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Heat transfer by free convection and radiation – simply shaped bodies in air and other fluids

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Item was monitored and guided by the following Working Party:

Dr R. Cheesewright	— Queen Mary College, London University
Dr G.H. Cowan	— HTFS, Atomic Energy Authority, Harwell
Mr J. Lis	— Central Electricity Research Laboratories

The Fluid Mechanics and Thermodynamics Committee, which first met in May 1964, has the following constitution:

Chairman	
Mr E.C. Firman	— Central Electricity Research Laboratories
	—
Vice-Chairman	—
Dr G.F. Hewitt	— HTFS, Atomic Energy Authority, Harwell
	—
Members	—
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Mr E.A.D. Saunders	— Whessoe Ltd
Mr R.A. Smith	— ICI Ltd
Mr M.A. Taylor	— BNOC (Developments) Ltd
Prof. V. Walker	— Bradford University
Mr N.G. Worley	— Babcock and Wilcox (Operations) Ltd.

The work on this Item was carried out in the Fluid Mechanics and Physical Properties Group of ESDU under the supervision of Mr N. Thompson, Group Head. The member of staff who undertook the technical work involved in the initial assessment of the available information and the construction and subsequent development of the Item was

Dr A. Acton	— Senior Engineer.
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HEAT TRANSFER BY FREE CONVECTION AND RADIATION – SIMPLY SHAPED BODIES IN AIR AND OTHER FLUIDS

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HEAT TRANSFER BY FREE CONVECTION AND RADIATION – SIMPLY SHAPED BODIES IN AIR AND OTHER FLUIDS

1. NOTATION AND UNITS

Excepting angles (degrees), pressure (bar) and temperature (degrees Celsius) the SI system of units is used throughout.

A	surface area	m^2
a, b, c	linear dimensions	m
c_p	fluid specific heat capacity at constant pressure	$\text{J}/(\text{kg K})$
D	diameter	m
F	view factor	
f	relative pressure factor, $(p/p_0)^2$	
$Gr(L)$	Grashof number based on characteristic length L , $\beta g L^3 \rho^2 \Delta T / \mu^2$	
g	local gravitational acceleration*	m/s^2
H	height or length	m
h	average heat transfer coefficient	$\text{W}/(\text{m}^2 \text{K})$
L	characteristic dimension of body; D , H or M	m
M	area/perimeter	m
$Nu(L)$	average Nusselt number based on characteristic length L , hL/λ	
Pr	Prandtl number, $c_p \mu / \lambda$	
p	pressure†	bar
q	average heat flux (average heat flow per unit area per unit time)	W/m^2
T	absolute temperature‡	K
t	temperature	$^{\circ}\text{C}$
V	velocity of draught in vicinity of body	m/s
W	width of vertical or inclined plate	m
β	fluid volumetric expansion coefficient	$1/\text{K}$
ΔT	temperature difference, $ t_w - t_{\infty} $	K

For footnotes, refer to end of Notation Section

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δ	boundary-layer thickness	m
ε	total hemispherical emissivity (emissivity integrated over all wavelengths of thermal radiation and over a solid angle of 2π sr)	
θ	angle made by plate to vertical	degrees
λ	fluid thermal conductivity	W/(m K)
μ	fluid dynamic viscosity	N s/m ²
ρ	fluid density	kg/m ³
σ	Stefan's constant – equal to 56.7×10^{-9} W/(m ² K ⁴)	
ϕ	angle between two plane surfaces	degrees

Subscripts

c	refers to free convection
f	refers to properties at arithmetical average temperature, $(t_w + t_\infty)/2$, denoted as film temperature
0	refers to standard conditions
r	refers to radiation
s	refers to surface located near body
w	refers to surface of body
∞	refers to surroundings of body

* Standard value at sea level, $g_0 = 9.81$ m/s².

† Standard air pressure at sea level, $p_0 = 1.013$ bar.

‡ Standard temperature of air surrounding body, $T_{\infty 0} = 288$ K.

