A STUDY OF HEAT-LOSSES BY FORCED CONVECTION

By

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Introduction

The problem of heat-transfer by convection has been the subject of many investigations during recent years. Although a considerable amount of experimental work has been done on the dissipation of heat from cylindrical tubes and ducts, relatively little data seems to be available on heat transfer during the flow of a fluid across a hot or cold body especially when the fluid is not bounded by the walls of a tube or channel. Again since our knowledge of the complex mechanism of heat losses by convection is still incomplete especially from the theoretical point of view, more experimental data is necessary for a satisfactory understanding of the thermodynamics of heat-exchange between a body and its surroundings under different ambient conditions.

Recently Kapadnis and Gogate (1952) and Kapadnis (1953, 1955) have investigated the variation of heat losses by convection from vessels of different shapes and sizes and tried to find out convection constants for different shapes of vessels. They have found that the convective heat-loss depends largely upon the shape of the vessel and varies approximately as the square root of the air velocity. Kapadnis (1953) studied the effect of fluid motion on heat transmission and determined the convective heat losses from vertical cylinders. He concludes from his experiments that the rate of heat transfer is proportional to 0.52th power of the air-velocity. In two subsequent investigations, Kapadnis (1953, 1955) has studied the heat losses due to forced convection from cylindrical, spherical and rectangular vessels and has tried to establish a quantitative relation between the Nusselt-Number and the Reynolds Number on the basis of his experimental data. Though his results cannot be said to be quite satisfactory, they appear to be fairly consistent within the range of air velocities and the limited range of Reynolds Numbers in which the observations were made. Most of his investigations were however limited to air-velocities above 240 cms./sec., there being very few regular observations for lower velocities ($<$ 240 cms./sec.) in the data published by him. We have tried to extend the investigation of heat dissipation by convection into the lower range of air velocities *i.e.* below 240 cm./sec. and have obtained some experimental results in the velocity range (47 to 293 cm./sec.) and within the range of Reynolds numbers 4468 to 33540. This has been made possible by the use of a very sensitive three cup anemometer (C. F. C. Anemometer Meteor No. 4) which could measure air velocities down to 15 cm./sec. Again Buttner (1934) and Winslow and others (1939) have treated the human body as a sphere of 15 cm. diameter or a cylinder of 7 cm. diameter for purposes of convective heat losses. We have therefore studied the heat losses from spheres and cylinders having nearly these values of diameters. The results of our investigation of Heat-transfer from spheres are described in this paper.

Theory

We know that in any problem of thermal convection, the heat transfer H can be denoted by a relation between the dependent variable dimensionless group ($Hl/k\triangle\theta$) known as the Nusselt number Nu and the three independent variable dimensionless groups known as the Reynolds number, Re, the Prandtl number, pr and the Grashof number, Gr (Fishenden and Saunders, 1950). Thus we can put

$$\frac{Hl}{k\triangle\theta} \propto \left(\frac{v\rho l}{\mu}\right)^{x_1}\left(\frac{c\mu}{k}\right)^{x_2}\left(\frac{ag\ \triangle\theta\ l^3\rho^2}{\mu^2}\right)^{x_3} \quad \ldots\ldots\ldots\ldots(1)$$

where the quantities within the three brackets on the right hand side represent Re, Pr and Gr. respectively. Here v denotes the forced velocity of the fluid, l the linear size of the vessel, $\triangle\theta$ the temperature difference between the fluid and the surface of the vessel, μ-viscosity, k-thermal conductivity, ρ-density, c-the specific heat of the fluid, ($a \times g$) – a product of coefficient of thermal expansion and acceleration due to gravity and x_1, x_2, x_3 are unknown indices.

For convection in gases, Pr may be taken as constant as it is nearly the same for all gases and over a wide range of temperature. For air, it is = 0.72. Again in the case of forced convection, the effects of buoyancy being negligible, the variable 'ag' may be dropped out since gravity no longer affects the problem. Putting $x_3 = 0$ in the general expression, the Grashof number is eliminated and Nu becomes dependent upon Re and Pr only, or, in the case of a gas, upon Re only. Thus we have

$Nu = B.\ Re^m$ ....(2) where B is a constant and $x_1 = m$.

