where

$$\theta = \frac{T - T_w}{T_\infty - T_w}; \quad P = \frac{T}{T_w}; \quad f = \frac{\psi}{\sqrt{\left(\frac{\mu_w X U_\infty}{\rho_w}\right)}};$$

$$\eta = Y\sqrt{\left(\frac{\rho_w U_\infty}{\mu_w X}\right)}$$

$$U_\infty \propto X^m; \quad m = \frac{X\left(\frac{dU}{dX}\right)}{U_\infty};$$

the prime denotes differentiation with respect to η and following the experimental work of Keenan and Kays [9]

$$\mu \propto T^{0.7}; \quad K \propto T^{0.85}; \quad C_p \propto T^{0.19}; \quad \rho \propto \frac{1}{T}.$$

The boundary conditions are

$$\eta = 0; \quad f = 0; \quad f' = 0; \quad \theta = 0$$

and, as the edge of the boundary layer is approached

$$\theta \to 1; \quad \theta' \to 0; \quad \theta'' \to 0$$

and

$$f \to \infty; \quad f' \to \frac{T_w}{T_\infty}, \quad f'' \to 0.$$

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TEMPERATURE DISTRIBUTIONS AND HEAT-TRANSFER CHARACTERISTICS OF A VEE-FIN ARRAY

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NOMENCLATURE

b,	half thickness of fin;
h,	mean convective film coefficient over fin surface;
k,	thermal conductivity;
Nu,	mean Nusselt number;
Re,	Reynolds number;
T,	surface temperature;
T_1,	temperature at the end of fin;
T_s,	the free stream temperature of air;
ϑ,	temperature difference $(T - T_s)$;
ϑ_1,	temperature difference $(T_1 - T_s)$.

INTRODUCTION

THE DESIGN of a compact heat exchanger relies on the high surface area density, which is obtained by compact heat-transfer surfaces. The one used extensively in radiator design is the plate fin design with triangular or square passages using interconnected fins [1].

The vee-fin design is favored for radiators, since it is easy to manufacture. A long strip of metal is formed into appropriate shape and then attached to the flattened tubes, so forming a continuous array of short triangular passages. This triangular passage consists of the flattened tube (primary surface) and the two lateral sides, i.e. the fins or the secondary surfaces.

The heat flux from the secondary surface, i.e. fin, is a function of the surface temperature of the fin, the fluid temperature and the flow characteristics. To evaluate the fin surface temperature distribution the fin material conductivity, the film coefficient and the fin geometry must be known.

The purpose of this investigation was to obtain more information about temperature distribution and film coefficient on the surface of a fin, forming a part of the triangular passage and exposed to air flow.

EXPERIMENTAL APPARATUS

The test section of 0·3048 × 0·1524 m (12 × 6 in) and 0·0254 m (1 in) deep consisted of an array of three triangular passages, each 0·1524 m (6 in) high with base of 0·1016 m (4 in). The fins separating the passages were 0·002 m ($\frac{5}{64}$ in) thick, heated at the base and at the apex of the passage.

The central triangular passages were used for temperature measurements.

The thermocouples were fixed at 0·019 m (0·75 in) intervals alternatively at top and bottom surface of the two fins forming the central triangular passage.

Each thermocouple was soldered into a small slit at the surface. The surface was then sanded smooth so as not to create a disturbance to the flow. The thermocouple leads were neatly tucked to form the trailing edge of the fin.

The test section was mounted into the test rig, consisting of a long rectangular duct, forming the experimental section. The base and the apex of the triangular array of test section were flush with the rectangular duct. The wall temperatures T_1 were measured at the base and at the apex of the triangular section. The first reading was taken after stable conditions had been maintained for 15 min and after interval of further 10 min the second reading.

The tests were made for four different wall temperatures for each of which six Reynolds numbers conditions were employed.

The differences between any two readings at the same point were of the order of approximately 0·25°C.

