ft² of A)
f - Fanning friction factor, dimensionless
G - mass velocity based on A_c, lbs/(hrft² of A_c)
h - unit conductance for thermal convection heat transfer,
Btu/(hrft^{2o}F)
h_m - steam side unit film conductance, Btu/(hrft^{2o}F)
h_{std} - unit conductance evaluated at certain standard gas properties,
Btu/(hrft^{2o}F)
K_c - contraction loss coefficient for air flow at test core entrance,
dimensionless
K_e - expansion loss coefficient for air flow at test core exit,
dimensionless
k - thermal conductivity, Btu/(hrft^{2o}F/ft)
L - flow length, ft, in
l - flow length of uninterrupted fin, ft, in
r_h - flow passage hydraulic radius based on free-flow area, A_c, ft
V - heat exchanger total volume, ft³
w_c - steam condensate rate, lbs/hr

Greek Letter Symbols

- α - ratio of total transfer area of one side of exchanger to total
volume of exchanger, ft²/ft³
β - ratio of total transfer of one side of exchanger to the volume
between the plates of that side, ft²/ft³

- η_o - temperature effectiveness of the heat transfer surface, dimensionless
 ρ - gas density, lbs/ft^3
 σ - ratio of free-flow to frontal area, dimensionless
 μ - gas viscosity, evaluated at bulk average temperature, lbs/(hrft)
 μ_f - gas viscosity, evaluated at film average temperature, lbs/(hrft)

Dimensionless Groupings

- St - Stanton's number, (h/Gc_p) , a heat transfer modulus
 Pr - Prandtl's number, $(\mu_f c_p/k)$, a fluid properties modulus evaluated at film average temperature
 $StPr^{2/3}$ - generalized heat transfer grouping. This factor versus Re_f defines the heat transfer characteristics of the surface.
 Re - Reynolds number, $(4r_h G/\mu)$, a flow modulus characterizing the type of flow.
 Re_f - $(4r_h G/\mu_f)$
 Re_l - $(2G/\mu_f)$
 f - Fanning friction factor. The $f - Re_f$ relation defines the friction characteristics of the surface.

INTRODUCTION

This report is the sixth of a series giving the basic heat transfer and flow friction characteristics of high performance heat transfer surfaces. Large transfer area per unit volume and a high heat transfer coefficient relative to the flow friction power requirement are the characteristics of "high performance surfaces" as used throughout this report. A complete description of the test apparatus and method of analysis of data is given in ref. (a), the first report of the series. Included here are the results of tests of one strip, or serrated fin and two ruffled or wavy fin plate-fin surfaces. The general characteristics of the fins are shown in the following sketches.

The heat transfer performance of these surfaces, using the dimensionless correlation, $(h/Gc_p)Pr^{2/3}$ versus Reynolds number, Re_f , is summarized in both tabular and graphical form. The friction performance is similarly presented employing the correlation, f versus Re_f . The Reynolds number range tested (Re based on flow passage hydraulic diameter) is 300 to 8000.

Test Surfaces

The three surfaces tested are designated according to a numbering convention as follows. For the strip-fin surface, the first number refers to the length of uninterrupted fin in the direction of flow, $1/8$ inch, and the second to the number of fins per inch, 15.2. Thus this surface is designated as $1/8-15.2$. For the ruffled fin surfaces, the first number refers to the number of fins per inch and the second to the wave length of the ruffle. The letter R is added to denote the ruffled character of the fins. Thus these surfaces are designated as $11.44-3/8R$ and $17.8-3/8R$.

The heat transfer surfaces are formed of 0.006 inch aluminum fin material brazed between 0.010 aluminum plates. They are built into steam-to-air test

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cores as described in ref. (a). All of the test core dimensions necessary to reduce the basic laboratory data are contained in Test Core Data. The detailed descriptions of the surfaces are contained in the drawings on the figures giving the basic heat transfer and flow friction characteristics, and in the caption sheets to these figures.

Summary and Conclusions

The results and conclusions of these tests are summarized as follows:

1. Both strip and ruffled fins exhibit similar heat transfer characteristics which suggests a similarity of the convective heat transfer mechanisms in the flow boundary layer. A fair correlation between the three surfaces and the heat transfer characteristics for flow over a flat plate with laminar boundary layer is obtained by plotting the heat transfer parameter, $StPr^{2/3}$, as a function of a Reynolds number based on the length of uninterrupted fin. A similar correlation is not possible for the friction factors because of parasitic eddy losses which are not directly associated with the boundary layer behavior.
2. The performance of strip-fin surfaces is dependent to a certain extent upon the character of the leading and trailing edges of the fins. A dull cutting tool will produce jagged and bent over edges which tend to increase both $StPr^{2/3}$ and f .
3. The heat transfer and flow friction data presented here are recommended for application to identical or geometrically similar surfaces within a tolerance of ± 5 percent. Geometrical similarity in the case of the strip-fin surface implies fairly rough fin edges, and the characteristics of surfaces with clean fin edges may differ somewhat from the data presented.
4. A comparison of the heat transfer--friction power performance indicates that just as high performance is obtainable with ruffled fins as with strip fins, and this performance is far superior to that obtainable with plain uninterrupted fins.
5. It is believed that the ruffled fin surface will have less tendency to become fouled than the strip-fin surface and therefore may possibly be attractive for the hot gas side of a gas turbine regenerator, or other applications where high performance is desired but fouling is a major consideration. This opinion requires verification by tests at actual or simulated service conditions.

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TEST RESULTS

Description of Tables and Graphs

All of the core dimensions necessary to reduce the basic laboratory test data are tabulated under TEST CORES. The dimensions characterizing surface geometry only, irrespective of the manner in which the surfaces happened to be incorporated into the test cores, are contained in the caption sheets to Figs. 1, 2, and 3.

The heat transfer and flow friction characteristics, respectively, are presented using the dimensionless correlations, $(h/Gc_p)Pr^{2/3}$ versus Re_f , and f (Fanning friction factor) versus Re_f . The results are given in tabular form and also in graphical form, Figs. 1, 2, and 3.

The curves representing the best interpretation of the friction factor data are faired through test points for the isothermal flow tests. The curves representing the best interpretation of the heat transfer data lie somewhat above the test points due to an allowance for an estimate of the steam side resistance. The basis for this estimate is given in the appendix.

