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Specification for

Air heating and cooling coils —

Part 1: Method of testing for rating of cooling coils

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Co-operating organizations

The Refrigeration, Heating and Air Conditioning Industry Standards Committee, under whose supervision this British Standard was prepared, consists of representatives from the following Government departments and scientific and industrial organizations:

Association of Consulting Engineers*
Association of Manufacturers of Domestic Electrical Appliances
Boiler and Radiator Manufacturers' Association*
British Gas Corporation
British Mechanical Engineering Confederation
British Oil and Gas Firing Equipment Manufacturers' Association
British Refrigeration and Air Conditioning Association*
Department of the Environment
Department of Health and Social Security*
Electricity Council, The Central Electricity Generating Board and the Area Boards in England and Wales
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The Government departments and scientific and industrial organizations marked with an asterisk in the above list, together with the following, were directly represented on the committee entrusted with the preparation of this British Standard.

Department of the Environment, Building Research Establishment
Greater London Council
Manufacturers' Association of Radiators and Convectors
Oil Appliance Manufacturers' Association
Unit Heater Manufacturers' Association

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Contents

Foreword

This Part of this British Standard has been prepared under the authority of the Refrigeration, Heating and Air Conditioning Industry Standards Committee in response to requests from industry.

The Committee acknowledge their debt to the Heating and Ventilating Research Association for the Association's work in formulating the methods of testing that appear in this standard.

This Part is the first of a series concerned with air heating and cooling coils.

It is intended to issue further Parts which will deal with methods of testing for rating of heating coils, with air cooling coils for chilled water and for direct expansion, and with construction and safety aspects of heating and cooling coils.

Where reference is made to British Standards for which no metric version is available then the appropriate British Standard in imperial units shall be used in conjunction with BS 350 "*Conversion factors and tables*"; attention is also drawn to BS 3763 "*The International System of units (SI)*".

A British Standard does not purport to include all the necessary provisions of a contract. Users of British Standards are responsible for their correct application.

Compliance with a British Standard does not of itself confer immunity from legal obligations.

Summary of pages

This document comprises a front cover, an inside front cover, pages i to iv, pages 1 to 50, an inside back cover and a back cover.

This standard has been updated (see copyright date) and may have had amendments incorporated. This will be indicated in the amendment table on the inside front cover.

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0 Introduction

The principal complexities in testing cooling coils arise from

a) small water temperature differences;

b) the accuracy of measurement of air wet-bulb temperature near saturation;

c) the non-uniform air temperature and non-uniform humidity profiles off the coil.

a) and b) can be overcome either by specifying sophisticated instrumentation or by the careful use of simple instruments, e.g. thermocouple and mercury in glass thermometer; the latter alternative has been adopted in this standard, errors being assessed by an overall heat balance. The non-uniform off-coil conditions can only be overcome by adequate air mixing and suitable mixers have therefore been specified.

In this standard fairly tight tolerances have been applied to the definition of steady state conditions. The tight tolerances are to allow for the probability of small water temperature differences and for the large effect of a small air wet bulb temperature change on the air total heat and dew point. The duration of the test period has been chosen to ensure an accurate average heat transfer figure.

The wide range of cooling coils produced by any one manufacturer makes it desirable that testing is carried out only on a relatively small number of representative coils. Such a technique is already in existence for hot water air heater batteries (see BS 3208) where a representative coil is defined as one having the same water tube arrangement, number of circuits, fin type and spacing as the coils in the production range.

It is logical to apply the same principle to cooling coils. However, the effect of dehumidification cannot be ignored and consequently the method of analysis employed for cooling coils is considerably more complex than that given for heating coils in BS 3208. The method of analysis used in this standard is based upon that given in ARI[1)] Standard 410-64, "*Standard for forced-circulation air-cooling and air-heating coils*", as this is considered to be the most satisfactory method available at present.

Tests have to be carried out on coils with both dry and fully wet surfaces in order to obtain the air-side thermal resistances. This data is then used to obtain the heat transfer, both latent and sensible, for a wide range of entering air conditions.

1 Scope

This standard gives a method of testing for rating of duct-mounted air cooling coils, with chilled water as the cooling medium, within the following range of variables:

Inlet air temperature: below 40 °C

Inlet water temperature: 2 °C to 20 °C

Inlet air wet bulb depression: greater than 4 °C

Sensible heat ratio (Q_s/Q_T): less than 0.8 (for dehumidifying coils)

Water flow: Reynolds number above 3 100

Air velocity at coil face: 1 m/s to 4 m/s

2 References

The titles of the British Standards referred to in this standard are listed on the inside back cover.

3 Definitions

For the purpose of this standard, the following definitions apply:

3.1
cooling coil

a water-to-air heat exchanger of the tubular type through which air is passed by mechanical means over the external surface; such as is normally connected into a system of ventilation ductwork

in this British Standard, the heat exchanger surface considered is of the extended surface, externally finned type

[1)] Air Conditioning and Refrigeration Institution, Arlington, Virginia, USA.

3.2
reference air conditions

temperature: 20 °C, absolute pressure: 1.013 bar [2)], relative humidity: 43 %, density: 1.200 kg/m^3

3.3
rows

the number of banks of tubes in the direction of the air flow (see Figure 1)

3.4
passes

the number of times the water in any one circuit crosses the air flow (see Figure 2)

3.5
similar coils

a range of similar coils is defined as coils having the same:

a) water tube size, spacing, arrangement (in-line or staggered, see Figure 2) and internal construction;

b) fin type, construction and spacing (see Figure 3);

c) water flow geometry, i.e. the general method of combining rows and circuits shall be the same for all coils in the range.

3.6
turbulator

a device inside a tube for increasing the turbulence and heat transfer

4 Nomenclature

Symbol	Description	Units
A_{dc}	Calculated dry coil external surface area	m^2
A_{De}	Surface area of ductwork between inlet sampling tube and centre of coil	m^2
A_{Do}	Surface area of ductwork between outlet sampling tube and centre of coil	m^2
A_F	Coil face area	m^2
A_i	Coil internal surface area ($N_t \pi d_i l_t 10^{-3}$)	m^2
A_o	Coil external surface area ($A_p + A_s$)	m^2
A_{oc}	Calculated coil external surface area	m^2
A_p	Area of exposed tubes	m^2
A_s	Total surface area of coil fins (both sides)[a]	m^2
A_t	Cross-sectional area of coil tube	m^2
A_{wc}	Calculated wet coil external surface area	m^2
B	Coil surface area ratio (A_o/A_i)	—
C	Thermal resistance ratio	kg K/kJ
c_{pa}	Specific heat capacity of moist air	kJ/(kg K)
c_{pw}	Specific heat capacity of water	kJ/(kg K)
c_{pam}	Specific heat capacity of air at mean air temperature	kJ/(kg K)
c_{pwm}	Specific heat capacity of water at mean water temperature	kJ/(kg K)
C_o	Heat transfer exponent	
d_i	Tube internal diameter	mm

[a] For A_s, total surface area of coil fins, where fin collars are employed, the collar is taken as part of the exposed tube area.

[2)] 1 bar = 10^5 N/m^2 = 100 kPa.

Symbol	Description	Units
d_o	Tube external diameter	mm
E	Effectiveness	—
F	Factor for corrected logarithmic mean temperature	—
f_a	Air side heat transfer coefficient	W/(m^2 K)
f_w	Water side heat transfer coefficient	W/(m^2 K)
f_{ws}	Water side heat transfer coefficient for smooth bore tubes	W/(m^2 K)
f_{wt}	Water side heat transfer coefficient for tubes fitted with turbulators	W/(m^2 K)
g	Acceleration due to gravity	m/s^2
h_{ai}	Enthalpy of inlet air	kJ/kg dry air
h_{aic}	Corrected enthalpy of inlet air	kJ/kg dry air
h_{ao}	Enthalpy of outlet air	kJ/kg dry air
h_{aoc}	Corrected enthalpy of outlet air	kJ/kg dry air
h_{si}	Enthalpy of saturated air at coil surface temperature at air inlet	kJ/kg dry air
h_{so}	Enthalpy of saturated air at coil surface temperature at air outlet	kJ/kg dry air
h_{am}	Mean air enthalpy $(h_{aic} + h_{aoc})/2$	kJ/kg dry air
h_{sm}	Enthalpy of saturated air at mean coil surface temperature	kJ/kg dry air
h_{sD}	Enthalpy of saturated air at inlet air dew point	kJ/kg dry air
h_B	Enthalpy of air at wet/dry boundary	kJ/kg dry air
h_s	Enthalpy of saturated air at apparatus dew point	kJ/kg dry air
Δh_m	Log mean enthalpy difference	kJ/kg dry air
k	Thermal conductivity of insulating material	W/(m k)
K_i	Air inlet section heat leakage coefficient	kW/K
K_o	Air outlet section heat leakage coefficient	kW/K
K_w	Hydraulic pressure drop correction factor	—
k_t	Thermal conductivity of tube wall material	W/(m K)
k_f	Thermal conductivity of fin material	W/(m K)
l_e	Length of one complete water circuit $(l_i = N_t l_t / N_c)$	m
l_t	Length of each water tube (one pass)	m
M	Air-to-water side heat capacity ratio	—
m_a	Air mass flow rate	kg/s
m_w	Water mass flow rate	kg/s
N	Number of rows in coil	—
N_c	Number of circuits	—
N_m	Number of rows in test coil	—
N_n	Number of rows in n row coil	—
N_t	Number of tubes in coil	—
P_b	Barometric pressure	bar[a]
P_F	Static pressure upstream of flow meter	Pa
ΔP_{sn}	Static pressure drop for n row coil	Pa

[a] 1 bar = 10^5 N/m^2 = 100 kPa.

Symbol	Description	Units
Δp_c	Coil air side pressure drop	Pa
Δp_{cr}	Coil air side pressure drop corrected to reference density	Pa
Δp_F	Air flow meter pressure drop	Pa
Δp_w	Hydraulic pressure drop (measured)	Pa
Δp_{wc}	Hydraulic pressure drop corrected for head differences between coil inlet and outlet	Pa
Δp_{wct}	Hydraulic pressure drop at reference temperature	Pa
Q	Heat transferred	kW
Q_s	Mean sensible heat transferred	kW
Q_{as}	Sensible heat transferred on air side	kW
Q_{aT}	Total heat transferred on air side	kW
Q_T	Mean total heat transferred	kW
Q_w	Heat transferred on water side	kW
R	Overall thermal resistance	m^2 K/W
R_{ad}	Air film thermal resistance for a dry surface	m^2 K/W
R_{aw}	Air film thermal resistance for a moist surface	m^2 K/W
R_f	Fin thermal resistance	m^2 K/W
R_{md}	Dry metal thermal resistance	m^2 K/W
R_t	Tube wall thermal resistance	m^2 K/W
R_w	Water film thermal resistance (based on A_o)	m^2 K/W
t_a	Ambient temperature, (dry bulb)	°C
t_{ad}	Apparatus dew point	°C
t_{ai}	Inlet air temperature, (dry bulb)	°C
t_{ao}	Outlet air temperature, (dry bulb)	°C
t_{aic}, t_{aoc}	t_{ai} and t_{ao} corrected for heat losses	°C
t'_{ai}	Inlet air temperature, (wet bulb)	°C
t'_{ao}	Outlet air temperature, (wet bulb)	°C
t_D	Inlet air dew point	°C
t_F	Flow meter temperature, (dry bulb)	°C
t'_F	Flow meter temperature, (wet bulb)	°C
t_{si}	Coil surface temperature at inlet	°C
t_{so}	Coil surface temperature at outlet	°C
t_{sm}	Mean coil surface temperature	°C
t_{wi}	Inlet water temperature	°C
t_{wo}	Outlet water temperature	°C
t_{wm}	Mean water temperature; $(t_{wi} + t_{wo})/2$	°C
Δt_m	Corrected logarithmic mean temperature difference between air and water	°C

