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SHORTER COMMUNICATIONS

PRESSURE DROP AND LOCAL HEAT TRANSFER IN AN IN-LINE, PARALLEL-TUBE HEAT EXCHANGER WITH INTER-TUBE FINS: AN EXPERIMENTAL STUDY

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NOMENCLATURE

- A, area;
- A', area per unit length in x-direction;
- c_p, specific heat at constant pressure;
- D, outside tube diameter;
- f, friction factor $\Delta P/[n(G^2/2\rho)]$;
- G, mass velocity per unit of minimum flow area;
- h, heat-transfer coefficient;
- $\bar{h}$, heat-transfer coefficient averaged over total heat-transfer surface;
- k, thermal conductivity;
- k', dimensionless thermal conductivity, (k_{metal}/k_{fluid});
- L, length of heat-transfer surface per tube row;
- m, mass flow rate;
- n, number of tube rows;
- P, static pressure;
- Pr, Prandtl number, $(c_p\mu/k)$;
- Re, Reynolds number, (GD/μ);
- S, pitch of tubes;
- St, Stanton number, $(\bar{h}/Gc_p)$;
- T, temperature;
- x, distance along heat-transfer surface in the direction of flow.

Greek symbols

- μ, absolute viscosity;
- ρ, density;
- η, ratio of total heat transferred by heat exchanger to that transferred if fins were of negligible thermal resistance;
- δ', dimensionless fin thickness, (δ/L).

Subscripts

- C, minimum flow cross section;
- G, bulk-gas condition;
- L, flow direction;
- S, surface condition;
- T, cross-stream direction.

1. INTRODUCTION

1.1 *Purpose of the present work*

THE COOLING tubes lining the furnace chamber of many recently constructed, large boilers are joined by fins, running parallel to the tube axes. The fins are welded to the tubes to ensure good thermal contact and to eliminate leakage, this process being made economical by modern resistance-welding techniques.

In the case of large combustion chambers, the heat transfer is almost entirely by radiation. But the resistance-welding methods could, of course, be applied to fabricate a low cost, in-line, parallel-tube heat exchanger with the tubes separated by fins oriented in the direction of flow. This geometry is sketched inset in Fig. 1. One of the aims of the present study is to determine the overall performance of such a heat exchanger and to compare it with that of the conventional, plain-tube heat exchanger.

The second objective of the present work is to obtain reliable data about the local heat-transfer rates for the finned-tube geometry. Considerable research is currently in progress towards providing a theory for the prediction of two-dimensional separated flows, see [1] for example. Reliable experimental information about local quantities is required to properly test the theory and this is particularly scarce.

2. DESCRIPTION OF EXPERIMENT

2.1 *Choice of tube pitch*

The performance of the plain-tube heat exchanger improves as the inter-tube space is reduced [2]. Practical difficulties associated with the fastening of tube ends preclude, however, the use of very small tube spacings in a commercial heat exchanger. These considerations led to the choice of inter-tube pitches, in both the flow (S_L) and cross-flow (S_T) directions, of $1\frac{1}{2}$ tube diameters for the finned-tube heat exchanger of the present work.

