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Heat Transfer and Flow Friction in Fine Porous Media

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HEAT TRANSFER AND FLOW FRICTION IN FINE POROUS MEDIA

by

W. A. Steyert

ABSTRACT

NOTICE
This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

Fluid-flow friction factors and heat-transfer coefficients are reported for fine screens (40, 80, and 100 mesh) and for packed fine copper spheres (70 μ m). The fluids used are nitrogen and helium gas and water. The observed values do not deviate greatly from the values determined by scaling previous measurements taken from much coarser structures.

I. INTRODUCTION

Most fluids are poor thermal conductors. To achieve good heat exchange between a flowing fluid and a solid surface, high fluid velocities are frequently employed. The resulting turbulent mixing provides the required heat transfer between the main fluid stream and the solid surface. Unfortunately, the turbulence causes frictional fluid-flow losses by converting mechanical energy into heat. Sometimes, large pumps and compressors are needed to provide the flow.

If the fluids are forced to flow through very small channels in the appropriately constructed solid, each molecule of the fluid is brought close to the solid, and heat need flow only short distances through the poorly conducting fluid. Pressure drops are kept small by choosing a suitable overall geometry so that fluid flow through the surfaces is slow.

II. OBJECTIVES

The aims of this research are to measure pressure drops in fluids moving through small channels in

metals and to measure heat-transfer coefficients in the same geometries, in preparation for magnetic-refrigeration tests. The literature gives pressure drops and heat-transfer coefficients for larger versions of some of our geometries and provides scaling laws with which to calculate values for some of our geometries. However, because fluid-flow scaling laws are not always accurate far outside of the measurement range, actual measurements are needed. Another objective is to learn the pitfalls involved in making accurate heat-transfer measurements, because heat transfer will be the main problem in magnetic-refrigeration technology.

The construction and testing of cryogenic counterflow heat exchangers with finely divided metal surfaces will demonstrate these heat-transfer methods before they are used in magnetic-refrigeration cycles.

III. SURFACES TO BE MEASURED

Pressure drops and heat-transfer coefficients were measured for 40-, 80-, and 100-mesh (US Standard) copper screen (Fig. 1). The finest mesh, low-cost copper screen available is 100 mesh from Buffalo

Fig. 1.

Screens of wire diameter d and opening size D .

Wire Works* (screen finer than 100-mesh copper requires specialized construction methods).

The pressure drop across a tube filled with spheres of 0.32- and 0.48-cm diam was measured, as was the pressure drop across a plate 0.43 mm thick of 60- μ m-diam sintered copper spheres** (Fig. 2).

Because thin plates of fine particles are very fragile, we used 60-mesh copper screen as a base and diffusion-bonded (sintered) to it rough 1.05- μ m-diam copper particles and 70- μ m-diam copper spheres** to form beds. Pressure drops were measured for the 1.05- μ m particles and the 70- μ m spheres, and heat-transfer coefficients were measured for the 70- μ m spheres.

After the above measurements were completed, we found that the screens were preferable to the particle beds when we compared the ratio of heat transfer, hA' (W/K), to pressure drop, P (Pa or dyn/cm²), for a given fluid mass-flow rate, G (g/cm²·s), and frontal area of surface, A . This is attributable to the open structure of the screen. We felt that the open nature of the screen could be maintained but the heat-transfer area increased if the screen had a light "dusting" of spheres diffusion-bonded to it. Some 100- and 40-mesh screens were dipped into a dilute solution of organic vacuum grease in toluene. When the toluene evaporated, the screens were dusted with 70- μ m copper spheres, which adhered to the residual grease in more or less of a monolayer. The screens were then heated to 1223 K in 0.013 Pa (10⁻⁴ torr) of dry-air pressure for about 4 h to bond the spheres to the screens (the air was dry to improve the low-temperature thermal

Fig. 2.

Idealized array of spheres of particle diameter d_p . The particles in our plate were randomly packed.

Fig. 3.

a) Spheres (70 μ m) bonded to a copper 40-mesh screen in an incomplete monolayer.

b) Spheres (70 μ m) bonded to a copper 100-mesh screen, filling many of the screen openings.

conductivity of the copper¹). The resulting screen (Fig. 3) had a little less than a monolayer on the 40 mesh, and a little more than a monolayer on the 100-mesh screen (spheres almost filling most of the screen openings). Heat-transfer coefficients were measured for the uncoated and coated 40-mesh

*P. O. Box 129-T, Buffalo, NY 14240.