Symbol	Description	Units
V_r	Air volume flow rate corrected to reference air density; the reference air density is 1.2 kg/m^3	m^3/s
v_r	Air velocity at reference conditions	m/s
v_w	Mean water velocity	m/s
ΔW	Change in moisture content across coil	kg/kg dry air
X_b	Inner radius of annular fin	mm
X_e	Outer radius of annular fin	mm
Y_f	Fin thickness	mm
Y_i	Thickness of insulating material	mm
y	Factor defined in **B.2**	kg K/kJ
Z	Ratio of slope of saturated air enthalpy curve to specific heat capacity of moist air	—
Greek Symbols		
β	Bypass factor	—
γ, δ	Parameters defined in **12.1.1.2**	—
η	Surface effectiveness	—
ϕ	Fin efficiency	—
ρ	Density	kg/m^3
ρ_r	Reference air density (1.2 kg/m^3)	kg/m^3
Subscripts		
a	air; apparatus	
B	boundary	
b	barometric; inner radius (of fin)	
c	calculated; circuits; corrected	
D	dew point; duct	
De, Da	duct dimensions	
e	outer radius (of fin)	
F	face; flow meter	
f	fin	
i	inlet; insulating; internal	
m	mean; metal; number of rows in test coil	
n	a number	
o	external; outlet	
p	exposed tubes; pressure	
r	reference condition(s)	
s	saturated; sensible heat; smooth; surface	
t	reference temperature; tube(s); turbulator	
T	total	
w	water; wet	

5 Instrumentation

5.1 Temperature

5.1.1 The measurement of temperature shall comply with the requirements of BS 1041. If liquid in glass thermometers are used, they shall be graduated in intervals not exceeding 0.1 °C, and shall comply with the requirements of BS 593, partial immersion ranges. All temperature measuring instruments shall be calibrated, against an NPL calibrated thermometer, to a precision of 0.1 °C.

5.1.2 The entry and exit water temperature shall be measured by means of instruments inserted into oil filled pockets similar to those shown in Figure 4.

5.1.3 For water temperature differences below 5 °C it will be necessary to take special care and the differential thermocouple system shown in Figure 5 is recommended. A separate measurement of inlet water temperature will be required.

Where a differential thermocouple system is employed, then the differential in millivolts shall be divided by half the number of junctions (i.e. 3 in the case of that shown in Figure 5), and added to or subtracted from the relevant measured temperature (converted to millivolts if measured by a direct reading instrument).

5.1.4 The entry and exit air wet bulb and dry bulb measurements shall be made by means of the system given in BS 4194:1967 (Appendix D). The measurement system shall be connected to sampling tubes similar to those shown in Figure 6.

5.2 Water flow measurement. The measurement of water flow should preferably be by means of direct weighing (although one of the methods described in BS 1042 may be used). The water leaving the test rig shall be collected in vessels of known weight, and weighed to an accuracy of 0.1 % over the range of the weights used in the test. The weight of each vessel used shall not exceed 50 % of its normal contents. The net weight of each charge shall be recorded by weighing the vessel both after emptying the previous charge and after filling.

5.3 Air flow measurement

5.3.1 Air flow measurement shall be in accordance with BS 1042-1 or, alternatively, a flow meter may be calibrated in-situ by the method given in BS 1042-2.

5.3.2 Leakage flow rates may be measured by means of a calibrated variable area flow meter or integrating air volume flow meter, see Figure 7.

5.4 Pressure measurement

5.4.1 Wall static pressures shall be measured with static taps complying with the specifications laid down in BS 1042-2. A minimum of four taps shall be symmetrically disposed about the duct walls and connected to form a piezometric ring.

5.4.2 Manometers shall have scale intervals no greater than 2 % of the indicated reading, with the exception of pressure below 50 Pa, in which case the maximum intervals shall be 1 Pa.

5.4.3 An inverted U-tube manometer, similar to that shown in Figure 10, shall be employed for the measurement of hydraulic pressure drop.

5.4.4 The minimum differential pressures for flow measurement shall be a) 25 Pa for inclined U-tube manometers and micro-manometers and b) 500 Pa for vertical U-tube manometers.

6 Air flow leakage test

The heat transfer tests described in this standard are to be carried out on coils that have been sealed against air flow leakage; this will not, however, always be the case in practice and consequently it is necessary to carry out air flow leakage tests to indicate possible degradation of coil heat transfer performance.

6.1 A standard coil shall be connected to an air supply system similar to one of those shown in Figure 7.

6.2 The air supply pressure shall be increased to that desired for the test and held for a minimum period of 15 min and then reduced to zero. The supply pressure shall then be increased to the previous level and held constant for a further 15 min, at the end of which the air flow rate shall be read and recorded as the coil leakage flow rate at the test pressure.

NOTE If an integrating flow meter is employed measurement may commence 10 min after the second pressurization.

7 Test apparatus

7.1 Waterside

7.1.1 There shall be available a means for providing and controlling a continuous supply of water at any temperature and flow rate that are required for the test. Suitable arrangements of equipment are shown in Figure 8 and Figure 9.

7.1.2 The pipe work shall be arranged to give an unobstructed straight run at entry and exit from the coil under test, the pipe diameter being equal to that demanded by the unit connections and of a length equal to five pipe diameters. The thermometer pockets shall be positioned such that the water flow is towards the base of the pocket, as indicated by the arrows in Figure 4.

7.1.3 Side wall tappings for measuring the hydraulic pressure drop shall be fitted adjacent to the connections to the coil. These tappings shall be as specified in Figure 10 and connected to form a piezometer ring.

7.1.4 The lengths of pipe between the temperature measurement positions, the coil connections and the coil casing shall be insulated with at least 40 mm thickness of insulating material having a thermal conductivity not exceeding 0.06 W/(m K)

7.2 Air side

7.2.1 The test coil shall be connected to an air supply and measuring system similar to that shown in Figure 11, the flow metering system being located downstream of the coil under test.

7.2.2 The air velocity, temperature and humidity profiles at the sampling tubes shall be uniform (to within 5 %) and therefore, mixers similar to those shown in Figure 12 shall be employed, together with the flow straightener, gauze screens, and changes in section indicated in Figure 11.

NOTE It is recommended that the profiles at the inlet and outlet sampling tubes are checked during the assembly of the test rig as any large deviations from the mean may give rise to difficulty in obtaining a heat balance. The outlet conditions should also be checked before the flow metering system is connected.

7.2.3 The test coil and ductwork between the inlet and outlet sampling tubes shall be insulated with at least 50 mm thickness of insulating material having a thermal conductivity not exceeding 0.06 W/(m K). Ductwork between the sampling tubes shall be constructed from metal and ductwork from the inlet sampling tube to the flow meter shall be sealed against air leaks. The test coil shall be sealed against air leakage and fitted with condensate drains.

NOTE If the ductwork between the outlet sampling tube and the flow meter is of non-absorbent material then it will not be necessary to measure the wet bulb temperature at the flow meter.

8 General test instructions

8.1 The coil condensate drains shall be connected to a condensation collection system similar to that shown in Figure 13. Care shall be taken to ensure that:

a) the tubes containing the sealing water do not rise above the level of the bottom of the coil drain tray;

b) there is a sufficient head of water within the drain tubes to prevent air leakage.

8.2 If the test is to be on a dry coil surface then the inlet air dew point shall not be higher than the entering water temperature.

8.3 Before commencing the test

a) bleed the water supply system to remove all air;

b) start the air supply system and set the desired flow rate;

c) circulate the water through the coil and regulate the flow and temperature to those desired for the test;

d) set the required air inlet temperature and humidity.

8.4 The test shall be carried out under steady state conditions. These shall be said to exist when the following measurements do not vary by more than the specified amounts from their mean value over a period of 60 min; readings being taken at 15 min intervals, or alternatively, using a data logger.

a)	Inlet air temperature, dry bulb	± 0.1 °C
	wet bulb	± 0.1 °C
b)	Inlet water temperature	± 0.1 °C
c)	Water flow rate	± 1 %
d)	Air flow rate	± 1 %

For wet coil tests, the condensate shall be seen to be flowing into the collecting vessel(s) indicated in Figure 13.

NOTE It is unlikely that the condensate will be discharged at a uniform rate. The form of discharge is likely to be periodic and if possible at least one cycle should be allowed to elapse before commencing testing.

8.5 The test shall occupy not less than 60 min, and complete sets of data shall be taken at intervals not exceeding 10 min with the exception of barometric pressure and ambient temperature which shall be recorded at start and finish. During the test, the mean values shall not vary by more than those for the steady state conditions (see **8.4**). A complete set of data shall comprise:

Symbol	Description	Units
t_{wi}	is the inlet water temperature	°C
t_{wo}	is the outlet water temperature	°C
t_{ai}	is the inlet air temperature, (dry bulb)	°C
t'_{ai}	is the inlet air temperature, (wet bulb)	°C
t_{ao}	is the outlet air temperature, (dry bulb)	°C
t'_{ao}	is the outlet air temperature, (wet bulb)	°C
Δp_c	is the coil air side pressure drop	Pa
Δp_F	is the air flow meter pressure drop	Pa
p_F	is the static pressure upstream of flow meter	Pa
t_F	is the flow meter temperature, (dry bulb)	°C
t'_F	is the flow meter temperature, (wet bulb)	°C
m_w	is the water flow rate from weighing or pressure drop measurements	kg/s
Δp_w	is the hydraulic pressure drop	Pa
p_b	is the barometric pressure	bar[a]
t_s	is the ambient temperature, (dry bulb)	°C

[a] 1 bar = 10^5 N/m^2 = 100 kPa.

8.6 The test measurements shall be averaged and the mean values used in all computations.

9 Calculations

9.1 Water side heat transfer. The heat transferred to the water shall be computed from the equation:

$$Q_w = m_w c_{pw}(t_{wo} - t_{wi}) \text{ (kW)}$$

where

m_w is the water mass flow rate (kg/s) obtained by direct weighing or by means of a flow meter

c_{pw} is the specific heat capacity of water at $(t_{wo} + t_{wi})/2$ [kJ/(kg K)]

t_{wi} is the inlet water temperature (°C)

t_{wo} is the outlet water temperature (°C)

9.2 Air side heat leakage

9.2.1 The heat leakage, coefficients K_i, K_o, (kW/K) for the sections between the inlet air temperature measuring station and the coil centre, and the coil centre and outlet air temperature measuring station respectively shall be computed from the equations:

Inlet section: $K_i = A_{De}k/Y_i$(kW/K)

Outlet section: $K_o = A_{Do}k/Y_i$(kW/K)

where

A_{De} is the surface area of ductwork between inlet sampler and centre of coil (m^2)

A_{Do} is the surface area of ductwork between outlet sampler and centre of coil (m^2)

Y_i is the thickness of insulating material (mm)

k is the thermal conductivity of insulating material [W/(mK)]

9.2.2 The following corrections shall be made for heat losses and gains to the air passing through the test coil:

Inlet air enthalpy at coil

$$h_{aic} = h_{ai} + \frac{K_i(t_a - t_{ai})}{m_a} \text{ (kJ/kg (dry air))}$$

Inlet air temperature (dry bulb) at coil

$$t_{aic} = t_{ai} + \frac{K_i(t_a - t_{ai})}{m_a c_{pa}} \text{ (°C)}$$

Outlet air enthalpy at coil

$$h_{aoc} = h_{ao} - \frac{K_o(t_a - t_{ao})}{m_a} \text{ (kJ/kg (dry air))}$$

Outlet air temperature (dry bulb) at coil

$$t_{aoc} = t_{ao} - \frac{K_o(t_a - t_{ao})}{m_a c_{pa}} \text{ (°C)}$$

where

h_{aic} is the corrected enthalpy of inlet air [kJ/kg (dry air)]

h_{ai} is the enthalpy of inlet air [kJ/kg (dry air)]

K_i is the air inlet section heat leakage coefficient (kW/K)

t_a is the ambient temperature (dry bulb) (°C)

t_{ai} is the inlet air temperature (dry bulb) (°C)

m_a is the air mass flow rate (kg/s)

t_{aic} is the inlet air temperature (dry bulb) corrected for heat losses (°C)

C_{pa} is the specific heat capacity of moist air [kJ/(kg K)]

h_{aoc} is the corrected enthalpy of outlet air [kJ/kg (dry air)]

h_{ao} is the enthalpy of outlet air [kJ/kg (dry air)]

K_o is the air outlet section heat leakage coefficient (kW/K)

t_{ao} is the outlet air temperature (dry bulb) (°C)

t_{aoc} is the outlet air temperature (dry bulb) corrected for heat losses (°C)

9.2.3 If the total heat leakage is less than 1 % of the test coil capacity (Q_w), no correction is necessary.

NOTE **Air mass flow rate.** The air mass flow rate shall be calculated from the flow meter calibration with the air density obtained from the barometric pressure, upstream static pressure, the wet and dry bulb temperatures at the flow meter. If, however, the duct between the flow meter and the off-coil sampling tube is constructed from a non-water-absorbing material, the necessary moisture content may be obtained from the off-coil measurements.

