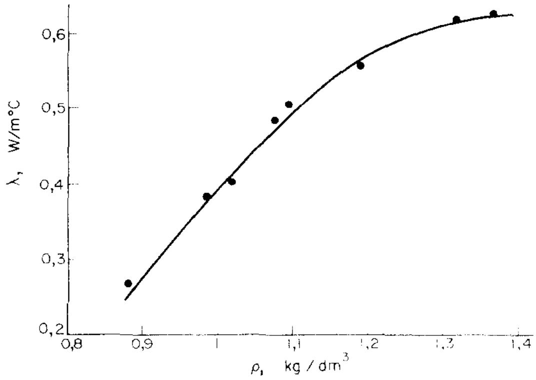

FIG. 4. Variation de la conductivité thermique du plâtre avec l'humidité.

supérieure à 200 J/°C). Elle est particulièrement intéressante pour mesurer la capacité thermique des matériaux humides. Nous avons observé que la capacité thermique est la somme de la capacité du matériau sec et de la capacité de l'eau d'humidification. Connaissant la capacité thermique et la valeur de m_1, nous avons pu représenter la conductivité thermique en fonction de la masse volumique du matériau humide Fig. 4.

CONCLUSION

L'étude détaillée de la dynamique des échanges calorifiques dans un milieu symétrique à trois couches a montré l'intérêt des modes normaux pour la représentation des évolutions thermocinétiques en régime variable. Les constantes de temps fondamentales des modes normaux ont été identifiées par un traitement adapté des évolutions fluxmétriques. Les coefficients obtenus caractérisent totalement l'échantillon étudié et peuvent être utilisés pour calculer les constantes thermophysiques. Dans le cas particulier des matériaux humides, l'évolution thermocinétique imposée doit être symetrique par rapport aux coordonnées d'espace. Ce traitement de ce type d'évolution conduit à la détermination de la capacité et la diffusivité thermiques.

Le traitement mathématique des évolutions thermocinétiques appliqué à la métrologie a le grand avantage d'éliminare le difficile problème expérimental de la conductivité en régime permanent. La représentation des échanges calorifiques par la mesure instantanée des flux calorifiques semble particulièrement adaptée à l'étude des évolutions thermocinétiques. Le développement des méthodes fluxmétriques en régime variable est lié d'une part à la fabrication de fluxmètres de faible capacité permettant de suivre les évolutions rapidement variables dans le temps, d'autre part à la mise au point de modèles mathématiques utilisables pour les faibles valeurs du temps.

Remerciements – Nous remercions Monsieur Marechal, Chef de la Division Thermique du Centre Expérimental du Bâtiment et des Travaux Publics qui nous a incité à développer de nouveaux fluxmètres calorifiques et avec qui nous avons des discussions très fructueuses sur les applications à la métrologie des mesures fluxmétriques.

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ON THE PENETRATION OF TURBULENCE THROUGH PERFORATED FLAT PLATES

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NOMENCLATURE

d, pore diameter;
E_1, E_2, turbulence spectra;
f, longitudinal correlation;
g, lateral correlation;
k_1, streamwise wave number;
k_t, threshold wave number;
P, penetration coefficient;
$\mathbf{r}$, position coordinate;
u, streamwise velocity fluctuation;
v, velocity fluctuation normal to the plate;
x, streamwise coordinate.

Greek symbols

α, numerical coefficient;
λ_f, longitudinal microscale;
η_t, non-dimensional threshold.

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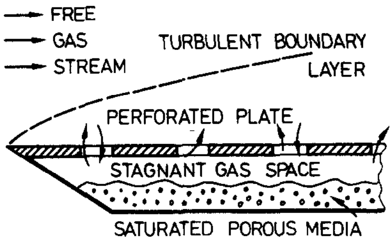

FIG. 1. Schematic diagram of the system.

