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Sensor for transient heat flux at a surface with throughflow

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Abstract—A permeable heat flux sensor of the thin-film surface thermometry type is developed. With this unique sensor instantaneous local values of transient heat flux can be measured at a permeable surface which is subject to a rapidly fluctuating heat transfer and at which there may be throughflow. The response time of the sensor is extremely small, $\sim 10^{-10}$ s. Sensitivity of the sensor with the present electronics is 459 μV $^\circ$C^{-1} which gives a surface temperature resolution of 0.0005°C with a 12-bit data acquisition system. Although validated for a particular case of transient local heat flux with throughflow, this new type of sensor is of general applicability.

INTRODUCTION

THE OBJECTIVE was to develop a sensor capable of measuring instantaneous local heat flux at a permeable surface at which there is throughflow and which is subject to a rapidly varying cyclical heat transfer. No sensor for these constraints has been reported. The closest approach to a heat flux sensor satisfying these requirements is that developed by van Heiningen *et al.* [1]. Their analysis of the features and limitations of various fast response heat flux gauges such as thin-film surface thermometry, thick-film calorimetry and hot-film probes [2–6] showed that a thin-film sensor based on a resistance thermometer is the preferred choice for measurement of a rapidly and widely fluctuating transient heat flux. With this type of sensor they could measure local values of such a transient heat flux, but only at an impermeable surface.

Their sensor, a thin-film resistance thermometer flush mounted on the heat transfer surface, was made by deposition of a gold filament about 0.27 μm thick on an electrically insulating substrate of appropriate thermal and mechanical properties. With a heat transfer surface and sensor substrate that is thermally 'semi-infinite', they demonstrated the validity of local transient heat flux measurements obtained with such a sensor.

The present work is devoted to developing a thin-film heat flux sensor which satisfies similar basic constraints, i.e. for cyclical local heat flux, but with the additional requirement that the sensor be applicable for a permeable surface at which there is throughflow. Development of a sensor with such capability would enable experimental investigation of an important class of problems in transport phenomena, those in which there is simultaneous flow and heat transfer at a permeable surface. One industrial problem with large economic consequences for which the development of such a sensor would be particularly relevant is drying paper with combined impingement and through drying. In such a process impingement heat transfer at the surface of a wet sheet of paper moving rapidly through a dryer is affected strongly by throughflow rate at the surface. Thus the objective of the present study relates to challenging problems in both theoretical transport phenomena and industrial process development.

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DEVELOPMENT OF A POROUS HEAT FLUX SENSOR

Selection of the porous sensor material

There are numerous constraints on the choice of material of the porous substrate for the sensor. The substrate throughflow characteristics and transient temperature response must match those of the throughflow surface in which the sensor is mounted; the porous media used must be homogeneous, machinable, durable to 100°C; it must have negligible electrical conductivity to carry a thin-film resistance thermometer. In the present case the ideal solution was possible, i.e. use of the identical material, porous glass, for the sensor substrate and for the permeable heat transfer surface in which the sensor is mounted. Experiments established that a continuous thin film of gold could be vacuum deposited on porous glass obtained from the 3M Company. To achieve the desired range of throughflow rates across the permeable heat transfer surface, a 0.48 m diameter, 13 mm thick porous glass of a uniform particle size, d_p, of 50 μm (3M Grade 55), of porosity 30%, was selected for the cylinder wall. Thus the thickness of the heat transfer surface and the sensor is about $260d_p$. In the application for which the sensor was validated this heat transfer cylinder was rotated rapidly while being subjected to a heating flow on one half, to a cooling

NOMENCLATURE

a	internal surface area per unit volume of the substrate
d_p	particle diameter
C_{Pa}	specific heat of air
C_{Pp}	specific heat of the substrate
h	heat transfer coefficient
H	nozzle-to-impingement surface spacing
h_p	heat transfer coefficient for throughflow air within the substrate per unit internal surface area of the substrate
h'_p	heat transfer coefficient for throughflow air within the substrate per unit volume of the substrate
k_{eff}	effective thermal conductivity of the substrate
k_j	thermal conductivity of air at the temperature of the jet nozzle exit
l	gold film thickness
Nu	Nusselt number, hw/k_j
q_s	surface heat flux
Re	Reynolds number at the jet nozzle exit, $u_j\rho_j w/\mu_j$; Re_h, of heating jet; Re_c, of cooling jet
t	time
Δt	time increment between consecutive samples
Δt_r	time for one revolution of heat transfer surface
T	temperature
T_j	jet nozzle exit temperature
T_p	substrate temperature
T_s	surface temperature of the sensor, of the heat transfer surface
u_j	jet nozzle exit velocity
u_s	throughflow velocity
v_s	heat transfer surface velocity
w	width of the slot jet
x	direction of heat transfer within the sensor
Δx	distance between finite difference layers
y	lateral direction along the heat transfer surface
Z	penetration depth.

Greek symbols

α	thermal diffusivity
μ_j	viscosity at temperature of jet nozzle exit
ρ_a	air density
ρ_j	air density at temperature of jet nozzle exit
ρ_p	substrate density
ρ_s	air density at temperature of heat transfer surface
$\Delta\tau_m$	time for a complete cycle of the fastest heat flux frequency.

flow on the other half. Thus any particular location on the periphery, including the sensor, is in a state of cyclical transient heat transfer.

