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HEAT TRANSFER AND PRESSURE DROP
CHARACTERISTICS OF DRY TOWER EXTENDED SURFACES

Part II: Data Analysis and Correlation

Prepared for

Dry Cooling Tower Program
Battelle Memorial Institute
Richland, Washington

under Special Agreement BSA-988

June 15, 1976

PFR ENGINEERING SYSTEMS, INC.
Marina del Rey, California

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INTRODUCTION

This report is a summary of the heat transfer and pressure drop characteristics of extended surfaces. The summary is based on the analysis of data that were compiled and reported in Part I of this project. The data was analyzed with the objectives of determining the effects of geometrical configurations of an extended surface tube bank on its heat transfer and pressure drop characteristics.

A generalized correlation of heat transfer and friction losses was developed on the basis of this analysis. These correlations can be used to predict the performance of an extended surface tube bank of specified geometry and flow conditions.

A criteria was developed which compares extended surfaces on a common basis. This criteria can be used for a comparative evaluation of surfaces and determine their relative merit. Although the final selection of an extended surface for a specified function will be determined by cost consideration, this criteria may be used for preliminary conceptual analysis. Graphs were prepared that illustrate the relative merit of various extended surfaces for dry tower applications.

Section 1 contains a discussion on the data sources, its classification, and the role of the theoretical fin efficiency cal-

ulation on the accuracy of the convective heat transfer coefficient.

Section 2 describes the effect of geometrical and design parameters on the heat transfer and pressure drop of the extended surfaces.

Section 3 presents the correlations and in Section 4 the comparative criteria of extended surfaces is developed.

1.0 HEAT TRANSFER AND FLOW CHARACTERISTICS

Heat transfer and pressure data was collected from various sources and presented in a uniform way in a previous report (31). This data was used for an analysis of the effect of the geometrical configurations of extended surfaces on their heat transfer and pressure drop analysis. The heat transfer data that was reported in the literature was not based on a consistent definition of the heat transfer coefficient. In some reports the apparent coefficient, including fin resistance or an overall coefficient, was presented. In order to facilitate the analysis, the data was classified and converted to convective heat transfer coefficients. A qualitative discussion on the data is given in the following.

1.1 Data Classification. Three basic configurations of extended surfaces were analyzed:

- a. Round tube, helical round smooth fins
- b. Round tube, helical round serrated/cut fins
- c. Round tube, continuous plate smooth fins

The bulk of the reported experimental data pertains to the above three types of extended surfaces. There also exists scattered data on different surfaces such as, crimped fins, elliptical tubes with rectangular fins, wire fins, and round or continuous fins with various surface modifications. These additional data are not

presented in a manner that would lend itself to a quantitative analysis. Consequently, only general qualitative observations were made here.

The experimental data for the surfaces classified in the above three configurations is based on tests in which the geometry was varied in a consistent manner. For example, data was reported for variations in the tube and fin diameters, fin height and number of fins per unit length, and other geometrical variables. The effect of such variation of the heat transfer and friction loss of an extended surface tube bank can thus be quantitatively studied. However, some differences also exist in the data for these surfaces. These differences exist because of minor variations in the finned tubes. For example, a helical smooth round fin can be of identical size and yet differ by the method of attachment to the tube. The fins can be extruded, welded, L-footed wound, etc. Assuming no bond resistance, these manufacturing differences between a fin of identical size (i.e., tube and fin diameters, number of fins per unit length, etc.) result in a difference of approximately between five to ten percent in their heat transfer and friction drop characteristics. Similar differences were found between the data on serrated or plate fin tubes. The fact that the data is collected from various sources employing different experimental techniques, compounds the problem and adds to the uncertainty. It is thus safe to conclude that the data of various sources differs by an average of $\pm 8\%$ even for finned tubes of identical overall geometries and

thus an ideal correlation will still have this inherent error spread.

1.2 Fin Effectiveness. Heat transfer data from various tests was reported as heat transfer coefficients. There was no consistency in these reported coefficients since some of them included the fin conduction resistance. In order to facilitate a rational analysis and correlation of the heat transfer characteristics of extended surfaces, the data was converted to a convective heat transfer coefficient with the aid of the fin effectiveness concept. The relation between the convective and apparent heat transfer coefficient is given as follows:

$$h_o = (h_c)(\eta_c)$$

1

$$\eta_c = 1 - \left(\frac{A_f}{A_t} \right) (1 - \eta_f)$$

where;

h_o is the apparent coefficient (including fin resistance)

h_c is the convection coefficient

η_c is the fin effectiveness

η_f is the fin efficiency

$\left(\frac{A_f}{A_t} \right)$ is the ratio of fin to total surface area

The fin efficiency η_f is a function of the fin dimension, fin thermal conductivity, and the convection heat transfer coefficient. A theoretical fin efficiency can be computed from the solution of the heat conduction equation. A compilation of such solutions for fins of various shape and form is given by Kern and Kraus (28).