THEORY AND EXPERIMENTAL RESULTS

Considering the radiator geometry as an array of short triangular passages, the sides of the passages being the fins. Each fin can be presented as rectangular plate of length $2l$ and thickness $2b$ of uniform thermal conductivity, k, receiving the heat from two heat sources located at ends and at same temperature T_1. The heat is dissipated from its two surfaces to an ambient air at temperature T_s. Assuming the value of mean film coefficient, h, to be the same for the two surfaces of the fin, the surface temperature, T, is given by the equation

$$\vartheta = \vartheta_1 \frac{\cosh mx}{\cosh ml} \tag{1}$$

where

$$\vartheta = T - T_s, \quad \vartheta_1 = T_1 - T_s \quad \text{and} \quad m = h/2kb \quad [2\text{–}4].$$

The surface temperature distribution was measured for the

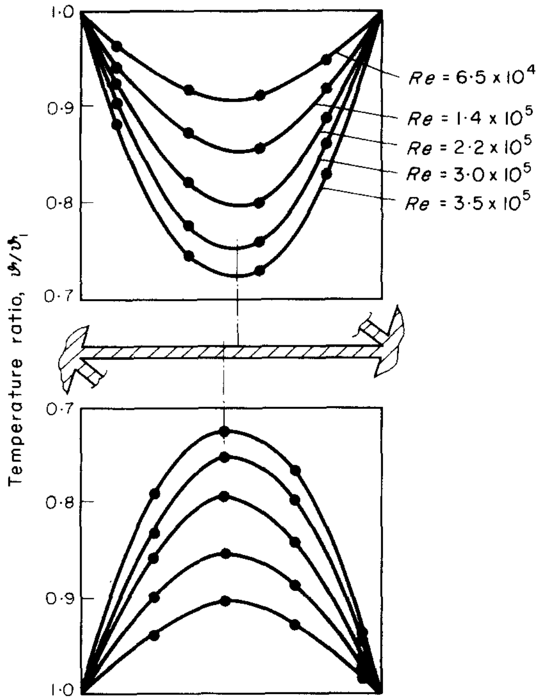

FIG. 1. Temperature distribution on the surface of the fin.

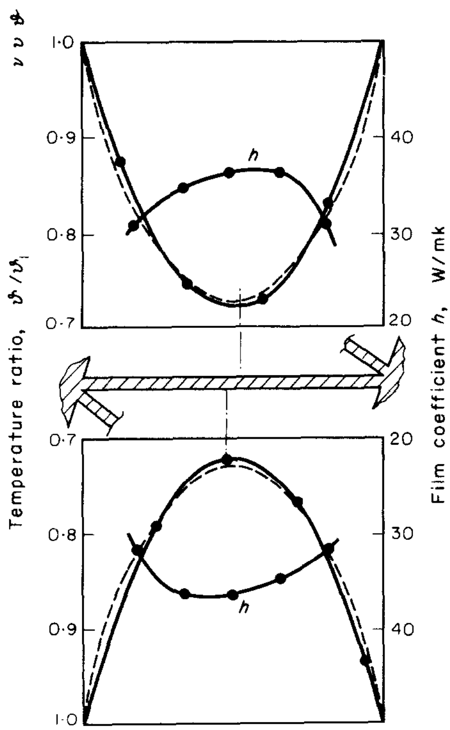

FIG. 2. Theoretical and experimental temperature distributions and the local film coefficient for $R_E = 3{\cdot}5 \times 10^5$.

two fins, forming the sides of the triangular passage. Both surfaces were considered in each fin, i.e. the internal surface, being part of the central triangular passage and the external surface, which forms part of the next triangular passage. The results show, that for a given base temperature, T_1, and Reynolds number, the surface distribution in the two fins is identical over the whole length of the fin. The surface temperature distribution on one surface is the mirror image of the surface temperature on the other surface, since the two surfaces of the same fin are exposed to the flow in two identical but inverted triangular passages.

Using a non-dimensional temperature difference $\vartheta/\vartheta_1 = (T-T_s)/(T_1-T_s)$, the surface temperature curves are independent of the base temperature and were plotted as function of Reynolds number only.

In all cases, the minimum surface temperature for a given surface was recorded at the same point, independent of Reynolds number effect. The larger values of Reynolds number resulted only in lower surface temperature (Fig. 1). The velocity distribution along the wall of a triangular passage is neither uniform nor symmetrically distributed [5–7]. The local film coefficient and the temperature distribution reflect this variation. Since the two surfaces of the fin are exposed to the flow in two identical but inverted triangular passages, and the temperature distribution on the two surfaces is identical but inverted, results that the position of minimum temperature on one surface does not correspond to the position of minimum temperature on the other surface.