Fabrication and mounting of the heat flux sensor

From the analysis of sensitivity of the thin-film resistance thermometer based heat flux sensor made by van Heiningen *et al.* [1] it was estimated that a gold film of resistance in the range 100–350 Ω and of thickness in the same order as their sensor (about 0.25 μm) was required for the desired accuracy of the transient heat flux measurement. Extensive development of techniques for depositing very thin, electrically continuous filaments of gold on a porous glass substrate was required. The following procedure evolved for fabricating a porous sensor consisting of a gold film of dimensions 1 mm × 70 mm, i.e. $20d_p$ wide and $1400d_p$ long.

(1) Sections, 10 mm × 100 mm, cut from the same piece of 13 mm thick porous glass used as the heat transfer surface were checked visually, then microscopically, to ensure no irregularities in the sensor substrate.

(2) A 1 mm × 70 mm Scotch tape mask was glued to the substrate surface. Photofabrication of the thin gold filament by the technique of van Heiningen *et al.* [1] was inapplicable because the chemicals damage the porous glass binder. The Scotch tape mask adhered well during deposition, then was easily removed.

(3) About 0.115 g of 0.999% purity gold was vacuum deposited (Edwards Vacuum Coater, Model 306) to produce films in the desired range of resistance before aging, 500–1000 Ω. Many times a seemingly continuous gold film had infinite resistance. Non-conducting and conducting sensors were indistinguishable under a microscope.

(4) Continuous gold filament sensors were 'aged' at 100°C for 6 h, during which time the resistance dropped by half.

(5) The gold film was connected with a high conductivity pure silver paint (Conductive Silver 200 from Degussa A.G.) to 1.3 mm diameter silver lead wires fixed with Easypoxy, an epoxy resin, in holes drilled through the substrate.

(6) For measurement of substrate surface temperature during *in situ* calibration, two thermocouples were glued (high thermal conductivity adhesive, Trabond 2151, Tra-Con Inc., Medford, Massachusetts) flush with the substrate surface, at 5 mm from the gold film ends.

As the gold film thickness was about 0.15 μm, i.e. about 1/300th of the average particle diameter, d_p, in a substrate about $260d_p$ thick, these are truly thin-film sensors.

The 100 mm long sensor was flush mounted centrally in the 200 mm long porous wall of the 0.48 m diameter heat transfer cylinder (Fig. 1). Teflon tape, covered on both sides by a very thin layer of a silicon based adhesive, was applied around the sensor circumference before insertion for ease of sensor replacement and to prevent bypassing of the throughflow between the sensor and the surrounding heat transfer surface.

Calibration of the heat flux sensor

The sensor and associated thermocouples were calibrated in a thermostated bath before installation. However, as the calibration was not sufficiently stable, possibly due to the complex substrate microstructure, *in situ* calibration was made before each run. During calibration the sensor resistance was measured with a 4-decade Wheatstone bridge. The unbalanced bridge output voltage was monitored on a storage oscilloscope. The surface temperature was measured with the two calibrated thermocouples at either end of the thin gold film. The thin-film temperature was taken as the average of the two calibrated thermocouples, which never differed by more than 0.25°C. Stability of calibration under throughflow conditions was also checked by measuring the thin-film resistance and temperature with throughflow. The maximum *in situ* calibration temperature, about 60°C, which exceeds the maximum attained during the runs, was always measured at the end of the experiments to confirm calibration stability.

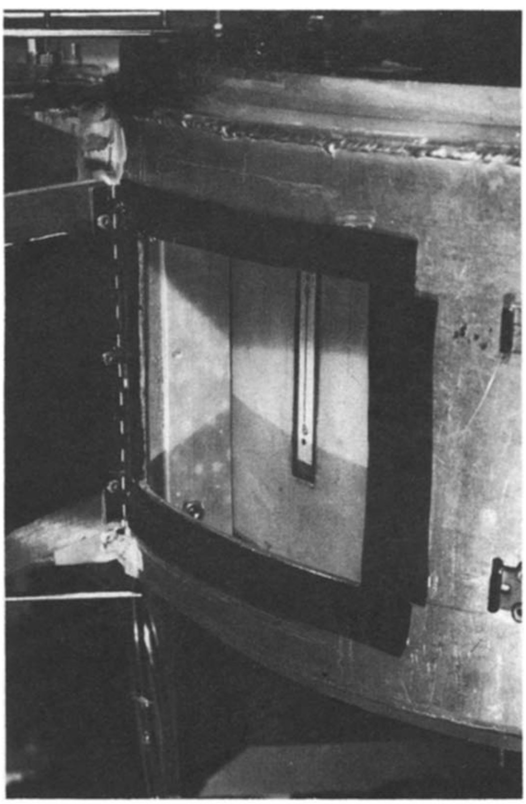

FIG. 1. Sensor in place.

Thermophysical properties of substrate

The specific heat of the porous glass was determined from DSC measurements (Perkin Elmer DSC7) as

$$C_P = 875.51 - 6.48T_p + 7.23 \times 10^{-2} T_p^2.$$

Its thermal conductivity, 0.414 W m^{-1} K^{-1}, was measured using two techniques detailed in ref. [7].

