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Gas–solid heat exchange in a fibrous metallic material measured by a heat regenerator technique

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Abstract—The convective heat transfer properties of a porous metallic fibre material used in gas surface combustion burners are studied. The important parameter governing the heat transfer between hot gas and metal fibre—the heat transfer coefficient—is measured using a non-steady-state method based on cyclic counterflow heat regenerator theory. The factors controlling the ranges of experimental conditions that can be used are studied. A correlation between gas flow rate and heat transfer is obtained for laminar flows, showing a rapid increase in heat transfer coefficient with increasing gas flow. The heat transfer coefficient is significantly lower than previously assumed.

1. INTRODUCTION

RECENT research [1] in the design of surface combustion gas burners has used metal fibre structures as the burner medium. In such devices a combustible mixture is forced through the permeable medium and burns within the surface layer, heating it to incandescence.

The sintered metal fibre material used, Bekitherm,† has been developed and is produced by N. V. Bekaert S.A. The fibres used in this study are of a refractory steel, Fecralloy‡ with a diameter of 22 μm. The alloy forms a protective alumina coating on the fibres. The structure of the material can be seen in Fig. 1, a scanning electron micrograph of the surface. The fibres are randomly oriented in layers. The material has a porosity of 80% and is produced in sheet form several millimetres thick.

Work is in progress developing a mathematical model to describe the performance of these burners and a prerequisite for this has been the determination of the important heat transfer parameters. In a recent paper [2] we described the measurement of the thermal diffusivity of the material. This paper concentrates on the study of convective heat transfer between the flowing gas and the fibre and we describe a novel application of heat regenerator theory to measure the heat transfer coefficient for this process.

We have attempted to obtain a correlation between flow rate and heat transfer for very low Reynolds number flows, $Re = 0.1$–2.0. These conditions correspond to typical flow rates of gas used in the burners which operate with specific thermal inputs in the range 100–5000 kW m^{-2}. Because of the scarcity of correlation data for flow through porous material and at such low Reynolds numbers, the same method was used to determine correlations for bronze and steel gauzes—materials for which some data were available.

In Section 2 we describe the regenerator theory and the model of the fibre structure to be used to determine the heat transfer coefficient. In Section 3 we describe the experimental method and in Section 4 the analysis of the experimental data. The results are presented and discussed in Section 5 and compared with a correlation for single wires taken from the literature. Finally the implications of our findings for surface combustion burner operation are discussed.

2. THEORY AND MODEL

A simple steady-state method to determine the heat transfer coefficients requires the measurement of the difference between the average gas and solid temperature during flow through the sample material. Using metallic fibre material in surface combustion gas burners [1], it is observed that although the downstream surface at which the burning occurs is very hot—typically 900°C—the upstream surface remains cool, around the temperature of the incoming gas. There is clearly a steep temperature gradient in the material arising from the low thermal conductivity parallel to gas flow. The metal fibre material has such a short thermal equilibration length that unless unrealistically thin samples are used, the solid and gas quickly equilibrate over a short distance for any entering gas temperature. Since it is not possible to maintain a constant temperature difference between gas and solid, a measurement varying with time is

† Bekitherm is a trademark of N. V. Bekaert S.A., Zwevegem, Belgium.
‡ Fecralloy is a trademark of UKAEA, Didcot, U.K.

NOMENCLATURE

B	pre-indexation coefficient in heat transfer relationship
C	heat capacity
d	fibre diameter
G	mass flux
h	heat transfer coefficient
h^*	heat transfer coefficient averaged over surface per unit volume
ΔH	enthalpy of combustion
L	regenerator length
n	index coefficient in heat transfer relationship
P	heating (and cooling) period duration
R	rate of combustion
S_t	transverse fibre mean spacing
t	time
T	temperature
u	gas velocity
V_s	solid volume
V_t	total volume
z	length.

Greek symbols

α	dimensionless heat transfer parameter
β	porosity
λ	thermal conductivity
Λ	non-dimensionalized length of regenerator
μ	dynamic viscosity
ξ	non-dimensional distance
η	non-dimensional time
Φ	correction function to heat transmission coefficient
Π	non-dimensionalized heating (and cooling) period
ρ	density
τ_c	thermocouple response time
τ_d	dead volume residue time.

Subscripts

g	gas
s	solid
1	cooling period
2	heating period.

Dimensionless groups

Nu	$hd/\lambda_g = B\,Re^n$
Re	Gd/μ
Ω	$d^2\rho_s C_s/P\lambda_g$.