The theoretical fin efficiency for helical round smooth fins is based on a solution in which the convective heat transfer is assumed to be uniform on the surface of the fin and equal to the average coefficient h_c . It is evident from considerations of flow characteristics across a tube bank that this assumption is incorrect. There exists experimental evidence in the literature that the non-uniformity of the heat transfer coefficient causes the fin efficiency to be lower than its theoretical value. Kuntysh et al (17) and Pshenishnov and Luzhnov (10) reported on heat transfer data for finned tubes of identical geometry and different material. The data reported is in the form of an apparent coefficient with the fin resistance included. The adequacy of the theoretical fin efficiency concept can be determined from this data.

Figures 1 and 2 show the J-factor convective coefficient as calculated from the data with the aid of the theoretical fin efficiency. The convective coefficient so calculated should be identical for all the finned tubes since they are identical in their geometry and method of manufacturing. Figures 1 and 2 show that

FIGURE 1: COMPARISON OF CONVECTIVE COEFFICIENTS FOR FINS OF DIFFERENT MATERIAL. DATA OF REFERENCE 10.

FIGURE 2: COMPARISON OF CONVECTION COEFFICIENTS FOR FINS OF DIFFERENT MATERIAL. DATA OF REFERENCE 14.

(Reynolds Number defined with base diameter)

there exists a difference between the convective J-factor lines for the different fin materials. This disparity is an indication that the theoretical fin efficiency for round helical smooth fins is deficient in converting apparent heat transfer coefficients to convective coefficients. From Figures 1 and 2 it can be concluded that an actual fin efficiency, i.e., a fin efficiency that can be used to calculate the conduction resistance of the fin, is always lower than the theoretical fin efficiency. As the theoretical fin efficiency decreases, i.e., by employing fins of a low thermal conductivity, the disparity between theoretical and actual fin efficiency also increases. A separate analysis is required to determine the deviation of actual fin efficiency from theoretical fin efficiency as a function of the flow conditions and fin geometry.

In analyzing data which is reported as an apparent coefficient, we found that when the theoretical fin efficiency was above 0.75 the difference between theoretical and actual fin efficiency was less than 8%. Thus, for the correlations development, only data with a theoretical fin efficiency above 0.75 was used. Since most of the finned tubes were manufactured with fins made of aluminum, which has high conductivity, only 15% of all the data was not included in the correlational work.

The extended surface data was analyzed in terms of the effects of the various geometrical configurations of the surfaces on the heat transfer and pressure loss. The next section is a discussion of these effects.

2.0 EFFECT OF EXTENDED SURFACE CONFIGURATIONS ON THEIR PERFORMANCE

Finned tube banks are constructed by arranging individual tubes into a module of different layout and number of rows. The finned tube can have different base diameters with fins of different size and form. The arrangement of the tube bank and the tube and fin geometry, affect both heat transfer and pressure drop. These effects are analyzed in the following.

2.1 Tube Bank Layout, Staggered vs. In-Line Arrangement.

A schematic diagram of the basic layout of a tube bank is given in Figure 3.

FIGURE 3

The only constructional difference between these two arrangements is that in the staggered arrangement each alternate row is shifted half a transverse pitch. The finned tube itself can be of various shapes and sizes for example, smooth fins, cut fins, continuous plate fins, etc. The staggered and in-line arrangement will differ on account of the flow dynamics. In an in-line arrangement the extent of transverse flow components are small in comparison to a staggered arrangement. A comparison of the characteristics of heat transfer and pressure drop of finned tubes in a staggered and in-line layout can be made on the basis of tests reported by Brauer (4), Carnavos (22), and Rabas and Eckels (23).

Figures 4, 5, and 6 show a plot of J and f-factors for tube banks of identical finned tubes arranged in staggered or in-line arrangements. Reynolds number is defined with the base tube diameter. These plots are essentially nondimensional presentation of measured heat transfer and pressure loss coefficients. Figures 4 and 5 show that the staggered arrangement results in heat transfer coefficients which are about 50% higher than the coefficients for in-line arrangement. The pressure drop is approximately 60% higher. This difference is more pronounced when employing serrated finned tubes as shown in Figure 6. For identical operating conditions, the heat transfer coefficients of staggered arrangement are about 125% higher than for in-line arrangement whereas the pressure drop is only about 80% higher. It is thus safe to conclude that within the range of flow conditions and finned tube variables reported here, staggered arrangement is superior to in-line arrangement.