The value of H—the heat loss due to convection alone, required for the evaluation of the Nusselt number is obtained by subtracting the heat loss H_2 by radiation [$H_2 = 1.73 \times 10^{-9} \times E\,(T_1^4 - T_2^4)$] from the total heat loss H_1 given by $H_1 = (M + Mo)\,\frac{d\theta}{dt}$. Here M is the mass of hot water in the vessel, M_0, the water value of the containing vessel, E, the emissivity of the surface taken as 0.55 for aluminium paint, T_1, the absolute temperature of the vessel and T_2 is the absolute temperature of the surroundings. For a temperature difference $\Delta\theta$ between the vessel and the surroundings, we therefore have $H = H_1 - H_2$.

Experimental

A spherical vessel of copper V, with a clean, white and uniform aluminium paint on its outer surface, was filled with some hot water and was placed at a distance of 40 cm. from an electric fan F (Fig. 1). A stream

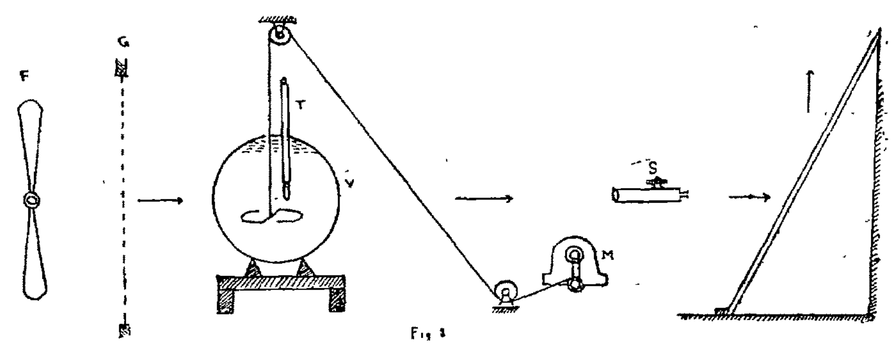

Fig. 1

of air proceeding from the fan was directed on to the vessel after allowing it to pass through a wire grid G situated at a distance of 15 cm. from the fan. The grid G served to render the divergent stream of air from the fan into a uniform parallel stream*. The top of the vessel was closed with a lid having two holes in it. Through one of the holes passed a sensitive thermometer and through the other a stirrer was kept working up and down in the vessel so that a uniform temperature was maintained throughout the whole mass of water at any instant. A very thin layer of oil was spread on the surface of water inside the vessel to help in preventing evaporation. The speed of the air stream proceeding from the fan could be

* In fact, the stream of air must be passed through a wind tunnel in order to make it sufficiently uniform and parallel, giving natural conditions. We are at present constructing a wind-tunnel and hope to report the effect of this refinement in a subsequent communication.

varied by changing the strength of the current in the circuit of the fan and the air velocity was measured by means of a sensitive three cup anemometer. The latter was so sensitive that it was possible to measure quite easily, air-velocities within the range 15 to 300 cm/sec., with sufficient accuracy. Special care was taken to avoid reflected air streams from the front wall by using various titled screens at different angles. These screens helped in diffusing and directing upwards, the oncoming air streams. The temperature of the water in the vessel was measured by focussing a small telescope S on the vertical thermometer T passing through the lid of the vessel. This procedure eliminated the possibility of affecting the temperature of the vessel by the breath of the observer. The stirrer was moved vertically up and down in the vessel by connecting it to a string passing over a pulley as shown in fig. 1. The other end of the string was passed round a pulley and then connected to an eccentric arrangement attached to the shaft of a low speed electric motor M. The air-velocity was measured by placing the sensitive anemometer exactly in the place of the vessel before and after every set of readings for the rate of cooling. The usual precautions were taken to minimize the heat losses due to conduction, radiation and evaporation. Losses due to natural convection were determined by separate experiments in still air and all subsequent observations were corrected for losses due to natural convection and radiation. The heat losses from the vessel, under these circumstances, were due to forced convection alone.