The Reynolds number, $Re_f = (4r_h G/\mu_f)$, is evaluated on the basis of the flow passage hydraulic radius and a "film average" viscosity. The hydraulic radius is defined as follows:

$$r_h/L = A_c/A$$

$$r_h = A_c L/A$$

For cylindrical tubes of any cross-section geometry, $A_c L/A$ is identical with the flow cross-section divided by the wetted perimeter, which is the conventional definition of hydraulic radius. The "film average" viscosity is defined as the viscosity corresponding to a temperature half way between the wall temperature and the arithmetic average of air inflow and outflow bulk temperatures.

Fig. 4 shows a comparison of the best interpretation of heat transfer and friction data plotted as a function of Reynolds number, Re_f , based on the length of uninterrupted fin, or in the case of the ruffled or wavy fin surfaces, the length of fin between each change of fin direction. Also included on this figure are the data for two previously tested strip-fin surfaces, 1/4-11.1 and 3/32-12.22, from refs. (b) and (c), and the characteristics for flow over a flat plate with sharp leading edge and laminar boundary layer.

The laminar-type boundary layer friction characteristics for the flat plate are taken from the drag coefficient data of ref. (d).

$$f = 1.328 Re_f^{-0.5}$$

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For air flow over a flat plate system with a sharp leading edge, since Pr is close to unity ($Pr \approx 0.7$), it has been demonstrated analytically by the momentum - heat transfer analogies, and by experiment, ref. (e), that

$$StPr^{2/3} \approx 1/2$$

Thus for the flat plate system,

$$StPr^{2/3} = 0.664 Re$$

The tabular Summary of Basic Heat Transfer and Friction Data is taken directly from the curves representing the best interpretation of the data. The heat transfer-friction power characteristics, h_{std} versus E_{std} are evaluated from these data. E_{std} and h_{std} are given in the Summary and in graphical form on Fig. 5. The plot of h_{std} versus E_{std} compares the heat transfer rate with the friction power expenditure on a unit transfer area basis. Fig. 6 shows a similar comparison on a unit volume basis in which βh_{std} is plotted as a function of βE_{std} .

Experimental Methods and Calculations

The description of the steam-to-air test apparatus and the method of analysis of data are included in Technical Report No. 2, ref. (a). The auxiliary data used in conjunction with the computing procedure is also given in this reference with the exception of the temperature effectiveness of the surface, η_o , the entrance and exit coefficients, K_c and K_e , and the steam side film coefficient, h_m . These terms vary with different test cores and consequently are given in the appendix to this report for the particular cores considered here.

Experimental Accuracy

The experimental accuracy is analyzed in Technical Report No. 2, ref. (a), and the estimate of ± 5 percent probable error for both the $StPr^{2/3}$ grouping and f is considered to be applicable here. The probable error in the Reynolds number determination is likewise believed to be within ± 1.6 percent. The use of a "film average" viscosity in evaluating the Reynolds number is a question of interpretation of results rather than of experimental accuracy, but may introduce some error in applications involving large temperature differences between the gas and the wall.

For the core incorporating the surface 1/8-15.2 some very poor energy balances were obtained at low air flow rates. This was found to be due to a deficiency in the steam system which would not affect either the heat transfer or friction results. Failure of the inlet steam separator system resulted in the carry over of liquid water with the slightly superheated core steam. This difficulty was corrected, and it will be noted that the energy balances for the two subsequent test cores are quite good. For this system the energy

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unbalances should be slightly on the positive side since the steam condensate must be separated from a considerable excess or "blow" steam (see ref. a), and the condensate separator cannot be expected to produce 100 percent separation.

Surface Geometry

The following geometrical relations are given to facilitate the extension of the test results reported here to heat exchanger design calculations. These relations hold for any surface for which the hydraulic radius is equal to $A_c L/A$, that is, surfaces with A_c constant throughout the flow length, L . For other surfaces an "equivalent" r_h may be defined as $A_c L/A$, where A_c is the minimum free-flow area.

The following geometrical factors are given for each particular surface tested as part of the basic data for the surface in question (see caption sheets for Figs. 1, 2, and 3).

- b - Plate spacing, ft
- r_h - Flow passage hydraulic radius, or equivalent hydraulic radius, ft
- β - Ratio of total transfer area of one side of exchanger to the volume between the plates of that side, ft^2/ft^3

For a particular heat exchanger application the following geometrical factors are required for each of the two sides of the core.

- A - Total transfer area of one side of the exchanger, ft^2
- A_c - Free-flow area of one side, ft^2
- A_{fr} - Frontal area of one side, ft^2
- L - Flow length of one side, ft
- V - Total exchanger volume, including both sides, ft^3
- α - Ratio of total transfer area of one side of the exchanger to total volume of the exchanger, ft^2/ft^3
- ϕ - Ratio of free-flow area to frontal area of one side of the exchanger

In addition, the plate thickness (a , ft) between the two sides of the exchanger must be specified.

The various surface and core factors are related as follows for side 1. Subscript 1 refers to one side of the exchanger and 2 to the other side. Factors without subscript are common to both sides. The same relations apply to side 2 with exchange of subscripts 1 for 2 and 2 for 1.

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$$\alpha_1 = \left(\frac{A_c}{A_{fr}} \right)_1 = \left(\frac{Ar_h}{A_{frL}} \right)_1 = \frac{(Ar_h)_1}{V} = \frac{b_1 \beta_1 r_{h1}}{b_1 + b_2 + 2a}$$

$$A_{c1} = (a A_{fr})_1 = \left(\frac{Ar_h}{L} \right)_1 = \left(\frac{Ac}{La} \right)_1$$

$$\alpha_1 = \frac{A_1}{V} = \left(\frac{A}{A_{frL}} \right)_1 = \left(\frac{c}{r_h} \right)_1 = \frac{b_1 \beta_1}{b_1 + b_2 + 2a}$$

$$\left(\frac{r_h}{L} \right)_1 = \left(\frac{A_c}{A} \right)_1 = \left(\frac{c}{La} \right)_1$$

These relations together with the basic data are required for the design of heat exchangers.