**Metco 55 copper spheres for flame-sprayed coatings. Metco, Inc., 1105 Prospect Ave., Westbury, NY 11590.

Fig. 4.

Pressure-drop apparatus (disassembled). Small-diameter tubes connect to the pressure and reference sides of a differential pressure gauge. Incoming gas goes through the copper coil for precooling if the experiment is carried out at liquid-nitrogen temperature.

screen. Pressure drops and heat-transfer coefficients were measured for the 100-mesh sphere-coated screen.

IV. PRESSURE-DROP MEASUREMENTS

A. Pressure-Drop Apparatus

The pressure-drop apparatus is shown in Fig. 4. An inlet tube supplies the fluid (nitrogen or helium gas or water) that flows through the material being tested. The flow rate is measured by running the exit gas through a meter or the water into a beaker of known volume. The flow rate is given by the volume divided by the elapsed time.

The pressure drop across the material is measured with a 200-mm or 50-mm Wallace and Tiernan* gauge or from the height differential in a U-tube of water. The inlet pressure is measured relative to the outlet pressure in static lines attached to the pressure lines 2 cm on either side of the sample.

*Division of Pennwalt Corp., 25 Main Street, Belleville, NJ 07109.

Fig. 5.

Pressure drops measured for one 75-K and three room-temperature fluids flowing through copper screen alone and screen coated with 70 μm spheres. For each screen $d = 0.011$ cm and $D = 0.014$ cm. Each layer is 0.025 cm thick. The porosity, of the screen alone, α_t , based on projected frontal area, is 0.31.

For the screens and the fine spheres the sample was a disk about 1.1 cm in diameter (exact areas, which differ slightly from material to material, are used in the data analyses). When multiple layers of screen were studied, they were randomly oriented and lightly pressed one on top of another. The interscreen spacing was typically half of the screen thickness. Separate tests showed that this distance was not significant. For the larger spheres, the pressure drop was measured in a tube 0.9 cm in diameter and 7.6 cm long filled with the spheres.

B. Pressure-Drop Results

Figures 5-9 present the pressure-drop results. The pressure drop P in the system before a sample was inserted was measured and found to be less than 5% of the pressure drop through any of the materials,

Fig. 6.

Pressure drops measured for two room-temperature gases flowing through copper screen. For each screen $d = 0.0135$ cm, $D = 0.0165$ cm, and the screen thickness is 0.035 cm. The porosity, α_t , is 0.30.

Fig. 8.

Pressure drops measured for room-temperature fluids flowing through copper screen filled with powder for two thicknesses of the resulting matrix. The surprisingly low flow impedance may be caused by leakage through the irregularities in the matrix and around the O-rings in the sample holder.

Fig. 7.

Pressure drops measured for three room-temperature fluids flowing through a tube of spheres of diameter d_p (see inset). The porosity $\alpha = 0.39$ for $d_p = 0.32$ cm and $\alpha = 0.49$ for $d_p = 0.48$ cm.

and so this correction was neglected. Thus, the pressure drops plotted are those read directly from the gauge or U-tube. A slight correction to the gas velocity was needed. The superficial velocity* v is given by

$$v \approx \frac{V}{At} \quad (1)$$

where V is the volume of gas that has flowed through an area A of the screen in time t . Actually,

$$v = \frac{V}{At(1 + \rho/1160)} \quad (2)$$

*As is common practice in heat-exchange work, we used superficial velocities; that is, velocities deduced assuming that the sample was not in the system.

Fig. 9.

Pressure drops measured for room-temperature fluids flowing through copper screen filled with 70- μ m spheres for two thicknesses of the resulting matrix. Porosity (based on the measured void fraction) of the screen-sphere composite, α , is 0.5. Also, pressure drops for sintered plates 0.43 mm thick of 60- μ m spheres; porosity, α , is 0.40 for these plates.

where the factor in parentheses corrects for the fact that, as the gas moves through the material, the average gas density is somewhat higher than it is at atmospheric pressure of 77 300 Pa (580 torr), and thus the gas velocity is lower. This correction is not needed for water [Eq. (1) becomes an equality].