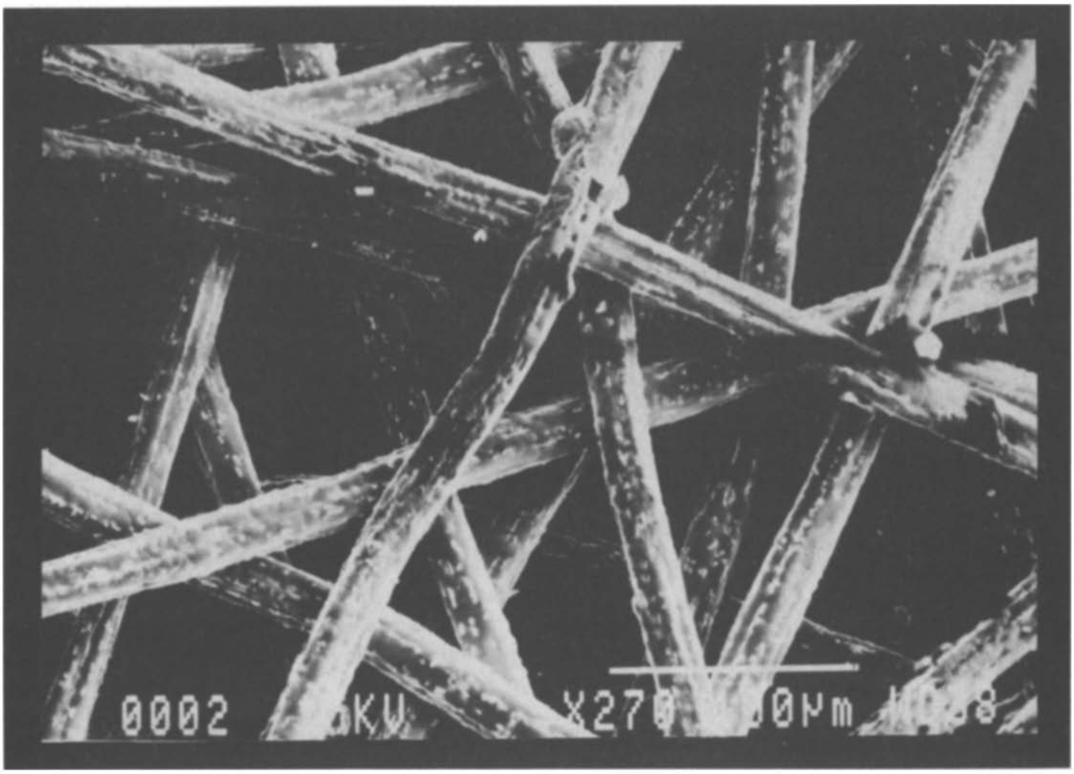


FIG. 1. Scanning electron micrograph of 22 μm metal fibre material showing layered structure of randomly laid fibres.

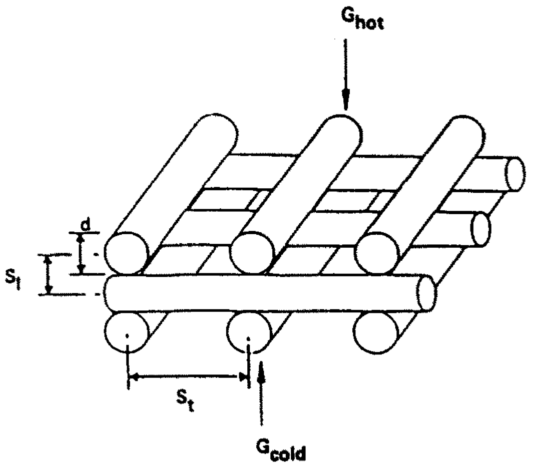


FIG. 2. Model of burner material as regular stacked array.

required—as in a regenerator. In addition, it is clearly impossible to measure solid (i.e. fibre) temperatures as the wires are small. In the regenerator, measurements of the cycled gas temperature are sufficient to characterize the heat transfer coefficient.

2.1. *Material structure*

The material is constructed from layers of sintered fibres of Fecralloy giving rise to an anisotropic conductivity which is large within a layer (typically 1 W $m^{-1}K^{-1}$) but much smaller (0.1 W $m^{-1}K^{-1}$) between layers [2]. The thermal conductivity of solid Fecralloy is 9.5 W m^{-1} K^{-1}. The structure may be viewed for heat transfer purposes as composed of mutually orthogonal layers of fibres consisting of repeated cells (Fig. 2(a)) any one of which forms a crystallographic type unit which characterizes the burner material [3]. The cell dimensions are (S_t, S_t, $2d$), and assuming the fibres to be cylindrical the porosity β depends on the ratio of solid volume V_s to total volume V_t

$$\beta = \frac{V_t - V_s}{V_t}. \tag{1}$$

This yields the mean fibre spacing within a layer S_t in units of the fibre diameter d, as

$$\frac{S_t}{d} = \frac{\pi}{4(1-\beta)}. \tag{2}$$

When heat is transferred from gas to solid, it occurs perpendicular to the direction of forced convection, so that the overall take-up of heat by the solid is governed by the relative thermal inertia of gas and solid.

2.2. *Fundamental equations*

In a counterflow regenerator, hot and cold gas flow alternately and in opposite directions through the material [4]. In the first half of the cycle, the hot gas gives up its heat to the cold material. The material heats up and the gas leaves the regenerator cooled. In the second period, the material releases the heat to the cold gas flow, cooling the material. A temperature–time profile can be obtained for this regenerative (hot and cold) cycle at various positions within the bulk of the sample. For a given cycle in the middle, once the regenerator has stabilized, it can be shown [4, 5] that the heat transfer coefficient is directly proportional to the ratio of the change in gas temperature over a hot (or cold) period to the temperature difference between corresponding points in time during the hot and cold periods.