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TEST CORE DATA

Core 1/8-15.2 (Mfgs. No. DL-901-2257)

Air Side

Plate spacing - 0.414 in
Plate metal thickness - 0.010 in
Fin metal thickness - 0.006 in
Fin and plate material - aluminum of thermal conductivity,
 $k = 110 \text{ Btu}/(\text{hrft}^2\text{of}/\text{ft})$
Frontal area - 0.567 ft²
Free-flow area - 0.242 ft²
Free-flow/frontal area - 0.427
Plate area - 5.16 ft²
Fin area - 35.2 ft²
Total area - 40.36 ft²
Hydraulic diameter of flow - $4r_h = 0.00870 \text{ ft}$
Air flow length - 4.35 in

Steam Side

Frontal area - 0.308 ft²
Free-flow area - 0.1104 ft²
Plate area - 5.52 ft²
Fin area - 21.98 ft²
Total area - 27.50 ft²
Hydraulic diameter of flow - $4r_h = 0.01048 \text{ ft}$

Core 11.44-3/8R (Mfgs. No. DL-901-2255)

Air Side

Plate spacing - 0.413 in
Plate metal thickness - 0.010 in
Fin metal thickness - 0.006 in
Fin and plate material - aluminum of thermal conductivity,
 $k = 110 \text{ Btu}/(\text{hrft}^2\text{of}/\text{ft})$
Frontal area - 0.567 ft²
Free-flow area - 0.249 ft²
Free-flow/frontal area - 0.439
Plate area - 5.204 ft²
Fin area - 28.140 ft²
Total area - 33.344 ft²
Hydraulic diameter of flow - $4r_h = 0.0106 \text{ ft}$
Air flow length - 4.26 in

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Steam Side

Frontal area - 0.308 ft^2
 Free-flow area - 0.115 ft^2
 Plate area - 5.572 ft^2
 Fin area - 21.798 ft^2
 Total area - 27.370 ft^2
 Hydraulic diameter of flow - $4r_h = 0.01102 \text{ ft}$

Core 17.8-3/8R (Mfgs. No. DL-901-2256)

Air Side

Plate spacing - 0.413 in
 Plate metal thickness - 0.010 in
 Fin metal thickness - 0.0060 in
 Fin and plate material - aluminum of thermal conductivity,
 $k = 110 \text{ Btu}/(\text{hr ft}^2 \text{ } ^\circ\text{F}/\text{ft})$
 Frontal area - 0.567 ft^2
 Free-flow area - 0.237 ft^2
 Free-flow/frontal area - 0.418
 Plate area - 4.976 ft^2
 Fin area - 43.67 ft^2
 Total area - 48.65 ft^2
 Hydraulic diameter of flow - $4r_h = 0.00695 \text{ ft}$
 Air flow length - 4.28 in

Steam Side

Frontal area - 0.308 ft^2
 Free-flow area - 0.108 ft^2
 Plate area - 5.55 ft^2
 Fin area - 21.56 ft^2
 Total area - 27.11 ft^2
 Hydraulic diameter of flow - $4r_h = 0.01037 \text{ ft}$

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HEAT TRANSFER AND FRICTION DATA - CORE 1/8 - 15.2 (Mfg. No. DL-901-2257)

Run No.	Air Flow Rates		Upstream Air State		Press. °H ₂ O abs. @ 70°F	Downstream Temp. °F	Core ΔP °H ₂ O @ 70°F	Steam Average Saturation Temp. °F	Convection Film Coeff., h Btu/(hr ft ² °F)	(h/Gcp)Pr ^{2/3}	Friction Factor f	Reynolds' No.		% Energy Unbal. (Q _{air} - Q _{sat}) Q _{air}
	lbs/hr	Core Mass Velocity lbs/hr ft ²	Humidity lbs/lb	Temp. °F								Re & Ave. Film Temp.	Re & Ave. Bulk Temp.	
25	7557	31229	.0085	86.0	402.73	193.2	33.87	228.9	76.6	.00790	.0491	5307	5590	+1.0
24	7212	29803	.0090	84.7	403.08	193.9	30.87	229.0	74.2	.00802	.0493	5064	5335	+ .5
23	6327	26146	.0095	83.8	403.82	196.7	24.40	229.0	68.3	.00842	.0513	4443	4681	-1.0
22	5610	23183	.0095	80.9	404.38	198.9	19.29	229.1	63.5	.00882	.0520	3947	4150	- .7
20	4959	20493	.0086	83.7	406.36	202.7	14.93	229.2	59.5	.00936	.0539	3475	3661	-3.0
21	4721	19510	.0091	79.8	405.01	202.3	13.93	229.1	56.8	.00938	.0534	3315	3493	+0
19	4292	17734	.0086	82.9	406.80	205.2	11.45	229.1	54.1	.00982	.0533	3008	3162	-2.1
18a	3948	16315	.0115	81.5	407.16	206.6	9.83	229.3	51.4	.01011	.0541	2767	2909	+3.2
18b	3935	16260	.0119	83.9	404.88	206.4	9.86	228.9	50.8	.01002	.0543	2758	2899	+ .2
18	3754	15512	.0088	82.7	407.10	207.5	9.02	229.1	49.4	.01027	.0551	2631	2765	-5.2
17	3256	13456	.0087	81.9	407.35	209.9	6.94	229.2	45.0	.01077	.0565	2278	2394	- .8
16	3255	13450	.0082	80.0	407.21	210.3	6.93	230.3	44.7	.01069	.0564	2277	2398	-3.9
15	2938	12142	.0078	78.2	407.36	211.7	5.82	230.2	41.9	.01109	.0582	2055	2165	-3.6
14	2541	10498	.0086	85.7	406.29	214.7	4.51	229.9	38.3	.01173	.0601	1773	1860	-3.1
13	2244	9271	.0087	85.5	406.39	216.1	3.72	230.0	34.7	.01205	.0637	1566	1643	-4.9
12	1955	8078	.0090	85.2	406.50	217.7	2.94	230.0	31.8	.01266	.0664	1365	1428	-7.0
12a	1952	8066	.0111	82.4	405.82	216.2	2.92	229.0	31.5	.01250	.0661	1362	1429	-2.1
11	1929	7970	.0106	89.0	404.00	216.7	2.821	228.7	31.1	.01257	.0653	1346	1407	+ .3
10	1670	6899	.0106	87.9	404.09	217.8	2.212	228.8	27.8	.01295	.0680	1163	1218	-2.5
9	1443	5961	.0109	87.2	404.16	218.8	1.753	228.8	24.7	.01336	.0724	1005	1052	-5.2
9a	1438	5943	.0097	85.3	406.17	219.0	1.728	228.8	25.0	.01354	.0723	1004	1051	+2.6
8	1278	5281	.0103	82.7	404.64	219.5	1.439	229.0	22.5	.01368	.0762	892	934	-2.6
7	1154	4770	.0104	82.1	404.67	220.0	1.221	229.1	20.5	.01386	.0794	806	843	-3.4
6	984	4066	.0112	86.1	406.64	221.4	.957	229.1	18.4	.01447	.0860	686	718	-4.6
5	839	3468	.0114	86.3	406.66	222.6	.757	229.2	16.4	.01516	.0937	585	611	-4.8
4	740	3058	.0114	85.7	406.68	223.1	.638	229.2	14.8	.01549	.1019	516	539	-7.7
4a	737	3045	.0100	85.1	406.34	222.7	.633	228.9	14.6	.01541	.1021	513	537	+ 1.1
3	643	2657	.0116	85.4	406.70	224.3	.522	229.3	13.6	.01641	.1106	448	468	-3.2
2	559	2308	.0115	84.8	407.23	224.6	.430	229.3	12.0	.01663	.1211	389	407	-11.4
2b	556	2299	.0101	84.9	406.37	224.6	.429	229.0	12.1	.01694	.1217	388	405	- .1
1	405	1675	.0110	84.0	407.25	226.2	.282	229.3	9.7	.01853	.1513	282	295	-20.1
1b	401	1660	.0098	84.2	406.39	226.1	.280	229.5	9.3	.01810	.1530	280	292	-1.5
Average														-1.5
Deviation														3.5

