HEAT TRANSFER IN A COLUMN PACKED WITH SPHERES

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Abstract The purpose of the study is to investigate average and local surface heat transfer coefficients in a cylinder packed with spheres. Here the term «local» applies to a single sphere within the bed. Averages are derived from the sixteen different spheres that were instrumented and distributed throughout the bed. The experimental technique consisted of introducing a step - wise change in the temperaturey of air flowing through the bed and recording the resulting air and ball temperature transients within the bed over a range of Reynolds numbers. An overall correlation of heat transfer with Reynolds number is derived and compared to others available in the literature, and comment is given on the observed patterns in the local coefficients. Comparison is made to theoritical transients.

چکیده   در این مقاله ضرایب محلی و میانگین انتقال گرما در سطح کره هایی که درون استوانه قرار داده شده اند مورد مطالعه قرار گرفته است. در اینجا کلمهٔ 'محلی' به کرهٔ تنها در یك بستر اطلاق می شود. میانگینها از شانزده کرهٔ مختلف که در تمام بستر جاسازی و توزیع شده اند به دست می آید. روش تجربی به این ترتیب است که در دمای هوایی که از بستر شارش می یابد به طور پله ای تغییراتی داده و دمای هوا و کره ها در داخل بستر در گستره ای از اعداد رنولدز ثبت می شود. یك رابطهٔ عمومی انتقال گرما نسبت به عدد رنولدز به دست آورده شده و با روابط موجود در نوشتار مقایسه گردیده است. در مورد ضریب محلی پیشنهاد لازم در نقشهای ملحوظ داده شده است در ضمن با جوابهائی که به طور نظری از گذارها به دست می آیند مقایسه به عمل آمده است.

INTRODUCTION

Many important processes require contact between a gas or a liquid stream and solid particles. These processes can be carried out in catalytic reactors, grain dryers, beds for storage of solar thermal energy, gas chromatography, regenerators, and desiccant beds. When fluid comes in contact with the surface of particle heat and / or mass between may be transferred between the fluid and the particle. The device may consist of a pipe, vessel, or some other containment for the particle bed through which the gas or liquid flows. The bed is heated during the charging cycle by pumping hot air or another heated working fluid through the bed. The particles, which comprise the packed bed, are heated to air temperature thereby storing heat sensibly. During the discharge cycle, cooler air is pumped through the bed, cooling the particles and removing the stored heat.

Correlations for heat or mass transfer in packed beds utilize a Reynolds number based on the superficial fluid velocity that is, the fluid velocity that would exist if the bed were empty. The length scale used in the Reynolds number and Nusselt or Sherwood number is generally the equivalent diameter of the packing. Since spheres are only one possible type of packing, an equivalent particle diameter must be defined which is based in some way on the particle volume and surface area. Such a definition may vary from one correlation to another, so some care is needed before attempting to apply the correlation. Another important parameter in packed beds is the void fraction, which is the fraction of the bed volume that is empty. The void fraction sometimes appears explicitly in correlations and is sometimes used in the Reynolds number. In addition, the Prandtl number may appear explicitly in correlation even though the original data may have been for gases only. In such a case, the correlations is probably not reliable for liquids [1].

LITERATURE SURVEY

The literature regarding heat transfer in packed columns is considerable and summaries are availabe which include hundreds of investigations. This paper is concerned with deep beds (height / diameter greater than 1) filled with solid spheres with air as the transfer fluid [2 - 6].

Most investigators have used steady state techniques to determine heat transfer coefficients [2, 4, 6, 7]. The bed is heated in some way; e. g., electrically [2] or by a surrounding steam jacket [7], and the heat transfer coefficients are derived from a simple balance between the enthalpy change of the gas as it goes through the bed and the heat generated within the bed. Denton, [4] heated a single sphere within the bed to a steady state while measuring adjacent gas temperature. Although he varied the position of the test sphere over a series of tests, no comment was made on whatever variation with position he may have noticed. Only a very few, notably Furnas [5], and Coppage and London [3], have applied transient techniques to a bed of the above description. Furnas, perhaps the experimental pioneer of the subject, used iron balls (1.8 to 4.8 cm dia.) in a column 1 m long by 2.5 cm diameter. He used three temperature probes, each at a different axial position. Each probe was cleverly designed to alternatively measure gas and solid temperature throughout the heating transient. The increase in solid temperature with time and the solid - gas temperature difference were used to derive the heat transfer coefficient. On the other hand, Coppage and London compare the temperature - time history of the fluid leaving the bed to an analytical solution by Schumann [8], though the exact method of comparison is not explained.