If the heat transfer between the gas and the solid surface is governed by the coefficient h, then from the cylindrical geometry of the unit cells, the heat transmission coefficient (i.e. heat flux exchange per unit length) is governed by

$$h^* = \frac{hA^s}{V^t} = \frac{4(1-\beta)}{d}h \tag{3}$$

where A^s is the surface area of fibrous heat storing mass within a unit cell. The gas and solid temperature equations are derived from a heat balance between heat flux and exchange between the gas and solid [7]

$$\frac{\partial T_g}{\partial z} = \frac{-h^*}{GC_g}(T_g - T_s) \tag{4a}$$

$$\frac{\partial T_s}{\partial t} = \frac{h^*}{\rho_s C_s(1-\beta)}(T_g - T_s) \tag{4b}$$

where we have longitudinally linearized heat transfer across a unit cell. If we define non-dimensionalized space and time variables ξ and η by

$$\xi = \frac{h^* z}{GC_g}, \quad \eta = \frac{h^* t}{\rho_s C_s(1-\beta)} \tag{5}$$

we obtain

$$\frac{\partial T_g}{\partial \xi} = -(T_g - T_s) \tag{6a}$$

$$\frac{\partial T_s}{\partial \eta} = (T_g - T_s). \tag{6b}$$

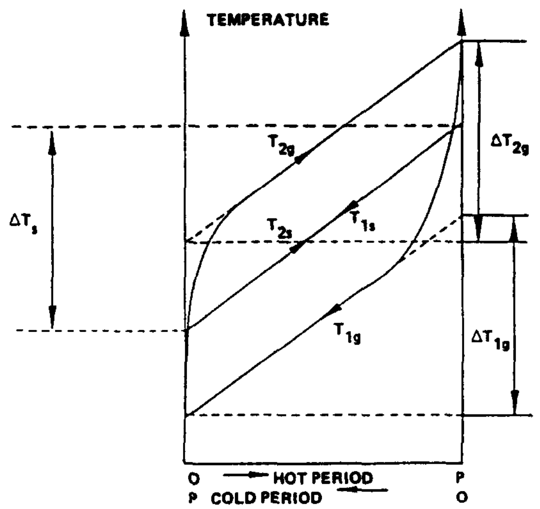
From equation (4) we define the non-dimensionalized regenerator length and period by [5]

$$\Lambda = \frac{h^* L}{GC_g}, \quad \Pi = \frac{h^* P}{\rho_s C_s(1-\beta)}. \tag{7}$$

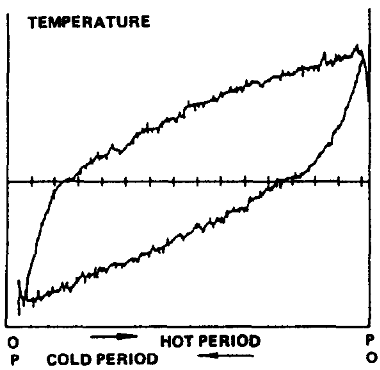
2.3. *Formal solution*

As hot air enters, the centre gas temperature increases, and decreases on switching to the cold period, in a linear fashion over the central part of the cycle. (Elsewhere in the regenerator, the hot and cold period gas temperatures may not be linear, with time.) If the variations in temperature with respect to time and space are nearly linear throughout the regenerator, then one can show that the treatment of the regenerator equations can be much simplified—the zero eigenfunction approximation [6].

When the thermal conductivity of the solid is small



(a)



(b)

FIG. 3. Gas temperature at the centre of a regenerator during one cycle: (a) ideal response; (b) experimentally observed response.

in the direction of gas flow, then at the centre of the regenerator, the gas and solid temperatures are linear functions of time. This response is shown in Fig. 3(a). During the cycle, at any point in time, the difference between wall and gas temperature is also constant as a consequence of the parallel temperature traces [5, 7]

$$T_{2g}-T_s = T_s-T_{1g} = 1/2(T_{2g}-T_{1g}). \qquad (8)$$

The only regions where equation (8) does not apply are at the switchover between periods when the temperature changes from hot to cold and vice versa are clearly not instantaneous, at the beginning of a new period. Accordingly, periods and switching have to be optimized to reduce these effects as will be seen below. In addition these rapid changes may be corrected by using a function Φ which depends on the dimensionless group

$$\Omega = \frac{d^2\rho_s C_s}{P\lambda_s} \qquad (9)$$

where d is the fibre diameter [6, 7].

Since the response curves are approximately linear and parallel, it can be seen that in the middle of the regenerator, over a period Π the magnitude of the solid temperature change (ΔT_s) is the same as that of the gas over a period although the signs for gas temperature change are opposite between a hot and cold cycle. The linearity of the graph shows that the temperature changes over the centre of each period are identical [5, 7]

$$\Delta T_{2g} = -\Delta T_{1g} = \Delta T_s. \qquad (10)$$

In order to obtain a value of h^*, the heat transmission coefficient, we may solve equation (6) to formally obtain

$$h^* = \frac{\rho_s C_s(1-\beta)}{(T_g-T_s)}\frac{\partial T_s}{\partial t}. \qquad (11)$$

Integration over a heating cycle P

$$h^* = \frac{\rho_s C_s(1-\beta)\Delta T_s}{P(T_{2g}-T_s)} \qquad (12)$$

where ΔT_s is the change in solid temperature over a cycle. We may now use relationships (11) and (12) pertinent to the zero eigenfunction solution to obtain

$$h^* = \frac{2\rho_s C_s(1-\beta)}{P}\frac{\Delta T_{2g}}{T_{2g}-T_{1g}}. \qquad (13)$$

Thus, measuring the temperature profile of the gas at the centre of the regenerator during a single cycle at equilibrium operation is sufficient to calculate the heat transfer coefficient. The value of ΔT_{2g} can be obtained by measuring the overall gas temperature change extrapolated over a period and the value of $T_{2g}-T_{1g}$ obtained by comparing the hot and cold temperatures at the corresponding points in time within the cycle. Alternatively, numerical simulation of regenerator operation using equations (6a) and (6b) can be used with h^* as a parameter [8]. However, previous workers have shown that in the middle of the matrix, assuming a reasonable approximation to the zero eigenfunction exists, then the analytical solution is adequate [6].