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HEAT TRANSFER AND FRICTION DATA - CORE 11.44 - 3/8R (Mfgs. No. DL-901-2255)

Run No.	Air Flow Rates		Upstream Air State			Downstream Temp. °F	Core ΔP °H ₂ O @ 70°F	Steam Average Saturation Temp. °F	Convection Film Coeff., h Btu/(hr ft ² °F)	(h/Gcp)Pr ^{2/3}	Friction Factor f	Reynolds' No.		% Energy Unbal. (Q _{air} - Q _{steam}) Q _{air}
	lbs/hr	Core Mass Velocity lbs/hr ft ²	Humidity lbs/lb	Temp. °F	Press. °H ₂ O abs. @ 70°F							Re @ Ave. Film Temp.	Re @ Ave. Bulk Temp.	
6	9370	37629	.0100	76.2	402.55	167.4	30.60	229.2	74.2	.00636	.0372	7883	8397	+2.7
5	8436	33881	.0099	75.1	403.76	170.2	25.36	229.3	70.1	.00667	.0386	7084	7545	+2.7
4	7368	29590	.0106	79.0	402.27	175.9	20.18	228.7	66.0	.00719	.0403	6174	6548	+3.5
3	6492	26072	.0107	79.6	403.08	179.3	16.56	228.8	60.8	.00752	.0432	5430	5758	+3.8
2	5761	23137	.0105	79.7	403.77	182.8	13.55	228.8	57.2	.00797	.0452	4818	5099	+4.5
7	5031	20205	.0109	78.3	407.25	186.3	10.66	229.0	53.1	.00847	.0474	4208	4453	+1.4
1	4711	18921	.0106	79.4	404.62	188.6	9.71	229.1	51.3	.00873	.0490	3933	4161	+1.5
8	4314	17323	.0103	79.1	407.75	190.8	8.32	228.8	49.0	.00912	.0507	3601	3802	+1.5
9	3760	15102	.0105	80.2	408.13	194.5	6.66	228.9	45.2	.00965	.0536	3133	3308	+1.4
10	3252	13061	.0107	80.5	408.41	198.1	5.30	229.0	41.6	.01026	.0571	2704	2849	+1.6
11	2965	11908	.0107	81.1	408.54	200.4	4.55	228.7	39.7	.01075	.0589	2465	2597	+1.9
12	2564	10297	.0106	81.1	408.73	203.5	3.62	228.9	36.3	.01136	.0626	2132	2241	+1.7
13	2282	9166	.0107	81.0	408.85	205.7	3.06	229.0	33.7	.01186	.0669	1894	1995	+1.5
14	2011	8076	.0103	75.9	409.19	207.6	2.415	229.4	31.2	.01244	.0680	1672	1758	+3.4
15	2000	8033	.0103	77.9	409.19	207.6	2.392	228.7	31.2	.01253	.0680	1663	1748	+3.8
16	1702	6834	.0101	78.9	409.30	210.6	1.844	228.6	28.4	.01341	.0724	1412	1485	+4.0
17	1503	6037	.0105	79.7	409.37	212.6	1.513	228.8	26.2	.01398	.0762	1247	1309	+3.0
18	1351	5426	.0106	80.3	409.42	213.9	1.270	228.8	24.2	.01437	.0793	1119	1176	+2.9
19	1167	4687	.0096	75.0	408.97	215.8	1.003	229.1	22.2	.01525	.0844	968	1018	+2.5
20	1002	4024	.0094	76.9	408.86	217.7	.781	228.8	20.3	.01620	.0889	830	872	+3.3
21	854	3428	.0095	77.6	408.89	219.1	.596	229.0	17.9	.01678	.0936	707	742	+3.8
22	745	2994	.0097	78.5	408.91	219.9	.484	228.8	16.1	.01729	.0999	617	648	+2.7
23	662	2658	.0096	79.1	408.92	220.5	.397	229.1	14.4	.01738	.104	547	574	+2.9
23a	658	2641	.0122	85.7	407.46	220.7	.397	228.9	14.3	.01739	.105	542	565	-1.5
24	571	2294	.0096	79.5	408.93	221.1	.309	228.9	12.8	.01790	.109	472	495	+3.0
25	453	1820	.0095	80.3	408.97	221.9	.210	228.9	10.4	.01844	.118	375	392	-3.9
25a	445	1789	.0118	86.5	407.49	222.0	.207	228.8	10.2	.01830	.120	367	384	-0
26	368	1477	.0097	80.3	408.98	222.3	.149	228.8	8.6	.01874	.128	304	318	+2.4

Average Deviation 2.5

187123

HEAT TRANSFER AND FRICTION DATA - CORE 17.8 - 3/8R (Mfg. No. DL-901-2256)