It has been found that the column to particle diameter ratio, D/D_p, is an important variable associated with what has been termed the «wall effect» [2, 4, 6, 7]. The average heat transfer coefficient is a maximum for $D/D_p = 6.67$ according to Leva and the effect diminishes as D/D_p becomes large [7]. For D/D_p greater than 17 the effect appears negligible [4].

DETAILED EXPERIMENT DESIGN

Initial design of the experiment was a series of compromises. A large D/D_p ratio minimizes the wall effect and was thus thought desirable. Also a large diameter for the column minimizes the effect of radial heat loss. A six inch diameter tube was considered at some length but abandoned in favor of a three inch tube when it was found that the required heating and air volume exceeded the scope of the project.

Desire to treat the sphere itself as a lumped item thermally led to the use of small 5.5 mm steel balls with air as the fluid. The Biot number has not in practice exceeded 0.01 and so this approximation is considered satisfactory. Air heating requirements, materials problems, and radiation effects are minimized by using a relatively small stepwise air temperature change to produce the desired temperature transients. Hence, in

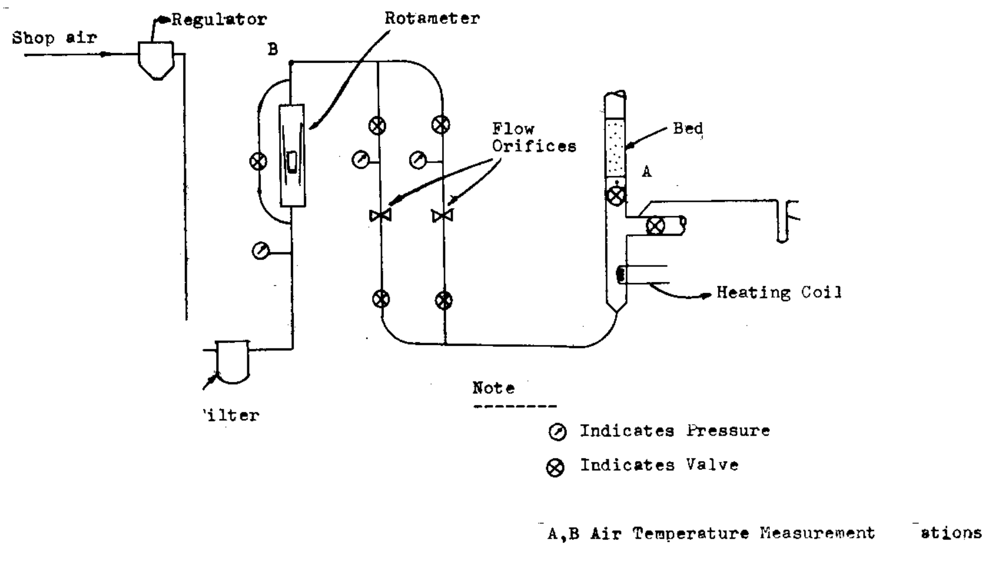


Figure 1. Piping Arrangement

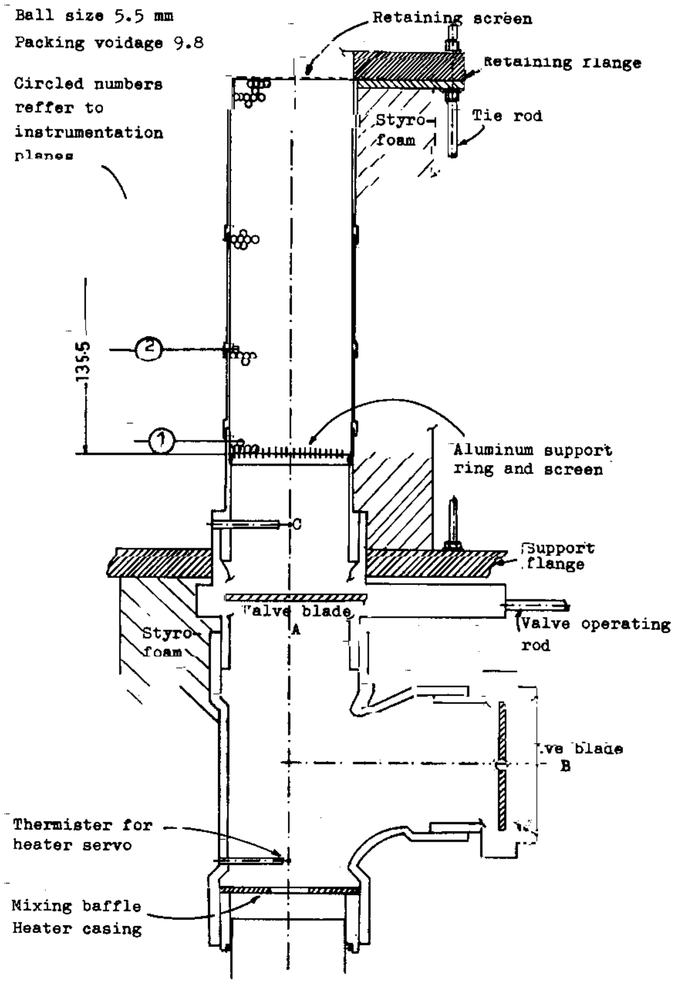


Figure 2. Bed detail

general, air was heated from room temperature to 65°C.

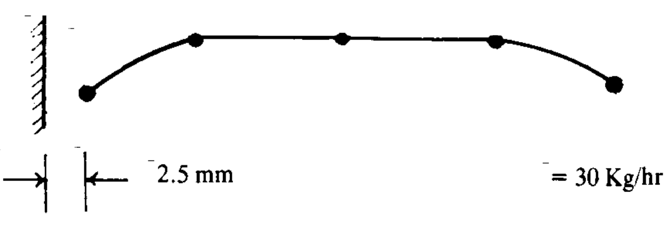
The piping arrangement is show in Figure 1. The twc flow orifices and by - pass around the rotameter were installed to handle high flow rates, (greater than 6C Kg) but this capability has not yet been used.