9.3 Air side sensible and total heat transfer

9.3.1 *Sensible heat transferred*

$Q_{as} = m_a c_{pa} (t_{aic} - t_{aoc})$ (kW)

9.3.2 ***Total heat transferred***

$Q_{aT} = m_a(h_{aic} - h_{aoc}) - m_a c_{pw} \Delta W t'_{ao}$ (kW)

where

Q_{as}	is the sensible heat transferred on air side (kW)
m_a	is the air mass flow rate (kg/s)
c_{pa}	is the specific heat capacity of moist air [kJ/(kg K)]
t_{aic}	is the inlet air temperature (dry bulb) corrected for heat losses (°C)
t_{aoc}	is the outlet air temperature (dry bulb) corrected for heat losses (°C)
Q_{at}	is the total heat transferred on air side (kW)
h_{aic}	is the corrected enthalpy of inlet i [kJ/kg (dry air)]
h_{aoc}	is the corrected enthalpy of outlet air [kJ/kg (dry air)]
pw	is the specific heat capacity of water [kJ/(kg K)]
ΔW	is the change in moisture content across coil [kg/kg (dry air)]
t'_{ao}	is the outlet air temperature (wet bulb) (°C)

9.4 Mean heat transfer

For sensible cooling onl $-(Q + Q_{as}/2$(kW)

For sensible and latent cooling $Q_T\ (Q_w + Q_{aT})/2$(kW)

where

Q_s	is the mean sensible heat transferred (kW)
Q_w	is the heat transferred on water side (kW)
Q_{as}	is the sensible heat transferred on air side (kW)
Q_T	is the mean sensible heat transferred (kW)
Q_{aT}	is the total heat transferred on air side (kW)

9.5 Heat balance

A test shall be void if the ratios:

Q_w/Q_{as} or Q_w/Q_{aT} are outside the limiting values 0.95 and 1.05

where

Q_w	is the heat transferred on water side (kW)
Q_{as}	is the sensible heat transferred on air side (kW)
Q_{aT}	is the total heat transferred on air side (kW)

9.6 Hydraulic pressure drop. The hydraulic pressure drop shall be corrected for any height (static head) difference between measuring points.

9.7 Air side pressure drop. The air mass flow rate shall be converted to the air volume flow rate (V_r) at reference conditions by the equation:

$V_r = m_a/\rho_r = m_a/1.2$(m^3/s)

and the pressure drop converted by the equation:

$\Delta p_{cr} = \rho_a \Delta p_c / \rho_r = \rho_a \Delta p_c / 1.2$ (Pa)

where

V_r is the air volume flow rate corrected to reference air density (m^3/s)

m_a is the air mass flow rate (kg/s)

ρ_r is the reference air density (1.2 kg/m^3)

Δp_{cr} is the coil air side pressure drop corrected to reference density (Pa)

p_a is the air density (kg/m^3)

Δp_c is the coil air side pressure drop (Pa)

10 Requirements for rating tests

10.1 To establish ratings for similar coils (see **3.5**), having four or more rows, the test coil shall have a minimum of four rows.

If the ratings are required for 1, 2 or 3 row coils, then separate tests shall be carried out on similar coils (see **3.5**) with the appropriate number of rows.

All test coils shall have a minimum face area of 0.5 m^2.

10.2 The tests shall be carried out within the range of variables given in clause **1**. For values of variables outside this range, further tests will be required.

11 Tests required for establishing a rating

The following tests shall be made.

11.1 Heat transfer Both sensible and latent cooling tests shall be made at one water velocity and a minimum of four air flow rates spaced approximately at equal logarithmic intervals throughout the required flow range.

Latent cooling tests shall be on a coil with a fully wet surface.

NOTE A fully wet coil condition can only be confirmed from the analysis of the test data.

11.2 Air side static pressure drop. The air side static pressure drop shall be measured during each heat transfer test.

11.3 Hydraulic pressure drop. The hydraulic pressure drop shall be measured during each heat transfer test. An additional series of measurements shall be carried out on one coil at a minimum of four water velocities covering the manufacturer's recommended range of water velocities. These measurements may be carried out without a thermal load. The water temperatures shall be recorded.

11.4 Air flow leakage. A 4 row coil shall be tested for air leakage. Tests shall be carried out at a minimum of four static pressures, spaced at equal logarithmic intervals, up to 1 500 Pa or the manufacturer's maximum recommended working pressure, whichever is the greater. The leakage flow rate per coil tube shall be plotted against static pressure on logarithmic graph paper. It will be possible to obtain the approximate leakage flow for any coil in the range from this graph.

12 Heat transfer calculations

The heat transferred from a coil surface is related to the fluid and coil properties by the equations:

$$Q_s = \frac{A_o \cdot \Delta t_m \times 10^{-3}}{R_{ad} + R_{md} + R_w}, \text{ for a dry coil surface (kW)}$$

$$Q_T = \frac{A_o \Delta h_m \times 10^{-3}}{c_{pa} R_{aw}}, \text{ for a wet coil surface (kW)}$$

where

Q_s is the mean sensible heat transferred (kW)

Q_T is the mean total heat transferred (kW)

A_o is the coil external surface area ($A_p + A_s$)(m^2)

A_p is the area of exposed tubes (m^2)

A_s is the total surface area of coil fins (both sides) (m^2)

R_{ad} is the dry air film thermal resistance (m^2 K/W)

R_{md} is the dry metal thermal resistance (m^2 K/W)

R_w is the water film thermal resistance (based on A_o) (m^2 K/W)

Δt_m is the corrected logarithmic mean temperature difference between air and water (see **12.3**) (°C)

c_{pa} is the specific heat capacity of moist air [kJ/(kg K)]

R_{aw} is the air film thermal resistance for a moist surface (m^2 K/W)

Δh_m is the logarithmic mean enthalpy difference between the inlet air and the enthalpy of saturated air at the coil surface temperature (see **12.4**) [kJ/kg (dry air)]

In order to employ the two equations, it is necessary to calculate the various thermal resistances. These shall be determined from theoretical and experimental data as described in the following clauses.

12.1 Metal thermal resistance. The dry metal thermal resistance of the coil (R_{md}) may be expressed as the sum of the tube wall (R_t) and fin thermal resistances (R_f). The fin thermal resistance is a function of the air side heat transfer coefficient and will therefore depend upon the state of the coil surface which may be either dry or wet. In order to avoid duplicating equations and calculations, a general heat transfer coefficient f_a is employed

12.1.1 ***Basic equations***

12.1.1.1 Tube wall thermal resistance:

$$R_t = \frac{B(d_o \quad d\ )}{{}_t(1 + d_o/d_i)} \times 10^{-3} (\text{m}^2\ \text{K/W})$$

where

R_t is the tube wall thermal resistance (m^2 K/W)

k_t is the thermal conductivity of the tube wall material [W/(m K)]

d_o is the tube external diameter (mm)

d_i is the tube internal diameter (mm)

B is the coil surface ratio (A_o/A_i)

A_o is the coil external surface area ($A_p + A_s$) (m^2)

A_i is the coil internal surface area ($N_t \pi d_i l_t \times 10^{-3}$) (m^2)

A_p is the area of exposed tubes (m^2)

A_s is the total surface area of coil fins (both sides) (m^2)

N_t is the number of tubes in coil

l_t is the length of coil water tube (m)

12.1.1.2 Fin thermal resistance:

$$R_f = \frac{(1-\eta)}{\eta f_a} (\text{m}^2\ \text{K/W})$$

where

f_a is the air side heat transfer coefficient [W/(m²K)]

R_f is the fin thermal resistance (m² K/W)

$\eta = \dfrac{\phi A_s + A_p}{A_o}$ is the surface effectiveness

A_s is the total surface area of coil fins (both sides) (m²)

A_p is the area of exposed tubes (m²)

A_o is the coil external surface area ($A_p + A_s$) (m²)

ϕ is the fin efficiency obtained from Figure 14 and Figure 15. The parameters γ and δ in these figures are determined from the following equations:

$$\gamma = X_e / X_b$$

$$\delta = (X_e - X_b)\ \sqrt{(2 f_a \times 10^{-3})/(k_f\ \ Y_f)}$$

k_f is the thermal conductivity of the fin material [W/(m K)]

$X_b = d_o/2$, or $d_o/2 + ½$(fin collar thickness) (mm) when fin collars are used.

for:

a) Continuous plate fins (use Figure 14 for ϕ)

$X_e = (\text{length of fin} \times \text{depth of fin} \times {}^1/\pi \times {}^1/N_t)^{0.5}$ (mm)

N_t is the number of tubes

Y_f is the fin thickness (mm)

b) Individually finned tubes (use Figure 15 for ϕ)
with rectangular fins:

$X_e = (\text{length of fin} \times \text{depth of fin} \times {}^1/\pi)^{0.5}$ (mm)

Y_f is the fin thickness (mm)

with circular fins of constant thickness:

X_e is the fin outside radius (mm)

Y_f is the fin thickness (mm)

c) Spiral fins or fins of constant cross-sectional area (use Figure 15 for ϕ)

X_e is the fin tip radius (mm)

Y_f is the fin root thickness (mm)

12.1.2 *Rating data*. The calculations indicated by the equations given in **12.1.1** shall be carried out for a minimum of six values of f_a. It will be necessary to assume values covering the expected range of coil performance. R_a (= $1/f_a$) shall be plotted versus ($R_a + R_m$) as shown in Figure 16. R_m (= $R_f + R_t$) shall be plotted versus f_a as shown in Figure 17.

12.2 Water side thermal resistance

For smooth bore tubes:

$R_w = B/f_{ws}$ (m^2 K/W)

For tubes with turbulators:

$R_w = B/f_{wt}$ (m^2 K/W)

where

R_w is the water side thermal resistance (m^2 K/W)

B is the coil surface area ratio

f_{ws} is the water film heat transfer coefficient for smooth bore tubes [W/(m^2 K)]

f_{wt} is the water film heat transfer coefficient for tubes with turbulators [W/(m^2 K)]

f_{ws} and f_{wt} are calculated for smooth bore tubes and for tubes with turbulators as indicated in **12.2.1** and **12.2.2**.

12.2.1 ***Smooth bore tubes***

$$f_{ws} = \frac{5600(1 + 0.015t_{wm})v_w^{0.8}}{d_i^{0.2}} \text{ (W/(m}^2\text{ K))}$$

where

t_{wm} is the mean water temperature $(t_{wi} + t_{wo})/2$ (°C)

v_w is the mean water velocity $(m_w/\rho_w N_c A_T)$ (m/s)

d_i is the tube internal diameter (mm)

m_w is the water mass flow rate (kg/s)

ρ_w is the water density (kg/m^3)

N_w is the number of circuits

A_T is the cross-sectional area of coil tube (m^2)

12.2.2 *Tubes with turbulators*. The following tests shall be carried out to determine the water side heat transfer coefficient for tubes with internal fins or turbulators, since this cannot be determined by an empirical formula.