Run No.	Air Flow Rates		Upstream Air State			Downstream Temp. °F	Core ΔP °H ₂ O @ 70°F	Steam Average Saturation Temp. °F	Convection Film Coeff., h Btu/(hr ft ² °F)	(h/Gc _p) Pr ^{2/3}	Friction Factor f	Reynolds' No.		% Energy Unbal. (Q _{air} - Q _{steam}) Q _{air}
	lbs/hr	Core Mass Velocity lbs/hr ft ²	Humidity lbs/lb	Temp. °F	Press. °H ₂ O abs. @ 70°F							Re _f & Ave. Film Temp.	Re @ Ave. Bulk Temp.	
19	8252	34819	.0108	78.3	403.84	190.1	32.40	229.1	66.8	.006178	.0298	4745	5010	+1.9
18	7307	30832	.0108	77.2	404.87	193.5	26.29	229.1	62.9	.006568	.0313	4202	4437	+3.9
17	6287	26528	.0119	76.1	405.74	197.2	20.64	229.2	57.9	.007000	.0337	3608	3809	+4.1
16	5821	24562	.0105	80.7	406.42	200.3	18.13	229.2	55.6	.007292	.0347	3334	3513	+ .6
14	4949	20883	.0108	85.0	404.11	205.1	13.97	228.8	51.6	.007955	.0368	2829	2974	+1.5
15	4840	20422	.0106	79.8	407.16	204.8	13.41	229.4	50.3	.007940	.0374	2767	2914	+1.0
13	4262	17984	.0111	84.9	404.55	208.2	11.03	229.1	47.2	.008409	.0395	2432	2556	+1.6
12	3688	15561	.0112	84.5	404.94	211.0	8.80	229.1	43.6	.008976	.0421	2104	2207	+1.3
11	3188	13450	.0107	84.0	405.23	213.7	6.99	229.0	40.2	.009625	.0448	1819	1908	+1.7
10	2894	12212	.0107	80.7	405.90	215.0	5.96	228.7	38.5	.01015	.0465	1651	1732	+1.4
9	2545	10739	.0104	79.8	406.06	217.2	4.90	229.1	35.7	.01071	.0496	1452	1523	+1.4
8	2264	9551	.0098	78.4	406.18	218.5	4.10	228.9	33.4	.01124	.0527	1291	1355	+1.5
1	1956	8253	.0105	78.3	406.32	220.7	3.31	229.1	30.9	.01206	.0572	1114	1168	+2.2
7	1919	8098	.0097	77.6	406.32	220.7	3.15	229.0	30.5	.01212	.0565	1095	1146	+2.5
2	1683	7102	.0104	79.3	406.42	222.3	2.62	229.1	28.3	.01283	.0613	958	1005	+2.7
3	1472	6211	.0107	80.2	406.49	223.6	2.11	229.1	26.2	.01357	.0645	838	877	+3.2
4	1278	5391	.0107	80.8	406.54	224.5	1.664	229.1	23.8	.01423	.0676	726	760	+3.1
5a	1140	4809	.0127	82.4	407.98	224.9	1.392	229.0	21.9	.01456	.0715	648	678	+3.2
5	1102	4652	.0118	85.9	406.09	225.4	1.331	229.0	21.8	.01499	.0726	627	654	+3.0
6a	1039	4383	.0122	82.9	408.02	225.4	1.208	228.9	20.7	.01514	.0748	590	617	+2.8
6	968	4082	.0116	86.1	406.13	225.7	1.093	229.1	19.3	.01513	.0777	550	573	+3.4
Average Deviation														2.3

1977173

ISOTHERMAL FRICTION DATA - CORE 1/8 - 15.2 (Mfgs. No. DL-901-2257)

Run No.	Air Flow Rates		Core ΔP "H ₂ O @ 70°F	Upstream Air State			Friction Factor f	Reynolds' No. Re
	lbs/hr	Core Mass Velocity lbs/hr ft ²		Press. "H ₂ O abs. @ 70°F	Temp. °F	Humidity lbs/lb		
26c	8809	36402	38.90	401.64	80.5	.0109	.0479	7069
25c	8374	34605	34.83	402.06	79.7	.0109	.0478	6720
24c	7278	30073	26.41	403.12	79.8	.0108	.0487	5840
23c	6552	27074	21.64	403.75	79.5	.0110	.0498	5258
22c	5644	23324	16.17	404.44	79.5	.0111	.0506	4529
21c	4959	20491	12.60	404.88	79.2	.0113	.0515	3979
8c	4704	19436	11.40	402.83	82.5	.0113	.0512	3758
7c	4080	16863	8.82	403.20	82.5	.0115	.0530	3260
6c	3755	15518	7.58	403.39	82.3	.0112	.0540	3000
5c	3313	13692	6.00	403.60	81.9	.0110	.0551	2647
4c	2885	11920	4.59	403.80	81.8	.0111	.0558	2310
3c	2520	10413	3.68	403.92	81.8	.0110	.0589	2018
2c	2216	9160	2.882	404.01	81.4	.0110	.0598	1775
20c	1997	8253	2.406	404.12	83.1	.0107	.0615	1596
1c	1810	7480	2.025	404.18	81.0	.0107	.0634	1449
19c	1727	7137	1.877	404.21	83.2	.0108	.0644	1377
18c	1511	6244	1.501	404.28	83.0	.0108	.0675	1207
17c	1303	5385	1.173	404.33	83.0	.0108	.0712	1041
16c	1150	4753	.962	404.36	83.0	.0107	.0752	919
15c	981	4057	.755	404.41	83.0	.0107	.0813	784
14c	834	3447	.580	404.43	82.9	.0108	.0868	667
13c	735	3037	.478	404.45	82.6	.0108	.0924	587
12c	657	2714	.403	404.48	82.6	.0107	.0974	525
11c	574	2372	.330	404.49	82.5	.0107	.1052	459
10c	493	2036	.269	404.52	82.1	.0106	.1169	394
9c	385	1590	.192	404.51	81.8	.0105	.1375	307

2011/13

ISOTHERMAL FRICTION DATA - CORE 11.44 - 3/8R (Mfgs. No. DL-901-2255)