The principal requirement is for chronological temperature linearity over at least the middle of the period. This allows extrapolation of the linear portion of the regenerator to the end and beginning of periods. The heat transmission coefficient so obtained $\langle h^*\rangle$ is defined with respect to the average temperature of the heat storing mass. The true value is dependent on the function Φ discussed above, which corrects for the rapid temperature changes occurring immediately after switchover. The correction gives [6]

$$\frac{1}{h^*} = \frac{1}{\langle h^*\rangle} - \frac{d}{\lambda_s}\Phi \qquad (14)$$

and it can be shown that for small values of the parameter $\Omega < 1$ (equation (9)) then Φ is 1/8. In this case using $\lambda_s = 9.5$ W m^{-1} K^{-1} and $d = 22\times10^{-6}$ m we obtain $\Omega = 2\times10^{-4}/P$, so restricting P to times greater than 0.2 ms.

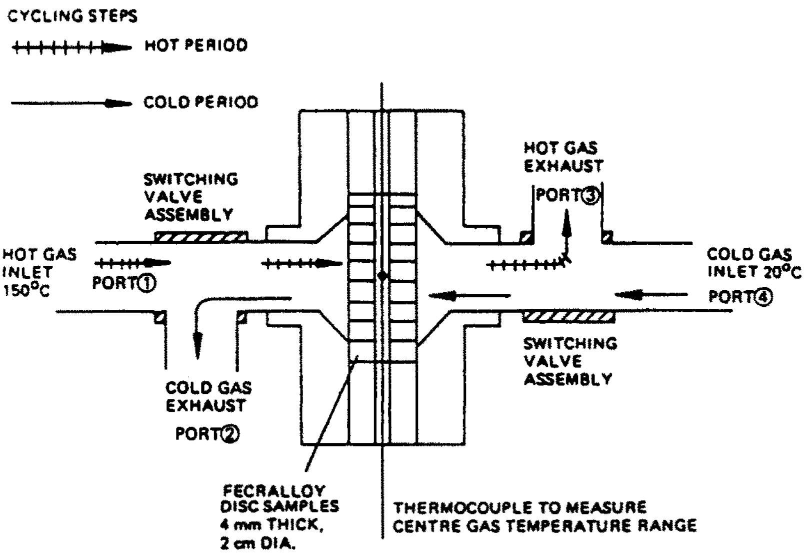


FIG. 4. Schematic diagram of regenerator assembly.

3. EXPERIMENTAL METHOD

The regenerator part of the apparatus comprises a nylon holder containing two 4 mm thick samples cut into 25 mm diameter discs, with a 40 μm diameter bare wire chromel/alumel thermocouple sandwiched in between to measure the gas temperature profile in the centre of an 8 mm thick sample (Fig. 4).

The cyclic gas flow is controlled by solenoid valves on either side of the sample. These three-way valves were switched out of phase using a relay circuit fed by a variable frequency square wave generator. The valves had a common regulated and metered gas supply of which one side was heated to 150°C and the other at 20°C.

The operation of the valves is also shown in Fig. 4. During the hot period, ports 2 and 4 are shut so that hot gas flows from 1, passes through and heats the sample and exits colder through 3. During the cold period ports 1 and 3 are shut and cold air passes from 4, through the sample and is exhausted warmer through 2. The apparatus was designed to minimize the switching volume between the ends of the matrix and the valve ports. The frequency of this regenerative cycle could be varied from 0.02 to 10 Hz.

Gas flow rates between 2 and 25 l min^{-1} were used. For most of the experiments the gas used was air but some results were also obtained using carbon dioxide. The thermocouple signal was amplified and recorded using a digital oscilloscope. In later tests thermocouples were also inserted to measure the cyclic profile of the gas temperature just before entering the sample on both hot and cold sides of the regenerator. Fine bronze gauze (200 mesh, wire diameter 50 μm) and geometrically identical stainless steel gauzes (200 and 100 mesh, wire diameters 50 and 110 μm) were later substituted for the Fecralloy discs to obtain comparative correlations at higher Reynolds number. These samples were prepared by stacking gauze discs in the holder on either side of the centre thermocouple to form a matrix of the same thickness as the Fecralloy sample.

4. ANALYSIS OF EXPERIMENTAL DATA

A typical time response of the thermocouple measuring centre gas temperature over a cycle is shown in Fig. 3(b). The slope of the curve is governed by heat transfer as can be seen in equation (4b). The time for the rapid changes following switchover is governed by (i) thermocouple response time, (ii) switching volume, and (iii) the voidage in the regenerator storing gas from the preceding period.

The response time of the thermocouple τ_c is partially responsible for the damped initial response at the start of a new period. This was found to vary over the range of flow rates used from 180 to 80 ms. Accordingly, switching rates were maintained at least an order of magnitude slower. The switching volume between the end of the matrix and the valve port was discussed above. Gas flowing into a valve at the beginning of a period has to displace the gas in the holder from the previous period and this had the effect of distorting the profile immediately after switchover. A residence time, τ_r, can be defined as the time required for this 'dead' volume to be flushed out of the system for a given flow rate.

