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Natural convection heat transfer on finned tubes in air†

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Abstract—In an experimental investigation temperature fields and mean heat transfer coefficients were obtained on a finned tube. The diameter of the fin was varied in order to find out about the effect of fin height. Such an effect on the heat transfer coefficient vanishes in a correlation given in the literature. By thermovision the usefulness of the fins can be observed: the smaller fin proved to be better than the larger one under given conditions. The heat transfer coefficient depends on fin height, with better heat transfer for smaller fins. The respective correlation based on thermovisually obtained fin temperatures is presented. For the more practical case, with the Grashof number being based on the temperature difference between central tube and ambient air, an improved correlation is presented with less than 6% scatter.

1. INTRODUCTION

FINNED tubes are widely used when heat is to be dissipated to air or other gases by natural convection. There is a large number of practically applied configurations of finned tubes. Here we want to investigate a horizontal tube with circular fins of constant thickness. There are not many experimental investigations found in the literature [1–4]. The results of these experimental works are correlated [5] in the equation:

$$Nu = \frac{hd_{\text{eff}}}{k} = 0.24\left(Gr \cdot Pr\frac{b}{d}\right)^{1/3} \tag{1}$$

with $d_{\text{eff}} = d+f$ and

$$Gr = \beta g d_{\text{eff}}^3 \Delta\vartheta / \nu^2.$$

This equation (1) correlates the results within $+25\%$. The temperature difference $\Delta\vartheta$ is taken between the surface of the central tube and the ambient air. The region covered is $10^3 < Gr \cdot Pr < 10^7$ for $Pr = 0.70$ (air). The applicability of this equation is restricted to tubes and fins made of the same metallic material and perfect contact between tube and fin.

From equation (1) it follows that the heat transfer coefficient is independent from d_{eff} which means from the fin height f:

$$h = 0.24 \cdot k\left(\frac{\beta g \Delta\vartheta}{\nu^2} Pr\frac{b}{d}\right)^{1/3}. \tag{2}$$

In order to learn about the temperature distribution across the various fins, thermography was used as infrared photography. This non-contactive method gives a picture of how useful the fins really are without disturbing the temperature field on the fin's surface.

† Dedicated to Prof. J. P. Hartnett upon his 70th birthday.

2. EXPERIMENTAL AND MEASURING SET-UP

The set-up is shown in Figs. 1(a) and (b). The test specimen is a finned tube with both central tube and fins made of copper; its total length is 45 mm. Five fins are soldered to the central tube at a distance of 8 mm to each other. The fin thickness is 1 mm. The fin height was varied: starting with a height of $f = 47$ mm, it was milled down to 37 and 27 mm after each series of test runs. Thus, the fin diameter D was varied from $D = d+2f = 110$ mm, to 90 and 70 mm. An electric heater was inserted inside the central tube; its power could be adjusted in various steps between 5 and 65 W. The accuracy of power measurements is 1%. The test specimen is hung up by thin steel wires so that natural convection is not impeded. For measurements of the mean wall temperature, six thermocouples (copper, constantan, 0.5 mm) were soldered to the finned tube on sites A and B as shown in Fig. 1(a).

The thermocouples were individually calibrated, the calibration uncertainty is 0.05 K.

For the thermovision experiments the front surfaces (towards the scanner, site A, and towards the powermeter, site B in Fig. 1) were coated with black paint in order to obtain a uniform emissivity close to 1. The thermovision system is by AGA, model 782, and consists of a scanner, monitor, computer and colour printer. The software DISCO 2.0 allows for the display of temperature distributions, temperature profiles, mean temperatures for the entire fin or selected cross-sections.

Temperature measurements

From the thermocouple measurements on six points of the model (Fig. 1) a mean fin temperature was determined as the arithmetic mean value. From thermovisual temperature readings at the same points,

NOMENCLATURE

A	area
b	distance between neighbouring fins
d	diameter of central tube
d_{eff}	effective diameter, $d+f$
D	diameter of fin
f	height of fin, $(D-d)/2$
g	acceleration of gravity
Gr	Grashof number, $\beta g d_{eff}^3 \Delta\vartheta/\nu^2$
h	mean heat transfer coefficient
k	thermal conductivity
L	tube length
n	number of fins
Pr	Prandtl number, ν/a
$\dot{Q}$	heating rate.