Run No.	Air Flow Rates		Core ΔP "H ₂ O @ 70°F	Upstream Air State			Friction Factor f	Reynolds' No. Re
	lbs/hr	Core Mass Velocity lbs/hr ft ²		Press. "H ₂ O abs. @ 70°F	Temp. °F.	Humidity lbs/lb		
27c	10925	43875	33.20	391.51	74.2	.0092	.0326	10451
26c	9680	38877	26.70	393.22	74.0	.0092	.0340	9281
25c	8591	34500	21.39	394.65	73.9	.0092	.0351	8237
24c	7548	30312	17.10	395.86	73.9	.0091	.0369	7237
23c	6590	26466	13.71	393.76	73.9	.0091	.0394	6319
22c	5643	22664	10.51	397.63	73.9	.0092	.0417	5411
21c	4782	19206	7.94	398.33	73.7	.0092	.0444	4585
20c	4666	18740	7.62	398.42	73.7	.0093	.0449	4474
19c	4143	16639	6.24	398.79	73.5	.0093	.0469	3972
18c	3629	14572	5.03	399.11	73.4	.0094	.0498	3479
17c	3299	13251	4.31	399.30	73.3	.0089	.0519	3164
16c	2873	11538	3.42	399.51	73.4	.0089	.0545	2755
15c	2530	10160	2.794	399.64	72.8	.0089	.0576	2426
14c	2212	8882	2.245	399.80	72.5	.0089	.0606	2125
12c	1982	7959	1.923	393.20	79.9	.0099	.0627	1883
13c	1867	7498	1.699	399.93	72.4	.0089	.0644	1794
11c	1729	6943	1.563	393.29	79.7	.0098	.0671	1643
10c	1481	5947	1.211	393.37	79.9	.0099	.0710	1407
9c	1370	5502	1.070	393.42	79.5	.0099	.0736	1302
8c	1134	4553	.788	393.47	79.5	.0099	.0797	1077
7c	961	3858	.599	393.54	79.5	.0099	.0848	913
6c	840	3372	.482	393.56	79.1	.0098	.0898	800
5c	738	2964	.389	393.57	79.1	.0098	.0941	703
4c	659	2645	.320	393.60	78.9	.0097	.0975	627
3c	570	2288	.249	393.62	78.7	.0097	.1017	543
2c	449	1804	.166	393.64	78.7	.0096	.1096	428
1c	370	1485	.120	393.65	78.4	.0095	.1174	352

217173

ISOTHERMAL FRICTION DATA - CORE - 17.8 - 3/8R (Mfgs. No. DL-901-2256)

Run No.	<u>Air Flow Rates</u>		Core ΔP "H ₂ O @ 70°F	<u>Upstream Air State</u>			Friction Factor f	Reynolds' No. Re
	lbs/hr	Core Mass Velocity lbs/hr ft ²		Press. "H ₂ O abs. @ 70°F	Temp. °F	Humidity lbs/lb		
6c	10026	42302	36.23	400.27	72.0	.0095	.0261	6637
5c	8557	36107	27.32	402.04	71.8	.0096	.0277	5665
4c	7481	31565	21.61	403.20	71.6	.0095	.0291	4952
3c	6667	28130	17.71	404.00	70.8	.0096	.0304	4423
2c	5960	25146	14.67	404.65	71.3	.0095	.0318	3945
1c	5014	21157	10.98	405.36	71.8	.0096	.0340	3319
25c	4831	20386	10.34	407.47	74.2	.0105	.0346	3184
24c	4214	17779	8.29	407.86	74.1	.0105	.0367	2783
23c	3774	15925	6.95	408.14	74.0	.0105	.0386	2493
22c	3305	13944	5.63	408.40	73.8	.0105	.0411	2183
21c	2937	12394	4.68	408.58	73.7	.0105	.0430	1940
20c	2573	10855	3.75	408.74	73.5	.0105	.0451	1699
19c	2224	9385	2.952	408.90	73.1	.0105	.0476	1469
17c	2033	8579	2.588	407.01	74.8	.0100	.0497	1340
18c	1922	8111	2.340	409.01	72.6	.0103	.0508	1270
16c	1749	7379	2.047	407.11	74.0	.0101	.0536	1155
15c	1512	6380	1.634	407.18	74.2	.0101	.0576	996
14c	1336	5636	1.344	407.25	73.9	.0102	.0610	882
13c	1178	4969	1.103	407.28	73.9	.0100	.0647	778
12c	980	4133	.832	407.35	73.9	.0100	.0710	647
11c	847	3572	.664	407.37	73.5	.0100	.0763	559
10c	760	3205	.558	407.38	73.0	.0098	.0799	502
9c	670	2828	.459	407.39	73.0	.0099	.0847	443
8c	580	2447	.369	407.41	72.8	.0099	.0914	383
7c	470	1984	.271	407.43	72.9	.0098	.1027	311

22-71173

SUMMARY OF BASIC HEAT TRANSFER AND FRICTION DATA

$(h/Gc_p)Pr^{2/3}$, f versus Re_f for smoothed curves

Core 1/8 - 15.2 (Mfgs. No. DL-901-2257)					Core 17.8 - 3/8R (Mfgs. No. DL-901-2256)				
Re_f	$(h/Gc_p)Pr^{2/3}$	f	h_{std}	E_{std}	$(h/Gc_p)Pr^{2/3}$	f	h_{std}	E_{std}	Re_f
6,000	.00850	.0487	128.7	1.769					6,000
5,000	.00896	.0498	113.0	1.047	.00675	.0293	106.6	1.20	5,000
4,000	.00959	.0516	96.78	.5554	.00740	.0320	93.5	.676	4,000
3,000	.01040	.0540	78.72	.2452	.00835	.0358	79.1	.319	3,000
2,500	.01110	.0558	70.01	.1466	.00900	.0385	71.1	.1986	2,500
2,000	.01177	.0584	59.39	.07858	.00982	.0421	62.0	.1112	2,000
1,500	.01267	.0628	47.95	.03565	.0110	.0478	52.1	.0532	1,500
1,200	.01327	.0676	40.18	.01965	.0120	.0530	45.5	.0302	1,200
1,000	.01373	.0726	34.64	.01221	.0129	.0579	41.1	.01911	1,000
800	.01427	.0800	28.80	.006889	.0142	.0643	35.6	.01087	800
600	.01520	.0913	23.01	.003317	.0158	.0738	29.9	.00526	600
500	.01580	.1010	19.93	.002123					500
400	.01675	.1145	16.90	.001232					400
300	.01810	.1390	13.70	.0006312					300

Core 11.44 - 3/8R (Mfgs. No. DL-901-2255)