Figure 10 shows the pressure-drop results plotted in dimensionless friction factor f . For a screen, *Chemical Engineers' Handbook*² Eq. 5-100 (translated to SI or cgs units*) gives

$$P = \frac{n}{C^2} \left(\frac{1 - \alpha_f^2}{\alpha_f^2} \right) \left(\frac{v^2 \rho}{2} \right) \quad (3)$$

where n is the number of layers of screen the fluid passes through, α_f is the porosity based on the projected free frontal area $D^2/(d + D)^2$ from Fig. 1, ρ is

*The pressure drop formulas used here are valid in any consistent set of metric units. In English units a factor of g (the acceleration of gravity at the earth's surface) is necessary.

Fig. 10.

Friction factor vs Reynolds number as defined by Ref. 2 for screen. Solid curve is from the results in Ref. 2.

the fluid density, and C is an empirically determined dimensionless constant. We define

$$f \equiv \frac{1}{C^2} \equiv \frac{P}{n \left(\frac{1 - \alpha_f^2}{\alpha_f^2} \right) \left(\frac{v^2 \rho}{2} \right)} \quad (4)$$

Figure 5-44 of Ref. 2 gives C as a function of the Reynolds number,

$$Re = Dv\rho/\alpha_f\mu \quad (5)$$

where μ is the fluid viscosity in Pa·s (SI) or poise = dyn·s/cm² = g/cm·s (cgs). Figure 10 shows f calculated from the measured values in Eq. (4) and also shows the curve of $f = 1/C^2$ from Fig. 5-44 of Ref. 2.

A convenient tabulation of pressure drops in particle beds is given in Ref. 3. Here the friction factor f is defined as

$$f = \frac{P_d p \alpha^3}{L \rho v^2 (1 - \alpha)} \quad (6)$$

where the porosity α is the ratio of void to filled volume in the bed, d_p is the spherical-particle diameter, and L is the dimension of the bed in the flow direction. In this formulation

Fig. 11.

Friction factor vs Reynolds number for small and large spheres with three fluids at room temperature. Curve is the data of Ref. 3.

$$Re \equiv \frac{\rho v d_p}{\mu(1-a)} \quad (7)$$

Figure 11 shows f , calculated from the measure values, versus Re .

V. HEAT-TRANSFER MEASUREMENTS

A. Heat-Transfer Apparatus

Figure 12 shows the heat-transfer apparatus. The sample being tested was heated by an electrical current (50-100 A). The fluid was heated as it flowed through the material. The fluid inlet and exit temperatures were measured by copper-constantan thermocouples (0.008-cm wire diameter) and recorded on a four-pen chart recorder (500 μ V full scale). The thermocouples measuring inlet and outlet temperatures were soldered to passive 100-mesh screens in the fluid flow. The screen positioning assured that the thermocouple and the fluid were the same temperature.

The sample temperature was measured at two locations by thermocouples soldered to the sample. The solder bits were small (less than 1 mm in diameter) to minimize flow disruption near the thermocouples. The silk-insulated thermocouple wires

Fig. 12.

Heat-transfer apparatus. The fluid is heated as it moves upward through the electrically heated screen. Fluid flows through a roughly 1-cm-diam disk of the material. Styrofoam cylinders 1 cm long with 1-mm walls insulate the gas stream on each side of the sample.

exiting the experimental area were glued to the sample to assure that the sample temperature was measured as accurately as possible. It was important to run the thermocouple wires roughly perpendicular to the applied voltage, which heated the sample, to minimize current leakage through the silk insulation. The leakage could cause erroneous signals. The current through the sample was reversed during each measurement to identify and correct for any such erroneous effect. The voltage applied to the sample precluded the use of the sample itself as the copper side of the thermocouple. An estimate of the sample temperature was also deduced by measuring the voltage across the screen as a function of current; the copper resistance is a measure of the sample temperature.

The fluid-flow rate was again measured by running the exit gas through a meter or the water into a beaker of known volume. The thermal heat-transfer coefficient h was deduced as a function of flow rate.

B. Heat-Transfer Data Analyses

Consider a fluid flowing through a uniform matrix at temperature T_m as shown in Fig. 13. Consider a frontal area A of the matrix with thickness L . Consider the matrix to have total heat-exchange area A' and to have heat-transfer coefficient h ($\text{W}/\text{cm}^2 \cdot \text{K}$). If the fluid is flowing with superficial velocity v and has specific heat C_p ($\text{J}/\text{g} \cdot \text{K}$) and density ρ , then for a gas temperature $T(x)$ at position x in the matrix,

$$C_p \rho v A \frac{dT(x)}{dx} = h \frac{A'}{L} [T_m - T(x)] \quad (8)$$

or

$$T(x) = T_m + (T_i - T_m) \exp(-hA'x/LC_p\rho vA) \quad (9)$$

where T_i is the fluid temperature at the input, $x = 0$, of the matrix. The output fluid temperature at $x = L$, T_o , is given by

$$T_o = T_m + (T_i - T_m) \exp(-hA'/C_p\rho vA) \quad (10)$$

or

$$h = \frac{C_p \rho v A}{A'} \ln \left(\frac{T_m - T_i}{T_m - T_o} \right) \quad (11)$$