Different values of air-velocity ranging from 47 to 293 cm./sec. were used thus obtaining a number of corresponding cooling curves for the vessel. From these cooling curves the rate of fall of temperature $d\theta/dt$ for any mean temperature θ could be calculated. The values of $d\theta/dt$ are then used to calculate the total heat losses by the vessel at various temperature and air-velocities.

Results and Discussion

The values of Re and Nu were determined for several values of the air velocity at different mean temperatures of the vessel. These are recorded in Table I from which it can be easily seen that Re lies between 5916 and 33540 for the range of air velocities 48 to 277 cm./sec. for the larger sphere of diameter 20.4 cm. and that for the smaller sphere of diameter 15.8 cm., the range of Re is from 4468 to 27560 for air velocities ranging from 47 to 293 cm./sec. Fig. 2 gives a plot of Nu against Re which

TABLE I.

Diameter of the spherical vessel in cms.	Air velocity in cms./sec.	Reynolds Number Re.	Log. Re.	Rate of Heat Loss due to convection in K. cal./sq. meter/hr.	Nusselt Number Nu.	Log. Nu.	Temperature of the vessel in C°
	47.6	4468	3.6504	229.3	48.54	1.6861	64
	75.6	7106	3.8517	215.4	62.98	1.7992	56
	109.7	8191	3.9134	359.5	70.92	1.8502	56
	157.3	14790	4.1700	506.8	106. 0	2.0253	64
15.8	185.3	17390	4.2403	581.6	112. 7	2.0492	64
	218.9	20570	4.3132	607.7	131, 1	2.1176	64
	248.7	23320	4.3678	702.6	141. 7	2.1419	64
	293.2	27560	4.4402	760.7	154.4	2.1886	64
	48.7	5916	3.7720	131.4	56. 5	1.7524	50
	94.5	11470	4.0594	224.8	87. 8	1.9436	52
	120.7	14640	4.1657	189.3	99. 3	1.9973	46
20.4	137.1	16640	4.2214	263.0	112. 7	2.0517	50
	166.4	20180	4.3051	344.2	127. 7	2.1065	52
	205.1	24890	4.3960	356.8	140. 9	2.1488	50
	227.0	27550	4.4402	349.5	152. 3	2.1827	48
	256.0	31060	4,4922	448.6	162. 5	2.2108	52
	276.4	33540	4.5255	427.8	169. 7	2.2299	50

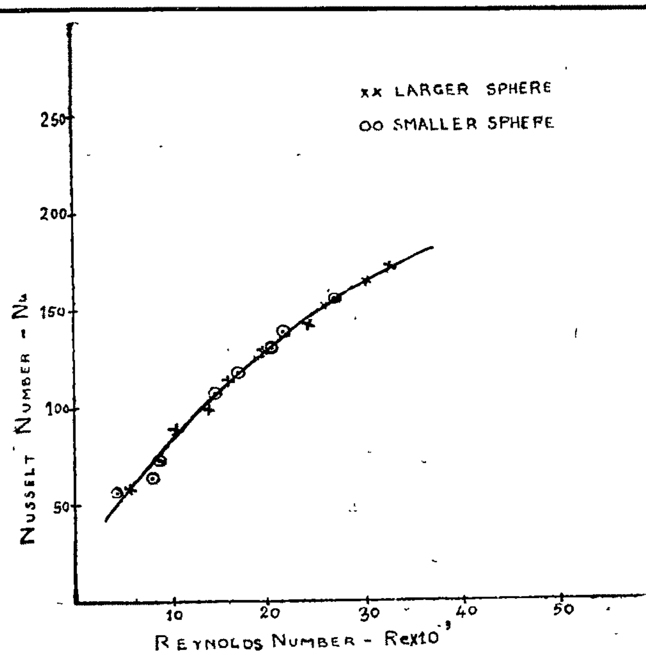

Fig 2

is a curve convex towards the Nu-axis. The Nu-Re curves for the two spheres with diameters 20.4 cm. and 15.8 cm. respectively, are very nearly coincident with each other which means that the size of the sphere has scarcely any effect on the nature of the graph. This conclusion is supported by the plot of log Nu against log Re in Fig. 3 where the two straight