FIGURE 4: COMPARISON OF "J" FACTOR, INLINE AND STAGGERED ARRANGEMENT.

FIGURE 5: COMPARISON OF F-FACTOR BETWEEN INLINE AND STAGGERED ARRANGEMENT

FIGURE 6: COMPARISON BETWEEN STAGGERED AND INLINE ARRANGEMENT FOR SERRATED FINNED TUBES. DATA OF REFERENCE 23.

No data exists for a comparison between in-line and staggered arrangement for continuous plate fins. However, from scattered data by Cox (27) the conclusion from round helical fins apply.

2.2 Effect of Number of Rows. A module or a bank of finned tubes can be arranged with varying number of tube rows in the cross flow direction. The addition of a row of tubes in the cross flow direction is a common way of increasing heat transfer area when the face or approach area of the tube bank is fixed. Such an addition of area increases the length of the flow path and may affect the dynamics of the flow with possible effects on heat transfer and pressure drop.

Information on the effect of the number of rows is reported by Brauer (3), Cox (26), Jameson (11), Mirkovich (20), Ward and Young (9), and Weirman et al (24). The row effect was studied experimentally in two ways;

- a. An average heat transfer coefficient is measured for a tube bank. The number of rows in each test is varied. This method was employed by Brauer (3), Cox (26), Jameson (11), and Weirman et al (24).
- b. The overall number of rows is fixed. The heat transfer coefficients are measured by measuring the heat fluxes and temperature difference across each row. This method was employed by Ward and Young (9) and Mirkovich (20).

Ward and Young (9) condensed steam in the tubes to heat air flowing across. Mirkovich used water to cool hot air flowing across.

Figures 7 to 10 show the test results of the row's effect as presented by Brauer. These graphs display a basic difference for the row effect between an in-line and a staggered arrangement. For an in-line arrangement, the heat transfer coefficient decreases with an increase in the number of rows. The one row bank heat transfer coefficient is approximately (50 - 80%) higher than the average for four rows. For a staggered arrangement, the four row tube bank is approximately 30 % higher than a one row tube bank.

The pressure drop decreases with the number of rows. For a staggered arrangement, the average pressure drop per row remains constant for two or more rows. This is in agreement with data by Jameson (11) who found identical pressure drop per row for a tube bank with two, four and eight rows.

Ward and Young (9) and Mirkovic (20) measured individual coefficients of each row in an eight row staggered tube bank. Both reported that the heat transfer coefficient increases with the row number. At the third row the flow stabilizes and the coefficient remains constant from the third row onwards. Ward and Young (9) found that the coefficient of the first row is about .85 - .65 of the coefficient of the last row, the second row is about .95 of the last row for a range of maximum velocities between 300 and 1600 ft/min.

FIGURE 7: THE EFFECT OF NUMBER OF ROWS ON AVERAGE J-FACTOR FOR A TUBE BANK IN AN INLINE ARRANGEMENT. DATA OF REFERENCE 3.

FIGURE 8: THE EFFECT OF NUMBER OF ROWS ON AVERAGE J-FACTOR FOR A TUBE IN STAGGERED ARRANGEMENT.

FIGURE 9: THE EFFECT OF NUMBER OF ROWS ON FRICTION FACTOR FOR AN INLINE ARRANGEMENT.

Friction
Factor
per Row

FIGURE 10: THE EFFECT OF NUMBER OF ROWS ON FRICTION FACTOR FOR TUBE BANK IN STAGGERED ARRANGEMENT.

Mirkovic (20) reports that the first row is approximately .66 and the second row about 0.84 of the last row for a maximum velocity of 1200 ft/min. These individual heat transfer coefficients can be used to develop a general formula for the effect of the number of tube rows on heat transfer.

$$Q_N = \frac{1}{N} \sum_{i=1}^N Q_i \quad 2$$

where,

Q_i is the heat flux of row i

Q_N is the total average heat flux

N is the total number of rows

Equation 2 can be written as:

$$(\bar{h}_N)(\bar{\Delta T}_m)_N = \frac{1}{N} \sum_{i=1}^N h_i (\Delta T_m)_i \quad 3$$

where,

$\bar{h}_N$ is the average outside heat transfer coefficient

h_i is the outside heat transfer coefficient of each row

ΔT_m is the mean temperature difference between the tube wall and the cross flowing gas

In the special case when the tube side arrangement is counter-flow to the outside flowing gas and the capacity ratio is equal to one, the mean temperature difference is identical for all rows and equal to the average ΔT_m of the tube bank. Equation 3 can be written as,

$$\bar{h}_N = \frac{1}{N} \sum_{i=1}^N h_i$$

4

This equation was used by Ward and Young (9) and Mirkovic (20) to generate graphs for the effect of the number of rows.