Re_f	$(h/Gc_p)Pr^{2/3}$	f	h_{std}	E_{std}	Re_f
8,000	.00712	.0359	118.0	1.710	8,000
6,000	.00794	.0401	98.7	.8058	6,000
5,000	.00846	.0430	87.6	.5000	5,000
4,000	.00920	.0469	76.2	.2792	4,000
3,000	.01025	.0524	63.7	.1316	3,000
2,500	.0110	.0563	56.9	.08184	2,500
2,000	.0119	.0615	49.3	.04577	2,000
1,500	.0132	.0691	41.0	.02170	1,500
1,200	.0144	.0758	35.8	.01219	1,200
1,000	.0153	.0819	31.7	.007619	1,000
800	.0165	.0888	27.3	.004230	800
600	.0175	.0984	21.7	.001977	600
500	.0179	.1045	18.5	.001215	500

232123

FIGURE 1

PLATE-FIN SURFACE HEAT TRANSFER AND FRICTION DATA

CORE 1/8-15.2 (Mfgs. No. DL-901-2257)

Surface Data, Air Side:

Fin pitch - 15.2 per inch

Plate spacing - $b = 0.414$ inches

Flow passage hydraulic radius - $r_h = 0.00217$ ft.

Fin metal thickness - 0.006 in

Total transfer area / air side volume between plates -
 $\beta = 117 \text{ ft}^2/\text{ft}^3$

Fin area / total area = 0.873

o - Hot core test points. Heat transfer data evaluated on the basis of zero steam side resistance.

x - Cold core friction factors.

Best interpretation of friction data based on cold core data.

Best interpretation of heat transfer data based on an allowance for steam side resistance.

247173

FIG. 1

25 n 173

FIGURE 2

PLATE-FIN SURFACE HEAT TRANSFER AND FRICTION DATA

CORE 11.44-3/8R (Mfgs. No. DL-901-2255)

Surface Data, Air Side:

Fin pitch - 11.44 per inch
Plate spacing - $b = 0.413$ inches
Flow passage hydraulic radius - $r_h = 0.00265$ ft
Fin metal thickness - 0.006 in
Total transfer area / air side volume between plates -
 $\beta = 351 \text{ ft}^2/\text{ft}^3$
Fin area / total area - 0.847

o - Hot core test points. Heat transfer data evaluated on the basis of zero steam side resistance.

x - Cold core friction factors.

Best interpretation of friction data based on cold core data.
Best interpretation of heat transfer data based on an allowance for steam side resistance.

267173

FIG. 2
 2.7×10^{13}

FIGURE 3

PLATE-FIN SURFACE HEAT TRANSFER AND FRICTION DATA

CORE 17.8-3/8R (Mfgs. No. DL-901-2256)

Surface Data, Air Side:

Fin pitch - 17.8 per inch

Plate spacing - $b = 0.413$ inches

Flow passage hydraulic radius - $r_h = 0.00174$ ft

Fin metal thickness - 0.006 in

Total transfer area / air side volume between plates -
 $\beta = 514 \text{ ft}^2/\text{ft}^3$

Fin area / total area - 0.892

o - Hot core test points. Heat transfer data evaluated on the basis of zero steam side resistance.

x - Cold core friction factors.

Best interpretation of friction data based on cold core data.

Best interpretation of heat transfer data based on an allowance for steam side resistance.

287173

FIGURE 4

COMPARISON OF HEAT TRANSFER AND FRICTION CHARACTERISTICS

Reynolds number based on fin flow length or length of ruffle.

Surface 1/4-11.1 is a strip fin surface from ref. (b).

Surface 3/32-12.22 is a strip fin surface from ref. (c).

307173

314113

FIG 4

FIGURE 5

HEAT TRANSFER -- FRICTION POWER CHARACTERISTICS

Comparison on a unit area basis.

h_{std} and E_{std} evaluated from best interpretation of heat transfer and friction data for the following "standard gas" properties:

	$c_p = 0.2477 \text{ Btu/lb}^\circ\text{F}$
	$\mu = 0.0678 \text{ lbs/hrft}$
Air at 500 °F	$Pr = 0.671$
	$\rho = 0.0413 \text{ lbs/ft}^3$

$$h_{std} = h(c_{p_{std}}/c_p)(\mu_{std}/\mu)(Pr/Pr_{std})^{2/3} \text{ Btu/(hrft}^2\text{)}^\circ\text{F}$$

$$E_{std} = E(\rho/\rho_{std})^2(\mu_{std}/\mu)^3 \text{ fhp/ft}^2$$

32-1173

337173

FIGURE 6

HEAT TRANSFER -- FRICTION POWER CHARACTERISTICS

Comparison on a unit volume basis.

β = transfer area per unit of air side volume.

$\beta_{h_{std}}$ Btu/(hr °F ft³ of volume)

$\beta_{E_{std}}$ fhp/(ft³ of volume)

Surface 3/8-11.1 is a louvered fin surface from ref. (f).

Surface 19.86 is a plain fin surface from ref. (g).

347173

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DISCUSSION

Previous reports, refs. (c) and (f), have demonstrated the basic advantage of employing interrupted heat transfer surfaces to reduce the boundary layer dimensions and thereby to improve the heat transfer--friction power characteristics of the surface. The results of tests of finned-flat-tube surfaces, ref. (h), demonstrated that a similar improvement in performance is possible without interrupting the surface if the surface is ruffled, or wavy. The ruffled surface is believed for some applications to be superior to the interrupted surface from a fouling-tendency point of view, since there are no jagged fin edges for the accumulation of deposits. For this reason it may be possible to employ surfaces with ruffled fins on the gas side of the gas turbine regenerator with a reduced fouling rate, and thus have a high performance surface on the gas side where ordinarily, due to the low density fluid, the heat transfer coefficients are low and the friction power expenditure high. In studies of gas turbine regenerator designs, ref. (i), it was shown that much of the advantage of high performance surfaces is lost if their use is restricted to the air side only. The three surfaces considered here include two examples of wavy fins and one of interrupted, or strip fins, so that it is possible to compare the performance of the two basic types and determine whether or not the performance of interrupted fins can be matched by ruffled fins. In the discussion following, the heat transfer and friction performance of the three surfaces will be examined separately in order to determine at least qualitatively the mechanisms involved, and finally the overall performance will be compared.

Heat Transfer Results

The heat transfer parameter, $StPr^{2/3}$, for the three surfaces is plotted as a function of a Reynolds number based on the hydraulic diameter of the flow passage, Re_h , Figs. (1), (2), and (3). This method of presentation is used only in order to be consistent with the presentation of the results of previous tests and does not indicate that the hydraulic diameter is necessarily the most characteristic dimension of the surface. In fact the correlation between the two ruffled fin surfaces, 11.4-3/8R and 17.8-3/8R, is very poor when compared on a hydraulic diameter basis.