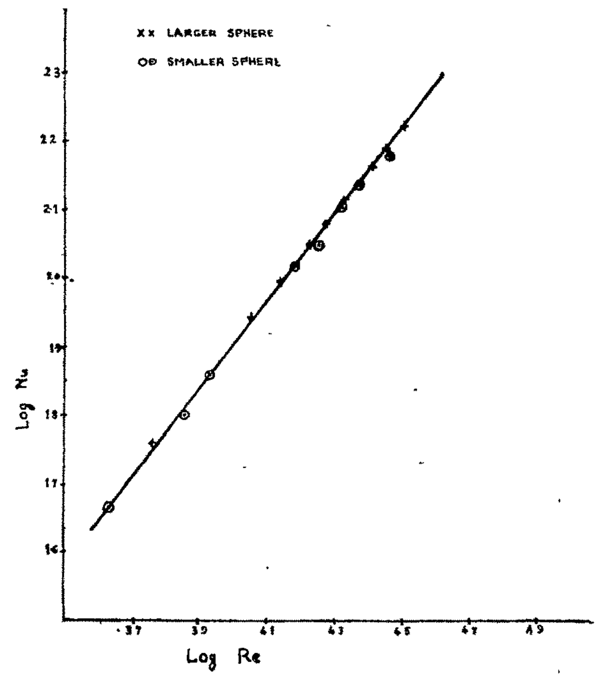

Fig 3

line graphs for the two spheres are nearly coinciding with each other. The values of B and m in the relation Nu = B. Re^m derived in equation (2) above, were calculated from Fig. 3. Here the slope of the straight line graph gives the value of m while the intercept on the log Nu-axis gives the value of log B. These values of B and m are then compared with those of other investigators as shown in Table II.

TABLE II

Re.	*B.*	*m.*	*Observers.*
4468–33540	0.295	0.61	Gogate, Patil and Desai.
47930–114000	0.203	0.65	Kapadnis and Gogate.
50–150000	0.340	0.60	McAdams.

The shape (Convection) constant C defined by the relation* :—Net heat loss $H = C A V^n \triangle \theta$ was found to be equal to 0.74 and 0.68 for the spheres of diameters 15.8 cm. and 20.4 cm. respectively for n = 0.61. These values may be compared with those obtained by Kapadnis and Gogate (1952) *viz.* :—1 27 and 1.21 for n = 0.5. If we take n = 0.61, the values of C according to the data of Kapadnis and Gogate work out to be 0.614 and 0.611 for the two spheres respectively.

Unfortunately, in the paper of Kapadnis and Gogate (1952), certain mistakes in the calculation of the shape constant C, have crept in. Thus for the value 1.629 in their paper, we get after recalculation, the value 1.266 for the smaller sphere while for the larger sphere, we get 1.214 instead of 1.568. We have carefully checked all the values of the shape constant C in that paper and have found that the C—values for the cylindrical and rectangular vessels appearing therein, are also not free from errors. We intend to discuss this point further in our next paper.

Acknowledgements :

Our thanks are due to Prof. K. R. Chaudhari for his help and useful suggestions during the course of this investigation.

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HEAT TRANSFER & REYNOLDS NUMBER

IT IS WELL KNOWN THAT THERE IS A CLOSE RELATION between heat transfer and momentum transfer near a solid surface over which a fluid at a different temperature is moving. The ratio of heat transfer per second, H, to the friction drag force, F, tangential to the surface is given by[1]

$$\frac{H}{F} = \frac{C\theta}{V}$$

where C is the specific heat of the fluid at constant pressure, θ is the temperature difference between the solid surface and the fluid and V is the velocity of the fluid. The drag coefficient, C_f, which is defined in aerodynamics as $\frac{2F}{\rho V^2}$, is then given by

$$C_f = \frac{2F}{\rho V^2} = \frac{2HV}{C\theta} \cdot \frac{1}{\rho V^2}$$

where ρ is the density of the fluid.