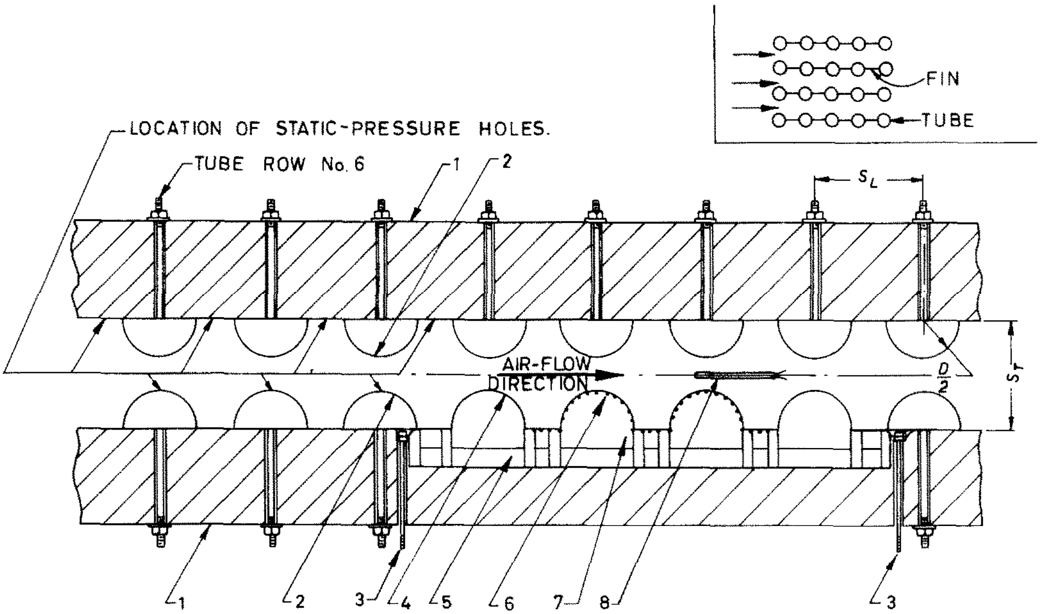

FIG. 1. Diagram of test section. 1. Insulation board; 2. Aluminium dummy tube; 3. Bus-bar terminal; 4. Stainless-steel heater; 5. Perspex ribbing; 6. Thermocouple; 7. Glass-fibre insulation; 8. Bulk-fluid temperature probe.

2.2 *Apparatus*

A typical flow path through the finned-tube heat exchanger was simulated using the apparatus shown in Fig. 1.

Aluminium half-tubes, $\frac{3}{4}$ in. dia., were secured to upper and lower sheets of insulation. Seven static pressure tappings were positioned, as shown, between the fifth and ninth tube rows downstream of the flow inlet. Heat-transfer measurements were effected, on the lower side, over the length comprising the subsequent three tube rows. The heat-transfer surface was formed from 0·005 in. thick, stainless-steel sheet, heated by the passage of an electric current. Copper–constantan thermocouples were glued to the undersurface of the stainless steel.

Heat-transfer coefficients were determined, at each thermocouple location, from measurements of the local heat-transfer rate and the difference between the temperature of the heated sheet and the bulk temperature of the air. The local heat-transfer rate was equal to the electrical-dissapation rate, which was constant over the heated sheet, corrected for conduction along the sheet and for heat lost downwards to the insulation; both of these corrections were small.

The difference in the heat-transfer coefficients for the two instrumented tubes was always less than the estimated experimental error of 5 per cent. Also, during development of the apparatus, experiments were made with the top side of the channel heated. This was found not to influence the heat-transfer coefficients obtained on the bottom side and was abandoned.

3. EXPERIMENTAL RESULTS

3.1 *Average heat transfer and pressure drop*

In Fig. 2, curves of friction factor and Stanton number versus Reynolds number for the present geometry are compared with those obtained by Pierson [3] for the plain-tube configuration. The finned-tube data lie 20–40 per cent below the plain-tube curves.

It is to be expected that, due to their damping effect, the incorporation of inter-tube fins will extend the range over which quasi-laminar flow is evident; this is indeed the case. Bergelin *et al.* [4] reported that turbulent flow for a plain-tube heat exchanger is fully established at a Reynolds number of about 5×10^3. The shapes of the curves in Fig. 2 for the finned-tube arrangement reveal the existence of transitional flow up to a Reynolds number of about 10^4.

3.2 *Local heat transfer*

The variation, along the length of a typical section, of the local, heat-transfer coefficient is given in Fig. 3 for a wide range of Reynolds numbers. Because of the finite separation of the temperature measuring stations along the heat-transfer surface, the probable sharpness of the peaks and valleys

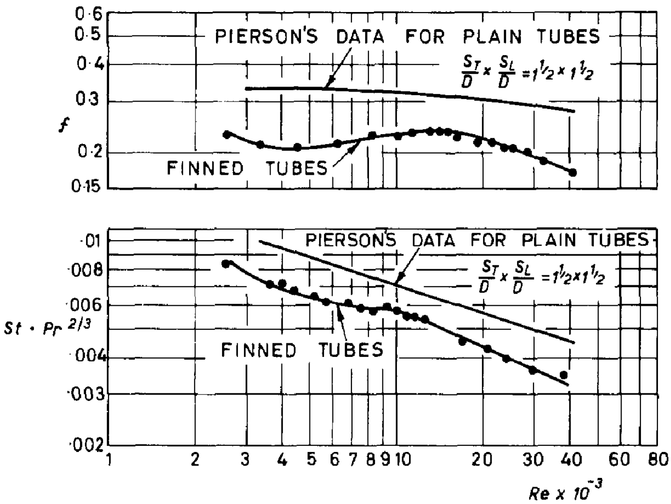

FIG. 2. Comparison of pressure drop and heat transfer of finned and plain tubes.