At low P, and especially at low flows, the switching volume distorts a considerable fraction of the cycle. This effect was minimized to less than 10% of cycle time by maintaining a high value of P. Hence the conditions employed throughout the experiment were such that

$$\tau_r < \tau_c < \tau \sim O(P).$$

It was desirable to run the experiments at a constant value of P to minimize variations in the effects due to the dead volume, the thermocouple response time, and the incidental heat losses to the gas valves—typical values chosen were in the range 1–5 s, with most runs at $P = 3.2$ s, providing linear temperature time profiles.

As 80% of the fibrous material is void, after a switchover, the entering gas must first expel any carryover gas left over from the previous cycle. The effect of this 'gas heat storage' is to retard regenerator operation at a point, the retardation increasing through the matrix. The time retardation is equal to the time for the gas flux with velocity u to reach the point z inside the regenerator. The new time variable (equation (5)) becomes [9, 10]

$$\eta = \frac{h^*}{\rho_s C_s(1-\beta)}\left(t - \frac{\rho_g z}{G}\right) \tag{15}$$

and the non-dimensionalized period (equation (7)) becomes

$$\Pi = \frac{h^*}{\rho_s C_s(1-\beta)}\left(P - \frac{\rho_g L}{G}\right). \tag{16}$$

From equation (16) it can be seen that for larger air flow, the gas storage term is insignificant. For example, over the range $Re = 0.2$–1.4, the carryover time as a fraction of total regenerative period $\delta\Pi/\Pi = L/uP$ fell from 2% to less than 0.5%. On a more rigorous quantitative level, Willmott and Hinchcliffe have shown [9] that for such small fractions of total throughput, the dead volume effects may be ignored if $\delta\Pi/\Pi < 40\%$ and $\Lambda/\Pi > 2$, where we have from equations (7) and (16)

$$\frac{\Lambda}{\Pi} = \frac{\rho_s C_s}{\rho_g C_g}\frac{(1-\beta)}{(uP-L)} \tag{17}$$

i.e. a parameter independent of the heat transmission coefficient itself. Thus over the range of Reynolds number studied Λ/Π falls from 9.5 to 1.3. When $\Lambda/\Pi = 2$, Re is typically 0.9. However, at such flows the carryover volume is sufficiently small to make any effects negligible, particularly as mixing of resident and incoming gases takes place [9, 10].

The operating conditions were chosen so that the cycling period P used was long enough for the regenerator to respond significantly, and for the effects of transients and thermal equilibration at the beginning of new periods to be small. Using equation (13) we obtain the heat transfer coefficient directly. It is governed by the thermal response time of the system and the characteristic length over which heat exchange takes place. For the calculation of results, a basic routine was used to sample the temperature trace and calculate the difference $T_{2g} - T_{1g}$ over the most linear part of the heating or cooling period, between $0.45P$ and $0.55P$. The overall change in gas temperature ΔT_g was obtained by extrapolation of the linear portions of the response curves back to the start of a period in order to remove the effect of the initial transient response of the temperature measurements in any period.

5. RESULTS AND DISCUSSION

Figure 5 shows the heat transfer coefficient as a function of flow plotted as Nusselt number vs Reynolds number defined by

$$Re = \frac{Gd}{\mu} \quad \text{and} \quad Nu = \frac{hd}{\lambda_g} \tag{18}$$

where μ and λ_g are gas viscosity and thermal conductivity, respectively. The dimensionless parameters are calculated with reference to a room temperature air supply which is appropriate because these are the values used to correlate Re and thermal input when the metallic fibre material is used as a burner. The size of the temperature changes over a period ΔT_{2g} and the temperature difference between corresponding parts in time of a hot and cold cycle $T_{2g} - T_{1g}$ are the parameters in equation (13) which are required to evaluate heat exchange. These of course vary with flow. In our experiments ΔT_{2g} varied from 8 to 50°C and $T_{2g} - T_{1g}$ from 4 to 10°C. The heat exchange is seen to increase rapidly with flow over the range of interest, $Re = 0.1$–1.0, $Nu = 0.01$–0.1. There is, however, a retarded increase in Nusselt number at lower flow rates which is a consequence of the distortion of the input gas profile due to the thermocouple response time and the switching volume as opposed to gas hold-up which has already been discussed. The consequence of this is that the measured hot cycle gas temperature is always less than that of the ideal case. The cold cycle gas temperature always correspondingly exceeds its ideal value. The net result is to decrease the value of $T_{2g} - T_{1g}$ in equation (15) from its true value and to increase the term in the numerator ΔT_{2g}. The effect is to give a value of h larger than the true one at low flow rates where transients distort a significant proportion of the cycle. On replacing the Fecralloy discs with stacks of bronze and steel wire mesh as the regenerator material, keeping the sample thickness constant, the same change in slope was observed at corresponding flow rates (Fig. 5(b)). A better estimate of the heat transfer coefficient at low flow rates is obtained by extrapolating back the higher flow correlation rather than relying on the actual experimental values. The correlations of Fig. 5 are all of the form

$$Nu = B\,Re^n. \tag{19}$$

Using air flow through the porous burner material (Fig. 5(a)) the values obtained were $B = 0.10$, $n = 1.64$, in good agreement with measurements obtained using an alternative single blow technique [11, 12]. The nearest literature values refer to single wires of a larger diameter and at higher flows [7] with $B = 0.89$ and $n = 0.33$.