The air heater was made from a helix of nickel - chrome wire in order to minimize its thermal mass for control purposes. The heater power control takes its error signal from a bridge circuit wherein one leg consists of a thermister placed downstream of the heater together with a precision potentiometer. The desired thermister resistance, which is a function of the existing air temperature, is read directly off the potentiometer. Optimum response of the controller to disturbances in incoming air temperature or velocity is affected by the value of the capacitor. The optimum value of this capacitor depends on the flow rate, however, for this experiment a single value, 2.2 μf, was used throughout. In practice air temperatures have not deviated more than + 0.25°C form the desired setting.

surrounding wall of the bed was machined from extruded PVC pipe with as thin a wall as practical (1.25 mm) to minimize the heat sink and then surrounded with styrofoam insulation. The column was made in four sections for ease in instrumentation. The balls in the column were supported by an aluminum screen which consisted of a series of slats so spaced as to give a perfect hexagonal packing arrangement for the first iayer of balls. The pressure drop through this support screen and the retaining screen (common window type) on the top of the bed without the balls in between, could not be measured throughout the flow range with the instruments used. Moreover, the mass of the support screen (18 gr) has been ignored in the computer simulation.

Step changes in air temperature are affected with valves V_1 and V_2 in Figure 2. Typically, the desired air flow and temperature were allowed to stabilize with valve V_1 shut and V_2 open, with the bed above at room temperature. The apparatus was run in this fashion as long as possible without disturbing temperature in plane 1 (Figure 2) at which valve V_1 was opened and V_2 shut simultaneously. It has been noted that the transient response of the thermocouple probe at C (Figure 2) produced by using this technique is essentially the same as that produced by inserting a similar probe physically into an identical hot air stream. Hence, in the computer simulation a perfect stepwise change in air temperature is used.