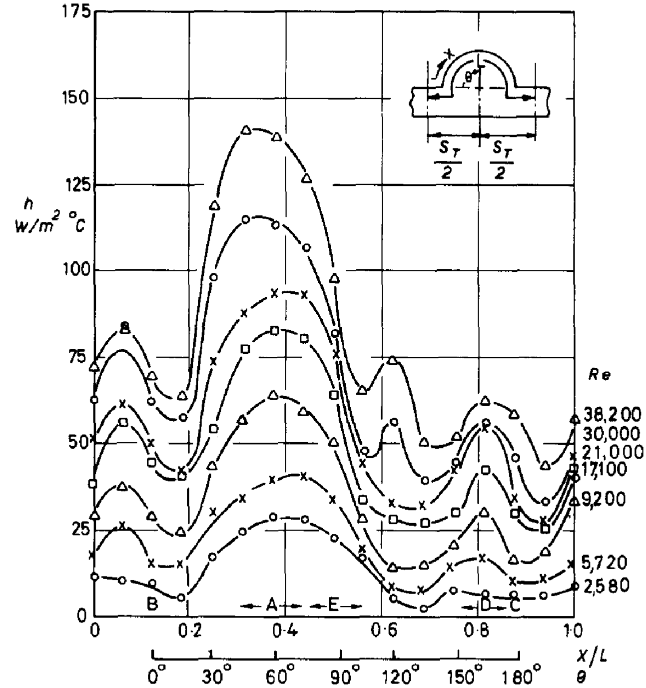

FIG. 3. Local heat-transfer coefficients.

is blurred. The shapes of the curves suggest the streamline pattern sketched in Fig. 4.

The large heat transfer at point A is associated with the mainstream stagnation streamline. Small peaks in the heat-transfer rate near B and D are indicative of two further stagnation streamlines shared by the main-vortex flow and secondary vortices in the corners B and C. At large Reynolds numbers, a small increase in heat transfer is evident just downstream of E. This corresponds, it is felt, to the establishment of a secondary vortex between the main flow and the main vortex in this region; such a vortex has not been sketched on Fig. 4.

Thus, for a specified duty, the required surface area is inversely proportional to the product of Stanton and Reynolds numbers.

Similarly, a one-dimensional treatment of the pressure loss shows that, for a given duty:

$$\Delta P_{\text{total}} \propto \frac{A_C f Re^2}{A_S' S_L St}.$$

The quantities $1/(St\,Re)$ and $(A_C f Re^2)/(A_S' S_L St)$ have been used in Fig. 5 to compare the performances of the finned-tube and plain-heat heat exchangers. We see that, for

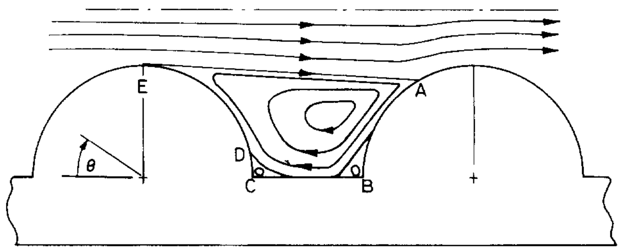

FIG. 4. Sketch of streamlines.

