Study Group Report: Plate-fin Heat Exchangers: AEA Technology

The problem under study concerned the apparent discrepancy between a series of experiments using a plate fin heat exchanger and the classical theory of Nusselt. A plate fin heat exchanger involves the flow of a stream of condensing nitrogen vapour in one flow channel and a counterflowing stream of liquid nitrogen in an adjacent flow channel. The liquid nitrogen is at a lower temperature and pressure and boils.

The classical theory of Nusselt treats the flow of the condensing vapour using lubrication theory. A liquid film of condensate forms on the walls of the flow channel, and drains under gravity. As the flow continues along the channel, further condensation occurs and the flux of liquid increases (figure 1).

Simple lubrication theory analysis suggests that the mass flux of the liquid per unit width on a flat plate is

$$M_l = \frac{\rho_l(\rho_l - \rho_v)gh^3}{3\mu}$$

This may be used to determine the quality, Q, which is a measure of the vapour flux divided by the total flux.

$$Q = \frac{(M - M_l)}{M}$$

where M is the total mass flux. Note the quality may alternatively be defined as a thermodynamic quality, in which case the ratio of the enthalpy (rather than mass) fluxes is used in the definition, with the liquid and vapour enthalpies being evaluated at the condensation temperature.

The relationship between the quality and the mass flux of liquid enables one to determine the heat transfer coefficient as a function of the quality. The heat transfer through the liquid film on the walls has the form, approximately,

$$F = \frac{K(T_c - T_w)}{h} = \alpha(T_c - T_w)$$

where T_c is the condensation temperature of the vapour and T_w is the wall temperature, and K is the thermal conductivity. The film thickness h may be recast in terms of the quality, using the above analysis, to yield the result

$$\alpha = (\frac{\rho_l(\rho_l - \rho_v)g}{3\mu M})^{1/3} \frac{K}{(1 - Q)^{1/3}}$$

This simple theory was applied to understand the variation of the heat transfer coefficient with quality along one of the rectangular plate fin passages in a test heat exchanger at

Harwell. The theory was adapted to include the effects of the geometry of the rectangular cavity and the reduced efficiency of the secondary fins, which are not directly in contact with the counterflowing channel. However, in spite of these corrections to the simple theory, the variation of the experimental heat transfer coefficients with quality differed from the scaling suggested above (figure 2). At high flow rates the dimensionless heat transfer coefficient varied somewhat more slowly than predicted by the Nusselt scaling, while at low flow rates the heat transfer coefficient decreased more rapidly than predicted by the Nusselt theory. The scaling parameter, which includes the effect of the geometry may be found from the data at imtermediate flow rates (figure 2).

At the study group, various explanations as to the source of the discrepancy between the Nusselt theory and the data were explored. Two approaches taken were to assess the data collection method and to assess the relevance of the model to the experiments.

Experimental assessment.

The quality is calculated by measuring the heat flux through the walls of the channel and then assuming that this heat flux is supplied by the latent heat released on condensation of the vapour. Therefore, at any distance down the pipe, the total flux of condensed vapour may be estimated.

However, in analysing the data collection methods, it was found that there is some superheating of the nitrogen vapour on entry to the channel. This superheat remains in the vapour for some time, and affects the estimate of the point at which condensation occurs since the superheat is conducted through the wall in addition to the latent heat. In turn this affects the estimate of the quality. However, from the enthalpy balance, this superheating only amounts to a quality error of a few percent.

Further, it was found that the heat flux in the wall was not in fact constant but varied significantly near the entry point. The heat flux measurements in the wall were quite far apart, and did not seem sufficient to resolve this variation and thereby accurately estimate the total heat flux through a given length of the wall. Indeed in some experiments, only one or two thermocouples were located in this entry region. One conclusion of the study group is that the accuracy of these estimates could be checked.

Theoretical Assessment.

In addition, during the study group week, we examined the assumptions implicit in the Nusselt theory, and their relevance to the flow in question. A major issue which was addressed was the role of any indentations in the walls, as arise for example near sites of brazing between adjacent channel walls. In such narrow grooves, the action of surface tension is to fill the groove with fluid, and thereby possibly create a flow path of lower resistance than on the smoother walls of the flow channel. This can increase the flux of liquid down the channel, but its efect upon the relationship between the heat transfer coefficient (HTC) and the quality was not clear.

In order to elucidate the possible impact of any grooves upon the HTC-quality relationship, we developed a simple model of a groove of size d embedded in a plane wall, and aligned with the flow. A simple scaling of the lateral surface tension forces and the downward gravitational force, suggests that the force ratio is

$$\frac{\gamma}{\rho g d^2} \sim 10^{-6} d^{-2}$$

and so any groove of size smaller than 1mm will draw in fluid on a short timescale compared to the downward flow timescale. Furthermore, the lateral flux of condensed liquid required to fill the groove may be supplied by the enhanced condensation around the groove which results from the thinning of the film around the groove under the action of surface tension (figure 3).