2. INTRODUCTION

This Data Item is intended to be mainly used for calculating the average steady-state heat transfer by free convection and radiation from internally heated, simply shaped, stationary bodies in still air at atmospheric pressure and ambient temperature. A flow-chart guide to the use of the Data Item is given on page 39.

The heat transfer can be calculated for the following body shapes:

- (i) vertical, inclined and horizontal plates (Section 4),
- (ii) vertical and horizontal cylinders (Section 5),
- (iii) spheres (Section 6).

Generally, these bodies are assumed to be isothermal and contained within a large isothermal free-space; allowance is made, however, for radiative heat transfer to and from a surface in close proximity to the body (Section 7.2). Factors that exert a controlling influence on the heat transfer are discussed in Sections 3 and 7, other factors that can sometimes be important are briefly discussed in Section 8.

The methods presented here are expected to yield convective and radiative heat transfer data for air which are accurate to within about ± 20 per cent. More detailed information concerning the accuracies of the correlation equations for free convection is included in Appendix A.

In the absence of condensation, the information presented can be used equally well for internally cooled bodies. Throughout the remainder of the text, internally heated bodies are referred to as hot bodies, and internally cooled bodies are referred to as cold bodies.

It is also possible to use the Data Item to calculate the heat transfer for these body shapes in other fluids*. An iterative process, shown in the flow-chart in Section 12, can be used to determine the surface temperature of the body if the average surface heat flux is specified.

3. CALCULATION OF HEAT TRANSFER – AIR AND OTHER FLUIDS

Free-convective fluid motion is generally confined to a boundary layer close to the body surface. The flow in this boundary layer is laminar for small bodies but for sufficiently tall bodies there is a development in the flow direction from laminar flow through a transitional state to fully turbulent flow. Evaluation of the heat transfer is relatively simple in the laminar and turbulent flow regimes, but difficult in the transitional regime. However, it is usually sufficiently accurate to evaluate the average heat transfer over the entire body surface by neglecting the transitional regime and considering the flow to be either laminar or turbulent, depending on the body size and the physical conditions. For vertical plates and horizontal cylinders, the data do not require the flow regimes to be identified; for other body shapes, distinguishing criteria are given.

The average free-convective heat transfer between an isothermal body and its fluid surroundings can be calculated from an average heat transfer coefficient defined by

$$h_c = \frac{q_c}{|t_w - t_\infty|} \quad (3.1)$$

* These methods are not applicable in the near-critical region or to the special case of water when the surface to bulk fluid temperature range includes the density maximum at 4°C.

In general, the average free-convective heat transfer coefficient can be related to the physical properties of the fluid, the forces controlling the fluid flow and the body geometry, in terms of $Nu(L)$, $Gr(L)$, Pr , H/D and θ . Both $Nu(L)$ and $Gr(L)$ are based on a length L that is characteristic for the body concerned and may be D , H or M ; Pr is a function of the fluid properties only. The factor H/D is the aspect ratio of a circular cylinder. General equations from which the free-convective heat transfer can be calculated are given in Appendix A and are applicable to most fluids.

Since in many free-convection problems, the body surface is at a moderate temperature and the surrounding fluid is air at atmospheric pressure and ambient temperature, the general equations have been simplified for ease of use in this particular application. The method of simplification, which introduces negligible error, is outlined in Appendix B. The resulting simplified equations apply to ambient air, defined as air at any naturally occurring pressure on the Earth's surface and satisfying the conditions

$$\text{and} \quad \left. \begin{array}{l} -10 < t_f < 85 \text{ ,} \\ -10 < t_\infty < 40 \text{ .} \end{array} \right\} \quad (3.2)$$

For bodies in ambient air, the appropriate method of obtaining h_c for the body shape in question should be selected from Sections 4 to 6 inclusive*. The thermal conductivity of air, which has to be evaluated in order to use these methods, is obtainable from either Figure 1 or Equation (C1.1) in Appendix C. The simplified equations contain a factor, f , that takes account of deviations of the air pressure from the standard value at sea level. At altitudes lower than 300 m above sea level, the air pressure will usually lie within the range

$$0.95 < p < 1.08 \text{ ,} \quad (3.3)$$

in which case f can be taken equal to unity. For air under conditions not covered by the simplified equations, or for fluids other than air, h_c can be calculated using the general equations in Appendix A.