12.2.2.1 Sensible cooling tests are carried out on two similar coils each having a minimum of 4 rows and each having identical heat transfer surfaces, but one coil having smooth bore tubes and the other having turbulators.

12.2.2.2 The tests shall be carried out at a minimum of four water flow rates, spaced approximately at equal logarithmic intervals throughout the recommended water flow range. Water flow rates and inlet temperatures for the tests on the coil fitted with turbulators shall be to within 2 % of those employed for the tests on the coil with smooth bore tubes. The tests shall be carried out on one air flow rate which shall not vary by more than 2 % from test to test.

12.2.2.3 The heat transferred in each test shall be calculated as described in **9.1**. The following additional calculations shall be carried out to determine f_{wt}.

a) Compute the overall thermal resistance, for each coil, and water flow rate from:

$$R = \frac{A_o \Delta t_m 10^{-3}}{Q_s}$$ (see also **12.3**)

b) Subtract the overall thermal resistance for the coil with turbulators from that for the coil with smooth bore tubes. This difference is equal to the change in thermal resistance due to the turbulators (ΔR).

c) Calculate the value of f_{ws} from the equation given in **12.2.1**.

d) f_{wt} is then given by the equation:

$$f_{wt} = (1/f_{ws} - \Delta R/B)^{-1}$$

12.2.2.4 $f_{wt}/(1 + 0.015\ t_{wm})$ shall then be plotted against v_w on logarithmic graph paper, and the best line drawn through the test points, as shown in Figure 18. This curve shall be used to determine f_{wt} in the analysis of the coil performance data.

12.3 Dry coil air side thermal resistance

$$R_{ad} = \frac{A_o \Delta t_m \times 10^{-3}}{Q_s} - R_{md} - R_w$$

where Δt_m is the corrected logarithmic mean temperature difference:

$$\Delta t_m = F \frac{(t_{aic} - t_{wo}) - (t_{aoc} - t_{wi})}{\log_e(t_{aic} - t_{wo}) - \log_e(t_{aoc} - t_{wi})}$$

F is obtained from Figure 19 for a one pass coil and is equal to unity for all other coils.

t_{aic} is the inlet air temperature (dry bulb) corrected for heat losses (°C)

t_{aoc} is the outlet air temperature (dry bulb) corrected for heat losses (°C)

t_{wi} is the inlet water temperature (°C)

t_{wo} is the outlet water temperature (°C)

The recommended calculation procedure is to first calculate

$R_w(= B/f_w)$ and $(A_o\Delta t_m \times 10^{-3})/Q_s$.

Subtracting R_w from $(A_o\Delta t_m \times 10^{-3})/Q_s$

will yield $R_{ad} + R_{md}$. R_{ad} can then be obtained from Figure 16 with $R_m = R_{md}$. This procedure is demonstrated by the example given in Appendix A. The above calculation shall be repeated for each air velocity and R_{ad} plotted against v_r (the coil face velocity at reference conditions) on logarithmic graph paper as shown in Figure 20.

v_r is calculated from the equation:

$$v_r = \frac{m_a}{\rho_r A_F} \text{ (m/s)}$$

where

m_a is the air mass flow rate (kg/s)

ρ_r is the reference air density (kg/m³)

A_F is the coil face area (m²)

12.4 Wet coil air side thermal resistance

$R_{aw} = (A_o\Delta h_m \times 10^{-3})/(c_{pa}Q_T)$

where Δh_m is the logarithmic mean enthalpy difference

$$\Delta h_m = \frac{(h_{aic} - h_{si}) - (h_{aoc} - h_{so})}{\log_e(h_{aic} - h_{si}) - \log_e(h_{aoc} - h_{so})}$$

h_{aic} is the corrected enthalpy of inlet air [kJ/kg (dry air)]

h_{aoc} is the corrected enthalpy of outlet air [kJ/kg (dry air)]

h_{si} is the enthalpy of saturated air at the temperature of the coil surface at air inlet [kJ/kg (dry air)]

h_{so} is the enthalpy of saturated air at the temperature of the coil surface at air outlet [kJ/kg (dry air)]

Calculate the coil surface temperature as follows (see Appendix A for example).

a) Calculate $R_w = B/f_{ws}$ or $R_w = B/f_{wt}$.

b) Obtain R_{ad} from plot of R_{ad} versus v_r (see Figure 20).

c) Calculate $f_{ad} = 1/R_{ad}$ and hence obtain R_{md} from plot of R_m versus f_a (see Figure 17).

d) Calculate thermal resistance ratio $C = (R_{md} + R_w)/(c_{pa}R_{ad})$.

e) Calculate mean surface temperature (t_{sm}) from the equation:

$t_{sm} = C(h_{am} + h_{sm}) + t_{wm}$

NOTE This equation can be solved by trial and error, or by means of a nomogram. Such a nomogram is shown in Figure 21 for a barometric pressure of 1.013 bar[3]. Nomograms for use at other barometric pressures may be drawn with the use of tables. The method of drawing the nomogram is shown in Figure 22. The method of using the nomogram is shown in Figure 23.

f) Obtain the ratio Z from Figure 24 at t_{sm} and the barometric pressure.

g) It is now necessary to assume a value for R_{aw}. R_{ad} is recommended for the first attempt.

h) The wet surface heat transfer coefficient (f_{aw}) is equal to Z/R_{aw}, and the wet metal thermal resistance (R_{mw}) is then equal to R_m at $f_a = f_{aw}$ (Figure 17).

j) Calculate the thermal resistance ratio for wet surfaces from the equation.

$C = (R_{mw} + R_w)/(c_{pa}R_{aw})$

and hence obtain the coil surface temperatures, and corresponding saturated air enthalpies at air inlet and outlet by either direct solution of the equations

$C = (t_{s1} - t_{wo})/(h_{aic} - h_{s1})$ $C = (t_{so} - t_{wi})/(h_{aoc} - h_{so})$

or from a nomogram (see Figure 21).

k) Calculate inlet air dew point, which shall not be below t_{si}, since, if below, the coil is not fully wet and the test is void.

l) Compute the logarithmic mean enthalpy difference (Δh_m) and hence the coil surface area (A_{oc}).

$A_{oc} = (c_{pa}R_{aw}Q_T \times 10^3)/\Delta h_m (m)$

If A_{oc} is not equal to A_o, then steps g) to l) shall be repeated for a minimum of two other values of R_{aw}. The values of R_{aw} shall be chosen such that the values of A_{oc} fall either side of A_o. A_{oc} shall be plotted against R_{aw} (on linear graph paper). R_{aw} for the test condition shall be obtained from this graph at point $A_{oc} = A_o$.

12.4.3 The calculations in **12.4.2** shall be repeated at each air flow rate employed in the tests and R_{aw} plotted against v_r (see **12.3.2**) on logarithmic graph paper, as shown in Figure 25.

13 Pressure drop calculations

13.1 Air side pressure drop, corrected to reference density

a) Calculate the reference air velocity (v_r) for all test readings from:

$r = m_a/\rho_r A_F = m_a/1\ 2A_F$

where

m_a is he air mass flow rate (kg/s)

ρ_r is the reference density (kg/m^3)

A_F is the coil face area (m^2)

b) Plot Δp_r drop against v_r on logarithmic graph paper. This graph shall be used to obtain the static pressure drop for all coils of the same number of rows as the test coil.

c) If data is required for coils with a greater number of rows than those in the test range, this data shall be obtained from the test coil with the greatest number of rows by means of the equation:

$\Delta p_{sn} = \Delta p_r N_n/N_m$, for each air velocity (Pa)

[3] 1 bar = 10^5 N/m^2 = 100 kPa.

where

Δp_{sn} is the static pressure drop for n row coil (Pa)

Δp_{r} is the air side pressure drop corrected to reference density obtained from the curve drawn under b) above (Pa)

N_{n} is the number of rows in n row coil

N_{m} is the number of rows in the test coil

13.2 Hydraulic pressure drop. The hydraulic pressure drop coefficient (pressure drop divided by dynamic head) is a function of the water tube Reynolds number. The water density may be assumed constant for the range of water temperatures given in clause **1**. The water viscosity, however, changes very rapidly with temperature in this range and it is therefore necessary to apply a viscosity correction. This correction is applied by converting all pressure, drops to the equivalent pressure drop at a reference water temperature of 10 °C. The following calculations shall be carried out.

a) Correct the measured hydraulic pressure drop for any head difference between the coil inlet and outlet water tubes, by means of the equation.

$$\Delta p_{wc} = \Delta p_{w} - gHp_{w}$$

where

Δp_{wc} is the corrected hydraulic pressure drop (Pa)

Δp_{w} is the measured hydraulic pressure drop (Pa)

H is the height of the inlet water supply tube minus the height of the outlet water supply tube (m)

ρ_{w} is the water density (kg/m^3)

g is the acceleration due to gravity (9.81) (m/s^2)

b) Correct Δp_{wc}, to the hydraulic pressure drop for a mean water temperature of 10 °C by means of the equation:

$$\Delta p_{wct} = K_{w} \Delta p_{wc}$$

where

Δp_{wct} is the hydraulic pressure drop at 10 °C (Pa)

K_{w} is the hydraulic pressure drop correction factor obtained from Figure 26 at the test mean water temperature.

c) Plot the hydraulic pressure drop per unit length of circuit ($\Delta p_{wct}/l_{c}$) against water velocity on logarithmic graph paper. The results for all test coils shall be plotted on one graph and the best line drawn through all test points. This line shall be used for the determination of the hydraulic pressure drop of any coil in the range.

14 Accuracy and calculations

The test results may not be extrapolated beyond the range of air velocities employed in the tests, or to conditions outside those given in clause **1**.

The test data may be applied to all coils in the range, providing that all the tests, described in clause **11**, have been carried out. The overall accuracy of calculations, under conditions other than those used in the rating tests, based on the test data is probably somewhat better than 10 % for all coils in the range and in the region of 5 % for the test coils.

Examples of cooling duty calculations are given in Appendix B. The preferred method of calculation of sensible cooling load from inlet conditions is given in Appendix C.

Typical values of thermal conductivity for some metals and alloys commonly used in the construction of air heating and cooling coils are given in Appendix D.