The effect of such a groove upon the Nusselt heat transfer theory is readily calculated, and shown to be important. The reason that a flow in a groove changes the scaling is that the liquid flux may be significantly enhanced by such a groove whereas the heat flux through the wall does not change significantly. The liquid flux is given by the sum of the lubrication flow down the wall, of lateral scale a say, plus the flow in the groove itself, giving the total liquid flux over a length a (figure 3)

$$M_l = \frac{\rho g}{6\mu}(2h^3 a + d^4)$$

A simple scaling analysis shows that for a groove of width 0.1 mm, a typical film thickness of 0.05 mm, and a channel width a of 1 mm, the flux ratio between the flow in the groove and that in the main film is

$$\frac{d^4}{2h^3 a} \sim O(1)$$

and so the flux is increased by O(1) owing to the presence of the groove.

We can then write down the relation

$$\frac{1}{h} = (\frac{\rho g}{a\mu M})^{1/3}(1 - Q - \frac{\rho g d^4}{a\mu M})^{-1/3}$$

In contrast, the heat transfer through the liquid film consists of the sum of the flux through the main film and that through the groove. This has the approximate value for the geometry of figure 3

$$Heatflux = k\Delta T(\frac{a}{h} + \frac{d}{(h+d)})$$

Now the ratio of the heat flux through the groove and through the film proper is

$$\frac{dh}{(d+h)a} \sim 0.1$$

and so the heat flux is not significantly altered by the presence of the liquid filled groove. As a result, the heat flux has the approximate value

$$Heatflux = \frac{K\Delta T a}{h}$$

and the heat transfer coefficient scales as $\frac{1}{h}(1 + O(\frac{hd}{(d+h)a}))$.

Using the revised formula for the liquid flux, we see that the heat transfer coefficient is related to the quality by the scaling

$$\alpha \sim (1 - Q - \frac{\rho_l(\rho_l - \rho_v)gd^4}{a\mu M})^{-1/3}$$

The term on the right hand side is of some import, as shown above, and corresponds to the enhancement of the liquid flux owing to the presence of the grooves. In figure 4, we show how this factor differs from the classical Nusselt theory in which the scaling is simply $(1-Q)^{-1/3}$. Note that the effect this correction is greatest at low flow rates for which the liquid in the film constitutes a large fraction of the total flux.

Experimental Design.

A further feature was noted about the present experimental configuration. This sheds some light upon the sensitivity of the apparatus to detect changes in the heat transfer coefficient with fin blade shape. It was identified that with the present experimental design, most of the temperature drop between the condensing vapour and the boiling liquid in adjacent flow channels actually occurs in the metal dividing the two flow channels rather than in the liquid film of condensate on the wall of the channel. In particular, if we balance the flux of heat through the wall with that through the liquid film, then

$$K_l \frac{T_c - T_w}{h} = K_a \frac{T_w - T_b}{h_w}$$

where h_w is the thickness of the wall and T_b is the boiling temperature on the far side of the wall. Since the wall thickness in the experimental set-up is approximately 6 mm, and the typical film thickness is 0.1 mm, then the temperature drop across the wall is about 10 times larger than that across the film of condensed vapour. As a result, the heat flux across the walls of the channel is insensitive to small changes in the mean thickness of the liquid film as the geometry of the channel is varied for example, since the heat flux is determined by the temperature drop across the wall. For a given position along the flow channel, in each experiment a comparable mass of liquid will condense in order to supply the heat flux through the wall. As a result, the quality at a given location along the wall will be relatively insensitive to differences in the detailed channel geometry, and it is difficult to accurately compare the efficiency of different channel designs.

In the real systems, the wall thickness is much smaller, and the heat flux is therefore more sensitive to changes in the thickness of the liquid film, since more of the temperature difference between the boiling and condensing flows occurs across the condensed liquid film.

A different approach might be to examine the properties of the flow at various points along the flow channel, by arranging a series of different length flow channels in series, and sampling the output from each channel.

A number of questions emerge from this report, including:

(i) is there a critical heat flux ?

(ii) what size/shape of groove is best ?

(iii) can models be tested using the existing experimental data ?