To optimize the uniformity of the incoming air stream, five temperature probes were spaced equidistantly across the bed diameter at the inlet of the bed and a series of experiments made with different mixing - baffle arrangements. The temperature time history of each of the five probes caused by turning on the heater was then an indication of the local velocity and temperature. Starting with a completely cold system, with the present baffle arrangement, (Figure 2 a single orifice immediately above the heater) the probes within 2.5 mm of the wall deviated a maximum of 5.5° from the center three, which were invariably within 0.5°C of each other:



However, this non-uniformity rapidly disappeared with time; all probes were within 1.5°C after three minutes and slowly approached identity thereafter. Also, the non-uniformity was minimized during a normal run since the system was normally allowed to warm for several minutes before changing valves.

INSTRUMENTATION

The four instrumented planes indicated in Figure 2 are shown in detail in Figure 3. In each case the goal was to measure, for each of four radial positions, the temperature of a sphere and the adjacent air space. Other than an attempt to get near the center, touch the edge and two positions in between, placement of the in-

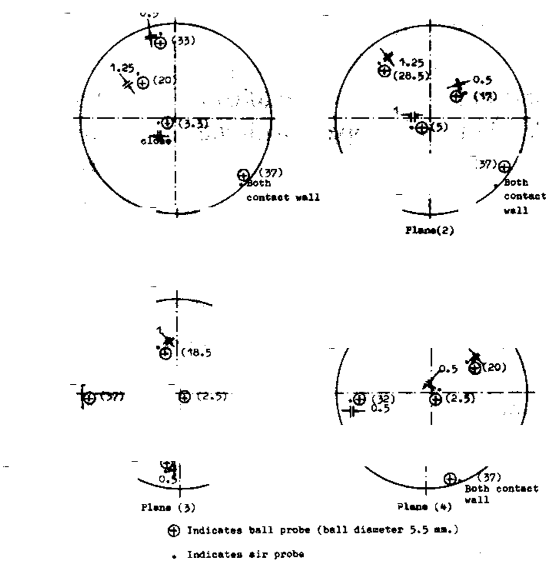


Figure 3. Bed detail location of temperature probes. (full scale)

strumented balls was random and depended in part on the lay of the bed underneath. For uniformity, a 0.25 mm, diameter copper - constantan thermocouple wire was used throughout. The balls to be instrumented were drilled to the center with a 1.25 m drill and the couple junction, jacketed with a plastic sheath so as not to touch the sides of the hole, was spot - welded to the bottom of the hole. The air temperature probes were placed as close as possible without touching a ball. Other balls were likewise gingerely placed to avoid contact with the air probe. Once the desired plane was instrumented, the next portion of the bed was filled by pouring balls in a few at a time and allowing them to seed their own position. It has been reported that this method of packing, results in a minimum practical void space [9]. Smaller void fractions can be obtained only by manually placing each sphere into a perfect regular array, as was done with the first two layers. Each thermocouple circuit was completed with a cold junction consisting of an ice - water bath. Two, twenty - point recorders were used simultaneously for data taking.

In addition to the four bed planes discussed above, air temperatures were measured at points A and B of Figure 1.

Pressure drop across the bed was measured by means of a manometer.

FLOW RATE CALCULATION

The rotameter scale was calibrated by comparison of two different scale readings to two known flow rates. The known flow rates were produced by dissipating a known power into the heater and measuring the temperture rise across the heater. Then,

$$\text{Flow rate} = \frac{\text{Power}}{C_p \times \Delta T}$$

where C_p is specific heat of air and ΔT is the temperature rise. Power was here supplied by a variable transformer and measured with calibrated volt and ampere meters. The correction necessary to account for the phase shift in volt and ampere sine waves due to the inductance of the heater coil was verified to be negligible. For each power reading the system was allowed to stabilize until no change in temperature rise could be detected, typically about 45 minutes. Several readings were then made and averaged to give a single value for temperature rise.

To take into account the radial heat loss several power inputs were made for each of the two rotameter settings. The flow rate at zero radial heat loss is then a simple extrapolation to a condition of zero power input.

To compensate for possible minute fluctuations in line voltage and air supply, the tests were made at a period of little activity at the laboratory. As it was, no such fluctuations were detected.