The measured bulk density of 1447 kg m^{-3} compared with the 1600 kg m^{-3} reported by the manufacturer.

INSTRUMENTATION FOR HEAT FLUX MEASUREMENTS

Signal conditioning

The sensor resistance cycles at the frequency of the heat flux being measured, which in the initial use of this sensor, ranged from 4 to 150 Hz. To obtain the resolution of local Nusselt number desired, the sampling frequency of the sensor resistance ranged from 167 to 2500 Hz. Sensor resistance was converted to voltage by a Wheatstone bridge with a 4-decade variable rheostat and a 7-decade multiplier ratio (J. C. Biddle Co., Cat. No. 601022). The bridge output voltage is amplified 2500-fold with a low noise (over the full bandwidth 2 μV RTI and 2 mV RTO) differential amplifier (DANA Model 2820). High frequency noise in the amplifier output is removed by a tuneable low-pass filter (Frequency Devices, Inc., Model 901F1). The low-pass frequency is selected by three decade switches, as well as the multiplier switch. The lower cut-off frequency was set to one half of the data acquisition sampling frequency (Nyquest criterion).

For short contact times the sensor response time to a step change in surface temperature was calculated assuming the gold film was supported on a perfect insulator (the thermal diffusivity of gold is 170 times that of porous glass). The average gold film temperature is within 1% of the new temperature when the Fourier number ($=\alpha t/l^2$) is 0.446. With $\alpha = 6.6 \times 10^{-5}$ m^2 s^{-1} for gold and $l = 0.15$ μm, the sensor response time is 1.5×10^{-10} s.

For a sensor of resistance 186 Ω at the operating temperature the sensor sensitivity is 459 μV °C^{-1}, which is one order of magnitude larger than that typical for thermocouples. Such high sensitivity, combined with a sensor response time of the order of 10^{-10} s justifies the selection of a thin-film sensor to measure local values of rapidly varying, cyclical transient heat transfer.

The self-heating heat flux of the sensor, an unavoidable characteristic, is about 5% of the lowest convective heat flux measured. This, then, is the maximum error from that source. The self-heating flux makes the measured heat transfer slightly too high when the surface is being heated, and vice versa.

Analog-to-digital (A/D) converter

The 27.5 kHz throughput A/D board (Data Translation, DT2801-A) is a high level, 12-bit analog to digital converter (ADC) system. During data acquisition, signals from the sensor and the optical switch for heat transfer cylinder position are sampled by the ADC in the differential mode. Depending on sensor voltage output, A/D board gains 1 or 2 are selected via the data acquisition software in addition to a gain of 2500 from the amplifier.

The minimum voltage change that can be differentiated by the ADC is 0.0012 V ($2^{12} = 4096$, variation from 0 to 5 V for gain 2). A variation of 0.0012 V corresponds to a change of sensor temperature of 0.0005°C, which is therefore the resolution of the heat flux sensor developed here.

COMPUTATION OF INSTANTANEOUS LOCAL HEAT FLUX

Without throughflow

Conditions for the heat flux sensor are given by the transient heat conduction equation

$$\frac{1}{\alpha}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial x^2} \tag{1}$$

where T and α ($= k_{eff}/\rho_p C_{p_p}$) are the porous substrate temperature and thermal diffusivity, respectively. This equation is integrated in t and x for the control volume shown in Fig. 2 by assuming an explicit behavior in time and a linear temperature variation between grids in the x-direction. After rearranging, the equation obtained is

$$T_i^1 = \frac{\alpha\Delta t}{\Delta x^2}(T_{i+1}^0 + T_{i-1}^0) + \left(1 - 2\frac{\alpha\Delta t}{\Delta x^2}\right)T_i^0. \tag{2}$$

For the numerical stability requirement that all equation (2) coefficients be positive, the minimum value of Δx must be

$$\Delta x = (2\alpha\Delta t)^{1/2}. \tag{3}$$

With this limiting condition, equation (2) becomes

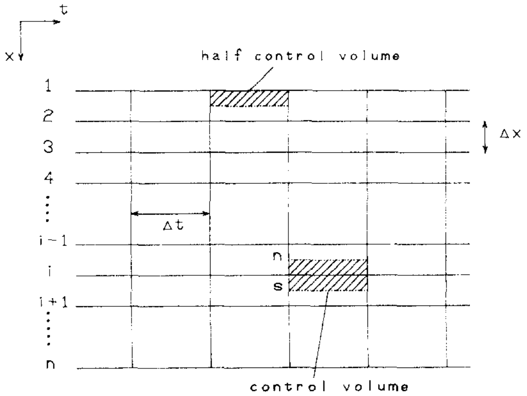

FIG. 2. Control volumes.

$$T_i^1 = \tfrac{1}{2}(T_{i+1}^0 + T_{i-1}^0). \tag{4}$$

The instantaneous surface heat flux, q_s, calculated from Patankar's [8] half-control volume concept, Fig. 2, is

$$q_s = \frac{k_{eff}}{\Delta x}(T_1^1 - T_2^0). \tag{5}$$

Since equations (4) and (5) are explicit, a simple marching technique is used to calculate the temperature distribution in the substrate at time 1 from the previous values at time 0. The initial and boundary conditions required for equations (4) and (5) are the measured surface temperature T_1 and the equality $T_{n-1} = T_n$ because, under the validation test condition of rapid cyclic transient heat transfer at the sensor surface, the sensor substrate can be considered thermally semi-infinite. The latter assumption was verified as follows, and by temperature measurements at that surface.