We have measured the amounts of heat transfer between solids (hot metal spheres, cylinders and rectangular vessels) and flowing streams of air for different velocities of air streams and have studied the relation between heat transfer H and Reynolds number $\text{Re} = \frac{\rho V l}{\mu}$, where l is the diameter of the sphere used and μ is the viscosity of air at the temperature at which the experiment is carried out. On plotting the Nusselt number, $\text{Nu} = \frac{Hl}{k\theta}$, against Re, we find that for values of Re up to nearly 10^5, the Nu-Re curve is concave to the Re axis, but as we approach the value of Re near about 10^5, the curve changes its form and becomes convex towards the Re axis. This change from concavity to convexity, towards the Re axis, of the curve giving the relation between heat transfer and Reynolds number has been confirmed by the results of experiments with spheres of different sizes (Fig. 1). This shows that the convective process of heat exchange undergoes a relatively sudden change near $\text{Re} = 10^5$.

An attempt has also been made to test the relation between the friction drag coefficient (C_f) and Reynolds number (Re) by plotting C_f against Re. It is found that the value of C_f goes on decreasing rather rapidly with increasing Re, attains a minimum value for $\text{Re} \sim 10^5$ and then increases again with increasing Re. This variation of C_f as a function of Re is associated with a change in the nature of the flow pattern as Re approaches its critical value, which is, in general, of the order of 10^5 (by the critical value of Re is meant the value for which C_f is minimum). For low Reynolds numbers, the flow in the boundary layer is laminar and it changes into turbulent flow when the Reynolds number exceeds the critical value. This value depends, to some extent, on the shape of the body and also on the nature of the surface, i.e. whether it is smooth or rough. Experiments on the direct determination of the drag coefficient of a sphere placed in wind tunnels have shown[2] that in the case of a sphere, the critical Reynolds number is of the order of 2×10^5 to 5×10^5.

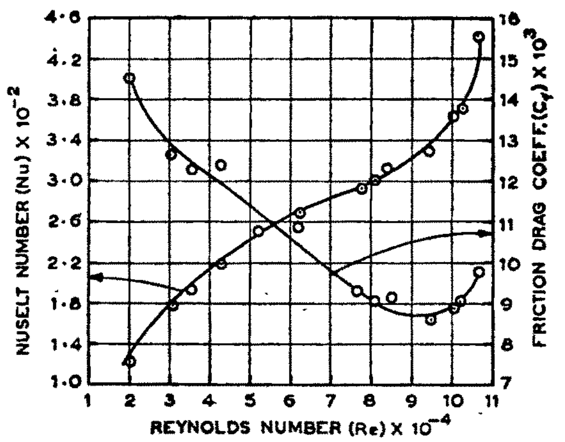

FIG. 1 — VARIATION OF REYNOLDS NUMBER (Re) WITH NUSSELT NUMBER (Nu) AND FRICTION DRAG COEFFICIENT (C_f)

We thus find, as is to be expected from the correspondence between the mechanisms of drag and convective heat loss, that the phenomenon of critical Re also manifests in heat exchange through the process of convection.

The details of this investigation will be published elsewhere.

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Heat Transfer by Forced Convection

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§ 1. Introduction

This note is concerned with the study of heat interchange between a solid body (metal sphere) and a moving fluid. It is an extension to higher Reynolds numbers of the work described in a previous paper (Gogate, Patil and Desai 1956). The main point that emerges from the present study is that the rate of heat exchange undergoes a relatively sudden change in the region of (Re) $= 10^5$, this being the region where the nature of the flow undergoes a change.