Note that, the extrapolation for the case of the lower flow rate (14.65 on the rotameter scale) may seem slightly artificial, it corresponds to exactly the same percentage scale correction as required at the higher flow rate and is justified in view of the difficulty of obtaining accurate power readings in the lower ranges of the instruments.

The correct flow rate is then the meter reading corrected for actual air density times the correction factor derived above (0.965). Hence:

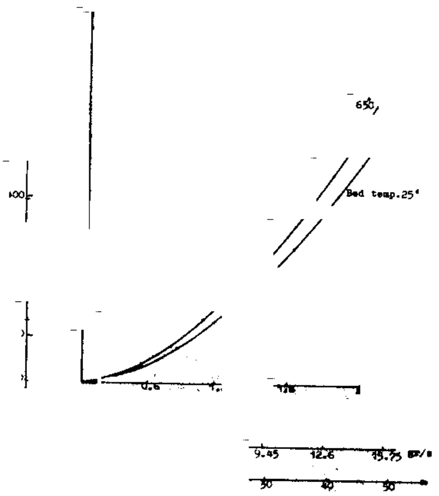


Figure 4. Pressure drop data

$$w_{actual} = w_{scale\ reading} \times 0.965 \left(\frac{\rho\ actual}{\rho\ standard} \right)$$

Where ω is the flow rate in Kg/m^3, ρ is density in kg/m^3, and air is treated as an ideal gas [10].

PRESSURE DROP

Pressure drop across the bed as a function of flow rate was taken in two runs. In the first, the air stream and bed were maintained at 65°C; in the second at 25°C. Raw data is summarized graphically in Figure 4, and compared to the work of Ergun [11] in Figure 5.

Ergun's correlation is an attempt to summarize a large amount of data, including beds of granular solids of mixed sizes, with a single equation. The deviation from his line is no worse than the scatter reported by Ergun himself.

It is also interesting to note that the data presented fall in the range considered neither turbulent nor laminar.

HEAT TRANSFER

Six runs were made which spanned the range of the rotameter. In each case the bed was heated from room temperature via the step change technique described above. The resulting data record is illustrated in Figure 6. Sixteen of these figures are produced for each run, corresponding to each of the ball-air probe thermocouple sets.

The heat transfer coefficient for each ball is then defined from

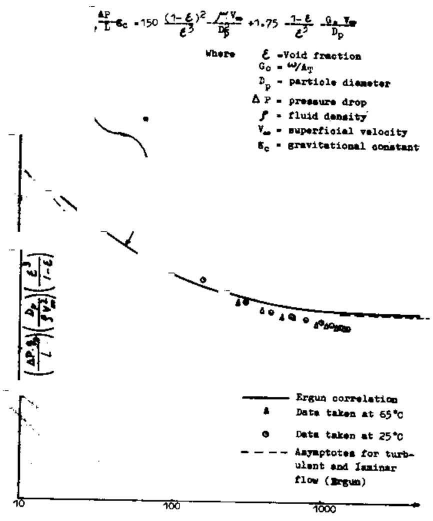


Figure 5. Packed bed pressure drop

$$\rho V C_p \frac{dH}{dt} = hA\,(T_a \quad T_b) \quad dT_b \quad dt$$

Where T_a is the air temperature, °C
T_b is the ball temperature
t is time, hr
V is ball volume, m^3
is the density kg/m^3
C_p is the specific heat, Kj/Kg°C
h is the heat transfer coefficient, W/(m^2.C)
A is the surface area of the ball, m^2

Now, T_a is the air temperature, and the data give only T_c, the temperature of the thermocouple. Thus, it was necessary to make a correction,

$$T_a = T_c + \left(\frac{\rho\ .V.C_p}{\rho\ \ hA}\ p\right)_{thermocouple} \times \frac{dT_c}{dt}$$