Figure 11 shows the effect of the number of rows on the average heat transfer coefficient as was given by Ward and Young (9) and Mirkovic (20). The Ward and Young (9) lines apply for tube banks arranged in a tight pitch, with the clearance between fins in the range of 10 - 20% of the pitch. The Mirkovic (20) lines apply to wider pitches with clearance between the fins of approximately 50% of the pitch between the tubes. These graphs apply only for pure countercurrent with capacity ratio of one.

The other extreme to the above condition is a pure cross flow with isothermal heat transfer of tube side fluid such as boiling or condensation. In this case, the mean temperature difference of the rows decreases with the number of rows and the first row will exhibit the highest heat flux even though it has the lowest coefficient. In this case the effect of the number of rows on the average

FIGURE 11: RATIO OF AVERAGE COEFFICIENT OF TUBE BANK WITH N ROWS OVER INFINITE NUMBER OF ROWS.

coefficient is small and is evident from Brauer's results; a bank with three or more rows has identical average outside coefficient.

2.3 Effect of Tube Pitch. Tube pitch is another variable commonly used in design to affect surface area and flow conditions. Its effect on the heat transfer and pressure drop is important especially for optimization studies. The effect of the tube pitch was studied in the works of Briggs and Young (7), Carnavos (22), Jameson (11), Kuntysh and Iokvedov (15), Mirkovic (20), and Robinson and Briggs(8).

Jameson (11) carried out an extensive study on the effect of the tube pitch on heat transfer and pressure drop. Figure 12 shows the results of Jameson's tests on one finned tube arranged in five various pitch layouts. The data is presented as a J-factor vs. a Reynolds number. This Reynolds number is based on the maximum flow velocity across the tube bank and a hydraulic diameter. Figures 13 14 show the same results using a Reynolds number which is based on the base tube diameter. It can be seen from Figure 13 that all the heat transfer data falls on a single line - no effect of the tube pitch is evident. The pressure drop decreases with an increase in the tube pitch. Jameson's (11) tests show that the pitch has no effect on the heat transfer coefficients on the basis of a Reynolds number defined with the base tube diameter and maximum flow velocities. The seemingly tube pitch effect shown in Figure 12 stems only from the effect of the pitch on the definition of the hydraulic diameter and

REFERENCE; Jameson

 $Re_h \times 10^{-3}$
 (Based on hydraulic diameter)

Tube outside diameter = 0.774 in.

Fin outside diameter = 1.463 in.

Fin pitch = 9.05 per in.

Fin thickness = 0.0120 in.

Number of rows in flow direction = 8

Heat transfer area / bare tube area = 10.107

	A	B	C	D	E
Flow passage hydraulic diameter	= 0.202 in.	0.322 in.	0.535 in.	0.21 in.	0.253 in.

Free flow area / frontal area	= 0.455	0.572	0.688	0.537	0.572
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Heat transfer area / frontal area	= 122.07	95.068	68.145	68.145	95.07
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Tube Material / Fin Material	= Admiralty/Copper
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Helically wound "L" type fins soldered to tube.

Convective heat transfer coefficient.

FIGURE 13: JAMESON'S HEAT TRANSFER DATA FOR VARIOUS TUBE PITCHES.

FIGURE 14: FRICTION COEFFICIENT F AS A FUNCTION OF THE PITCH FROM JAMESON DATA.

not on the actual flow conditions. The results by Jameson agree completely with similar studies by the other investigators, namely - no effect of the pitch on the heat transfer coefficient when described as a function of the maximum velocity but a strong effect on the pressure drop.

Similar effects were shown by Kuntysh and Iokvedov for cut fins. However, no data exists for this effect on continuous plate fins. Briggs and Young (7) found no difference in the heat transfer coefficients of the two tube banks arranged with a fin tip clearance of 3/16 in. and 3.5 in. identical maximum velocities. Carnavos (22) shows no effect of the tube pitch on the heat transfer with a strong effect on the pressure drop. Kuntysh and Iokvedov (14) show no effect of the pitch for closely spaced finned tubes. Robinson and Briggs (8) show that the pressure drop decreases with an increase in the tube pitch for identical maximum flow velocities.

2.4 Effect of Finned Tube Variables. The finned tube variables consist of fin height, spacing, and thickness. An increase in surface area of the tube bank can be achieved by employing finned tubes with higher fins and denser spacing. Increasing the height of the fin will not only decrease its fin effectiveness but may result in lower convective coefficients than on shorter fins. Denser fins, i.e., increase in their number per unit tube length may affect the heat transfer by laminarization of the flow conditions between the fins.

