the finite length fins under the conditions that the fin tip temperature is nearly equal to the ambient fluid temperature and the heat transfer near the tip is negligible compared to the total fin heat transfer.

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On the heat transfer characteristics of constrained air jets impinging on a flat surface

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1. INTRODUCTION

IMPINGING jets is a well-known and widely used technique for realizing high heat transfer rates between a fluid and a surface. The low cost and the fine degree of control that gas jets permit have made them particularly attractive for cooling applications. Specific applications include the cooling of the leading edge of turbine blades, the cooling of electrical equipment, the annealing of metal and plastic sheets, the tempering of glass, and the drying of textiles, veneer, paper and film materials. Different jet-impingement surface configurations varying from a single circular jet to arrays of round or slot nozzles are described in ref. [1] which is the most comprehensive survey of jet-impingement heat transfer available at the present time. Reference [1], however, does not contain any mention of constrained jets, which are the focus of the present work. By constrained jet, it is meant that the flow is forced to flow back after impinging on the surface rather than spreading and flowing over the surface. It is used in applications where only localized cooling is desired as shown in Fig. 1.

In order to efficiently design jet-cooling systems, one needs to know the dependence of heat transfer rates on variables which fully characterize the system. For single round and slot nozzles, this dependence can be described in the following dimensionless form [1]:

$$\overline{Nu} = f(Re, Pr, r/D, H/D). \tag{1}$$

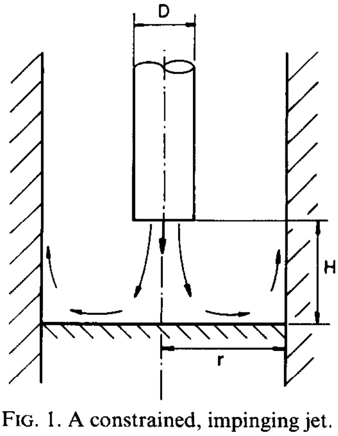

FIG. 1. A constrained, impinging jet.

Schlünder and Gnielinski [2] correlated their measurements and those of other researchers for free impinging single round jets by the following empirical equation:

$$\frac{\overline{Nu}}{Pr^{0.42}} = \frac{D}{r}\,\frac{1-1.1D/r}{1+0.1(H/D-6)D/r}\,F(Re). \tag{2}$$

The function $F(Re)$ may be represented by the following expression:

$$F(Re) = 2\,Re^{1/2}\left(1+\frac{Re^{0.55}}{200}\right)^{0.5}. \tag{3}$$

Equations (2) and (3) are valid in the ranges

$$2000 \leqslant Re \leqslant 400{,}000$$
$$2.5 \leqslant r/D \leqslant 7.5$$
$$2 \leqslant H/D \leqslant 12.$$

Equation (2), in general, would not give satisfactory results for constrained jets because of the difference in flow characteristics between free and constrained impinging jets. At present, there are no references in the literature concerning the case of impinging gas jet heat transfer with a solid boundary to constrain the radial flow of gas. The present work is aimed at examining the influence of Re, H/D and r/D on the heat transfer coefficient in the case of a constrained circular air jet impinging on a surface.

2. EXPERIMENTAL ARRANGEMENT AND TEST CONDITIONS

The experimental set-up, which allowed the determination of the average heat transfer coefficient between constrained air jets and a flat heated surface is schematically shown in Fig. 2. The heat transfer surface was that of a heated cylindrical copper block. The copper block was heated using a cartridge heater and was heavily insulated to ensure one-dimensional heat flow normal to the heat transfer surface. The copper block was instrumented with four copper–constantan thermocouples located on the centerline 3, 16, 28.5 and 41 mm away from the surface. The surface temperature was obtained by extrapolating the measured linear axial temperature distribution. The heat flux was obtained by measuring the input power to the heater or by using the known thermal conductivity of the copper specimen and the measured axial temperature gradient. The two methods yielded essentially the same result (within 5%).

NOMENCLATURE

D jet diameter
H distance between the jet and heat transfer surface
Nu Nusselt number
Pr Prandtl number
Re Reynolds number
r radius of the heat transfer surface.