Or

$$T_a = T_c + \frac{1}{X} \times \frac{dT_c}{dt}$$

Where the quantity X was derived as follows: In a separate test, a trace of T_c vs. time was made by physically inserting a typical thermocouple junction into the hot air stream. Then X was calculated for several points along the trace from the relation,

$$\frac{T_c - {}_a}{\quad} = e^{-Xt}$$

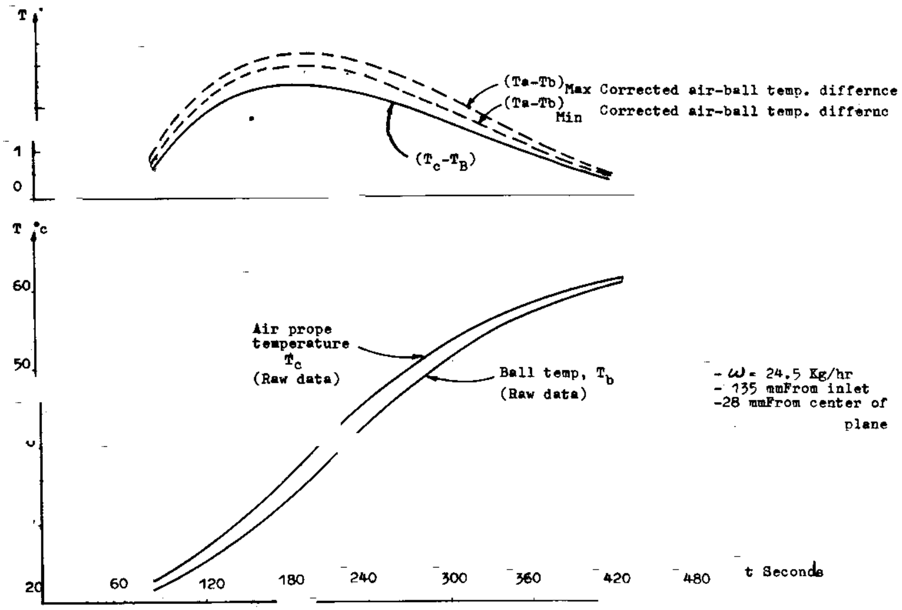


Figure 6. above T VS. time, bottom T VS. time.

Where T_i is the initial temperature. For the flow rate used, (20 Kg/h) it was determined that $562 < x < 802$. Now typically, heat transfer data for forced convection over spheres and cylinders are correlated as

$$Nu = c_1 Re^{c_2}$$

Where c_1 and c_2 are constants.
Then one can likewise say for this experiment,

$$X = c_3 \omega^{c2}$$

c_2 is taken as an average reported slope of .55 Kreith [12, 1] and one can then solve for c_3 with the result,

$$68.9 < c_3 < 98.3$$

for the extremes in X reported above. Thus it is possible to make maximum and minimum estimates of the true air temperature adjacent to each ball throughout the range of flow rates.

Figure 7 is an attempt to compare X with existing data. For such comparisons it is necessary to choose values for equivalent diameter and volume to surface area ratio. The data point in the Figure represents logical extremes for each case.

For data reduction, the time - temperature histories of each ball and air probe were digitized and a computer employed to calulate h for each position in time. The slopes were estimated by successively fitting a cubic polynomial to the data points. In calculalation to air temperature, the extreme values of c_3 were used, resulting in two values of T_a and h for each point in time. Inasmuch as extremes for c_3 were used, it is believed that the calculated values for h are extreme and bracket reality. The relative magnitudes of the true air temperature correction compared to the raw data are typified by Figure 6.

The time - wise variations of h were averaged and the results are in Figure 8. Gross averages for each plane are plotted in Figure 9. In these figures, Reynolds number is based on local conditions; i. e., an equivalent diameter and flow area defined as follows:

$$D_e = \frac{4(\text{wetted area})}{(\text{wetted perimeter})}$$

$$= \frac{4A_c\,(1\text{-p.f})}{D\,(1\text{-p.f.}\,\frac{D}{D_p})}$$

Where p.f is packing fraction (V_{balls} / V_{total})
A_c is cross sectional bed area
D is bed diameter
and $A_f = A_c\,(1\text{-p.f})$

The formulation for wetted perimeter is taken from Brown [13].