Carnavos (22) investigated the effect of the fin spacing on both heat transfer and pressure. Figures 15 & 16 show the performance of three tube banks with the tubes having 16, 24, and 30 fins/inch. Figures 15 & 16 show that the heat transfer decreases and the pressure drop increases with the increase in the density of the fins. Figures 15 & 16 also show that the difference in performance decreases with an increase in the number of fins and may become independent of fin density. The fin height in these experiments remained unchanged and equal to 3/16 inch.

Briggs and Young (7) and Kuntysh and Iokvedov (15) have determined from analysis of their data that the fin height over fin spacing ratio is a dominant variable affecting both the heat transfer and the pressure drop.

Rich (19) investigated schematically the effect of fin density on the performance of a continuous plate finned tube. The fin density varied from 2.92 to 20.6 fins/inch. Figures 7 and 8 show the data by Rich as presented in the standard form using hydraulic diameter. Figure 19 shows the J-factor as a function of the Reynolds number defined with the base tube diameter. Figure 20 shows the friction factor F as a function of the Reynolds number. Figure 19 shows that the fin spacing has a small effect on the heat transfer; J-factor decreases by about 15% as the number of fins/inch increased from 2.92 to 20.6. This effect is quite smaller than the effect of fin spacing on round fins which may be partially due to the fact that the plate fin retards the development of fully three-dimensional turbulent flow.

FIGURE 15: EFFECT OF THE NUMBER OF FINS/INCH ON HEAT TRANSFER

FIGURE 16: EFFECT OF NUMBER OF FINS/INCH ON FRICTION FACTOR F

REFERENCE; Rich

 $Re_h \times 10^{-3}$
(Based on hydraulic diameter)

Tube outside diameter = 0.525 in.

Fin outside diameter = _____

Fin pitch = _____ per in.

Fin thickness = 0.060 in.

Number of rows in flow direction = 4

	A	B	C	D	E
Heat transfer area / bare tube area =	11.65	13.7	17.24	21.1	29.6
Flow passage hydraulic diameter =	.157"	0.133"	.109"	.084"	.0574"
Free flow area / frontal area =	.551	0.545	.536	.525	.502
Heat transfer area / frontal area =	58.41	68.85	86.46	106.0	148.4
Tube Material / Fin Material =	CU/CU	CU/CU	CU/CU	CU/CU	CU/CU

Round tube, continuous plate smooth tube expanded fin
Convective heat transfer coefficient

REFERENCE; Rich

 $Re_h \times 10^{-3}$

Tube outside diameter = 0.525 in. (Based on hydraulic diameter)

Fin outside diameter = _____

Fin pitch = _____ per in.

Fin thickness = 0.060 in.

Number of rows in flow direction = 4

	A	B	C
Heat transfer area / bare tube area =	5.78	7.14	10.26
Flow passage hydraulic diameter =	.379 in.	.266 in.	.182 in.
Free flow area / frontal area =	.569	.563	.555
Heat transfer area / frontal area =	28.98	35.78	51.45
Tube Material / Fin Material =	CU/CU	CU/CU	CU/CU

Round tube, continuous plate smooth expanded fin
 Convective heat transfer coefficient

FIGURE 19: RICH⁽¹⁹⁾ DATA OF FIGURES 18 & 19 AS A FUNCTION OF REYNOLDS NUMBER BASED ON TUBE DIAMETER

FIGURE 20: COMPARISON OF FRICTION FACTORS FOR CONTINUOUS FINS
(DATA FOR RICH⁽¹⁹⁾).

2.5 Finned Tubes with Surface Modifications. Data was compiled on the performance of various fin tube surfaces with "surface modification." The term surface modification refers to finned tubes of which the extended surfaces are mechanically "deformed" to various shapes or forms. In this category we include; crimped and corrugated fins, cut, slotted or serrated fins, and continuous plate fins with surface interruptions.

Taranyan et al (16) and Kuntysh et al (17) studied the effect of cutting round helical fins on the resultant heat transfer and pressure drop. Figures 21 & 22 show a comparison between the performance of cut and smooth fins. Both the heat transfer and the pressure drop for cut fins are higher than for the equivalent smooth fin. In reviewing all the data from these reports, the pressure drop of cut fins is approximately 40% higher and the heat transfer is 25% higher than the equivalent smooth fin for identical maximum flow velocities.

Preece et al (6) compared the performance of a round smooth L-footed fin with an identical fin tube which was modified by slotting and "crippling" the surface. Preece et al (6) found that these surface modifications resulted in an increase of pressure drop by an average of 70% while the heat transfer increased only nominally by 10%. This agrees closely with experiments by Vampola (12) who found a doubling of pressure drop with a nominal increase in the heat transfer for finned tubes which the fin surface was crimped. Scattered data on other types of surface modifications are given by Preece et al (6), Kuntysh et al (17), and Cox (25,26,27).