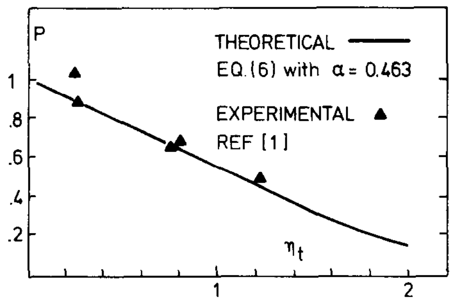

FIG. 2. The penetration coefficient.

1. INTRODUCTION

RECENTLY Cannon *et al.* [1] proposed a model for calculating the mass transfer from a saturated porous media covered by a rigid perforated thin flat plate, through the turbulent boundary layer of a gas flowing over the upper surface of that flat plate, into the environment above as shown in Fig. 1. An important feature of the proposed model is the penetration coefficient which describes the percentage of the eddies able to penetrate the pores of the plate. The experiments reported in [1] support the model and a few values for the penetration coefficient were obtained. In this note we present a phenomenological explanation for these coefficients and a theoretical method for calculating them.

2. THE PENETRATION COEFFICIENT

The physical model is shown schematically in Fig. 1. To evaluate the penetration coefficient, P, quantitatively, we shall assume that the pores act as a threshold in the spectral space, chopping the spectra and letting only eddies associated with wave numbers larger than k_t enter the pores from the turbulent gas stream, or

$$P = \int_{k_t}^{\infty} E_2(k_1)\,dk_1 \tag{1}$$

where k_t is the lower threshold wave number.

In this note the spectrum E_2 is represented by the equation

$$E_2(k_1) = 2/(\pi \overline{v^2}) \int_0^{\infty} g(x)\cos(k_1 x)\,dx \tag{2}$$

where $g(x)$ is the lateral correlation $\overline{v(\mathbf{r})v(\mathbf{r}+x)}$.

Here v is the velocity fluctuation perpendicular to the plate, and k_1 is the wave number associated with the streamwise direction x. This choice seems to represent the physical situation realistically as the two features of turbulence (i) the velocity fluctuations perpendicular to the wall and (ii) the eddies dimensions in the streamwise direction are considered.

Although, in general, the evaluation of E_2 is not simple, using isotropic turbulence as a model, we may expect [2] that

$$E_2(k_1) = \tfrac{1}{2}[E_1(k_1) - k_1 \partial/\partial k_1 E_1(k_1)] \tag{3}$$

and derive E_2 from E_1, which is defined by the relation

$$E_1(k_1) = 2/(\pi \overline{u^2}) \int_0^{\infty} f(x)\cos(k_1 x)\,dx. \tag{4}$$

Assuming a plausible longitudinal correlation $f(x) = \overline{u(\mathbf{r})u(\mathbf{r}+x)}$ and using $f(x) = e^{-(x/\lambda_f)^2}$, we obtain

$$E_2(k_1) = (\tfrac{1}{2}\lambda_f/\sqrt{\pi})(1 + \lambda_f^2 k_1^2/2)\,e^{-(\lambda_f k_1/2)^2} \tag{5}$$

where λ_f is the longitudinal microscale. Although the validity of equation (5) is doubtful in the inertial range, it is probably a reasonable approximation for the energy containing range [3] of interest here. With these simplifications we obtain

$$P = 1 - (2/\pi)^{1/2} \int_0^{\eta_t} e^{-\eta^2/2}\,d\eta + (\eta_t/\sqrt{(2\pi)})\,e^{-\eta_t^2/2} \tag{6}$$

where $\eta_t = k_t\lambda_f/\sqrt{2}$.

To complete the model, it is necessary to relate the threshold wave number to the pore diameter d, e.g. let $k_t = \alpha/d$, where α is a numerical coefficient of the order of unity. This gives

$$\eta_t = (\alpha/\sqrt{2})(\lambda_f/d) \tag{7}$$

resulting in a theoretical model given by equation (6) which depends on the longitudinal microscale only.

3. EXPERIMENTAL EVIDENCE AND REMARKS

Using the values of $\alpha = 0.463$ and $\lambda_f = 914\,\mu m$ reported by Cannon [4] equation (6) predicts $P = 0.67, 0.66, 0.89$ and 0.90 for the four plates given in Table 3 of [1] where $P = 0.67, 0.71, 0.88$ and 1.01 were found experimentally, as shown in Fig. 2 and Table 1.