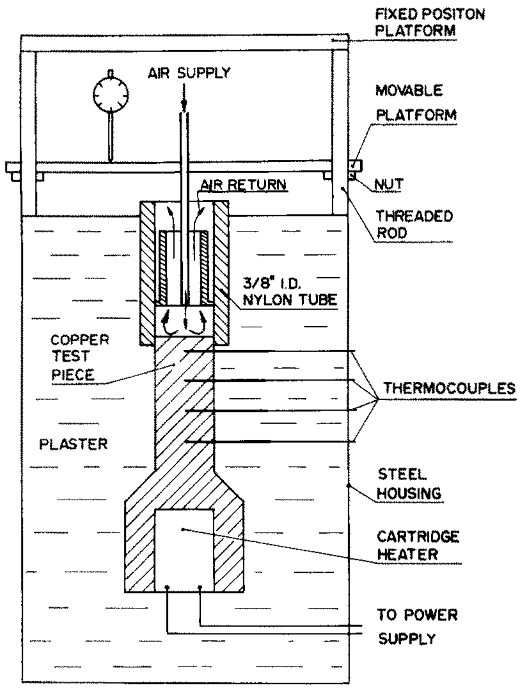

FIG. 2. Test apparatus.

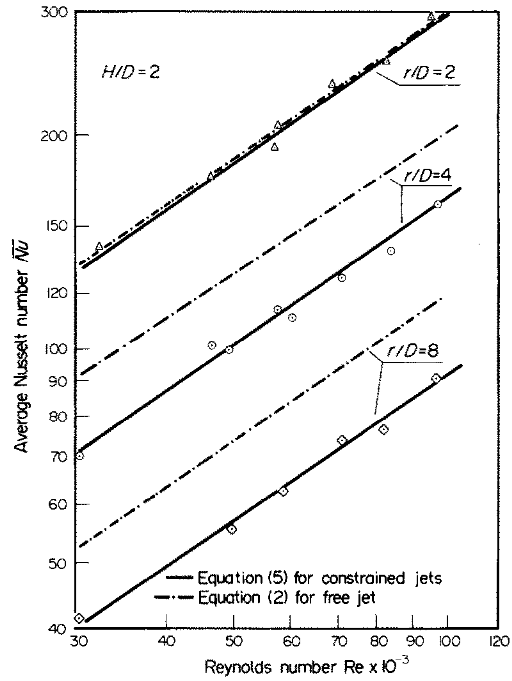

FIG. 3. Experimental results, $\overline{Nu}$ vs Re.

The upper surface of the test piece section, which measured 0.95 cm in diameter (3/8 in), was exposed to the air jet from a single round nozzle. Both the convection surface and the nozzle were enclosed by a 0.95-cm-I.D. nylon tube which constrained the flow of gas in the radial direction. The temperature of the air was measured by a copper–constantan thermocouple.

The air nozzle was mounted on a movable platform and the distance between the nozzle and the heat transfer surface was measured by a dial type micrometer. The air flow was supplied from the building air supply. It was filtered and passed through a pressure regulator. The air flow was measured by two calibrated rotameters, one for the range of 0.05–0.5 and the other for the range 0.5–7.5 $dm^3\ s^{-1}$.

Three sets of experiments were conducted for three different nozzle diameters: 0.06, 0.12 and 0.24 mm which corresponded to r/D equal to 8, 4 and 2, respectively. Each series consisted of runs in which only one variable was changing while the others remained constant. For each series of experiments, the ratio of the distance between the jet and the surface to the jet diameter, H/D, was varied between 2 and 6 and the Reynolds number Re was varied between 30,000 and 100,000. For each test condition, the heat flux was varied, by controlling the power input to the heater, to establish the effect of heat flux on the results.

The test conditions can accordingly be summarized as

$$30{,}000 \leqslant Re \leqslant 100{,}000 \quad (4a)$$

$$2 \leqslant H/D \leqslant 6 \quad (4b)$$

$$2 \leqslant r/D \leqslant 8. \quad (4c)$$

3. RESULTS AND DISCUSSION

For each experiment the average heat transfer coefficient was calculated based on the temperature difference between the wall and the air jet, i.e. $h = (q/A)/(T_w - T_{air})$. The choice of $(T_w - T_{air})$ as the driving force is consistent with other investigators [1–6]. However, it must be noted that, for heating applications, this temperature difference may not be appropriate, as the hot gas jet loses a significant amount of heat as it approaches the solid surface. In the cooling applications, as in our case, the air jet is essentially at the ambient temperature.

The data are presented in terms of Nu as a function of Re, r/D and H/D. The Nusselt and Reynolds numbers are based on the jet diameter D. This is consistent with the procedure used in ref. [1] and by most other investigators. The results were found to be independent of the heat flux.