For the semi-infinite analysis, the cyclical variation of temperature at the surface of the rotating heat transfer cylinder (and sensor) may be approximated as sinusoidal. For a sinusoidal surface temperature variation of frequency $1/\Delta t_r$ s^{-1}, the thermal penetration length [9] is

$$Z = -\ln(0.001)/(\pi/\Delta t_r\alpha)^{1/2} \tag{6}$$

where Z is the depth where the amplitude of the temperature fluctuation is 0.1% of that at the heat transfer surface. For even the slowest r.p.m. used, Z from the above equation is 3 mm (i.e. $60d_p$), much less than the sensor substrate thickness, 13 mm (i.e. $260d_p$). Thus the semi-infinite substrate assumption is justified. The number of finite difference layers 'n' is calculated as the integer value of the ratio $Z/\Delta x$. For about 500 samples per rotation, $n = 44$.

The initial condition chosen for the solution of equation (4) is the average temperature of the heat transfer cylinder. The effect of this initial temperature disappears completely after two revolutions of the heat transfer cylinder. After the correct temperature distribution in the substrate is obtained, equations (4) and (5) are solved consecutively to obtain the local flux distribution at the heat transfer surface, i.e. the heat flux at the surface of the heat flux sensor.

The surface temperature boundary condition for equations (4) and (5) is that corresponding to the sensor resistance averaged over 50 rotations of the heat transfer cylinder.

The errors in computation of heat flux amplitude and phase shift when equations (4) and (5) are used instead of the exact solution to the unsteady heat conduction equation estimated by Baines [4], are:

error in amplitude

$$1 - \left(\frac{\sin\left(\dfrac{2\pi\Delta t}{\Delta\tau_m}\right)}{\dfrac{2\pi\Delta t}{\Delta\tau_m}}\right)^{1/2} \tag{7}$$

error in phase shift

$$\frac{\pi\Delta t}{3\Delta\tau_m} \tag{8}$$

where $\Delta\tau_m$ is the time for a complete cycle of the fastest heat flux frequency. In the validation tests, during which the surface of the rotating heat transfer cylinder, including the sensor, is subjected to an alternation of heating and cooling by jets, the fastest complete cycle occurs within about 50 mm of the heat transfer surface. This condition, for the fastest intended rotational speed of the heat transfer cylinder, gives $\Delta\tau_m = 0.0067$ s. To obtain a resolution of the instantaneous Nusselt number corresponding to 16 values of Nu over 50 mm of the surface moving at this speed, the fastest sampling frequency used was 2500 Hz, i.e. $\Delta t = 0.0004$ s. From equations (7) and (8) the corresponding errors are very small, 1.3% in heat flux amplitude and 3.6° in phase shift.

With throughflow

The energy equations in the heat flux sensor, including heat transfer between throughflow air and the substrate, become:

substrate

$$\rho_p C_{Pp}\frac{\partial T_p}{\partial t} = k_{eff}\frac{\partial^2 T_p}{\partial x^2} - h'_p(T_p - T_a); \tag{9}$$

air

$$\rho_a C_{Pa}\frac{\partial T_a}{\partial t} = -\rho_a C_{Pa} u_s \frac{\partial T_a}{\partial x} + h'_p(T_p - T_a) \tag{10}$$

where subscripts p and a denote the substrate and air, respectively.

If within the substrate the local average substrate temperature equals the local air temperature, i.e. $T_p = T_a = T$, the energy accumulation term for air may be omitted as negligible relative to that for the substrate and equations (9) and (10) reduce to the single energy equation

$$\rho_p C_{Pp}\frac{\partial T}{\partial t} = k_{eff}\frac{\partial^2 T}{\partial x^2} - \rho_a C_{Pa} u_s \frac{\partial T}{\partial x}. \tag{11}$$

To test whether this simplification applies, h_p was estimated from the equation (12) correlation for packed beds and porous media [10]

$$J_h = 0.84 Re_p^{-0.51}, \quad 0.01 < Re_p < 50 \tag{12}$$

where

$$Re_p = \frac{\rho_a u_s}{a\mu} \quad \text{and} \quad J_h = \frac{h_p}{\rho_a C_{Pa} u_s} Pr^{2/3}.$$

For the minimum (0.09 m s^{-1}) and maximum (0.5 m s^{-1}) throughflow air velocities the predicted values of h_p are 215 and 500 W m^{-2} K^{-1}.

From experiments made to check the uniformity of permeability of the porous glass heat transfer surface, the interfacial area, a, for the porous material used is estimated as 20000 m^2 m^{-3} using the Kozeny–Carman relation [11] for laminar throughflow. Thus the predicted values of $h'_p(=h_p a)$ are 4.4×10^6 and 10.2×10^6 W m^{-3} K^{-1} for the minimum and maximum throughflow rates.