On Fig. (4) the three surfaces are compared using a Reynolds number in which the length dimension is the fin flow length (1/8 inches) in the case of the strip-fin surface, and the length of relatively straight fin (3/16 inches) in the case of the two ruffled fin surfaces. Superimposed upon this representation are the characteristics for flow over a single flat plate with sharp leading edge and a laminar boundary layer, and the characteristics for two strip-fin surfaces previously reported in refs. (b) and (c). Considering the very different geometry of these surfaces, the similarity of the heat transfer characteristics appears significant, especially when comparing the three surfaces considered here with the flat plate characteristics. A similarity of basic heat transfer mechanisms is indicated, which in the case of the ruffled fin surfaces may mean that the boundary layer breaks down at each change in

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surface direction and a new laminar boundary layer builds up along the relatively straight section. The actual flow is probably a good deal more complex than this. Because of fin curvature, a thin boundary layer would be expected on the concave side of the fins and either a thick boundary layer or separation on the convex side. The correlation of the two ruffled fin surfaces on this basis indicates that the fin spacing transverse to the flow direction, at least for the range of fin spacings considered, is not a significant variable.

The reason for the change of slope of the $StPr^{2/3}$ curve for surface 11.44-3/8R at $Re_f = 700$ ($Re_f = 1100$) is not apparent. Because of test system limitations, it was not possible to test surface 17.8-3/8R at sufficiently low Reynolds numbers to determine whether this surface would have the same characteristics. This may be due to a shift in the point of boundary layer separation at the curved portion of the fins, or it may have something to do with the complex flow pattern at the fin roots. It should be noted here that this discussion is of necessity qualitative since from 10 to 20 percent of the transfer area is wall or prime area subject to much more complex flow mechanisms than the fin area at the center of the flow tube.

The poor correlation between the strip-fin surface, 1/8-15.2, and the two previously tested strip-fin surfaces, 1/4-11.1 and 3/32-12.22 may be attributable to the character of the leading and trailing edges of the fins. Surface 1/8-15.2 had very jagged and bent over edges due evidently to a dull cutting tool. The effect of a blunt leading edge as compared to a sharp leading edge was shown in ref. (e) to increase $StPr^{2/3}$ and decrease the slope of the $StPr^{2/3}$ vs. Re curve in a manner very similar to the way in which these curves compare.

The heat transfer data of Figs. (1), (2), and (3) is recommended for use with identical or geometrically similar surfaces, although the reliability of the data for surface 1/8-15.2 may depend to a certain extent upon the character of the fin edges. For interpolation between the geometry of surfaces 11.44-3/8R and 17.8-3/8R, as well as for small extrapolations of geometry of all the surfaces, the use of Fig. (4) is recommended since the dimension l appears to be most characteristic of the surface.

Friction Results

The friction factor curves for surfaces 11.44-3/8R and 17.8-3/8R (Figs. 2 and 3) are nearly parallel to their respective heat transfer curves which would seem to indicate a similarity between the momentum and heat transfer mechanisms for these surfaces. On the other hand the lack of such similarity for the strip-fin surface, 1/8-15.2, indicates that there are parasitic friction losses in this case which are undoubtedly eddy losses contributed by the leading and trailing edges of the fins.

The friction factor characteristics for the two ruffled fin surfaces do not correlate at all well with each other on a hydraulic diameter basis. On a ruffle or wave length basis, Fig. (4), the correlation is much better but the discrepancy is still 15-20 percent which would bespeak caution in using these data for extrapolation to variations in geometry. It is difficult to account for the very high friction factors for these surfaces, since if they

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APPENDIX

Steam Side Resistance

The steam side unit film conductance, h_m , has been evaluated analytically using the analysis of Nusselt as given in ref. (1). The following assumptions are made:

- (a) The entire resistance occurs in a film of condensate on the walls of the steam side surface.
- (b) The condensate film flows in laminar motion.
- (c) The condensate film is moved by the force of gravity and the drag of the excess or "blow" steam.

The results of this analysis are only an approximation, but since the steam side resistance is only a small percentage of the total resistance (< 10 percent), high accuracy in this estimate is not necessary to render the $StPr^{2/3}$ results accurate to within ± 5 percent as estimated in ref. (a). The following is a tabulation of h_m as a function of the steam side condensate rate for the three cores reported here.

w_g lbs/hr	h_m Btu/(hrft ² °F)
25	5250
50	4000
75	3400
100	3000
150	2750
200	2550
250	2300

Temperature Effectiveness of the Surface

The method of evaluating η_o , the temperature effectiveness of the surface, is explained in Technical Report No. 2; ref. (a). Since η_o differs for each test core, it is not given in the appendix to Report No. 2. The following tabulation gives the values used in these tests.

$h \approx U$	η_o 1/8-15.2	η_o 11.44-3/R	η_o 17.8-3/8R
0	1.000	1.000	1.000
5	.985	.986	.985
10	.970	.971	.970
20	.943	.945	.941
30	.919	.921	.915
40	.896	.899	.892
50	.874	.878	.870
60	.853	.858	.849
70	.835	.840	.829
80	.819	.823	.811

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Entrance and Exit Coefficients

The entrance and exit coefficients, K_c and K_e , have been evaluated using the data of ref. (k). The following tabulation gives the values of K_c and K_e for the three test cores reported here as a function of Reynolds number, Re_f .

Re_f	K_c	K_c	K_c
	1/8-15.2	11.44-3/8R	17.8-3/8R
300	0.317	0.760	0.780
600	(constant)	0.760	0.780
1000		0.760	0.780
1500		0.730	0.748
2000		0.620	0.604
3000		0.433	0.434
4000		0.406	0.412
6000		0.398	0.405
8000		0.393	0.400
10000		0.390	0.395

Re_f	K_e	K_e	K_e
	1/8-15.2	11.44-3/8R	17.8-3/8R
300	0.318	0.150	0.180
600	(constant)	0.150	0.180
1000		0.150	0.180
1500		0.157	0.193
2000		0.193	0.243
3000		0.270	0.300
4000		0.288	0.310
6000		0.293	0.314
8000		0.296	0.319
10000		0.300	0.320

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