Another way to arrive at h is to calculate

$$hA' \left[T_m - \left(\frac{T_i + T_o}{2} \right) \right] = C_p \rho v A (T_o - T_i) \quad (12)$$

where the left-hand side of this equation is the heat flow from the matrix to the fluid at average temperature $(T_i + T_o)/2$, and the right-hand side is the power input to heat the matrix. Equations (11) and (12) are nearly the same, as a series expansion of h in the measured temperatures, T_i , T_o , and T_m , shows. However, because Eq. (12) is only an approximation, we used Eq. (11) to analyze the data.

C. Heat-Transfer Results

Figures 14-17 present the measured heat-transfer coefficients h in dimensionless form as a function of the superficial mass-flow velocity, G ($\text{g}/\text{cm}^2 \cdot \text{s}$). The measurements were made at room temperature; each plotted point typically represents data taken at several different power levels.

Figures 14-17 show the heat-transfer results in plots similar to those of Figs. 7-8 and 7-10 of Ref. 4. Here $hPr^{1/4}\alpha/GC_p$ is plotted against the Reynolds number, defined as

$$Re = D_h G / \mu \alpha \quad (13)$$

Pr is the Prandtl number $C_p \mu / \kappa$, where κ is the thermal conductivity; h/GC_p is sometimes called the Stanton number; and

$$G = \dot{m}/A = v\rho \quad (14)$$

Fig. 13.

Schematic diagram of heat transfer to a fluid flowing at superficial velocity v through a matrix that is of frontal area A , thickness L , and uniform composition and temperature.

Fig. 14.

The dimensionless quantity $hPr^{2/3}\alpha/GC_p$ vs Reynolds number for 100-mesh screens ($d = 0.011$ cm and $D = 0.014$ cm). The screen porosity, α , is 0.49 [Eq. (15)], and the hydraulic diameter, D_h , is 0.021 cm [Eq. (16)]. The line is from the results of Ref. 5, which were quoted by Ref. 4 for 60-mesh screen with $\alpha = 0.6$. One layer of screen was used, except as indicated; the work with three layers was not as accurate because, with the better heat transfer, the small ΔT between the screen and the outlet fluid was difficult to measure.

where $\dot{m}$ is the total mass-flow rate through superficial area A . Because α is not fully defined in Ref. 4, we go to the original reference⁵ where α is defined as the ratio of the total volume of void to the total volume of matrix. For a screen we take the thickness as $2d$ to calculate

$$\alpha = 1 - \frac{\pi}{4} \frac{d}{D + d} \quad (15)$$

Reference 5 gives the hydraulic diameter as $D_h = 4\alpha/\beta$, where β is the ratio of the surface of voids to the total volume of the matrix, which we calculate as $\pi/(D + d)$ for screen. Then

$$D_h = 1.27D + 0.27d \quad (16)$$

Fig. 15.

The dimensionless quantity $hPr^{2/3}\alpha/GC_p$ vs Reynolds number for 80-mesh screen ($d = 0.0135$ cm and $D = 0.0165$ cm), $\alpha = 0.65$ and $D_h = 0.025$ cm. The roughened screen was treated with emery cloth in a futile attempt to increase h .

We use Eqs. (14) and (16) in our plots of heat transfer from screen (Figs. 14 and 15).

Heat transfer through packed beds of spheres is estimated in Ref. 3 as

$$Nu = 0.80Re^{0.7}Pr^{1/3} \quad (17)$$

for $\alpha = 0.37$ and Re between 500 and 50 000. Nu is the dimensionless Nusselt number, hd_p/κ , and $Re = Gd_p/\mu$. We measure for Re between 0.16 and 40 and compare our results with those of Ref. 5, which go to Re as low as 20 in beds of packed spheres. Reference 5 has

$$\frac{h\alpha}{GC_p} = 0.21Re^{-0.31}Pr^{-1} \quad (18)$$

where

$$Re = D_h G/\mu\alpha \quad (19)$$

and

$$D_h = 4\alpha/\beta \quad (20)$$

Fig. 16.