Radiative heat exchange between a body and its surroundings is frequently significant compared with the heat transfer due to free convection. The methods in Section 7 should be used to calculate the average radiative heat flux for a body immersed in air or in any other gas that does not significantly absorb radiation (many gases fall into this category, but ammonia, carbon monoxide, carbon dioxide and steam have significant absorption characteristics). Most liquids are virtually opaque to thermal radiation, and consequently the net radiative heat exchange within them is negligible in comparison with free-convective heat transfer.

The total heat flux is obtained by addition of the average free-convective heat flux, q_c , obtained from h_c using Equation (3.1) and the average radiative heat flux, q_r .

4. CONVECTIVE HEAT TRANSFER – FLAT PLATES IN AIR

The equations in this Section apply to single surfaces of plates, but the heat transfer for double-sided plates can be calculated by considering the two surfaces independently. The heat transfer for a rectangular-sided box with its faces orientated in vertical and horizontal planes is often estimated by the ad hoc method of

* Data in Derivations 3, 9, 13 and 34 show that those equations which apply to laminar flow in Sections 4 to 6 can be used with little loss of accuracy up to t_f as high as 400°C. The air flow can be considered to be laminar provided that $f\Delta TL^3/\lambda_f^4$ is less than 2×10^7 (i.e. $Gr(L)Pr < 10^9$, see Appendix B) except where inclined or horizontal surfaces are involved. Appropriate laminar flow limits for these surfaces can be obtained from Sections 4.2 and 4.3.

adding individual contributions for each of its six faces. These contributions can be calculated using equations in Sections 4.1 and 4.3, Equation (3.1) and the appropriate surface areas. The neglect of interactions between the boundary layers at the edges of the box (especially the vertical edges) is a weakness of this method, but in the absence of more rigorous methods, it is the best available.

4.1 Vertical Plates

The average heat transfer coefficient for free convection along a vertical rectangular flat plate in ambient air is calculated using the equation

$$h_c = [0.825(\lambda_f/H)^{1/2} + 0.605(f\Delta T/\lambda_f)^{1/6}]^2 \quad (4.1)$$

This equation, applicable over the laminar, transitional and turbulent flow regimes, is derived from Equation (A2.1) in Appendix A. Strictly, Equation (4.1) is valid for

$$\left. \begin{aligned} &7 \times 10^{-3} < f\Delta TH^3/\lambda_f^4 < 2 \times 10^{10}, \\ &\text{and } H/W < 2. \end{aligned} \right\} \quad (4.2)$$

However, either or both of these criteria may be exceeded for tall plates. Equation (4.1) is expected to apply up to $H/W = 5$ without appreciable loss of accuracy; limited use of the equation beyond the upper bound of $f\Delta TH^3/\lambda_f^4$ is permissible, but may give lower accuracy.

4.2 Inclined Plates

The average heat transfer coefficient for free convection from an inclined rectangular plate is calculated using one of three equations depending on whether the plate faces upwards or downwards and on whether the flow is laminar or turbulent. As a consequence of the direction of the component of the free-convective driving force perpendicular to the surface of the plate in each case, an upward facing hot plate is equivalent to a downward facing cold plate, and a downward facing hot plate is equivalent to an upward facing cold plate.

The value of the parameter $f\Delta TH^3/\lambda_f^4$ can be used to determine the approximate limit between the turbulent and laminar flow regimes for ambient air. These regimes are illustrated in Figure 2; a positive value of the angle θ corresponds to a hot plate facing upwards (or a cold plate facing downwards) and a negative θ corresponds to a hot plate facing downwards (or a cold plate facing upwards).

In the laminar flow regime beneath the line in Figure 2, the average heat transfer coefficient in ambient air is calculated for both upward and downward facing plates using the equation

$$h_c = 0.68\lambda_f/H + 1.31[(f\Delta T/H)\cos\theta]^{1/4}, \quad (4.3)$$

where the characteristic dimension H is measured along the length of the plate (see Figure 2).