Greek symbols

β	coefficient of thermal expansion
δ	fin thickness
$\Delta\vartheta$	temperature difference
ε	emissivity
ϑ	temperature
ν	kinematic viscosity.

Indices

amb	ambient
eff	effective
tc	thermocouple
tv	thermovision.

the emissivity of the black paint coating was determined as $\varepsilon = 0.95$.

With this emissivity, the temperature distribution on the fins and the real mean temperature of the fin with regard to the various temperatures and respective surface areas could be determined by thermovision. This mean temperature ϑ_{tv} better represents the real mean temperature than the one obtained from thermocouple readings ϑ_{tc}.

The local thermocouple readings give lower mean temperatures than the thermovisual values which take account of the temperature areas. The largest deviations appear for small heating rates (10 or 20 W). For the calculation of the heat transfer coefficients the mean values of the fin temperature as obtained from the thermovisual measurements are used.

Thermovisual investigation

The temperature distribution on a fin indicates its usefulness. For cooling purposes a fin should dissipate

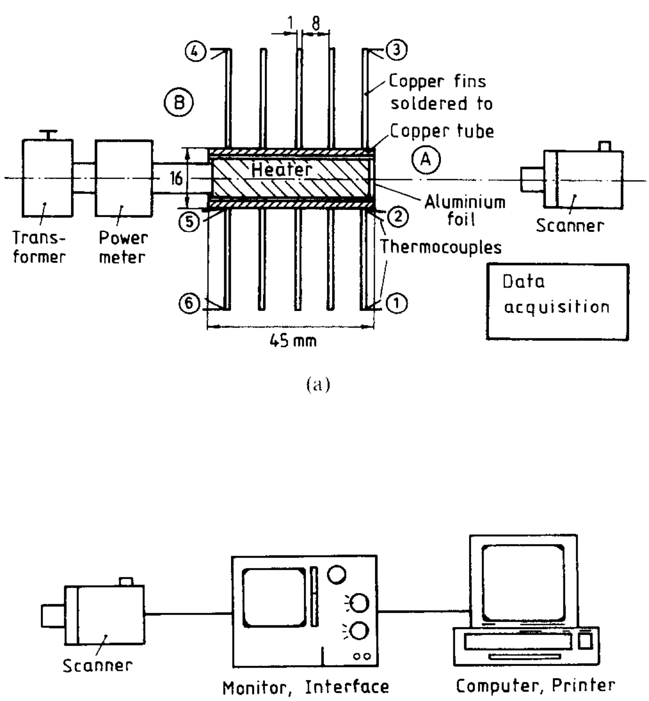

FIG. 1. Experimental and measuring set-up: (a) experimental set-up, (b) thermovision system.

a large amount of heat, this it would do best when the temperature gradient within the fin is constantly large and uniform in all directions. By thermovision it is relatively easy to obtain a picture of the temperature distributions. In Figs. 2 and 3 such distributions are shown, these were transferred from coloured photographs.

In Figs. 2(a)–(c) the distributions for a D = 110 mm fin for various heating rates are presented. A distinct asymmetry between the lower and the upper half can be observed in all pictures. Temperature gradients in the lower half are more or less constant across the fin while in the upper half gradients are small close to the central tube and increase rapidly close to the upper rim. This indicates that the fin is better used, i.e. dissipates more heat, in the lower half than in the upper.