FIGURE 21: COMPARISON OF "J" FACTOR BETWEEN A SMOOTH AND CUT FIN

FIGURE 22: COMPARISON OF F - FACTOR BETWEEN A SMOOTH AND CUT FIN

It can be concluded that any surface modification will increase the pressure drop as compared to the identical smooth fin. However, some modification such as serrated or cut fins will show an increase in heat transfer, other types of surface modifications will show a nominal or no increase in heat transfer.

2.7 Conclusions. The analysis of the data which was presented in the previous sections can be summarized as follows:

- * Theoretical fin efficiency for round fins is deficient in properly accounting for the conduction resistance of the fins. It is still possible to use the fin efficiency concept, however, an actual efficiency is lower than the theoretical efficiency. If the theoretical fin efficiency is higher than 0.75, the correction is less than 8%.
- * Staggered arrangement displays higher coefficients than in-line arrangement with a correspondingly higher pressure drop.
- * In a staggered tube bank of a fixed number of rows, the first two rows have lower heat transfer coefficients than the rest of the rows. From the third row onwards, the heat transfer coefficient remains unchanged. The average pressure drop per row of a staggered tube bank with two or more rows remains unchanged with the increase in the number of rows.

The effect of the number of rows is more pronounced in the in-line arrangement. For serrated tubes, the row effect continues up to the seventh row.

- * Tube pitch has no effect on the heat transfer coefficient for the same maximum velocities. The pressure drop is strongly affected by the pitch. It decreases with an increase in the tube pitch.
- * Finned tube variables, i.e., fin height and spacing, affect both the heat transfer and pressure drop of helical round smooth fins. For serrated or continuous plate fins, this effect is greatly reduced.
- * Any surface modification of a smooth fin will tend to increase the pressure drop. In some cases, the corresponding increase in heat transfer is nominal.

3.0 HEAT TRANSFER AND PRESSURE DROP CORRELATIONS

Because of the number of geometric variables and the complex flow conditions, the correlation of the performance of extended surfaces is a formidable task. In this work, an attempt for a mathematical modeling of the momentum and energy transfer in the tube bank did not result in reliable correlations. A different approach was thus pursued; variables which affect the performance were lumped in dimensionless form and a regression analysis of the data was performed. The analysis of the previous section served as a guideline for selecting the correlational form and the meaningful variables which affect the performance. These correlations are discussed in the following.

3.1 Heat Transfer - Staggered Arrangement.

- a. Round Tube, Helical Round Smooth Fins. For these tubes the analysis showed that when the heat transfer coefficient is expressed as a function of a Reynolds number defined with base diameter, no effect of the tube pitch is evident. The only geometrical variables affecting the heat transfer were the fin tube variables.

The fin tube variables were lumped into the area ratio variable, Ar which expressed the ratio of the overall extended surface area to the area of the base tube. Ar can be calculated from the following formula;

$$Ar = 1 + 2(N_f)(l_f)\left(1 + \frac{l_f + t_f}{d_r}\right) \quad 5$$

where,

N_f is the number of fins/inch

l_f is the fin height

t_f is the fin thickness

d_r is the base tube diameter

Figure 23 shows the range of the J-factor as a function of Reynolds number for all of the finned tube data. Figure 23 shows that the data from various sources falls on single lines according to the respective Ar ratio. From this observation the following general form was used for the correlation:

$$J = C Re^{-m} Ar^{-n} \quad 6$$

A regression analysis of all the data yielded the following results:

$$J = 0.29 Re^{-0.367} Ar^{-0.17} \quad 7$$

This correlation predicts 80% of the data within $\pm 10\%$ and 95% of the data within $\pm 18\%$.

b. Serrated or Cut Fins. The correlation for these fins is:

$$J = 0.195 \text{ Re}^{-.3} \text{ Ar}^{-.17}$$

8

80% of the data was within $\pm 15\%$ and 95% of the data within $\pm 20\%$.

The correlation is valid for the following range:
Module with six rows.

$$1000 < \text{Re} < 40,000$$

$$\frac{3}{8}'' \leq \text{Tube diameter} \leq 2''$$

$$300 \left[\frac{\text{ft}}{\text{min}} \right] \leq \text{Flow velocity} \leq 1500 \left[\frac{\text{ft}}{\text{min}} \right]$$

$$4.0 < \text{Ar} \leq 34$$

$$.0175 \text{cp} \leq \text{Viscosity} \leq .021 \text{cp}$$

A graphical presentation of the correlation for smooth and serrated fins is given in Figure 24.

c. Continuous Plate Fins. The data for continuous plate fins showed little effect of the finned tube variables. The data by Rich (19), Cox (27), and the Trane Company data which was presented by Kays and London (1) was correlated with the following formula:

$$J = 0.13 \text{ Re}^{-.33}$$

9

80% of the data were predicted within $\pm 12\%$.