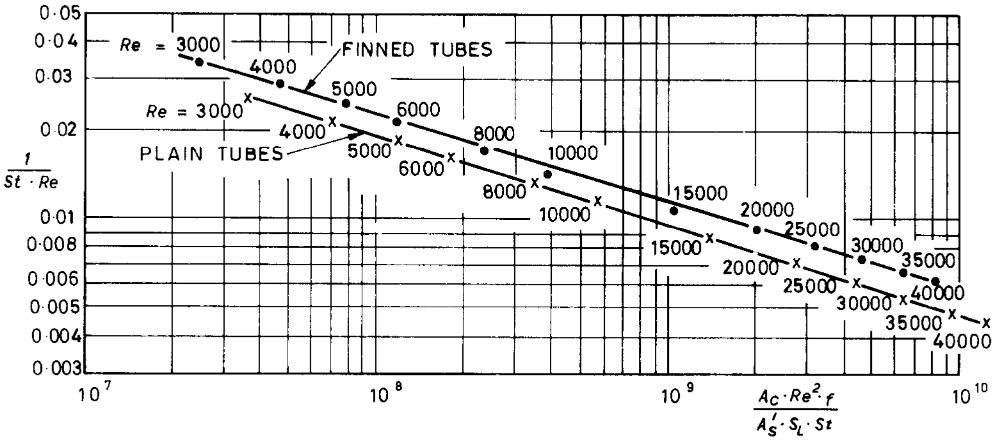

FIG. 5. Performance comparison of finned and plain-tube heat exchangers.

4. DISCUSSION OF PERFORMANCE

4.1 *Comparison of finned-tube and plain-tube geometries*

Consider the steady, one-dimensional flow of gas through a heat exchanger. An energy balance leads to:

$$h(T_G - T_S)\,A_S'\,\mathrm{d}x = -\dot{m}c_p\,\mathrm{d}T_G.$$

Assuming, for ease of illustration, that T_S is constant, this equation yields, on integration, the total required heat transfer surface area:

$$A_{\text{total}} = \frac{\dot{m}D}{\mu}\ln\frac{T_{G,\text{in}} - T_S}{T_{G,\text{out}} - T_S}\,\frac{1}{St\,Re}.$$

a given pressure drop, the finned-tube configuration requires some 17–22 per cent more surface area to transfer the same amount of heat.

4.2 *Effect of thermal resistance of fin*

The thermal resistance of the fins in the ideal fin-tube heat exchanger would be negligible that is, the temperature of the fins would be the same as that of the tubes. The performance of the real heat exchanger will decrease as the thermal resistance of its fins is increased; and for constant fin efficiency, the performance will decrease with increasing load as characterized by Reynolds number.

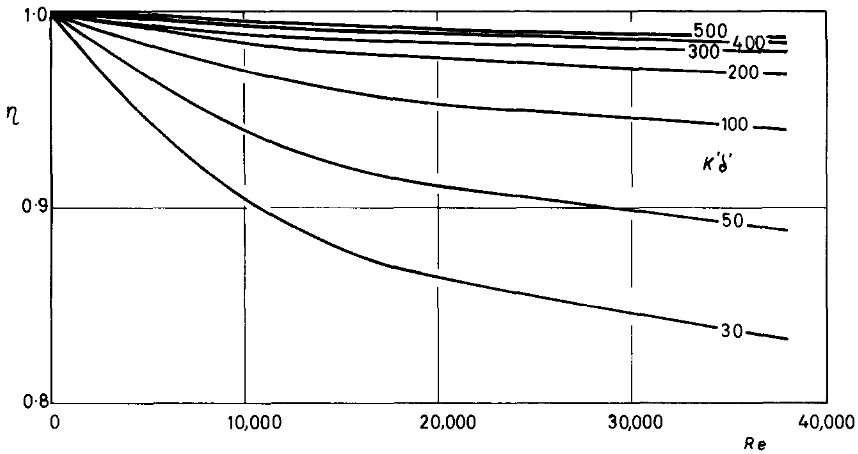

FIG. Effect of fin efficiency on heat exchanger performance.

These effects have been determined from the computer solution of the fin equation using the measured local heat-transfer coefficients; they are displayed by Fig. 6. The ordinate, η, is the ratio of the heat transferred by the real heat exchanger to that transferred by the ideal one.

5. CONCLUSIONS

The local heat-transfer data which have been obtained provide further material for testing recent theoretical procedures for predicting two-dimensional separated flows.

The greater constructional complexity of the finned-tube heat exchanger, relative to the conventional plain-tube heat exchanger, is not compensated by improved performance. The performance of the former is, in fact, significantly inferior to that of the latter. The fin-tube arrangement does not appear, therefore, to be a commercially attractive proposition.

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HEAT AND MASS TRANSFERS IN SOLID HELIUM

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NOMENCLATURE

b, a constant;
t, time;
x, distance.

Greek symbol
α, thermal diffusivity.

Abbreviations
b.c.c., body centred cubic;
h.c.p., hexagonal close packed;
L, prefix used before a chemical symbol to denote the appropriate liquid.

INTRODUCTION

HELIUM forms a most interesting solid *at low densities* be-