In Figs. 3(a)–(c) the temperature distributions for a

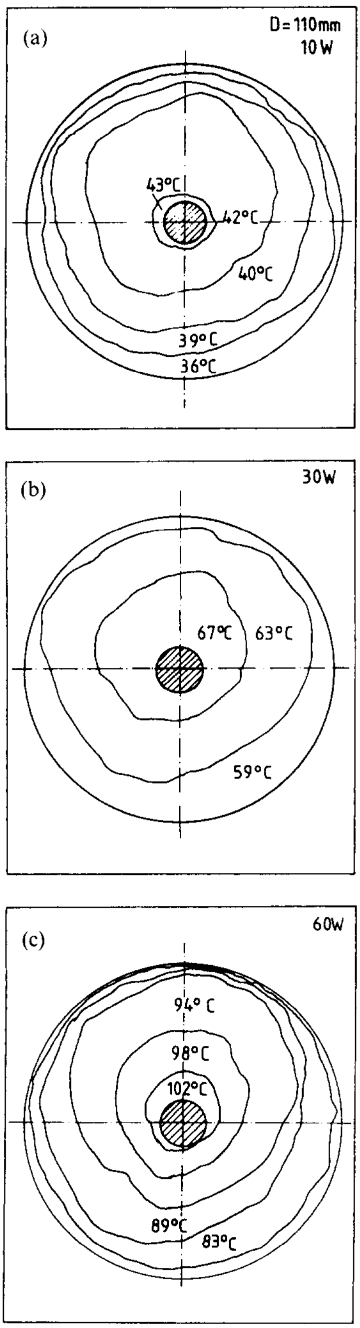

FIG. 2. Temperature distribution in a fin of D = 110 mm for various heating rates.

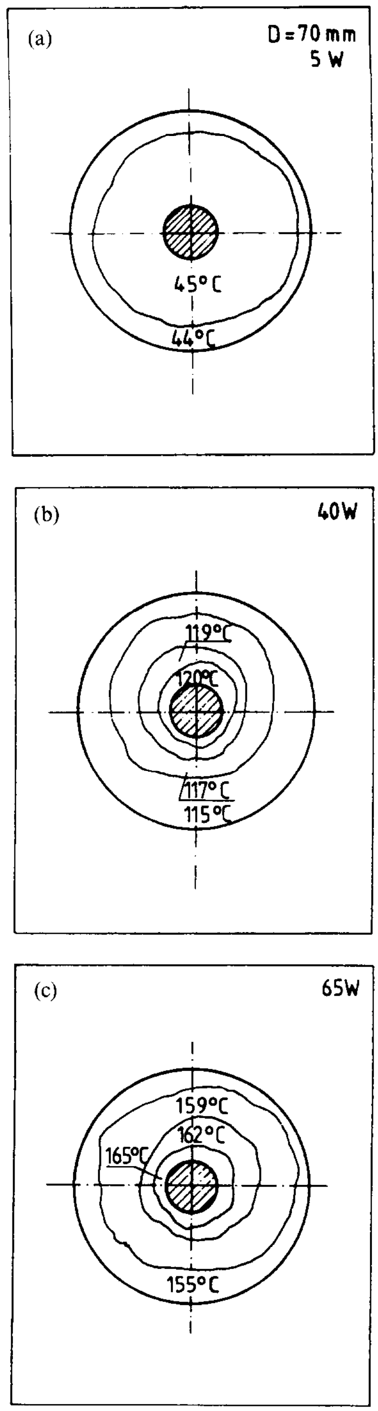

FIG. 3. Temperature distribution in a fin of D = 70 mm for various heating rates.

Table 1. Mean heat transfer coefficients for finned tubes of different height

Heating rate W	h in W m^{-2} K^{-1}		
	D = 110 mm	D = 90 mm	D = 70 mm
	A_{total} = 97 163 mm^2	A_{total} = 65 423 mm^2	A_{total} = 39 986 mm^2
10	6.47	6.98	8.25
20	7.71	8.36	9.67
30	8.67	8.92	10.33
40	8.76	9.57	11.09
50	9.09	10.13	11.78
60	9.54	10.51	12.29

fin of D = 70 mm show a fairly good symmetry. This fin dissipates heat equally well in all directions under the given conditions of natural convection to air and given heating rates.

Heat transfer coefficients

The heat transfer coefficients h are calculated according to equation (3):

$$h = \frac{\dot{Q}}{A_{total}(\vartheta_{tv} - \vartheta_{amb})}. \tag{3}$$

These represent mean heat transfer coefficients for the fin/tube arrangement.