Nusselt number distributions were calculated twice, once from equation (11) and once from equations (9) and (10) using the estimated h'_p values with the assumption that air enters the substrate at the substrate surface temperature. As these Nu distributions are completely indistinguishable, use of the simplification $T_p = T_a = T$ in the present case is justified.

The assumption that air enters the substrate at the exterior surface temperature of the substrate is verified as follows. Considering air flow in the pores of the sensor and heat transfer cylinder to be laminar, an 'adaptation time' of the fluid to the thermal boundary condition imposed by the solid may be expressed in terms of the Fourier number

$$Fo_t = 4\frac{\alpha t}{d_p^2}. \tag{13}$$

According to the manufacturer of the porous glass, supported by our microphotographs, mean pore size is about the same as the mean particle size, $d_p = 50$ μm. Therefore, $d_p/2$ is used as the characteristic length in equation (13). For a porous cylinder of mean temperature about 50°C, α is 2.8×10^{-5} m^2 s^{-1}. At the maximum superficial throughflow velocity, 0.5 m s^{-1}, pore air velocity for $\varepsilon = 0.3$ is 1.7 m s^{-1}. Therefore, air travels $1d_p$ in 3×10^{-5} s. The corresponding Fourier number is 1.3. At a Fourier number greater than 1, adaptation to the new thermal boundary condition is complete. Adaptation may even be faster since the flow at the pore entrance may be transitional or turbulent, not laminar.

The discretized form of equation (11) is

$$T_i^1 = \left(\frac{\alpha\Delta t}{\Delta x^2} + \frac{\rho_a C_{Pa} u_s}{\rho_p C_{Pp}}\frac{\Delta t}{\Delta x}\right)T_{i-1}^0 + \left(\frac{\alpha\Delta t}{\Delta x^2}\right)T_{i+1}^0 + \left(1 - 2\frac{\alpha\Delta t}{\Delta x^2} - \frac{\rho_a C_{Pa} u_s}{\rho_p C_{Pp}}\frac{\Delta t}{\Delta x}\right)T_i^0. \tag{14}$$

For numerical stability, the coefficient of T_i^0 was set equal to zero, which specifies the relation between Δx and Δt as

$$\Delta x = \frac{C\Delta t + ((C\Delta t)^2 + 8\alpha\Delta t)^{1/2}}{2}$$

where

$$C = \frac{\rho_a C_{Pa} u_s}{\rho_p C_{Pp}}.$$

Thus the finite difference equation with throughflow is

$$T_i^1 = \left(\frac{\alpha\Delta t}{\Delta x^2} + \frac{\rho_a C_{Pa} u_s}{\rho_p C_{Pp}}\frac{\Delta t}{\Delta x}\right)T_{i-1} + \left(\frac{\alpha\Delta t}{\Delta x^2}\right)T_{i+1} \tag{15}$$

which, for $u_s = 0$, reduces to the no-throughflow equation.

Finally the value of heat flux, q_s, at the surface of the sensor in the presence of throughflow at the surface of the sensor, the objective in this development, is obtained from equation (5).

Uniformity of throughflow velocity

Equation (15) assumes uniformity of throughflow velocity. This assumption depends on the degree of uniformity of permeability and of local pressure over the heat transfer surface. Uniformity of permeability was checked by measuring throughflow rate and pressure drop across the heat transfer surface with various fractions of the surface blocked to throughflow by plastic sheets. The linear relation obtained confirmed the uniformity of permeability of the porous glass. The variation of throughflow velocity due to local static pressure variation under the impinging jets used in the validation tests was less than $\pm 13\%$.

POROUS SENSOR AS A HEAT FLUX MEASURING INSTRUMENT

With a heat flux sensor for a throughflow heat transfer surface, fabrication of a thin-film surface thermometry sensor by deposition of a gold film on the surface of a porous substrate is accompanied unavoidably by some gold deposition on substrate particles below the surface layer. Examination under a microscope revealed areas of gold film at as much as three particle diameters below the surface. However, the great difficulty, noted earlier, in obtaining an electrically continuous gold film should be emphasized. Thus even at the surface layer of the particulate substrate, where the surface of the particles is completely accessible during gold deposition, achieving a thin film without a discontinuity over the 70 mm ($1400d_p$) length of the sensor filament is extremely difficult. It is therefore not remotely probable that a continuous gold film could form in the interior of the substrate where particles are shielded from gold deposition by the layer of particles at the exterior surface. Such regions of gold film below the surface, electrically isolated in the lateral direction, would have a negligible effect on the resistance of the filament at the sensor surface and hence on the value of heat flux at the surface obtained with this thin-film sensor.