Appendix A Examples of heat transfer test data calculations

A.1 Dry coil surface

Step	Calculation	Symbol	Values	Units
	Basic data for 2 row coil	A_o A_F B A_tN_c	7.4 0.141 (fitted with turbulators) 29 0.00076	m^2 m^2 — m^2
	Text data	t_{aic} t_{aoc} m_a Q_{as} t_{wi} t_{wo} m_w Q_w Q_s	31.4 19.1 0.28 3.47 4.8 7.9 0.263 3.42 3.45	°C °C kg/s kW °C °C kg/s kW kW
a)	Mean water velocity $= \dfrac{mw}{\rho_w A_T N_c}$	v_w	0.346	m/s
b)	Water film heat transfer coefficient, from Figure 18.	f_w	2 450	$W/(m^2\ K)$
c)	Water film thermal resistance $= B/f_w$	R_w	0.0118	$m^2\ K/W$
d)	Corrected logarithmic mean temperature difference between air and water (for 2 rows, $F = 1$)	Δt_m	18.53	°C
e)	Overall thermal resistance $= (A_o\Delta t_m \times 10^{-3})/Q_s$	R	0.0396	$m^2\ K/W$
f)	Air film thermal resistance for a dry surface plus dry metal thermal resistance, $= R - R_w$	$R_{ad} + R_{md}$	0.0278	$m^2\ K/W$
g)	Air film thermal resistance for a dry surface, from Figure 16	R_{ad}	0.024	$m^2\ K/W$

A.2 Wet coil surfaces

Step	Calculation	Symbol	Values			Units
–	Basic data for 2 row coil	A_o A_F B $A_t N_c$	7.4 0.141 29 0.00076			m^2 m^2 – m^2
–	Test data Sensible heat ratio	t_{aic} t_{aoc} h_{aic} h_{aoc} m_a t_{wi} t_{wo} m_w Q_s Q_t Q_s/Q_t	22.5 14.9 49.9 38.6 0.254 4.6 7.0 0.276 1.95 2.73 0.715			°C °C kJ/kg kJ/kg kg/s °C °C kg/s kW kW —
a)	Water film thermal resistance B/f_w at v_w = 0.35 m/s	R_w	0.0121			m^2 K/W
b)	Air film thermal resistance for dry surface at v_r = 1.5 m/s from Figure 20	R_{ad}	0.0255			m^2 K/W
c)	$1/R_{ad}$ Dry metal thermal resistance from Figure 17 at f_{ad}	f_{ad} R_{md}	39.2 0.004			W/(m^2 K) m^2 K/W)
d)	$(R_{ad} + R_w)/(c_{pa}R_{ad})$	C	0.622			kg K/kJ
e)	Mean coil surface temperature from Figure 21 and as shown in Figure 23	t_{sm}	12.1			°C
		h_{am} t_{wm}	44.25 5.8			kJ/kg °C
f)	From Figure 24 at t_{sm}	Z	2.45			—
g)	Assume air film surface thermal resistance for a moist surface	R_{aw}	0.0255	0.022	0.030	m^2 K/W
h)	Wet surface heat transfer coefficient $= Z/R$ Hence wet metal thermal resistance from Figure 17	f_{aw} R_{mw}	96 0.0038	111 0.0038	82 0.0038	W/(m^2 K) m^2 K/W
j)	$(R_{mw} + R_w)/(c_{pa}R_{aw})$ Thus from Figure 21	C	0.615	0.715	0.525	kg K/kJ
		t_{si} t_{so} h_{si} h_{so}	13.8 10.1 38.8 29.7	14.2 10.4 39.8 30.6	13.4 9.7 37.4 28.8	°C °C kJ/kg kJ/kg
k)	Inlet air dew point (from tables) $t_D > t_{si}$ and coil is wet	t_D	15	15	15	°C
l)	Coil surface area from $A_{oc} = (c_{pa}R_{aw}Q_T \times 10^3)/\Delta h_m$	Δh_m A_{oc}	9.95 7.10	9.10 6.70	11.05 7.49	kJ/kg m^2
m)	Plot A_{oc} versus R_{aw} as shown in Figure 27 Thus R_{aw} at $A_{oc} = A_o = 7.4\ m^2$	R_{aw}	0.0290			m^2 K/W

Appendix B Examples of cooling duty calculations

For the purpose of these examples, it is assumed that the typical graphs Figure 24 to Figure 27 represent the results of tests.

B.1 Dry coil surfaces

B.1.1 *Performance calculation*. It is required to calculate if a 4 row coil with a face area of 0.141 m², with an external surface area (A_o) of 13 m² and surface area ratio (B) of 25 will be suitable for cooling 0.44 kg/s of air from 28.5 °C to 17 °C with an inlet water temperature of 6 °C and a mean water velocity of 0.15 m/s (corresponding to a water mass flow rate of 0.27 kg/s). The coil is fitted with turbulators.

Step	Calculation	Symbol	Value	Units
a)	Mean face air velocity $= m_a/1.2\ A_F$	v_r	2.6	m/s
b)	Air film thermal resistance for a dry surface (from Figure 20)	R_{ad}	0.015	m² K/W
c)	Air film thermal resistance for a dry surface plus dry metal thermal resistance (from Figure 16)	$R_{ad} + R_{md}$	0.0188	m² K/W
d)	Heat taken from air $= m_a c_{pa} (t_{ai} - t_{ao})$	Q_a	5.1	kW
e)	Water outlet temperature $= t_{wi} + Q_a/c_{pw} m_w$	t_{wo}	10.5	°C
f)	Water film thermal resistance $= B/f_w$ (from Figure 18 at t_{wm} = 8.25 °C and v_w = 0.15 m/s)	R_w	0.0164	m² K/W
g)	Coil thermal resistance is $R_{ad} + R_{md} + R_w$	R	0.0352	m² K/W
h)	Corrected logarithmic mean temperature difference	Δt_m	14.2	°C
j)	Calculated heat transferred $= \dfrac{A_o \Delta t_m}{R 10^3}$	Q	5.25	kW

The calculation therefore indicates that the coil will be suitable for the required performance.

B.1.2 *Calculation of performance from inlet conditions*. It is required to calculate the performance of the coil specified in **B.1.1** when operating at the following conditions:

0.5 kg/s of air at 32 °C with a dew point of 8 °C and 0.5 kg/s of water at 10 °C.

Step	Calculation	Symbol	Value	Units
a)	Mean face air velocity $= m_a/1.2\ A_F$	v_r	2.96	m/s
b)	Air film thermal resistance for a dry surface (from Figure 20)	R_{ad}	0.0135	m^2 K/W
c)	Air film thermal resistance for a dry surface plus dry metal thermal resistance (from Figure 16)	$R_{ad} + R_{md}$	0.0173	m^2 K/W
d)	Water film thermal resistance (where the inlet water temperature is estimated as t_{wm} in Figure 18, and at a water velocity of 0.278 m/s) $= B/f_w$	R_w	0.0104	m^2 K/W
e)	Coil thermal resistance $= R_{ad} + R_{md} + R_w$	R	0.0277	m^2 K/W
f)	Air to water side heat capacity ratio (see Appendix C) $= m_a c_{pa}/m_w c_{pw}$	M	0.242	—
g)	Heat transfer exponent (see Appendix C) $= A_o 10^{-3}/c_{pa} m_a R$	C_o	0.925	—
h)	Effectiveness (see Appendix C) $= \dfrac{1 - e^{-C_o(1-M)}}{1 - Me^{-C_o(1-M)}}$ Therefore, heat transferred $= Ec_{pa} m_a (t_{ai} - t_{wi})$ Leaving water temperature Leaving air temperature NOTE If the water temperature difference is greater than 5 °C, it will be necessary to repeat the calculation from step d) using the calculated mean water temperature just found.	E Q t_{wo} t_{ao}	0.572 6.36 13.1 19.4	— kW °C °C

B.2 Coils with both latent and sensible heat loads

Before the heat load calculations can be made it is first necessary to determine the condition of the coil surface, i.e. all wet, partially wet or dry. The inlet air dew point will give an indication of the likely state of the coil surface, but it is necessary to have a more exact criterion to determine a wet/dry boundary. The air enthalpy at the wet/dry boundary (h_B) may be determined from the equation:

$$h_B = (t_D - t_{wo} + y h_{ai} + C h_{sD})/(y + C)$$

where

$y = (t_{wo} - t_{wi})/(h_{ai} - h_{ao})$ (kg K/kJ)

and

t_D is the inlet air dew point (°C)

h_{sD} is the enthalpy of saturated air at the inlet air dew point and absolute pressure (kJ/kg).

The application of this equation is demonstrated in the following examples.

B.2.1 *Performance calculation*. It is required to calculate if a two row coil with a face area of 0.141 m^2, an external surface area of 7.4 m^2 and surface area ratio of 29, fitted with turbulators, will be suitable for cooling 0.25 kg/s of air from 23.0 °C (65 % r.h.) to 15 °C (94 % r.h.), when supplied with 0.27 kg/s of water at 4.0 °C (water velocity = 0.356 m/s).

Step	Calculation	Symbol	Value	Units
a)	Total heat load $= m_a(h_{ai} - h_{ao})$	Q_T	3.02	kW
b)	Sensible heat load $= m_a c_{pa}(t_{ai} - t_{ao})$	Q_s	2.02	kW
c)	Air film thermal resistance for a dry surface at mean face velocity of 1.48 m/s (from Figure 20)	R_{ad}	0.026	m^2 K/W
d)	Air film thermal resistance for a wet surface (from Figure 25)	R_{aw}	0.027	m^2 K/W
e)	Water side heat transfer coefficient from Figure 18 at mean water inlet temperature and mean water velocity	f_w	2 500	W/(m^2 K)
f)	Water film thermal resistance $= B/f_w$	R_w	0.0116	m^2 K/W
g)	Air side heat transfer coefficient $1/R_{ad}$	f_{ad}	38.4	W/(m^2 K)
h)	Normally it would be necessary to carry out steps d) to h) of example **A.2**. In this case this is not necessary as examination of Figure 17 leads to the conclusion that $R_{mw} = 0.0038$ (f_{aw} is always greater than $2f_{ad}$)	R_{mw}	0.0038	W/(m^2 K)
j)	Coil characteristic $= (R_{mw} + R_w)/(c_{pa}R_{aw})$	C	0.57	kg K/kJ
k)	Temperature factor $(t_{wi} - t_{wo})/(h_{ai} - h_{ao}) = m_a/(c_{pw}m_w)$	y	0.222	kg K/kJ
l)	Inlet air dew point	t_D	16.7	°C
m)	Enthalpy of saturated air at inlet air dew point	h_{sD}	47.03	kJ/kg
n)	Water outlet temperature	t_{wo}	6.7	°C
p)	Boundary air enthalpy: $= \dfrac{(t_D - t_{wo} + yh_{ai} + Ch_{sD})}{(y + C)}$	h_B	61.2	kJ/Kg
q)	Compare h_B with h_{ai}, in this case h_B is greater than h_{ai} therefore coil surface is totally wet NOTE If h_B is between the inlet and outlet air enthalpies then the coil surface is only partially wet, and if h_B is below the outlet air enthalpy then the coil surface is dry.			
r)	Coil surface saturated air enthalpies at inlet and outlet from Figure 21 at $C = 0.57$, $h_{ai} = 52.6$, $t_{wo} = 6.7$, $h_{ao} = 40.5$, $t_{wi} = 4$	h_{si} g_{so}	39.5 29.5	kJ/kg kJ/kg
s)	log. mean enthalpy difference	Δh_m	12.07	kJ/kg
t)	Total heat transferred $= (A_o \Delta h_m \times 10^{-3})/(c_{pa}R_{aw})$	Q_T	3.26	kW
u)	Comparing the desired [see step a)] and computed heat transfers it is concluded that the coil is suitable for the required duty			

Step	Calculation	Symbol	Value	Units
v)	*Outlet air dry bulb* 1) Bypass factor $= \exp(-A_o \times 10^{-3}/c_{pa}R_{ad}m_a)$ 2) Enthalpy of saturated air at the mean coil surface temperature $= h_{ai} - \dfrac{Q_c}{m_a(1-\beta)}$ where $Q_c = Q_T$ from (t). 3) Corresponding temperature for saturated air from tables 4) Outlet air dry bulb $= t_{ad} + (t_{ai} - t_{ad})\beta$ 5) Sensible heat load $= m_a c_{pa}(t_{ai} - t_{ao})$ NOTE Although the coil selected in this example is capable of the total cooling load, the sensible heat ratio is marginally different from requirements, therefore the actual coil leaving conditions will be slightly different from the calculated values.	t_{ao} β h_s t_{ad} t_{ao} Q_s	 0.325 33.3 11.8 15.4 1.91	°C – kJ/kg °C °C kW

B.2.2 *Calculation of duty from inlet conditions*. The calculation method given in this example may also be employed for checking the duty of a coil when the surface is partially wet.

It is required to calculate the duty of a 4 row coil, a face area of 0.141 m^2, an external surface area of 15 m^2 and a surface area ratio of 25, fitted with turbulators, when supplied with 0.24 kg/s of air at 28.7 °C with a total heat of 51.87 kJ/kg, and 0.26 kg/s water at 4.9 °C and a mean water velocity of 0.145 m/s.