FIGURE 23: THE VARIATION OF J-FACTOR WITH THE AREA RATIO Ar BASED ON DATA FROM VARIOUS SOURCES.

FIGURE 24: HEAT TRANSFER CORRELATION FOR SMOOTH AND CUT FINS

3.2. Inline Arrangement. No sufficient data exists for a complete correlation. From the existing data the general recommendation is as follows:

$$\text{Smooth Fins} \quad J_{\text{inline}} = 0.67 J_{\text{staggered}}$$

$$\text{Serrated Fins} \quad J_{\text{inline}} = 0.5 J_{\text{staggered}}$$

The error for these formulas is estimated as $\pm 20\%$

3.3 Pressure Drop - Staggered Arrangement.

a. Round tube, helical round smooth fins. The pressure drop is both a function of the tube pitch and the finned tube variables. The friction factor is not a simple function as the f vs. Re line changes with Reynolds number. Furthermore, it was found that the hydraulic diameter is more suitable than the base diameter for correlation. The following correlation was derived:

$$F = (150 Re_h^{-1.0} + 1.8 Re_h^{-0.2}) (P_\ell / d_h)^{0.35} \quad (P_\ell / d_h)^{>4.0}$$

10

F is the friction factor defined as follows:

$$F = \frac{\Delta P / N_t}{\rho_{av} G_{max}^2 / 2g_c}$$

11

where,

ΔP is total tube bank pressure drop.

N_t is number of tube rows in flow direction.

G_{max} is the maximum mass flow velocity based on free flow area.

P_ℓ is the longitudinal pitch.

ρ_{av} is the average density of the air in the tube bank.

g_c is the gravitational constant 32.2.

Re_h is the Reynolds number defined on the basis of hydraulic diameter d_h .

$$Re_h = \frac{(G_{max})(d_h)}{\mu_{av}}$$

μ_{av} is the average viscosity of the air in the tube bank.

The range of variables for this correlation is:

$$400 < Re_h < 10,000$$

$$1.5 \leq \frac{P_\ell}{d_r} \leq 3.8$$

For all the cases when $(P_\ell/d_h) \leq 4.0$ a different correlation was developed:

$$F = 13.6 Re_h^{-.3} (P_\ell / d_h)^{-.42} \quad (P_\ell / d_h) \leq 4.0 \quad 12$$

b. Serrated or Cut Fins. Multiply the smooth fin correlation by a factor of 1.4.

c. Continuous Plate Fins.

$$F = (110 Re_h^{-1.0} + 1.35 Re_h^{-.2}) (P_\ell / d_h)^{.35} \quad 13$$

3.4 Inline Arrangement. No sufficient data exists for correlation. However, the inline friction factor can be as follows:

$$f_{inline} = 0.6 f_{staggered}$$

This is valid for both smooth and serrated finned tubes.

4.0 COMPARATIVE EVALUATION OF EXTENDED SURFACES

The multiplicity of finned tube variables poses the problem of their relative merit and the selection of a proper finned tube for a specific application. In any industrial application the selection of finned tubes is dictated by the combination of operating and capital costs as well as a variety of technical constraints. For a finned tube module, the capital cost is primarily related to the size of the equipment whereas, the operating cost is primarily related to the fan and pumping power.

In dry tower application, modules of finned tubes are being constructed as standard units whose size is dictated by shipping constraints. Normally, a module of 14.5 ft. wide and 80 ft. long is presently considered maximum size from shipping considerations. These modules though, can be constructed with finned tubes of various shape and form having various tube diameters, fin height and spacing, tube pitch, layout and number of rows. Each module will transfer a different amount of heat load as a function of the auxiliary fan and pumping power and water flow rate.

A measure of the efficiency of a module of a particular design will be its heat load under identical operating conditions. This efficiency must be compared on the basis of the cost of the module and thus identical operating criteria is not necessarily the final economic criteria. Consequently, the formulas derived in this section show the parametric variation of the performance of extended surface modules with changes in their design.

The general equation of heat transfer is written as follows:

$$Q = (MCp)_{air}(ITD)(p) \quad 14$$

where,

Q is the heat load of a module.

(MCp) is the product of the air flow rate and its heat capacity.

ITD is the initial temperature difference.

p is the effectiveness of the module.