The dimensionless quantity $hPr^{2/3}\alpha/GC_p$ vs Reynolds number for 60-mesh copper screen filled with 70- μ m copper spheres 0.63-mm- and 0.40-mm-thick matrix results are the same within the scatter and are not distinguished). The porosity, α , is 0.5, and the hydraulic diameter, D_h , is 0.0048 cm [Eq. (20)]. The dashed line is an extrapolation of data from Ref. 5 that were measured with 0.21-cm spheres for Re between 25 and 500. Experimental uncertainty in these results is large because the excellent heat transfer to the fluid makes the screen-to-outlet-fluid temperature difference small and thus difficult to measure accurately. We do not consider the large difference between the measured and the expected h at lower Re necessarily real.

We calculate and use for spheres

$$D_h = 2d_p \frac{\alpha}{3(1-\alpha)} \quad (21)$$

For the heat-transfer results from small spheres (Fig. 16) we plot $hPr^{2/3}\alpha/GC_p$ versus Re [Eq. (19)] as in Reference 4. The scaling law here differs by a factor of $Pr^{1/3}$ from that of Eq. (18). Because the dimensionless Pr is ~ 1 for most cases of interest, we ignore this small difference. (The available data are unable to distinguish between $Pr^{2/3}$ and Pr^1 scaling).

In addition to comparing our measurements with the results in Ref. 5, we compare them with the estimate⁸ that

Fig. 17.

The dimensionless quantity $hPr^{2/3}\alpha/GC_p$ vs Reynolds number for 40-mesh screen, $\alpha = 0.52$, $d = 0.036$ cm, $D + d = 0.063$ cm, and $D_h = 0.040$ cm. The 40-mesh screen coated with an incomplete monolayer of 70- μ m copper spheres (sintered to the wires) was also measured. The roughly 25% increase in h is probably, but not certainly, real. The values used for α , D_h , area, etc. for the coated and uncoated screen were taken as identical to allow a direct comparison of the results.

$$Nu = hd_p/\kappa = 2 \quad (22)$$

for low Re . (This is estimated as a minimum; for $Re \geq 1$, Nu should increase appreciably). Thus, for low Re we expect (using $Pr = C_p\mu/\kappa$ and Eqs. (19) and (21))

$$\frac{hPr^{2/3}\alpha}{GC_p} = \frac{2\kappa}{d_p} \frac{Pr^{2/3}\alpha}{GC_p} = \frac{4}{3} \frac{\alpha}{RePr^{1/3}(1-\alpha)} \quad (23)$$

Taking $\alpha = 1/2$ and $Pr \approx 1$, we get the dotted curve shown in Fig. 16.

Figure 17 shows the heat-transfer data for the 40-mesh screen and the screen coated with an incomplete monolayer of 70- μ m copper spheres. It was not possible to measure actual heat contact area A' with the spheres on the screen wires. Because we were primarily interested in a direct comparison with the

uncoated screen, h was calculated from the experimental values of hA' taking A' equal to the area of the screen wires.

The extremely good heat transfer from the sphere-coated 100-mesh screen made it impossible to obtain coefficients for this material (see following discussion of errors).

D. Errors in the Heat-Transfer Results

Because of the finely divided porosity in most of the materials being tested, the heat transfer is very good. Thus, the temperature of the fluid exiting the matrix is nearly equal to that of the matrix. [This is a particular problem at low flow velocities (low Re) for matrices of the small spheres.] It is difficult to determine this very small temperature difference, ΔT , accurately.

Small heat leaks can cool the outlet fluid making ΔT appear larger than it really is. Electrical heating of the screen near the thermocouple attachments can give a high screen-temperature reading (fluid cooling cannot occur there because the screen passages are blocked by the solder). Both of these effects cause underestimation of h . Fluid cooling of the thermocouples measuring screen temperature would cause underestimation of ΔT and over estimation of h .

For the sphere-coated 100-mesh screen we expected ΔT to be very small (≈ 0.02 K). The measurements gave a value near -0.5 K (implies the exiting fluid was hotter than the screen). It was thus impossible to deduce h for such excellent heat transfer.

If the heat-transfer experiment were run again, it would be helpful to cool the incoming fluid so that it is almost room temperature after being heated by the matrix; this would minimize heat leakage into the outlet fluid, making the measurement of ΔT more accurate.