It is suggested that the calculation take the following form:

Step	Calculation	Symbol	Value	Units
a)	Obtain $R_w R_{ad} R_{aw} R_{mw}$ by the method described in **B.2.1** steps c) to h) inclusive	R_w R_{ad} R_{aw} R_{mw}	0.0209 0.0230 0.0230 0.0038	m^2 K/W m^2 K/W m^2 K/W m^2 K/W
b)	Assume a value for the total heat transferred (if this was a duty check, Q_T would be taken from the specified performance)	Q_T	4	kW
c)	The outlet air enthalpy is $h_{ai} - Q_T/m_a$	h_{ao}	35.2	kJ/kg
d)	Outlet water temperature $(Q_T/m_w c_{pw}) + t_{wi}$	t_{wo}	8.58	°C
e)	Coil characteristic $(R_{mw} + R_w)/c_{pa}R_{aw}$	C	1.06	kg K/kJ
f)	$y = m_a/c_{pw}m_w$	y	0.221	kg K/kJ
g)	Inlet air dew point (from tables)	t_D	12.4	°C
h)	Enthalpy of saturated air at 12.4 °C (from tables)	h_{sD}	35.2	kJ/kg
j)	Boundary air enthalpy $\dfrac{t_D - t_{wo} + yh_{ai} + Ch_{sd}}{(y + C)}$ NOTE h_{ao} is greater than h_B which is greater than h_{ai} and therefore the coil has a partially wet surface.	h_B	41.06	kJ/kg

Step	Calculation	Symbol	Value	Units
k)	*Dry surface* 1) Capacity $m_a(h_{ai} - h_B)$	Q_d	2.59	kW
	2) The air dry bulb at the boundary $t_{ai} - Q_d/_{pa}m_a$	t_B	18.0	°C
	3) Water temperature at the boundary $t_{wo} - y(h_{ai} - h_B)$	t_{wB}	6.19	°C
	4) $R_{ad} + R_{md}$ from Figure 16	$R_{ad} + R_{md}$	0.025	m² K/W
	5) Calculated dry surface area $Q_d(R_{ad} + R_{md} + R_w)10^3/\Delta t_m$	A_{dc}	6.85	m²
l)	*Wet surface* 1) Capacity $Q_T - Q_d$	Q_w	1.41	kW
	2) The calculated wet surface area is $(c_{pa}Q_wR_{aw} \times 10^3)/\Delta h_m$ where $\Delta h_m = \dfrac{(h_B - h_{sD}) - (h_{ao} - h_{so})}{\log_e(h_B - h_{sD}) - \log_e(h_{ao} - h_{so})}$ where h_{so} is obtained from Figure 21 at t_{wi} and h_{ao}	A_{wc}	6.065	m²
m)	The total coil area is $A_{dc} + A_{wc}$ This is less than $A_o = 15$, the actual area, *therefore the process is to be repeated from step b)*	A_{oc}	12.915	m²
n)	Assuming $Q_T = 4.5$ kW	A_{oc}	15.8	m²
	$Q_T = 4.3$ kW	A_{oe}	14.65	m²
p)	Plot Q_T versus A_{oc}; then at $A_{oc} = A_o = 15$ m²	Q_T	4.36	kW
q)	The sensible heat capacity may be calculated using a method similar to that described in **B.2.1**.			

Appendix C Calculation of sensible cooling load from inlet conditions

The heat transferred to a dry coil surface may be calculated from the coil air and water inlet conditions by trial and error using the logarithmic mean temperature difference or directly from the air-side effectiveness.

The latter method is to be preferred and the calculations should be carried out as described below:

The heat transferred $(Q_s) = c_{pa}E\, m_a(t_{ai} - t_{wi})$

where

$$E = \frac{1 - \exp[-M(1 - \exp C_o)]}{M}$$, for a one water pass coil

and

$$E = \frac{1 - \exp[-C_o(1 - M)]}{1 - M\exp[-C_o(1 - M)]}$$, for a two or more water pass coil

where

$$M = \frac{m_a c_{pa}}{m_w c_{pw}}$$

$$C_o = \frac{A_o \times 10^{-3}}{c_{pa} m_a R}$$

$$R = R_{ad} + R_{wt} + R_{md}$$

Appendix D Typical values of thermal conductivity

Material	ISO designation	Nearest BS designation	Thermal conductivity W/(mk)		
			At 20 °C	At 100 °C	At other temperatures
Aluminium	Al 99.5 Al 99.0 Al Mnl	1B 1C N3	— — —	— — —	227–230 (0 °C) 227 (0 °C) 151–176 (0 °C)
Copper Electrolytic tough-pitch	Cu-ETP	C.101	393	386	—
Fire refined tough-pitch high conductivity	Cu-FRHC	C.102	393	386	—
Phosphorous-de-oxidized copper (high residual phosphorous)	Cu-DHP	C.106	292–365	—	—
Brass 70/30 brass, (deep drawing brass)	Cu Zn 30	CZ 105 and CZ 106	121	—	147 (200 °C)
Aluminium brass	Cu Zn 20 Al2	CZ 110	100	—	126 (200 °C)
Admiralty brass (inhibited brass)	Cu Zn 28 Snl	CZ 111	109	—	138 (200 °C)
Cupro-nickel 90/10 Cupro-nickel	Cu Ni 10 FelMn	CN 101	—	—	45 (temperature unspecified)
70/30 Cupro-nickel	Cu Ni 30	CN 107	—	—	29 (temperature unspecified)
Stainless steel		302 304 316 321 347	— — — — —	15.9 14.2 15.9 15.9 15.9	— — — — —
Carbon steel	Ts 2 Ts 3 Ts 9H	320 360 410	56.6 56.6 49.5	55.4 55.4 48.4	— — —