The effectiveness p is a function of the number of transfer units NTU, capacity ratio R and the pass arrangement.

$$NTU = \frac{(U)(A)}{(MCp)_{air}} \quad 15$$

U is the overall coefficient and A is the total surface area.

$$R = \frac{(MCp)_{water}}{(MCp)_{air}} \quad 16$$

For a given module, the following relations exist for heat transfer and pressure drop:

$$h_{\text{water}} = C_1 M_{\text{water}}^{0.8} \quad 17$$

$$\Delta P_{\text{water}} = C_2 M_{\text{water}}^{1.8} \quad 18$$

$$h_{\text{air}} = C_3 M_{\text{air}}^{\alpha} \quad 19$$

$$\Delta P_{\text{air}} = C_4 M_{\text{air}}^{\beta} \quad 20$$

The constants in the above equations can be compiled from the correlation for flow inside tubes and across extended surfaces as developed in Section 3.

The pumping power E can be similarly expressed in a general form:

$$E_{\text{air}} = C_5 M_{\text{air}}^{1+\beta} \quad 21$$

$$E_{\text{water}} = C_6 M_{\text{water}}^{2.8} \quad 22$$

From the above formulas, the following general expression applies for a module of prescribed design:

$$\frac{Q}{ITD} = \text{function} (HP, M_{\text{water}}) \quad 23$$

HP is the auxiliary power - the sum of pumping and fan power.

The above analysis was programmed to yield parametric results of equation 23 for any type of extended surface. Figures 25 to 33 show the results of the program for a bay containing three modules constructed with helical, smooth round fins. These graphs were developed for modules with four, five, and six rows with two passes each.

The following observations can be made:

1. When the number of rows remains unchanged, an increase in pitch will result in a decrease in the heat load.
2. As the number of rows increases, the heat load increases. This is a consequence of the fact that more heat transfer area is added. However, the benefit of increasing the number of rows is reduced as the number of rows increases. This is because the tube side velocities decrease for constant water flow rate.
3. The heat load increases as the water flow rate per bay increases especially for high values of pumping power.

A comparison was made of the performance of three modules constructed with three different tubes. 5/8 inch, 1 inch, and 2 inch base diameter finned tubes were used (see enclosed graphs). The 5/8 inch module was constructed with four rows one pass, whereas, the 1 inch and 2 inch tubes were constructed as four rows two pass.

The comparison is tabulated below for a water flow rate of 2.16×10^6 lb/hr water flow per bay.

HORSEPOWER	(Q/ITD) $\times 10^6$ (BTU/hr $^{\circ}\text{F}$)		
	5/8"	1"	2"
300	1.03	1.14	1.04
400	1.09	1.2	1.1
500	1.13	1.24	1.13

The table shows that the module constructed with 1 inch tubes is superior to the others. In a cost analysis we made for each module, it was determined that the 2 inch tube module costs about 13% more than the 1 inch module, and the 5/8 inch module costs about 7% less than the 1 inch module.

A final analysis on the optimization of the tube modules must be done within the context of power plant economics. Nevertheless, these attached graphs serve as an important guideline on the comparative evaluation of extended surfaces for dry tower application.

FIGURE 25

HEAT LOAD IN DRY TOWER MODULE VS, AUXILIARY POWER, WATER FLOW RATE AS PARAMETER.

FIGURE 26

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER. WATER FLOW RATE AS PARAMETER.

FIGURE 27

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER. WATER FLOW RATE AS PARAMETER.

FIGURE 28

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER, WATER FLOW RATE AS PARAMETER.

FIGURE 29

HORSEPOWER PER BAY

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER. WATER FLOW RATE AS PARAMETER.

FIGURE 30

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER, WATER FLOW RATE AS PARAMETER,

FIGURE 31
HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER. WATER FLOW RATE AS PARAMETER

FIGURE 32

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER. WATER FLOW RATE AS PARAMETER.

FIGURE 33

HEAT LOAD IN DRY TOWER MODULE VS. AUXILIARY POWER, WATER FLOW RATE AS PARAMETER,

NOMENCLATURE

A_r	The ratio of extended surface area to base tube area.
C_1, C_6	Constant in equations 17 - 22.
d_r	Base tube diameter.
d_h	Hydraulic diameter.
E	Fan or pump power.
f	Friction factor defined on the basis of hydraulic diameter.
F	Friction factor defined in equation 11.
G_{max}	Maximum mass flow velocity through the constricted area of the tube bank.
h	Heat transfer coefficient.
M	Mass flow rate.
N	Total number of rows in flow direction.
p	Heat exchanger effectiveness.
ΔP	Total tube bank pressure drop.
P_ℓ	Longitudinal pitch in flow direction.
Re	Reynolds number defined with base tube diameter.
Re_h	Reynolds number defined with hydraulic diameter.
Q	Heat load.

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