In general, the scarce data available suggests higher

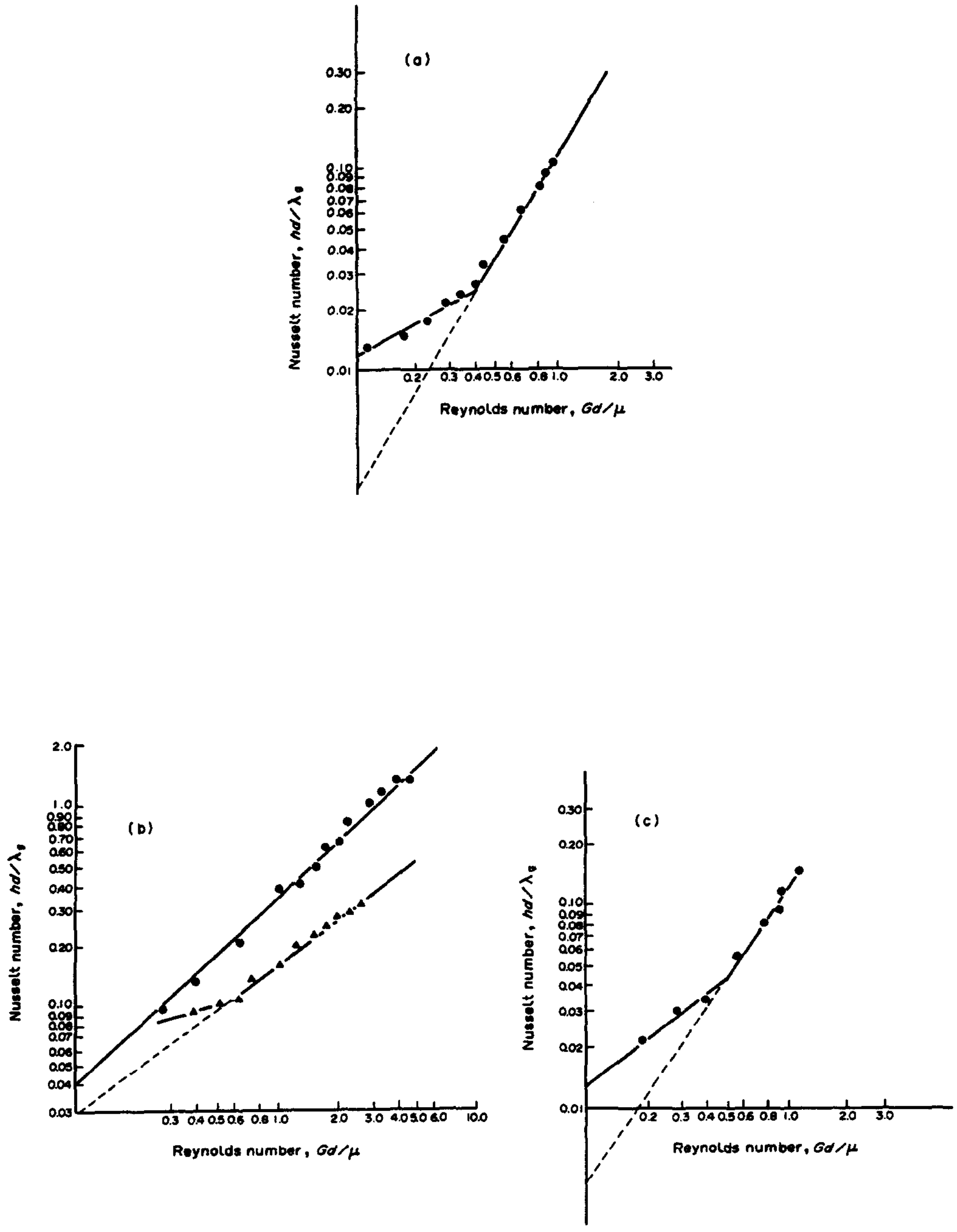


FIG. 5. Correlations between Nusselt and Reynolds numbers: (a) air flow in 80% fibrous material; (b) air flow in bronze (▲) and steel (●) wire meshes; (c) CO_2 flow in 80% fibrous material.

pre-Reynolds coefficients by a factor of ten, but lower indices by a factor of five for single wires. For the region we are interested in, the pre-Reynolds coefficient is more important since in the region $Re = 0.1$–2.0, the index has little significance close to $Re = 1.0$. Hence, overall the heat transfer coefficients are lower by a factor of ten for the porous material than for heat transfer to single wires. As these correlations for single wires in the literature vary with Reynolds number range, the change in slope is not surprising, as the flow around our much more complex system will be rather different. Valve wall heat storage and conduction will also be more strongly competing processes with heat transfer at low flows [12].

For bronze and steel gauzes (Fig. 5(b)) the values of B and n are 0.16 and 0.75 for bronze and 0.33 and 0.94 for steel. The likely reason for the difference between the bronze and steel results is the large difference in thermal conductivities of the two materials;

typically being 180 W m^{-1} K^{-1} for bronze and 16 W m^{-1} K^{-1} for steel. This difference in thermal conductivity is of practical significance. The analysis of the heat transfer equations which allows for the simplified form of equation (13) requires the assumption of zero solid conductivity parallel to gas flow. This is valid for materials such as steel or Fecralloy. The latter has a solid conductivity of 9.5 W m^{-1} K^{-1} and we have measured the porous longitudinal conductivity parallel to gas flow to be 0.15 W m^{-1} K^{-1} [4]. While the conductivity for the steel gauze will also be small, this is a much less accurate assumption for bronze with a conductivity more than 10 times as great. Thus the *Nu–Re* correlation is not independent of solid properties when the assumptions of the analysis—in this case vanishing longitudinal conductivity—break down [15].