In § 2 a brief description of the apparatus is given and the results are described in § 3. After giving the relation between heat transfer, friction-drag force and the drag coefficient, the values of drag coefficient are exhibited for different Reynolds numbers. The nature of the change of flow for Reynolds numbers in the region of 10^5 is briefly described and its relation to the experimental results described in this note is then discussed in § 4.

§ 2. Experimental Arrangement

The apparatus used in this investigation was a slight modification of that used in the previous experiments (Kapadnis and Gogate 1952, Gogate, Patil and Desai

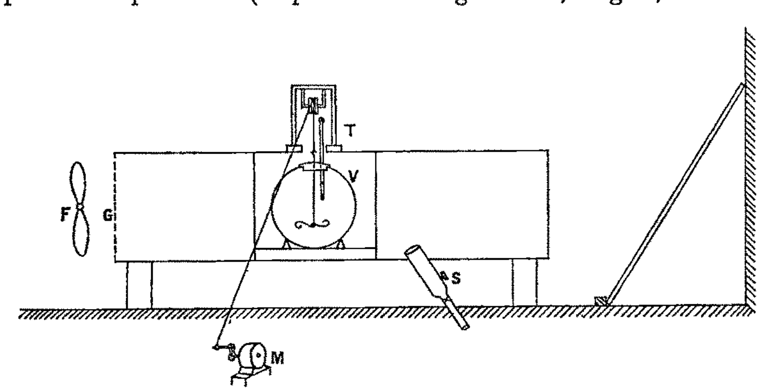

Figure 1.

1956). A spherical copper vessel V (figure 1) containing hot water was placed in a small wind tunnel at a distance of about 34 cm from an electric fan F and a stream of air proceeding from the fan was directed on to this vessel through the wind tunnel after allowing it to pass through a wire grid G situated at a distance of about 9 cm from the fan. The wind tunnel consisted of a rectangular wooden case 165 cm long with a square cross section 38 cm × 38 cm and open at both ends. The top of the vessel was closed with a lid having two holes in it. Through one hole passed a sensitive thermometer T and through the other a stirrer was kept working up and down by means of a string connected to an eccentric arrangement attached to the shaft of a low speed electric motor. The speed of the air stream could be varied by changing the strength of the current in the circuit of the d.c. fan F and the air velocity was measured by means of a sensitive three cup anemometer. The latter was placed exactly in the place of the vessel V before and after every set of observations for the rate of cooling of the vessel. Special care was taken to avoid reflected air streams from the front wall by using tilted screens at different angles. These screens helped in diffusing and directing upwards the oncoming air streams. The temperature of the water in the vessel was measured by focusing a small telescope S on the vertical thermometer T passing through the lid of the vessel. This procedure eliminated the possibility of affecting the temperature of the vessel by the breath of the observer. The usual precautions were taken to minimize the heat losses due to conduction, radiation and evaporation. Losses due to natural convection were determined separately in still air and all subsequent observations were corrected accordingly.

§ 3. Results

A number of cooling curves were obtained for the vessel using different values of air velocity and the rates of fall of temperature $d\theta/dt$ for different mean temperatures θ were calculated. These values of $d\theta/dt$ were used to calculate the total heat losses by the vessel at different temperatures and air velocities.

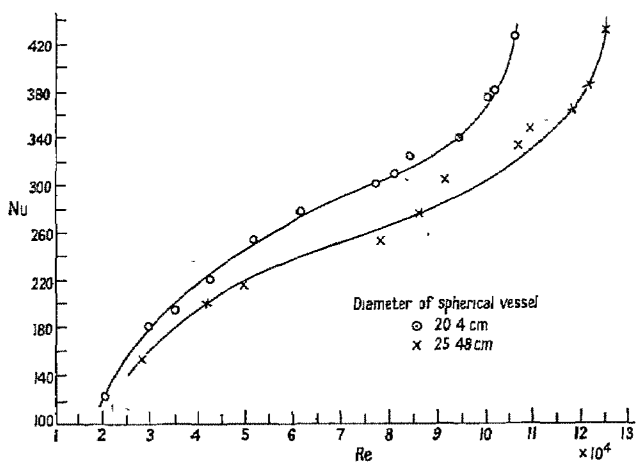

Figure 2.