From the experimental temperature distribution the values of local film coefficients have been calculated and are shown on Fig. 2, jointly with the theoretical and experimental temperature distribution for one $Re = 3{\cdot}5 \times 10^5$. The mean value of the film coefficient, whilst not predicting accurately temperature at any point, will give a satisfactory value of overall heat dissipation by the whole fin.

The mean film coefficients were calculated from the values of the local film coefficients for each Reynolds number. Using the mean film coefficients, the Nusselt number was obtained, based on bulk flow, and valid for Reynolds numbers $Re = 6{\cdot}5 \times 10^4$ to $Re = 3{\cdot}8 \times 10^5$, which is (Fig. 3)

$$Nu_{\mathrm{Fin}} = 0{\cdot}01085\,Re^{0{\cdot}8}. \tag{2}$$

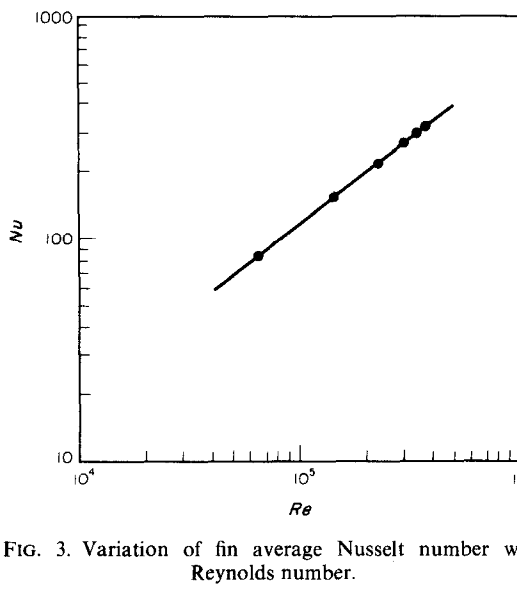

FIG. 3. Variation of fin average Nusselt number with Reynolds number.

Comparing the results with those of other researchers in the field of heat transfer in triangular ducts, it should be borne in mind, that the bulk of the references deal with tests in long ducts with fully, or nearly fully developed flow and in all cases, either uniform wall temperature or uniform heat flux. In the present case only a short triangular passage was used and the heating was only at the two ends of the fin. Furthermore, the value for the Nusselt number expressed in the equation (2), does not apply to the whole passage, but only to the surface of the fin.

It should be pointed out here, that the value of Nusselt number defined by equation (2) can at this stage be accepted as valid only for a fin in an array of triangular passages with apex angle of 36·9°. Any change of the geometry of the triangular passage may change the flow and film coefficient distribution along the wall and consequently the value of mean film coefficient.

Further testing would have to be carried out on a range of apex angles and fin geometries to establish a more universal validity.

CONCLUSIONS

1. The temperature distribution on the surface is not symmetrical and the position of minimum temperature on the surface is independent of the Reynolds number.
2. The temperature distribution on the two surfaces are identical, but are of mirror image form, thereby creating a displacement between the positions of minimum temperatures on the two surfaces.
3. The value of the average Nusselt number for the fin has been found to be $Nu = C\,Re^{0.8}$, the value of the constant being $C = 0.01085$. The validity of this Nusselt number correlation for different geometries requires further investigation.

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A TWO-DIMENSIONAL ANALOG SIMULATING THE HELMHOLTZ EQUATION FOR HEAT FLOW

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NOMENCLATURE

A_f,	fin surface area [m²];
A_m,	mean area between *iso*-potentials [m²];
C,	a constant;
E,	electrostatic field strength [V/m²];
h,	heat-transfer coefficient [W/m²K];
i_z,	current density in the z direction [A/m²];
I,	current [A];
k,	thermal conductivity [W/mK];
P,	constant voltage potential [V];
q,	flow;
r,	radius [m];
S,	current density [A/m²];
t,	paper thickness [m];
V,	voltage potential [V];
w,	characteristic fin length [m].

Greek symbols

γ,	electrical conductivity [Ω^{-1}/m];
δ,	fin thickness [m];
δ_l,	laminate thickness [m];
θ,	potential;
ϕ,	fin efficiency;
λ,	a constant;
μ,	a constant;
∇^2,	two-dimensional Laplacian operator $= \dfrac{\partial^2}{\partial x^2} + \dfrac{\partial^2}{\partial y^2}$.

Subscripts

0,	fin root;
m,	mean;
x, y, z,	refers to directions of Cartesian co-ordinate axes.