In a detailed consideration there must be different heat transfer coefficients on a finned tube since there occur quite different heat fluxes Q/A and different temperature differences $(\vartheta_{finned\ tube} - \vartheta_{amb})$ locally. In comparative calculations with the different surface areas (such as fin surface, central tube surface, fin rim surface and central tube end surfaces) being combined with their respective temperatures it was found that these more precise results only differ by less than 1% from results for a total surface area A_{total} combined with the mean fin temperature ϑ_{tv} determined from thermovision.

The total surface area is obtained from

$$A_{total} = 2n(A_1 - A_2) + nA_3 + A_4 + A_5 \tag{4}$$

with

$$A_1 = D^2\pi/4; \quad A_2 = d^2\pi/4; \quad A_3 = D\pi;$$
$$A_4 = dL\pi - nd\delta\pi; \quad A_5 = d^2\pi/2.$$

The mean heat transfer coefficients for the finned tubes with different fin height are presented in Table 1 for different heating rates.

In Fig. 4 these and some more results are shown. There is a clear effect of the fin diameter upon the heat transfer coefficient, i.e. for $D = d+2f$.

Consequently, the heat transfer coefficient is influenced by the effective diameter $d_{eff} = d+f$. This is in contradiction to equation (2) where the effective diameter cancels out.

3. NUSSELT CORRELATIONS

Since the heat transfer coefficients increase when the fin height decreases (Table 1 and Fig. 4), the exponent of the Grashof number in equation (1) has to be less than 1/3.

When the experimental results for the heat transfer coefficient h and the mean temperature difference $\Delta\vartheta = \vartheta_{tv} - \vartheta_{amb}$ are inserted into Nu and Gr together with thermophysical data taken at $(\vartheta_{tv} + \vartheta_{amb})/2$ the following correlation is obtained:

$$Nu = hd_{eff}/k = 0.376(Gr_{deff}\,Pr\,b/d)^{0.308}. \tag{5}$$

Our results are correlated by equation (5) to within $\pm 3\%$.

For most cases in practical applications it is impossible to forecast a mean fin temperature as observed here by the thermovision system. It is more convenient to use the temperature of the central tube and form a $\Delta\vartheta = \vartheta_{tube} - \vartheta_{amb}$ for Gr and for thermophysical data the reference temperature $(\vartheta_{tube} + \vartheta_{amb})/2$. With these

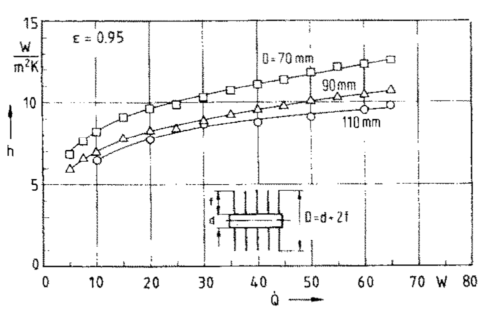

FIG. 4. Mean heat transfer coefficients for various fin heights and heating rates.

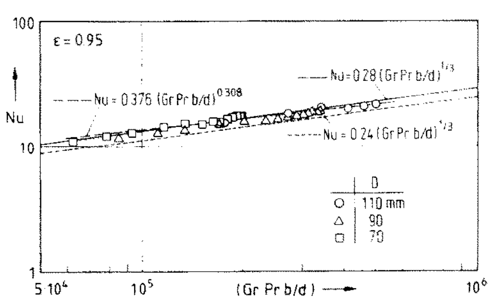

FIG. 5. Various Nusselt correlations for natural convection of air on finned tubes.

conditions we obtain the correlation

$$Nu = hd_{\mathrm{eff}}/k = 0.28\,(Gr_{d\mathrm{eff}}\,Pr\,b/d)^{1/3} \quad (6)$$

and the effect of different fin heights on h vanishes again. The experimental results are correlated to within 6%.

In Fig. 5 the various correlations are presented together with the experimental results. The region of application tested here is

$$5 \times 10^4 = Gr \cdot Pr\,b/d = 5 \times 10^5.$$

4. CONCLUSIONS

When thermovisual mean temperatures of the fins are used, a clear effect of the fin height upon the heat transfer coefficient can be found. With the practically better known tube temperature for the characteristic temperature difference an equally simple correlation, as given in literature, is obtained with a scatter of $\pm 6\%$.

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