To study the relation between heat transfer and the velocity of the air stream, a graph was plotted (figure 2) between the dimensionless quantity $Hl/k\Delta\theta$ known as the Nusselt number (Nu) and another dimensionless variable $\rho Vl/\mu$ known as the Reynolds number (Re) where H is the total heat loss due to forced convection, l the linear dimension of the vessel used, k the thermal conductivity of air, $\Delta\theta$ the temperature difference between the vessel and the surroundings, ρ the density and

μ the viscosity of air at the temperature of the experiment. It is clear from figure 2 that for values of (Re) up to nearly 8×10^4, the curve is concave to the (Re) axis but as we approach the value of (Re) near about 10^5, the curve changes its form and becomes convex towards the (Re) axis. This change from concavity to convexity towards the (Re) axis of the curve giving the relation between (Nu) and (Re) has been confirmed by experiments with spheres of different sizes as is evident from the two curves in figure 2. This shows that the convective process of heat exchange undergoes a relatively sudden change near (Re) $= 10^5$.

§ 4. Discussion

We know that the skin drag-coefficient C_f which is defined in aerodynamics as $2F/\rho V^2$ is given by (Fishendon and Saunders 1952)

$$C_f = \frac{2F}{\rho V^2} = \frac{2HV}{C\theta} \cdot \frac{1}{\rho V^2} = 2\,(\text{Stanton number})$$

where H represents the amount of heat-transfer per second, F is the friction drag force, θ the temperature difference between the surface of the vessel and the main fluid stream, C the specific heat of the fluid at constant pressure, V the velocity of the fluid stream and ρ its density at the temperature of the experiment. In figure 3, we have plotted C_f against the Reynolds number (Re). It will be

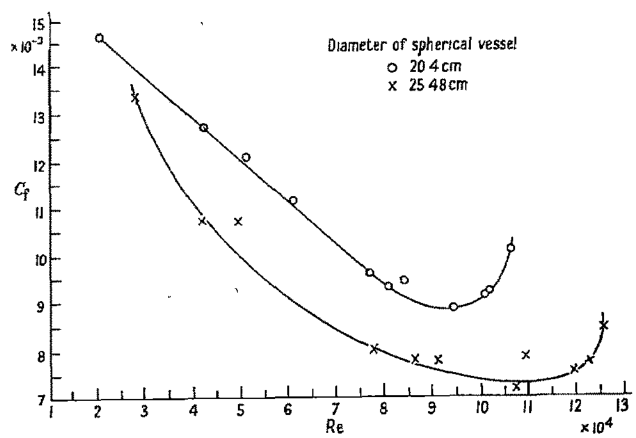

Figure 3.

observed that the value of C_f goes on decreasing rather rapidly with increasing (Re), attains a minimum value for (Re) $\sim 10^5$ and then it begins to increase again with increasing (Re). This behaviour of C_f as a function of (Re) is associated, as is well known (Goldstein 1938), with a change in the nature of the flow pattern as Re approaches its critical value which is, in general, of the order of 10^5; by the critical value of (Re) we refer to the value for which C_f is minimum. For low Reynolds numbers the flow in the boundary layer is laminar and it changes into turbulent flow when the Reynolds number exceeds the critical value. This value depends, to some extent, on the shape of the body and also on the nature of the surface, i.e. whether it is smooth or rough. Experiments on the direct determination of the drag coefficient of a sphere placed in wind tunnels have shown (Goldstein 1938) that in the case of a sphere the critical Reynolds number is comparable with $(2\text{–}5) \times 10^5$.

We thus find, as is to be expected from the correspondence between the mechanisms of drag and convective heat loss, that the phenomenon of critical (Re) also finds expression in heat exchange through the process of convection. It would be of interest to extend the experiments described in this note to the case of heat exchange between a solid body and a flowing liquid.

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