ACCURACY

As an aid in judging the accuracy of the averages,

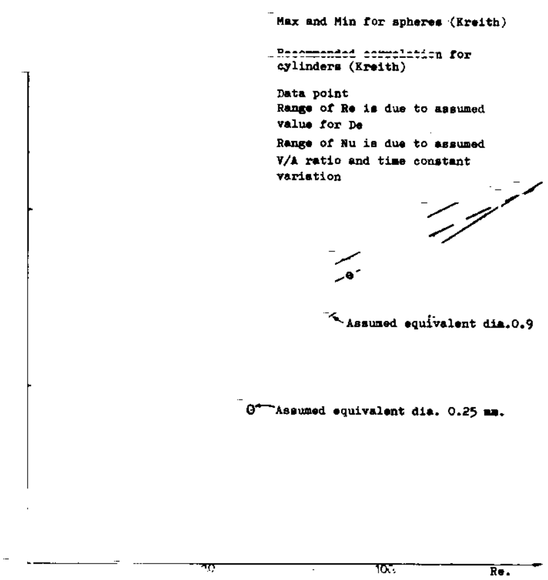


Figure 7. Comparing X with existing data

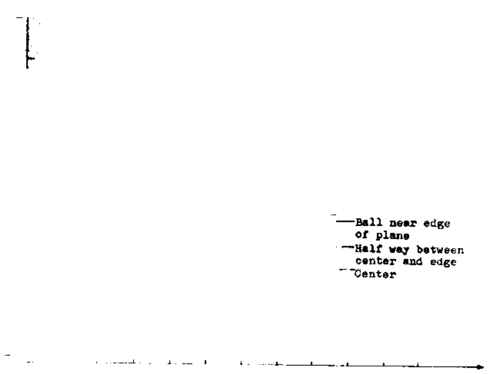


Figure 8. Average maximum and minimum values for h

Figure 8 , the number of positions in time for which each value was determined is indicated at the bottom of the figures. In addition, the absolute extreme maximum and minimum values found for each run are plotted in Figure 10.

It is obvious from Figure 6 that the validity of the calculation becomes more tenuous as T_b approaches T_a. In general, calculations were made only for $T_a - T_b = 0.5^\circ C$. Hence, the number of calculations becomes more limited for high flow rates and near the entrance of the bed.

It was also found that T_c for the balls toward the outside of a plane typically became lower than T_b as time approaches infinity. This is not surprising since there is some radial heat loss, and the air probes were typically placed at a slightly greater radius than the corresponding ball probes. However, for the balls at the wall this problem was too pronounced so these four edge balls were not used for heat - transfer calculations.

COMPARISON OF RESULTS TO OTHER INVESTIGATIONS

The Colburn j-factor is plotted against Reynolds number in Figure 11 for several investigations including this one. Here the triangles represent averaged for the three planes furthest from the entrance, and are best summarized by:

$$j_h = 1.55\, Re^{-.43}$$

It is difficult to judge the reason for the deviation of the results of one investigation from another in each particular case, but the spread is not surprising in view of the varition in measuring techniques and experimental conditions. The results of Furnas [5] for example, must certainly be influenced by the heavy walls of his container and the high temperatures (up to 760 °C) employed. The results from investigators who obtained data from a few probes buried within the bed [4 - 6] may suffer from not properly taking account of the wide variations which occur within a bed, which has been here illustred (Figure 8). Baumeister [2] took data (ball and gas temperature) only at the very entrance and exit of the bed, and his results may be abonormally high due to the higher heat transfer rates at the entrance which has here been illustrated in Figure [9].

COMPARISON OF RESULTS TO THEORY

The basis for mathematical comparison is stated by the differential equations below:

1. For the air $\qquad T_a(x,t)$

$$\frac{\sigma T_a}{\sigma t} + u\,\frac{\sigma T_a}{\sigma x} + \frac{hP}{P_a C_{pa} A_p}(T_a - T_b) = 0$$