In the turbulent flow regime for negative θ , the average heat transfer coefficient in ambient air is given by

$$h_c = 1.1[f\Delta T\cos^2\theta]^{1/3}, \quad (4.4)$$

whilst for turbulent flow with positive θ ,

$$h_c = 1.2[f\Delta T]^{1/3} . \quad (4.5)$$

Equations (4.3), (4.4) and (4.5) are derived from Equations (A2.2), (A3.1) and (A3.2) in Appendix A. Strictly, Equations (4.3), (4.4) and (4.5) are valid for

$$\left. \begin{aligned} 2 \times 10^3 < f\Delta TH^3/\lambda_f^4 < 10^{10} , \\ -80 < \theta < 60 , \\ \text{and } H/W < 2.5 . \end{aligned} \right\} \quad (4.6)$$

However, the equations are expected to apply up to $H/W = 5$ without appreciable loss of accuracy.

4.3 Horizontal Plates

The average free-convective heat transfer coefficient for a horizontal plate of arbitrary shape* in ambient air is obtained using one of two equations depending on whether the flow is laminar or turbulent. Upward facing hot plates are equivalent to downward facing cold plates, and downward facing hot plates are equivalent to upward facing cold plates.

For laminar flow

$$h_c = C_1 (f\Delta T/M)^{1/4} , \quad (4.7)$$

where M is a characteristic dimension that is equal to the surface area of the plate divided by its perimeter (one quarter of the side length of a square plate, or one quarter of the diameter of a disc).

For turbulent flow

$$h_c = C_2 (f\Delta T)^{1/3} . \quad (4.8)$$

Equations (4.7) and (4.8) are derived from Equations (A4.1), (A4.2) and (A4.3) in Appendix A. Values of the parameter $f\Delta TM^3/\lambda_f^4$, which indicates the ranges of validity of Equations (4.7) and (4.8), are given in the table together with values of the constants C_1 and C_2 .

* These data are not appropriate to a highly elongated shape. For an upward facing hot (or downward facing cold) surface, the data should be applied only for a plate that is either rectangular with the ratio of its side lengths lying in the range from 1 to 4, or for a shape that can be totally enclosed by a minimum area rectangle of this type. Experimental data for downward facing hot (or upward facing cold) surfaces are very sparse, and strictly only applicable to approximately square plates. However, cautious extrapolation to other shapes is usually permissible in view of the low values of h_c encountered.

	Laminar Regime – Equation (4.7)			Turbulent Regime – Equation (4.8)		
	$\frac{f\Delta TM^3}{\lambda_f^4}$ range	C_1	Derivation	$\frac{f\Delta TM^3}{\lambda_f^4}$ range	C_2	Derivation
Upward facing hot or downward facing cold	70 to 2×10^4	1.27	30 and 38	2×10^4 to 10^7	1.71	5 and 38
Downward facing hot or upward facing cold	10^2 to 10^7	0.46	5	No data, but region of little practical importance.		

5. CONVECTIVE HEAT TRANSFER – CIRCULAR CYLINDERS IN AIR

The equations in this Section apply to the curved surfaces of cylinders; if the ends of the cylinders are included in the heat transfer surface, end effects must be considered for short cylinders, *i.e.* for

$$H/D < 5. \quad (5.1)$$

For short vertical cylinders, it is recommended that the total free-convective heat transfer should be estimated by addition of individual contributions for the curved surface and the two end surfaces. These contributions are calculated from equations in this Section and Section 4.3, Equation (3.1) and the appropriate surface areas. An overestimate of the heat transfer may be incurred using this procedure, but underestimation is unlikely.

For short horizontal cylinders, the flows over the cylindrical and end surfaces are likely to interact with unknown effects on the total heat transfer. However, if the area of the curved surface is less than about 10 per cent of the total area of the two end surfaces ($H/D < 0.05$), the body can be treated as a double-sided vertical plate and it is recommended that the heat transfer should be calculated using Equation (4.1) with H replaced by D .

5.1 Vertical Cylinders

The average free-convective heat transfer coefficient for a vertical cylinder in ambient air is calculated using one of two equations depending on whether the flow is laminar or turbulent.