Moreover, one may determine the sensitivity of Nu profiles to any such averaging effect of gold film over more than the surface layer of the 50 μm diameter substrate particles. Thus a Nu profile may be recalculated taking the sensor temperature as the average of the surface temperature as normally determined and the temperature at one grid node ($\sim 1d_p$) below the surface. A Nusselt number profile as normally determined and for this hypothesis of 'averaged' surface temperatures are compared in Fig. 3. This is an extreme case as it assumes the gold film distributed uniformly on the first and second layers of particles. Even so, as the effect of averaging is not large, Fig. 3,

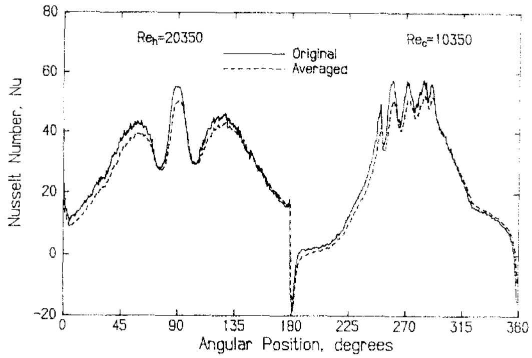

FIG. 3. Nu profiles: normally determined and obtained with averaged surface temperature.

this test confirms that the effect of having some areas of gold film below the surface layer of substrate particles is indeed insignificant.

VALIDATION OF THE HEAT FLUX SENSOR

Any heat flux sensor of a new type and with novel capabilities requires validation. The sensor developed here for local transient heat transfer for a surface at which there is throughflow was subjected to validation tests with the sensor mounted in a heat transfer surface being alternately heated and cooled by moving rapidly under impinging slot jets of heating and cooling air, in an experimental facility described in ref. [7]. In brief, the test facility involved a 0.48 m diameter rotating, cylindrical, permeable heat transfer surface, with throughflow. The external surface of this cylinder was subjected on one side to cooling by multiple slot jets and, at a position about 180° away, to heating by a single slot jet. The shortest time for a complete cycle of the transient heat flux between adjacent multiple jets was about 0.0067 s. However, the sensor is not restricted to the study of impingement flows but is generally applicable for throughflow heat transfer surfaces subject to cyclical heat transfer. For $h = q_s/\Delta T$ in the validation test system, the relevant ΔT is $T_j - T_s$, and jet nozzle width, w, is the characteristic dimension for Nusselt and Reynolds number. The results of this sensor test program are now presented.

A basic test is to establish that a new sensor produces heat transfer coefficients which are independent of the temperature difference. This test was made by varying the temperature difference for impingement heat transfer, $T_j - T_s$, by a factor of 2.7 for the heating slot jet, and by a factor of 2 for the cooling jets. The results of these tests, expressed as average Nusselt number for a heat transfer surface $16w$ wide for the single jet, Fig. 4, and as average Nusselt number for the multiple jets, Fig. 5, confirm that this heat flux sensor provides Nusselt number data which are independent of the temperature driving force.

A second validation test was to compare published results with local profiles of impinging jet transient heat transfer obtained with the new sensor. Two ref-

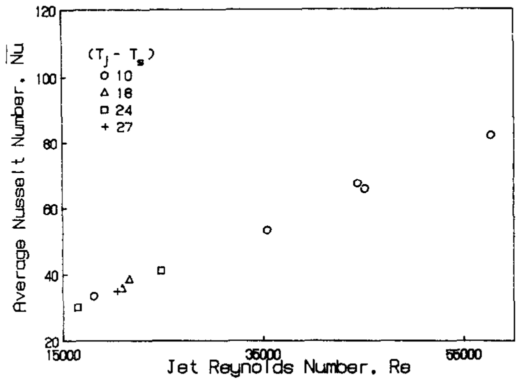

FIG. 4. Independence of Nusselt number from ΔT: test with heating jet.

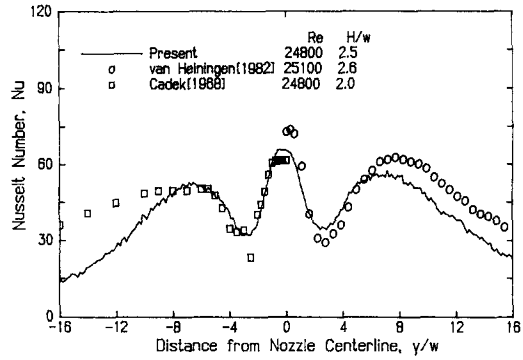

FIG. 6. Sensor test: profiles of local Nu for a slot jet at $Re = 24\,800$.

erence studies were used. Cadek [12] employed a thin-disk heat flux sensor, known as a Gardon foil. His sensor, of diameter 0.9 mm, gave a resolution of 0.28 of the slot jet nozzle width for profiles of local Nusselt number at a stationary impermeable impingement surface. Due to the difficulty of accurate calibration of the Gardon foil, Cadek used it to measure heat transfer profiles relative to the stagnation heat transfer. The impermeable thin-film heat flux sensor of van Heiningen *et al.* [1], described earlier, differs basically from the new sensor of the present study, which is porous. Their sensor, proven to be very sensitive and fast responding, on calibration profiles of local heat transfer of higher resolution and accuracy with a moving impingement surface than those of Cadek with a stationary surface.

The profiles of local Nusselt number under a single impinging slot jet obtained with the new type of sensor, Figs. 6 and 7, are for conditions of jet Reynolds number, Re, and nozzle-to-impingement surface spacing, H/w, matching as closely as possible those of the two best reference studies available. In order to bring all results to a common Reynolds number, Cadek's results were slightly adjusted for this effect using standard procedures, as detailed in ref. [7]. For these test conditions the Nusselt number under highly turbulent jets reflects complex flow phenomena at the impingement surface. The central peak corresponds to a thin, laminar boundary layer, while the off-stagnation minima and maxima correspond, respectively, to the start and completion of transition to a turbulent boundary layer. Local Nusselt number profiles of this complexity constitute a demanding validation test of any sensor.