2. For the balls $\qquad T_b(x,t)$

$$\frac{\sigma T_b}{\sigma t} - \frac{hP}{P_b C_{pb} A_b}(T_a - T_b) = 0$$

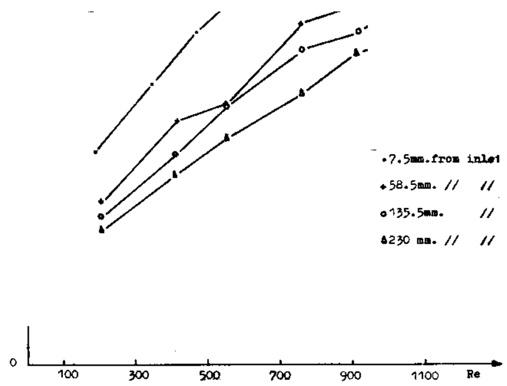


Figure 9. Heat transfer coefficient gross averages for each plane

3. Initial and boundary conditions

$T_a(x, O) = T_i$ the initial temperature

$T_b(x, O) = T_i$

$T_a(O, t) =$ the inlet air temperature

where p is the wetted perimeter [13]

A_b is the cross sectional area for the balls

Here it is noted that heat transfer occurs only via the convective coefficient h, constant material properties are assumed, and radial insulation is perfect. Analytic solutions are available [14, 8] in the form of dimensionless charts.

CONCLUSIONS

1. Perhaps the most striking fact is illustrated in Figure 12, which is a plot of the raw data for the ball temperatures of one run. For each plane in each run the balls toward the outside rose in temperature more quickly than the balls toward the center. The trend is monotonic and unbroken.

2. Conclusion number 1, above, is not reflected in any consistent fashion by the calculated local heat transfer coefficient (see Figure 12). The cause must therefore be found elsewhere, perhaps in analysis of local air velocities. Zabrodsky [9] reports mass velocities at the wall 30% to 70% higher than the average for the bed).

3. Calculated heat transfer coefficients are higher near the entrance of the bed (Figure 9). There seems, in addition, to be some falling off from the second to third planes and some recovery in the fourth. This is perhaps associated with classical convection theory which predicts higher transfer rates toward the leading edges of objects, and also the possible changing nature of the flow which is here neither clearly turbulent nor laminar, (Figure 5).

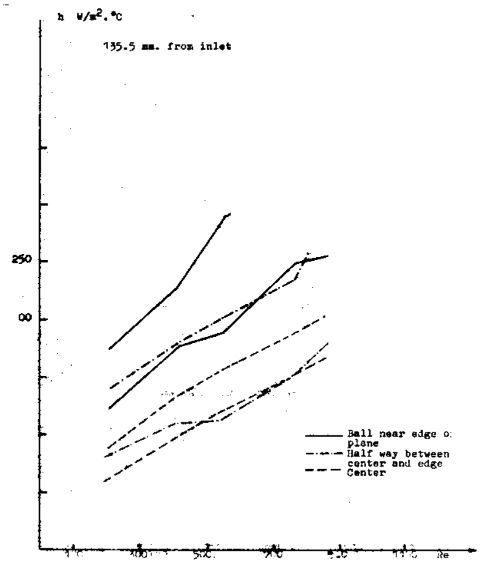


Figure 10. Maximum and minimum values for h.

4. There is considerable variation in the heat transfer coefficient in any given plane, (Figure 8).

5. The experimental results fall within the range reported by others, (Figure 11).

6. The agreement of the data with the numerical solution, «Rock Bed», is in general good with the greatest discrepancy at the beginning of each transient. It is thought that this is mostly due to the radial heat loss which is not treated by «Rock Bed».

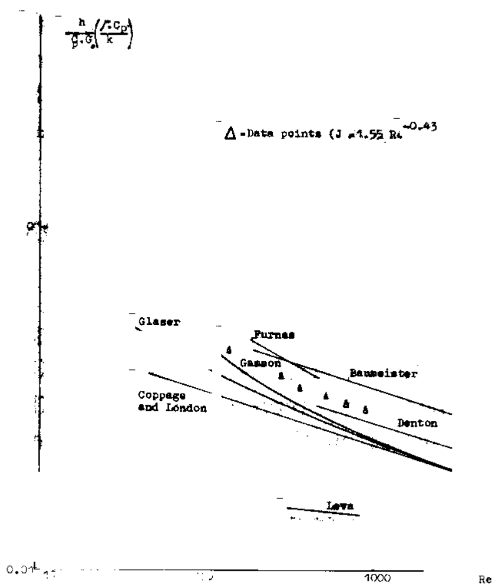


Figure 11. The Colburn J-factor against Reynolds Number