Typical experimental results are shown in Figs. 3 and 4. These figures show Nu to increase with increasing Re and decreasing r/D or H/D. It is also clear that the influence of the dimensionless heating surface radius r/D is much more pronounced than that of the dimensionless distance between the jet and the heating surface H/D. These trends are generally similar to those observed by other investigators for unconstrained jets except for the effect of H/D. For unconstrained jets, the literature shows a somewhat different effect of H/D. Most of the investigators observed a maximum heat transfer coefficient at H/D in the range of 6–7 [3–6]. The data shown herein for constrained jets does not show this trend.

All the data obtained in the present work (160 data points) were correlated using multiple linear regression. The result-

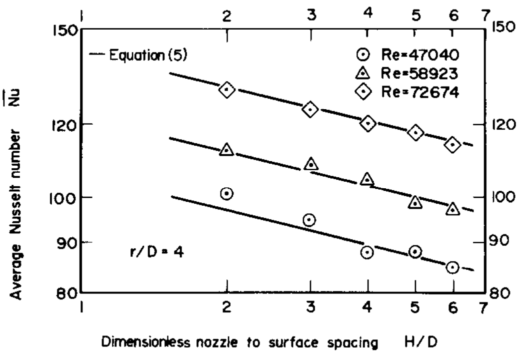

FIG. 4. Experimental results, $\overline{Nu}$ vs H/D.

ing empirical correlation is in the form,

$$\overline{Nu} = 0.216\,Re^{0.685}(H/D)^{-0.12}(r/D)^{-0.85}. \tag{5}$$

In Figs. 3 and 4, the predictions of equation (5) are given by the solid lines.

The experimental results obtained in the present work for unconstrained jets were compared with those available in the literature for unconstrained jets [1–7]. It was generally found that, for high r/D values, lower heat transfer rates were obtained for constrained jets as compared to unconstrained jets for the same values of Re and H/D. Superimposed on Fig. 3 is the prediction of equation (2) for unconstrained jets for H/D equals 2 and r/D equals 8, 4 and 2, respectively. These observations can easily be explained in terms of the resulting flow field near the heat transfer surface in each case.

For free impinging jets, the flow out of the nozzle will entrain air from the still surroundings forming a widening mixing region where an intensive momentum exchange will take place. As the flow approaches the stagnation region near the center of the heat transfer surface, it turns, forming what is essentially a wall jet separated from the surface by a boundary layer where the heat exchange with the wall takes place. For a constrained jet, the widening of the jet leaving the nozzle exit and its ability to entrain the surrounding air are limited by the backward flow leaving the surface. Accordingly, as the air reaches the surface its effective momentum and the width of the jet are somewhat lower than that of the free impinging jet. Also, the development of the boundary layer is slowed down by the presence of the constraining wall. Accordingly, boundary-layer stagnation and separation will take place near that wall. It is obvious that this change of the flow field will result in less heat removal from the surface. However, for low values of r/D this effect diminishes. As shown in Fig. 3, for r/D equals 2, equations (2) and (5) predict similar values of Nu.

4. CONCLUDING REMARKS

An experimental study was carried out to investigate the heat transfer characteristics of constrained air jets impinging on a flat surface. The results showed that, the heat transfer coefficients increased with increasing the jet Reynolds number Re and with decreasing the radius of the heat transfer surface r and the distance between the surface and the nozzle exit H. The data were successfully correlated by an empirical relationship, i.e. equation (5).

The results showed that, except for low values of r/D, constrained impinging jets are less capable of removing heat from flat surfaces compared to free, unconstrained, impinging jets under similar conditions. This was explained in terms of the resulting flow field in each case.

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Analysis of interdiffusion in film absorption

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1. INTRODUCTION

THE COMBINED heat and mass transfer process taking place in film absorption has received growing attention in recent years and has been analyzed in several articles [1–6]. In the absorption process, encountered in numerous applications in chemical technology, the mass transfer is often accompanied by a significant heat effect, particularly when the absorbate is a vapor with a large latent heat. As was demonstrated in the above works, the heat and mass transfer processes are coupled in this case, and the concentration and temperature distributions are interdependent.

In all the above studies, the thermal effects due to diffusion [7] (also known as interdiffusion) were assumed to be negligible. Indeed, this assumption is quite realistic in most cases of practical importance. Yet, interdiffusion may become important for films with short exposure time [8] where large temperature and concentration gradients are present. Then, an additional term in the energy equation, expressing the energy transport due to the diffusive mass flux, may affect the solution.

This analysis considers the falling film model with the governing equations in their complete form, to evaluate the contribution of the interdiffusion. A short-exposure time solution is obtained by a similarity method.

2. MODEL AND EQUATIONS

Figure 1 describes schematically the system under consideration. A film of liquid solution, composed of substances