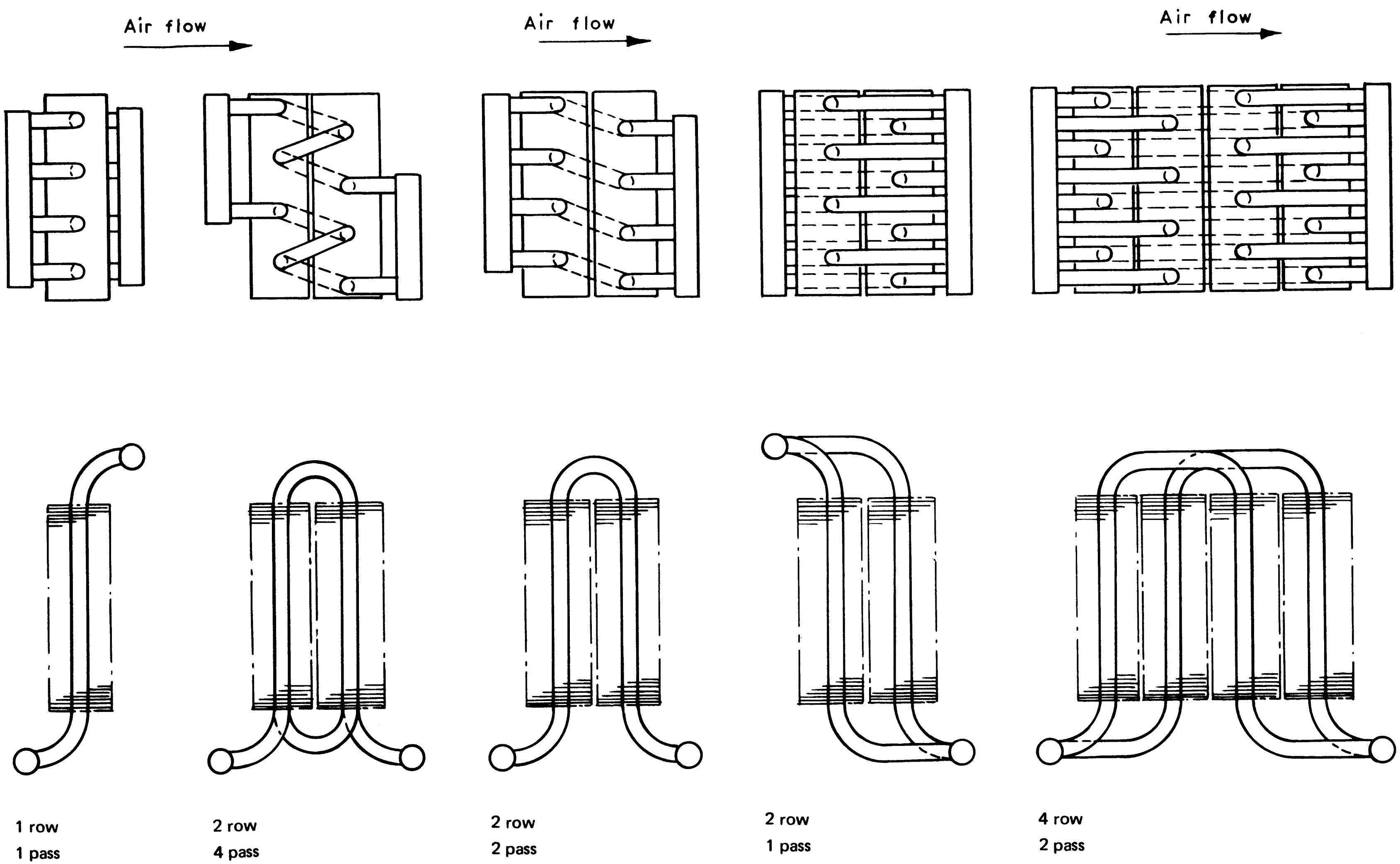


Figure 1 — Typical arrangements of passes

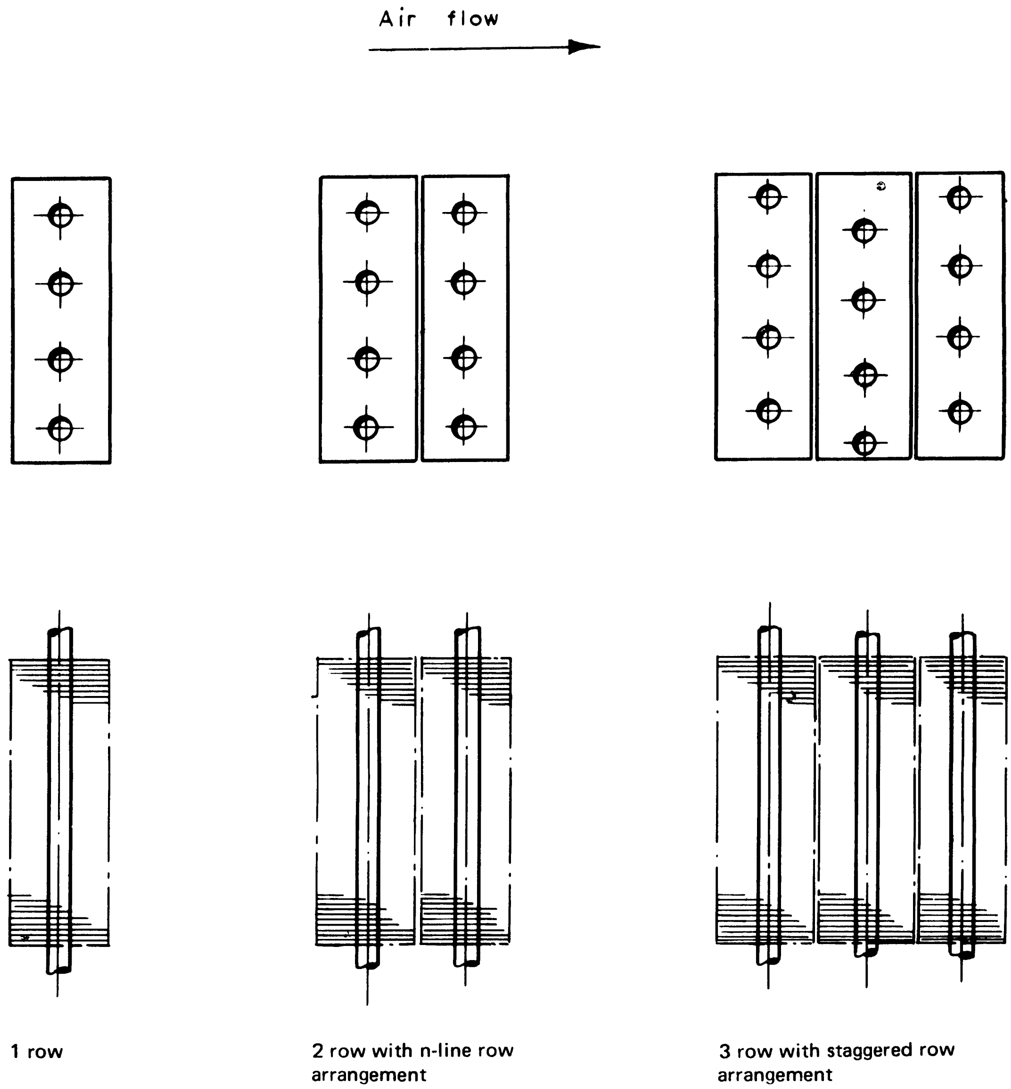


Figure 2 — Typical arrangements of rows

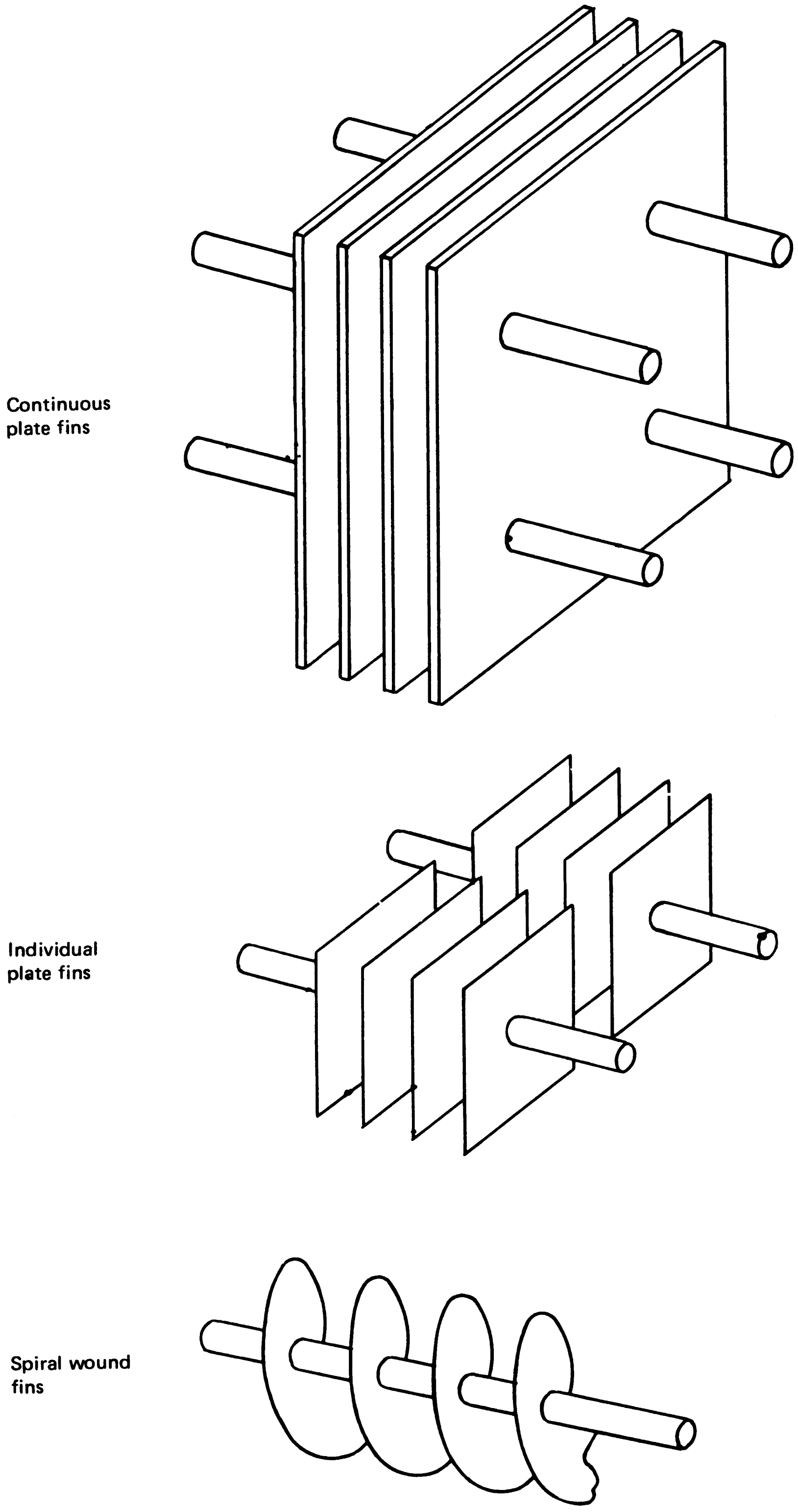
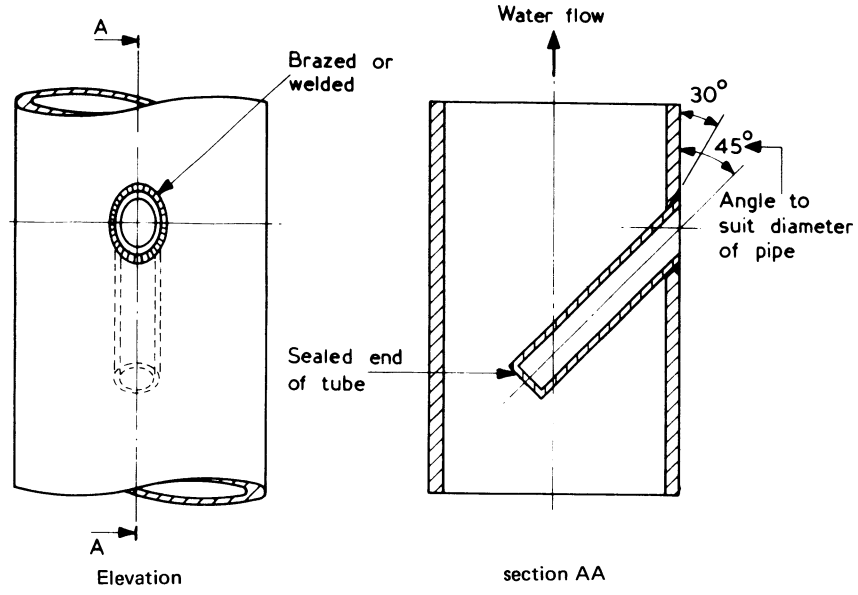
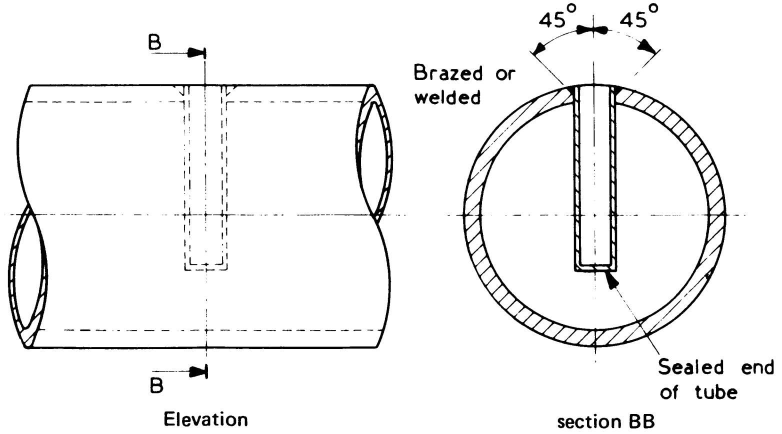


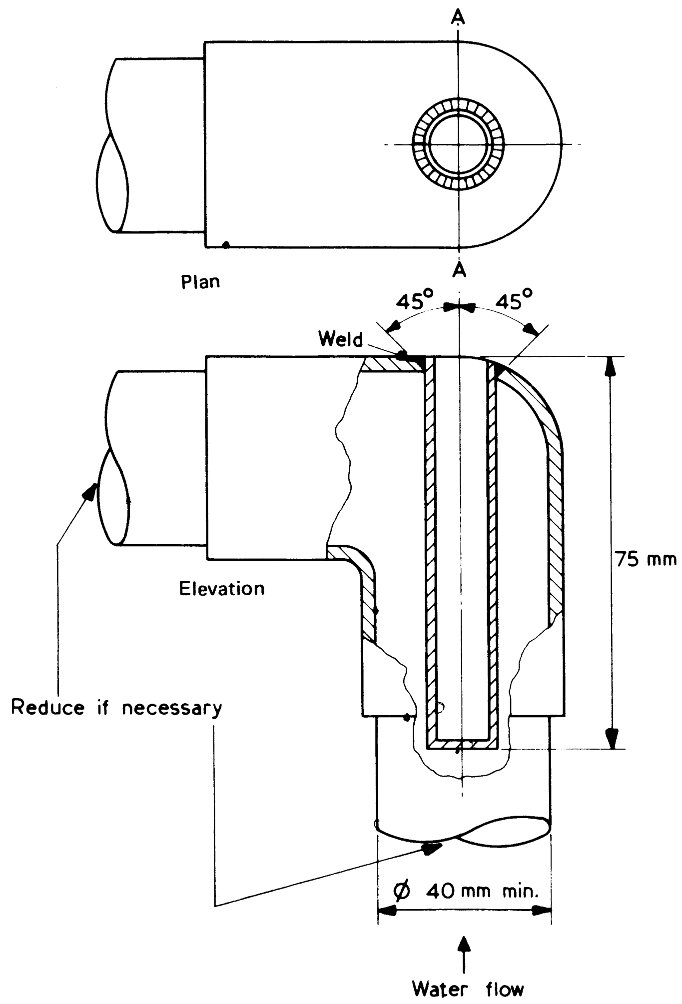
Figure 3 — Types of fins



(a) For use in vertical pipes over 75 mm diameter



(b) For use in horizontal pipes over 75 mm diameter



(c) Pipe less than 75 mm diameter

NOTE For (a) and (b) the tube should have a minimum length of 100 mm.