For laminar flow, the following equation derived from Equation (A5.1) in Appendix A is applicable,

$$h_c = 0.52\lambda_f/D + 1.4(f\Delta T/H)^{1/4}. \quad (5.2)$$

Equation (5.2) is valid for

$$\text{and} \quad \left. \begin{aligned} 2 \times 10^2 < f \Delta T H^3 / \lambda_f^4 < 2 \times 10^7, \\ H/D < 40 (f \Delta T H^3 / \lambda_f^4)^{1/4}. \end{aligned} \right\} \quad (5.3)$$

For turbulent flow, vertical cylinders and plates are equivalent as far as the free-convective heat transfer is concerned when the following conditions are satisfied:

$$\text{and} \quad \left. \begin{aligned} f \Delta T H^3 / \lambda_f^4 > 2 \times 10^7, \\ H/D < 50. \end{aligned} \right\} \quad (5.4)$$

In this case, Equation (4.1) should be used to calculate h_c . Limited data for cylinders with H/D exceeding 50 show an increase of heat transfer in comparison with data for vertical plates.

5.2 Horizontal Cylinders

The average heat transfer coefficient for free convection around a horizontal cylinder in ambient air is calculated using the equation

$$h_c = 0.36 [(\lambda_f / D)^{1/2} + (f \Delta T / \lambda_f)^{1/6}]^2. \quad (5.5)$$

This equation, applicable over the laminar, transitional and turbulent flow regimes, is derived from Equation (A6.1) in Appendix A. Equation (5.5) is valid for

$$7 \times 10^{-7} < f \Delta T D^3 / \lambda_f^4 < 2 \times 10^{11}. \quad (5.6)$$

6. CONVECTIVE HEAT TRANSFER – SPHERES IN AIR

The average heat transfer coefficient for predominantly laminar free-convective flow of ambient air about a sphere is calculated using the equation

$$h_c = 2.0 \lambda_f / D + 1.10 (f \Delta T / D)^{1/4}. \quad (6.1)$$

This equation, derived from Equation (A7.1) in Appendix A is valid for

$$2 \times 10^{-5} < f \Delta T D^3 / \lambda_f^4 < 2 \times 10^7. \quad (6.2)$$

Whenever the upper limit of this criterion is exceeded, the flow is likely to be turbulent over a significant portion of the surface of the sphere, but there are no known experimental data with which to establish an equation for this condition.

7. RADIATIVE HEAT TRANSFER

Heated or cooled bodies immersed in gases will generally exchange a significant amount of heat with their surroundings by thermal radiation. The heat exchange may be simply calculated by the methods given here if the gas is only a weak absorber of radiation and a diffuse-grey approximation can be made. This approximation requires the emissivities of the surfaces involved in the radiative interchange to be independent of the direction and wavelength of the radiation. All these requirements are reasonably well satisfied for unpolished surfaces in air. Polished surfaces do not show diffuse-radiation behaviour, but their emissivities are small, and consequently the radiative heat exchange between these surfaces and their surroundings is usually negligible in comparison with the free-convective heat transfer.

7.1 Body in Large Enclosure

The average heat flux, q_r , from an opaque body to an enclosure that is large compared with the body can be calculated from the equation

$$q_r = \varepsilon_w \sigma (T_w^4 - T_\infty^4). \quad (7.1)$$

This equation is applicable to black enclosures, *i.e.* enclosures that absorb all of the radiation emitted by the body. This is strictly true only if the walls of the enclosure are black ($\varepsilon_\infty = 1$). However, an enclosure made up of lower emissivity walls can still be treated as black provided that it is large enough for most of the radiation emitted by the body to be lost to the enclosure walls by a process of multiple reflection and absorption without returning to the body. The total hemispherical emissivity ε_w should be evaluated at the body surface temperature T_w whenever the data permit. Emissivity data for a number of commonly used surfaces are given in Table 11.1; further data may be obtained from Derivations 8, 14 and 26.