In the region of the central peak the heat transfer measured by the new sensor is intermediate between the reference studies at the lower Re, Fig. 6, but is somewhat higher at the higher Re, Fig. 7. In the latter case, stagnation region heat transfer for a high stagnation pressure is enhanced with a porous sensor due to transient local throughflow. This small difference is of no practical significance because a permeable sensor is not needed for studies at an impermeable heat transfer surface.

The design features necessary to obtain a porous sensor applicable with throughflow result in a sensor having slightly less lateral resolution along the heat transfer surface than the previous impermeable sensors, those of Cadek [12] for a stationary surface and van Heiningen *et al.* [1] for a moving surface. Thus the new sensor slightly overestimates the Nusselt number at the off-stagnation minimum and correspondingly underestimates the off-stagnation maximum. As these point values of Nu are of limited

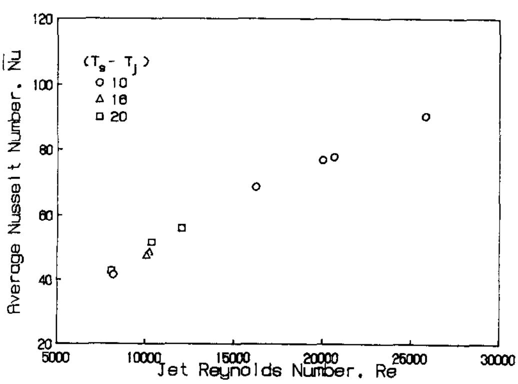

FIG. 5. Independence of Nusselt number from ΔT: test with cooling jet.

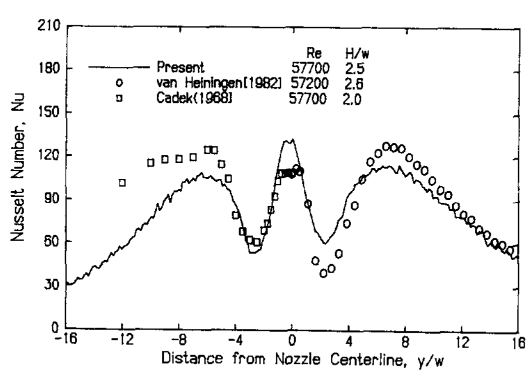

FIG. 7. Sensor test: profiles of local Nu for a slot jet at $Re = 57\,700$.

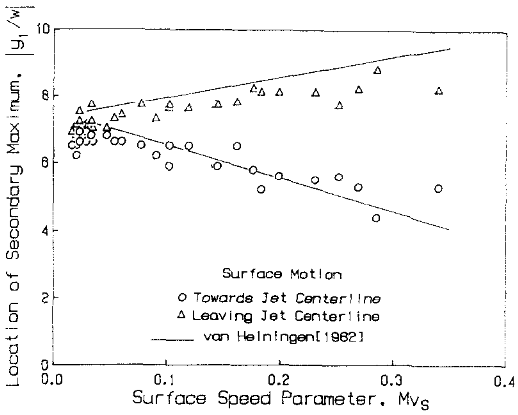

FIG. 8. Sensor test: effect of heat transfer surface speed on position of off-stagnation maxima.

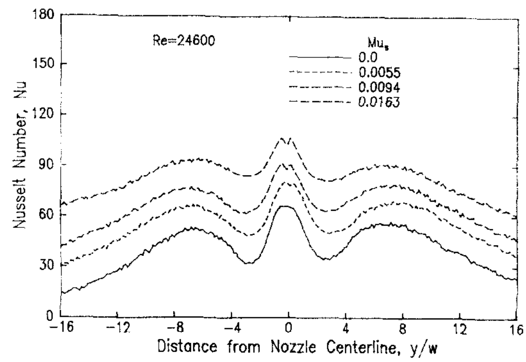

FIG. 9. Sensor demonstration: effect of throughflow on transient local Nusselt number.

interest, and as these effects compensate, the new sensor provides sufficiently precise resolution of this complex heat transfer profile.

The third type of validation was to test the aspects of high sensitivity and fast response of the new sensor by monitoring heat transfer at an impingement surface moving rapidly under an impinging jet. The only heat flux sensor with this capability for impermeable heat transfer surfaces is that of van Heiningen *et al.* [1], which they used only for a single impinging jet. Their single jet results with this sensor showed that Nu profiles are depressed on the side where surface motion is towards the jet nozzle but are enhanced on the leaving side. The Nu profiles at a rapidly moving impingement surface obtained with the porous sensor developed in the present study are essentially identical with their profiles for comparable conditions. A particularly sensitive characteristic of such Nu profiles is the extent of shift in position of the off-stagnation maxima in the direction of heat transfer surface motion. The shifts in position of the off-stagnation maxima on the approaching and leaving side of the slot jet nozzle as determined with the new sensor and by the impermeable sensor of van Heiningen [13] are compared on Fig. 8. Heat transfer surface speed is characterized by the non-dimensional parameter, $Mv_s = \rho_s v_s/\rho_j u_j$, described by ref. [1], with $Mv_s = 0$ for a stationary surface. The good agreement for this sensitive feature of Nusselt number profiles for a jet impinging on a rapidly moving surface provides further validation that the permeable heat flux sensor developed in the present study has excellent sensitivity and fast response characteristics, and provides reliable measures for rapidly changing transient heat transfer.