Air was replaced by carbon dioxide as the regenerator gas in order to test the universality of the correlation and to see if this could be related to gas properties. Figure 5(c) shows the Nusselt number–Reynolds number relationship. The irregularity in the slope is repeated at the lower flow region, as for air, but the corresponding correlation is of the form of equation (17) with $B = 0.12$, $n = 1.38$.

The primary interest in the fibrous material is as a radiant surface combustion burner in which energy is transferred from the gas to the solid material. If conduction in the gas and chemical reaction rate are included, then equation (4a) must be modified to balance the generated combustion heat in the gas

$$-GC_g = \frac{dT_g}{dz} + \lambda_g \frac{d^2T_g}{dz^2} - h^*(T_g - T_s) = -\beta \Delta H R \tag{20}$$

and the balance in the solid is (equation (4b))

$$\lambda_s \frac{d^2T_s}{dz^2} + h^*(T_g - T_s) = 0 \tag{21}$$

where ΔH is the enthalpy of combustion and R the rate of reaction.

Equations (20) and (21) can be simplified if constants are removed by the process of nondimensionalization, whereupon they become

$$-\frac{dT_g'}{dz'} + \frac{d^2T_g'}{dz'^2} - \alpha(T_g' - T_s') = \frac{-\beta\lambda_g R}{G^2C_g} \tag{22}$$

$$\frac{d^2T_s'}{dz'^2} + \frac{\lambda_g}{\lambda_s}\alpha(T_g' - T_s') = 0 \tag{23}$$

using the substitutions

$$T_g' = \frac{T_g}{T_A}, \quad T_s' = \frac{T_s}{T_A}, \quad z' = \frac{GC_g z}{\lambda_g}$$

and where the adiabatic temperature T_A is defined from the ratio of enthalpy of combustion to heat capacity. The thermal input TI is defined as the fuel input into the burner

$$TI = G\Delta H = GC_g T_A. \tag{24}$$

α is defined using equation (3) as

$$\alpha = \frac{4\lambda_g(1-\beta)h}{G^2C_g^2 d} = 4B(1-\beta)\, Re^{n-2}\, Pr^{-2} \tag{25}$$

where the standard form, equation (19), is used.

Hence, it can be seen that the heat transfer from gas to solid depends on porosity, fibre diameter, and on thermal input. It is this parameter α, the heat transfer coefficient in the non-dimensional form, which is to be determined to help develop the burner

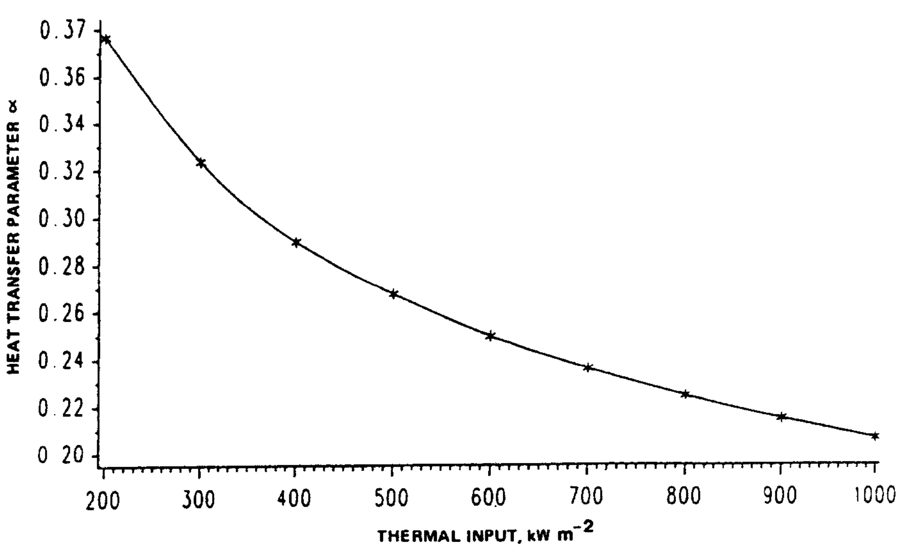


FIG. 6. Heat transfer from gas to solid as a function of thermal input.

model discussed above. If a correlation of the form of equation (19) does exist, α can be obtained by determining the appropriate values of B and n.

Figure 6 shows the non-dimensionalized heat transfer parameter α as a function of thermal input. From equations (22) and (23), it can be seen that this parameter balances influx of heat due to gas convection and conduction, against losses arising from conduction in the material. These equations are also applicable when the material is used as a heat sink.

6. CONCLUSIONS

The heat transfer coefficient for the gas to solid convective heat exchange in a metal fibre burner has been measured to give a Nu/Re correlation of the form of equation (19), using the matrix as a heat regenerator.

While no previous data exist for a correlation at such low Reynolds numbers, the experiments carried out using wire gauzes suggest that for the porous material, the heat transfer coefficients are lower by a factor of ten than for heat transfer to single wires, in this flow range. The consequence of this for the burner is to reduce the dimensionless heat transfer coefficient by a factor of 100 from the standard values usually applied to thin wires. The effects of dead volume space within the regenerator and the thermocouple response time have been shown to be significant in attempting to measure heat transfer using this method. While these have been minimized to produce acceptable results, a much more intricate formulation of the problem and experimental design would be required in order to totally nullify these effects for all heating periods at low flow rates.