7.2 Body and Nearby Surface in Large Enclosure

The information in this Section applies to the situation in which a simply shaped body is in close proximity to another surface. The effect of a nearby surface on the radiative heat transfer can be considerable. The free-convective heat transfer can also be affected under certain circumstances, briefly discussed in Section 8.4.

In the following simplified analysis, both the body surface and the adjacent surface are assumed to be opaque and diffuse-reflecting, and it is further assumed that neither surface is re-entrant. These conditions will be met in most practical situations in which one of the body shapes dealt with in this Data Item is located near to a plane surface (such as a wall) or a convex surface.

The total net average radiative heat flux from the body to surroundings that comprise the nearby surface in an effectively black enclosure is given by

$$q_r = \frac{\varepsilon_w \sigma \left\{ \varepsilon_s F_{ws} (T_w^4 - T_s^4) + \left[1 - F_{ws} + (1 - \varepsilon_s) F_{ws} \left(1 - \frac{A_w}{A_s} F_{ws} \right) \right] [T_w^4 - T_\infty^4] \right\}}{1 - (1 - \varepsilon_w)(1 - \varepsilon_s) \frac{A_w}{A_s} F_{ws}^2}. \quad (7.2)$$

The view factor, or configuration factor, F_{ws} , in this equation is defined as the fraction of the radiation from the surface of the body which strikes the nearby surface. View factor data for the body shapes dealt with in this Data Item and adjacent plane surfaces are given in Figures 3 to 6. The usefulness of these

Figures can be extended by employing the algebraic techniques of Appendix D. If the combination of body and adjacent surface shapes under consideration cannot be dealt with by the use of these Figures and techniques Reference 2, which contains a useful catalogue of known view factors, should be consulted.

In many practical situations, either or both of ϵ_w and ϵ_s will be close to unity and the denominator of Equation (7.2) will approximate to unity. A further simplification is possible if the nearby surface may be considered to be black ($\epsilon_s = 1$) giving, for any ϵ_w

$$q_r = \epsilon_w \sigma \left\{ T_w^4 - F_{ws} T_s^4 - (1 - F_{ws}) T_\infty^4 \right\}. \quad (7.3)$$

8. OTHER FACTORS THAT MAY AFFECT THE HEAT TRANSFER

8.1 Non-uniform Surface Temperature

The equations in this Data Item apply to body surfaces at uniform temperature, which is the boundary condition most often approximated in practice. In some applications, such as electrical heating, the heat flux may be nearly uniform over the surface of the body. For laminar flow, the surface temperature will then be non-uniform.

It is shown in Derivation 10, that for laminar flow along a vertical plate the same equation for the average free-convective heat transfer is applicable for both uniform temperature and uniform heat flux boundary conditions, provided that, in the latter case, the surface temperature at the mid-height of the plate is used to form the temperature difference $\Delta T = |t_w - t_\infty|$. If the flow is fully turbulent, h_c is effectively independent of vertical height and the two boundary conditions are equivalent. The temperature at the mid-height of the plate is also an appropriate temperature for radiative heat transfer calculations in the uniform heat flux case.

Except for a horizontal plate and as a first approximation, it is therefore reasonable to use the equations in this Data Item for uniform heat flux conditions by setting t_w equal to the surface temperature at the mid-height of the body. At the present time, the data for horizontal plates are extremely sparse, and the distributions of surface temperature are unknown. It is therefore only possible to deal with horizontal plates if the integrated average surface temperature (or a reasonable approximation to it) is known, or if it is required to calculate this average temperature.

In some applications, neither the surface temperature nor the surface heat flux can be considered to be uniform. Whenever variation in ΔT (or q_c) becomes appreciable compared with the average value, such a situation would be expected to arise, but its analysis is difficult and there are few experimental data. However, methods reviewed in Derivation 20 do provide guidance for free-convective heat transfer calculations with laminar flow and various distributions of temperature and heat flux over the height of a vertical plate.

8.2 Draughts

If air movement other than that caused by the heated or cooled body occurs, the convective heat transfer may be appreciably modified. However, in the case of ambient air, provided that

$$L\Delta T/(30V^2) \gg 1, \quad (8.1)$$