VI. ANALYSES OF RESULTS

The pressure drops through the fine screen-and-sphere matrices seemed to be the same or a little smaller than those expected based on the scaling of results obtained for larger screen-and-sphere matrices.²

The heat-transfer measurements on the screens show a slightly higher h than scaling from the larger sizes^{2,5} would predict. For spheres, the very large values for hA' make the measurements difficult, particularly at low flow rates; and so we believe the values of h at low Re in Fig. 16 are in error.

As a first step toward applying finely divided porous materials to magnetic refrigerators, a counterflow heat exchanger using these materials has been built. It has a value of $G \approx 1$ for helium gas at 77 300 Pa (1 atm) [$\rho = 0.008$ g/cm³, $\mu = 4 \times 10^{-6}$ Pa·s (40×10^{-6} poise), $C_p = 5.5$ J/g·K, and $Pr \approx 1$]. As an example, let us compare the figure of merit hA'/P for n layers, 1 cm² each, of 100-mesh screen and a bed of 70- μ m spheres, 1 cm thick and 1 cm² in area.

Using the definitions and the data in Fig. 10 for 100-mesh screen, we find $Re = 1150$, $f \approx 0.5$, and $P = 294n$. In Fig. 14, $Re = 895$; extrapolating or using Ref. 5 we find $hPr^{2/3}\alpha/GC_p = 0.04$, or $h = 0.67$. Because A'/A is $2.4n$, we find for $A = 1$ cm²

$$hA'/P|_{100 \text{ mesh}} = 5.5 \times 10^{-3} \quad (24)$$

Similarly, using Fig. 11 for spheres we find $Re = 350$, $f \approx 1.7$, and $P = 1.21 \times 10^5 L$. Also, using Fig. 16 we find $Re = 240$; extrapolating the dashed curve or using Ref. 5 we find $hPr^{2/3}\alpha/GC_p = 0.038$, or $h = 0.4$. Because $A'/A = 6L(1 - \alpha)/d_p = 430L$, then for $A = 1$ cm²

$$hA'/P|_{\text{spheres}} = 1.4 \times 10^{-3} \quad (25)$$

This is a factor of 4 worse than the figure of merit for the screen. The G dependence of P and h for the screens and spheres is similar for G values near 1, so 100-mesh screen will generally give better heat transfer than the 70- μ m spheres.* However, Eq. (4) or Fig. 11 shows the strong dependence of P on α for the spheres. If we increase α to 0.75 from 0.5, P would drop by a factor of roughly $(1 - \alpha)/\alpha^3 = 0.68$. However, hA' would drop only by a factor of $(1 - \alpha)/\alpha = 0.33$, thus increasing hA'/P by a factor of 2.

Let us compare the figures of merit for the 40-mesh screen with and without sphere coating.

*Changing the particle size, d_p , does not improve the figure of merit because it is almost independent of d_p for G near 1 (from Figs. 16 and 17 and $A' \propto d_p^{-1}$).

Figure 17 suggests (but does not prove) a roughly 25% improvement in hA' because of the sphere coating. The incomplete monolayer sphere coating decreased the value of α , only slightly, so from Fig. 10 we expect only a slight increase in P . Thus hA'/P does not differ much.

Let us also compare the figures of merit for the 100-mesh screen with and without sphere coating. Figure 5 shows that P for 4 layers of coated 100-mesh screen is a factor of roughly 1.8 larger than that for 10 layers of the uncoated material; thus, P per layer of coated screen is larger by a factor of 4.5. We were unable to deduce h because of the excellent heat transfer. Thus we must estimate hA' . By weight, we find that there are 0.058 g/cm^2 of spheres on the 100-mesh screen. This is an area A' of 5.5 cm^2 per cm^2 of screen area A . But the screen area A' is 2.4 times A . Therefore, we have increased the total heat-exchange area A' by a factor of $(5.5 + 2.4)/2.4 = 3.3$. The discussion preceding Eqs. (24) and (25) suggests that h for the screen is somewhat higher (by a factor of $0.67/0.4 = 1.7$) than that for the spheres, and that we might expect to increase hA' by a factor of $(2.4 \cdot 0.67 + 5.5 \cdot 0.4)/2.4 \cdot 0.67 = 2.4$. Thus, the spheres have probably decreased hA'/P by a factor of $2.4/4 = 0.6$.

In general, we found that screen, with its more open structure, is a better heat-transfer matrix than packed beds of spheres. As a result, fine screens are the primary heat-exchange media in the exchangers being used in further work.⁷

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