The three types of tests described above complete the validation of the new type of sensor. No direct validation is possible for the unique feature of this permeable sensor, its ability to measure rapidly changing heat flux when there is throughflow at the heat transfer surface, as no comparative data exist because no previous sensor had this capability. This potential of the new sensor is displayed on the sample set of profiles, Fig. 9, of transient local Nusselt number under a slot jet impinging on a fast moving surface with a range of rates of throughflow at the heat transfer surface. Throughflow rate is characterized by the non-dimensional parameter, $Mu_s = \rho_s u_s/\rho_j u_j$. The maximum value of the throughflow parameter, $Mu_s = 0.0163$, for the results of Fig. 9 corresponds to a very high throughflow. Specifically, this Mu_s value corresponds to 52% of the nozzle exit flow leaving the system as throughflow for a heat transfer surface extending 16 nozzle widths on either side of the nozzle centerline. A detailed discussion of these results, which is beyond the scope of the present communication, appears in ref. [7]. The enhancement of local heat flux by throughflow for jets impinging on a moving heat transfer surface is thus documented for the first time with the porous, thin-film flux sensor developed in the present study.

SUMMARY

A permeable, high sensitivity, fast responding thin-film heat flux sensor was developed which makes it possible to measure rapidly changing local heat flux at a surface in the presence of throughflow at the heat transfer surface. This sensor and the associated measurement system are shown to produce accurate and reliable results for a specific case, one where the rapidly changing heat flux is due to jets impinging on a moving surface with throughflow. A sensor of this new design is however not restricted to that particular case but is of general applicability for measurement of rapidly changing heat flux at permeable surfaces with throughflow. This sensor is currently being used in the study of transport phenomena not previously amenable to experimental measurement.

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CAPTEUR DE FLUX THERMIQUE TRANSITOIRE POUR UNE SURFACE AVEC ECOULEMENT TRAVERSANT

Résumé—On a mis au point un capteur de flux thermique perméable, du type de la thermométrie des films minces de surface. Ce capteur unique permet de mesurer la valeur locale d'un flux thermique transitoire en fluctuation rapide à une surface perméable soumise à un écoulement traversant. Le temps de réponse de capteur est extrèmement court, soit de 10^{-10} s. La sensibilité du capteur, avec l'équipement électronique actuel, est de 459 μV $°C^{-1}$, ce qui permet une résolution de la température de la surface de 0,0005°C, à l'aide d'un système d'acquisition de données à 12 bits. Bien que verifié pour un cas particulier d'un flux thermique transitoire avec air traversant à la surface, ce capteur d'un genre nouveau a un champ d'application général.

TEMPERATUR-MESSFÜHLER FÜR SICH ZEITLICH ÄNDERNDE WÄRMESTRÖME AN OBERFLÄCHEN MIT DURCHFLUSS

Zusammenfassung—Ein durchlässiger (permeabler) Wärmestrom-Messfühler wurde auf der Basis der Dünnschicht-Oberflächen-Temperaturmessung entwickelt. Mit diesem speziellen Messfühler Können augenblickliche, lokale Werte eines sich zeitlich ändernden und rasch schwankenden Wärmestromes an einer durchlässigen Oberfläche gemessen werden. Die Einspielzeit des Messfühlers ist sehr Kurz, $\sim 10^{-10}$ s. Die Ansprechempfindlichkeit mit der momentan verwendeten Elektronik beträgt 459 μV $°C^{-1}$ was einem Oberflächentemperatur-Auflösungsvermögen von 0,0005°C mit einem 12-bit Datenerfassungssystem entspricht. Obwohl für einen ganz speziellen, sich zeitlich ändernden Wärmestrom mit Durchfluss entwickelt, so bietet dieser neue Messfühler einen breiten Anwendungsbereich.

ДАТЧИК НЕСТАЦИОНАРНОГО ТЕПЛОВОГО ПОТОКА У ПОВЕРХНОСТИ СО СКВОЗНЫМ ТЕЧЕНИЕМ

Аннотация—Разработан проницаемый датчик теплового потока типа датчика, работающего на принципе поверхностной термометрии тонкой пленки. С использованием этого оригинального датчика можно измерять мгновенные локальные значения нестационарного теплового потока у проницаемой поверхности, подверженной действию теплопереноса, характеризующегося высокочастотными пульсациями, в которой может осуществляться сквозное течение. Время отклика датчика предельно мало, $\sim 10^{-10}$ с. В рассматриваемых условиях чувствительность датчика составляет 459 мкВ $°C^{-1}$, что обеспечивает разрешающую способность определения температуры поверхности 0,0005°C при 12-битной системе сбора данных. Несмотря на то, что эффективность разработанного датчика нового типа проверяется в частном случае нестационарного локального теплового потока при наличии сквозного течения, он имеет широкий диапазон применения.