The limited experimental data supports the phenomenological model outlined above, but further experimental research will be required, extending the threshold wave number range, to verify the analytical prediction of the penetration coefficient by equation (6) in this note.

Table 1. Penetration coefficients

Plate no.	5	4	8	9	12
Pore diameter d (μm)	244	394	376	1090	1180
Theoretical penetration coefficient equation (6)	0.45	0.67	0.66	0.89	0.90
Experimental penetration coefficient [1]	0.52	0.67	0.71	0.88	1.07

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INTERFEROMETRIC MEASUREMENT OF HEAT TRANSFER DURING MELTING FROM A VERTICAL SURFACE

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NOMENCLATURE

c, specific heat;
Fo, Fourier number, $\alpha t/L^2$;
Δh_f, latent heat of fusion;
L, height of heat exchanger;
Nu, local Nusselt number defined by equations (1) or (2);
Pr, Prandtl number, ν/α;
Ra_x, local Rayleigh number, $[g\beta|T_w - T_f|x^3/\nu^2](\nu/\alpha)$;
Ste, Stefan number, $c(T_w - T_f)/\Delta h_f$;
T, temperature;
t, time;
α, thermal diffusivity;
β, thermal expansion coefficient;
ν, kinematic viscosity.

Subscripts
f, refers to fusion;
i, refers to interface;
0, refers to initial (uniform) conditions;
x, refers to local;
w, refers to wall.

INTRODUCTION

THE PRESENT study was conducted to obtain evidence on the detailed temperature distribution and local heat transfer coefficients at both the heated vertical surface and the solid–liquid interface during melting of an initially isothermal solid near its fusion temperature. The work was motivated by the need to gain understanding of heat transfer processes in latent heat-of-fusion thermal energy storage systems [1]. Cost effective and thermally efficient storage systems which take advantage of the latent heat-of-fusion of a phase-change material are prerequisite for economic utilization of alternate energy sources such as solar, waste heat recovery and load levelling.

Photographic observations of melting from a vertical cylindrical heat source [2] and a vertical isothermal wall [3] have indicated marked deviations in the shape of the melt region from the solid–liquid interface positions that would be predicted by a pure conduction model. More melting took place at the top than near the bottom of the heated vertical surfaces. The change of the melt shape with time provided conclusive evidence of the importance of natural convection in the melted region. However, temperature distributions in the melt and local heat-transfer coefficients have not been reported. The analysis of melting from a vertical cylindrical heat source, including the effects of natural convection induced by temperature differences in the melt, similarly indicates marked deviations in the shape of the melt region from that which would be predicted by a pure conduction model [4]. Natural convection was also found to play a very important role in the melting of material contained in a rectangular cavity [5].

Experiments described in this paper provide detailed information about the heat-transfer processes which occur when a solid is melted from a vertical surface in contact with the material. Knowledge of heat transfer at the interface, for example, is essential in predicting the motion of the phase-change boundary during melting. The interferograms recorded furnish evidence of the role of natural convection in the melt and yield qualitative information about the flow field in the liquid.

EXPERIMENTS

A Mach–Zehnder interferometer of typical rectangular design, having 7.3 cm dia optics, was used to measure the temperature distribution in the liquid. A 10 mW He–He laser produced a beam which was expanded by a system of lenses and served as a light source. The use of a laser eliminated the need for a compensation in the reference path of the interferometer.

The test cell used in the experiments consisted of two independent units – a Plexiglass container, with inside dimensions of 10.8 cm high, 8.51 cm wide and 5.07 cm deep, having optical quality glass windows to hold the test fluid and a separate flat plate heat exchanger which was installed in the test cell. Care was taken in the construction of the test cell to insure that the faces were parallel. The vertical flat surface heat exchanger was machined from a copper block by milling channels in one face and then soldering a thin copper plate over the channels. The width of the heat exchanger was such