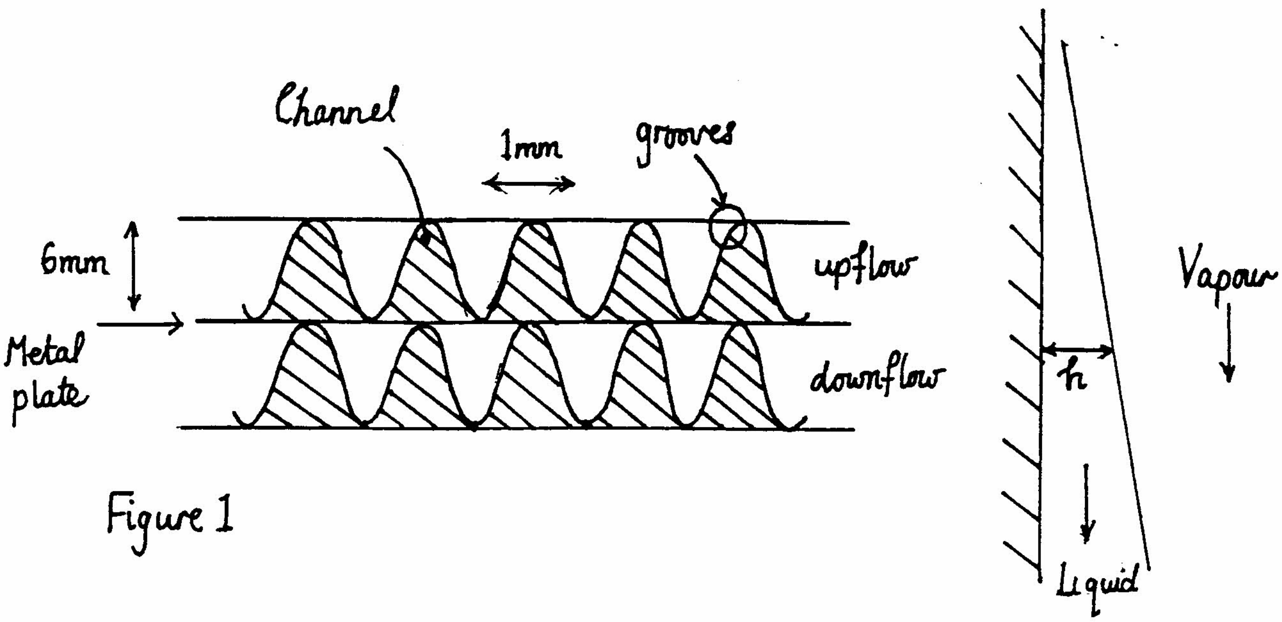


Figure 1

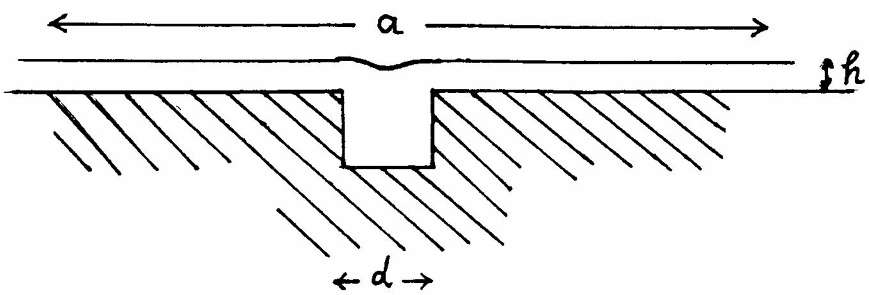


Figure 3

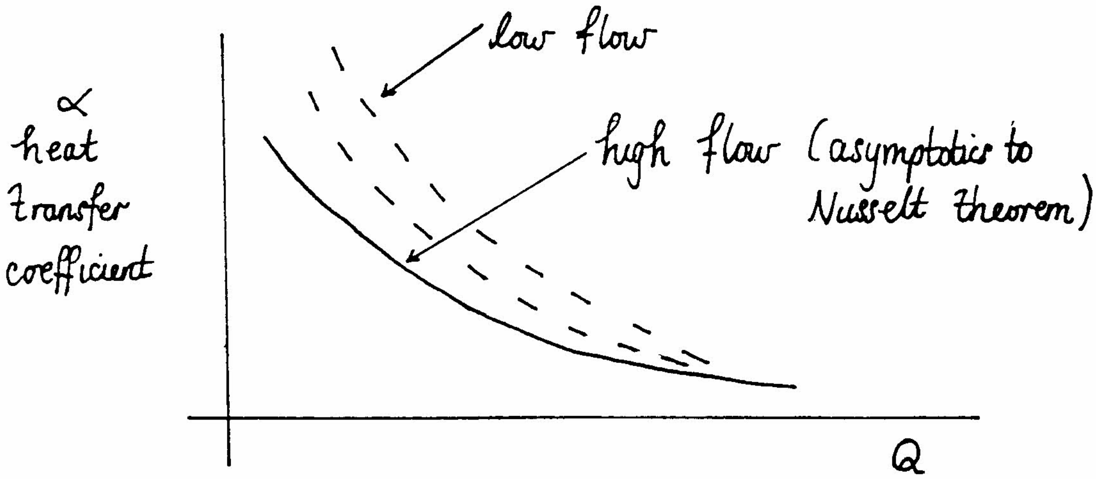
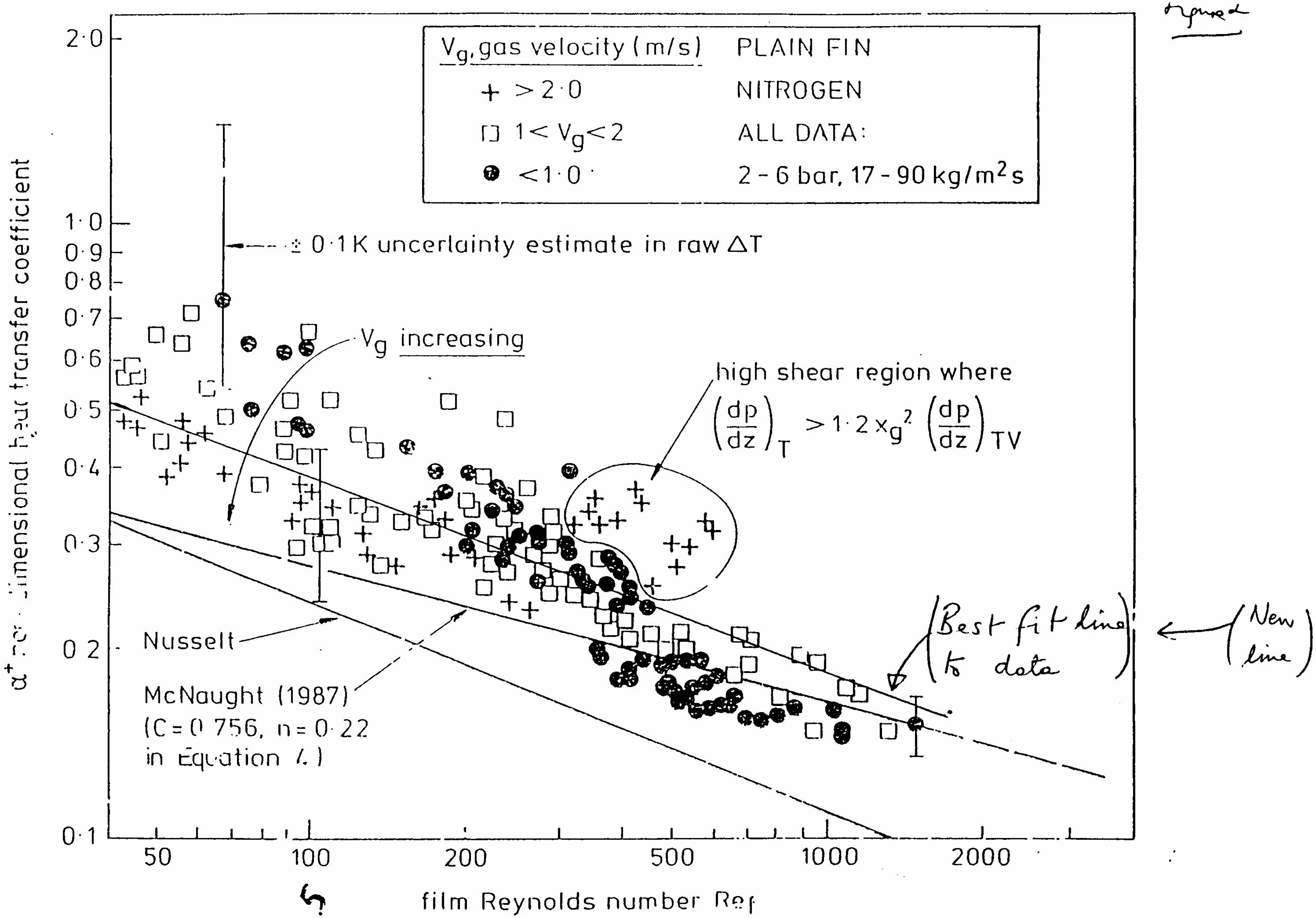


Figure 4

Fig 2



Vg, gas velocity (m/s)
+ > 2·0
1< Vg<2
<1·0
PLAIN FIN
NITROGEN
ALL DATA:
2 - 6 bar, 17 - 90 kg/m2s
± 0·1K uncertainty estimate in raw ΔT
Vg increasing
high shear region where
(dp/dz)T > 1·2 xg2 (dp/dz)TV
Best fit line to data
New line
Nusselt
McNaught (1987)
(C = 0·756, n = 0·22
in Equation 4)
film Reynolds number Ref
2·0
1·0
0·9
0·8
0·7
0·6
0·5
0·4
0·3
0·2
0·1
50
100
200
500
1000
2000

AWW