REFERENCES

1. L. C. Shirvill, Metallic fibre surface—combustion radiant gas burners, *Proc. Int. Gas. Res. Conf. 1986*, pp. 837–844. Gov. Inst. Inc., Rockville (1987).
2. M. Golombok and L. C. Shirvill, Laser flash thermal conductivity studies of porous metal fibre materials, *J. Appl. Phys.* **63**(6), 1971–1976 (1988).
3. M. Golombok and L. C. Shirvill, Emissivity of layered fibrous materials, *Appl. Opt.* **27**(18), 3921–3926 (1988).
4. M. P. Dudikovic and P. A. Ramachandran, Evaluation of thermal efficiency for heat regenerators in periodic operation by approximate methods, *Chem. Engng Sci.* **40**(9), 1629–1639 (1985).
5. M. Jakob, *Heat Transfer*, Vol. 2, pp. 261–341. Wiley, New York (1957).
6. H. Hausen, *Heat Transfer in Counterflow, Parallel Flow and Cross Flow* (Translated by M. S. Sayer; edited by A. J. Willmott). McGraw-Hill, New York (1983).
7. W. H. McAdams, *Heat Transmission*. Wiley, New York (1960).
8. A. J. Willmott, Digital computer simulation of a thermal regenerator, *Int. J. Heat Mass Transfer* **7**, 1291–1302 (1964).
9. A. J. Willmott and C. Hinchcliffe, The effect of gas heat storage upon the performance of the thermal regenerator, *Int. J. Heat Mass Transfer* **19**, 821–826 (1976).
10. P. J. Heggs and K. J. Carpenter, The effect of fluid hold-up on the effectiveness of contraflow regenerators, *Trans. Instn Chem. Engrs* **54**, 232–238 (1976).
11. B. S. Baclic, P. J. Heggs and H. Z. Z. Abou Ziyan, Differential fluid enthalpy method for predicting heat transfer coefficients in packed beds, *Proc. Int. Conf. Heat Transfer 1986*, Vol. 5, pp. 2617–2622 (1986).
12. P. J. Heggs, Private communication (1988).
13. A. Ambrosio, C. D. Coulbert, R. P. Lipkis, P. E. O'Brien and F. E. Romie, Heat transfer and pressure drop characteristics of four regenerative heat exchanger matrices, ASME 51-SA-34 (1951).

ECHANGE DE CHALEUR SOLIDE–GAZ DANS UN MATÉRIAU MÉTALLIQUE FIBREUX PAR UNE TECHNIQUE UTILISANT UN APPAREIL D'ÉCHANGE DE CHALEUR

Résumé—Les propriétés de transfert thermique par convection d'une matière métallique fibreuse poreuse utilisée dans les brûleurs radiants au gaz à combustion en surface ont été étudiées. Le coefficient de transfert thermique entre la fibre métallique et le gaz chaud a été mesuré en utilisant une méthode en régime non permanent fondée sur une théorie utilisant un appareil d'échange de chaleur à contre-courant cyclique. Les conditions expérimentales dans lesquelles les hypothèses de la théorie utilisant un appareil d'échange de chaleur sont valides ont été étudiées. Une corrélation entre le débit du gaz et le transfert de chaleur a été obtenue pour des débits hautement laminaires, présentant une rapide augmentation du coefficient de transfert de chaleur avec un débit de gaz croissant. Le coefficient de transfert de chaleur est sensiblement inférieur à celui que l'on avait supposé précédemment.

MESSUNG DES WÄRMEAUSTAUSCHS ZWISCHEN GAS UND FESTSTOFFEN IN EINEM METALLFASERSTOFF MIT EINEM WÄRMEREGENERATIONSVERFAHREN

Zusammenfassung—Es wurden die Eigenschaften eines porösen Metallfaserstoffes für gasgefeuerte Strahlbrenner bei Wärmeübertragung durch Konvektion untersucht. Der Wärmeübergangskoeffizient zwischen heißem Gas und Metallfasern wurde unter Anwendung einer Methode auf der Basis der Theorie der zyklischen Wärmeregenerierung im Gegenstrom außerhalb des Beharrungszustandes gemessen. Die Versuchsbedingungen, bei denen die Annahmen für die Regeneratiostheorie Gültigkeit besitzen, wurden untersucht. Es wurde eine Beziehung zwischen Gas-Strömungsgeschwindigkeit und Wärmeübergang für stark laminare Strömungen ermittelt, bei der der Wärmeübergangskoeffizient mit wachsender Gasgeschwindigkeit rasch ansteigt. Der Wärmeübergangskoeffizient ist dabei wesentlich niedriger als früher angenommen wurde.

ИЗМЕРЕНИЕ ТЕПЛООБМЕНА МЕЖДУ ГАЗОМ И ТВЕРДЫМ ТЕЛОМ В ВОЛОКНИСТОМ МЕТАЛЛИЧЕСКОМ МАТЕРИАЛЕ МЕТОДОМ РЕГЕНЕРАЦИИ ТЕПЛА

Аннотация—Исследуются свойства конвективного теплопереноса в пористом волокнистом металлическом материале, применяющемся в газовых беспламенных горелках. С использованием нестационарного метода, основанного на теории циклического противоточного регенератора тепла, измерен важный параметр, определяющий теплоперенос между горячим газом и металлическим волокном, а именно, коэффициент теплоотдачи. Исследуются факторы, определяющие диапазоны возможных экспериментальных условий. Получено соотношение между расходом газа и теплопереносом для ламинарных течений, показывающее быстрое возрастание коэффициента теплоотдачи с увеличением потока газа. Коэффициент теплоотдачи оказался намного ниже, чем предполагалось.