Figure 4 — Thermometer pockets

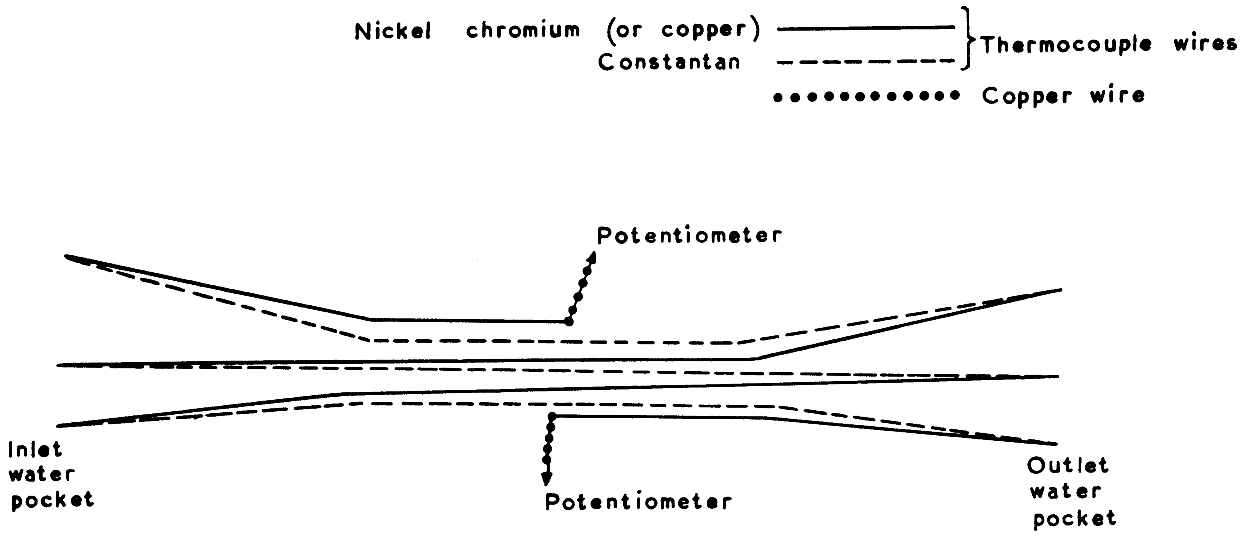


Figure 5 — Multi-junction differential thermocouple

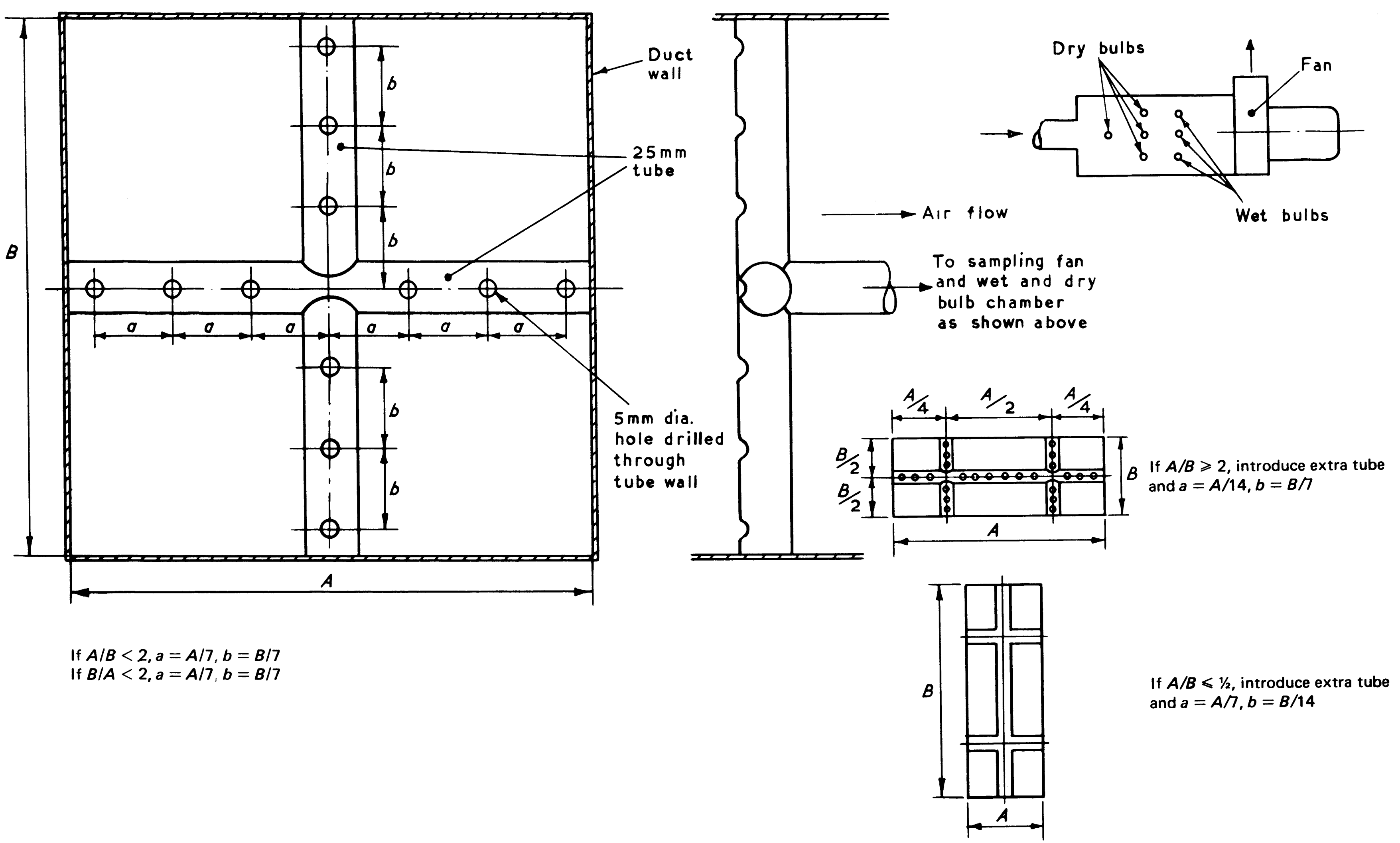


Figure 6 — Air sampling tubes

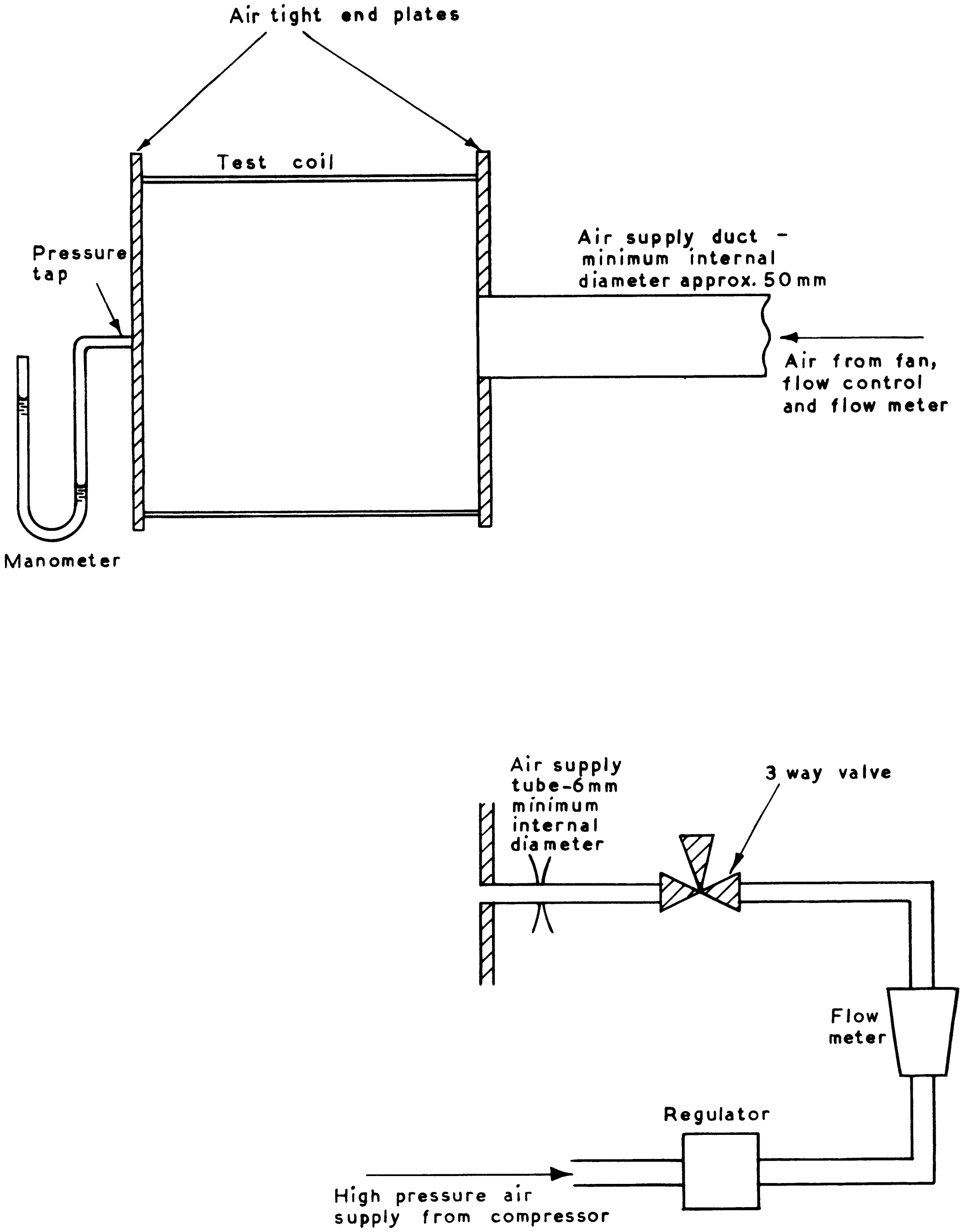


Figure 7 — Apparatus for air flow leakage test (showing alternative air supply systems)

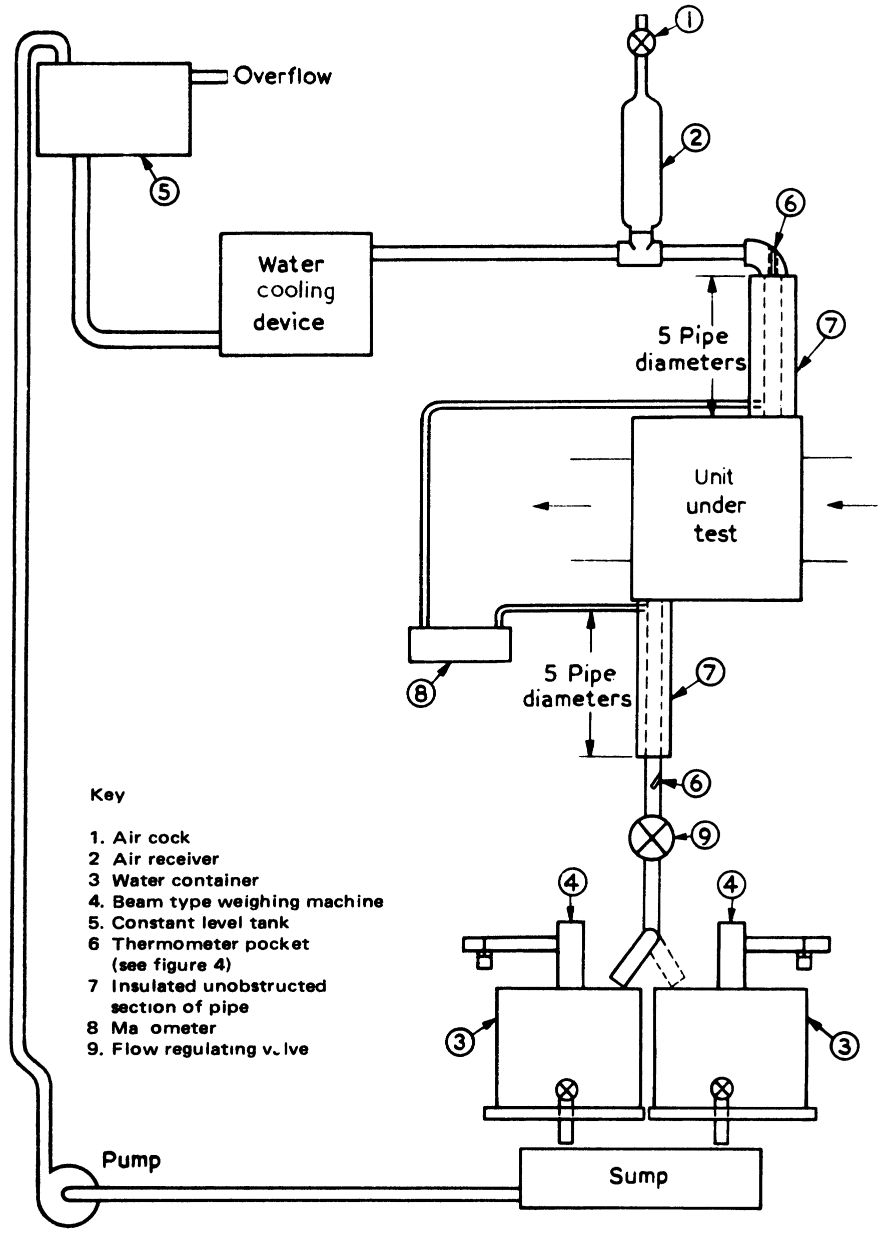


Figure 8 — Water supply system

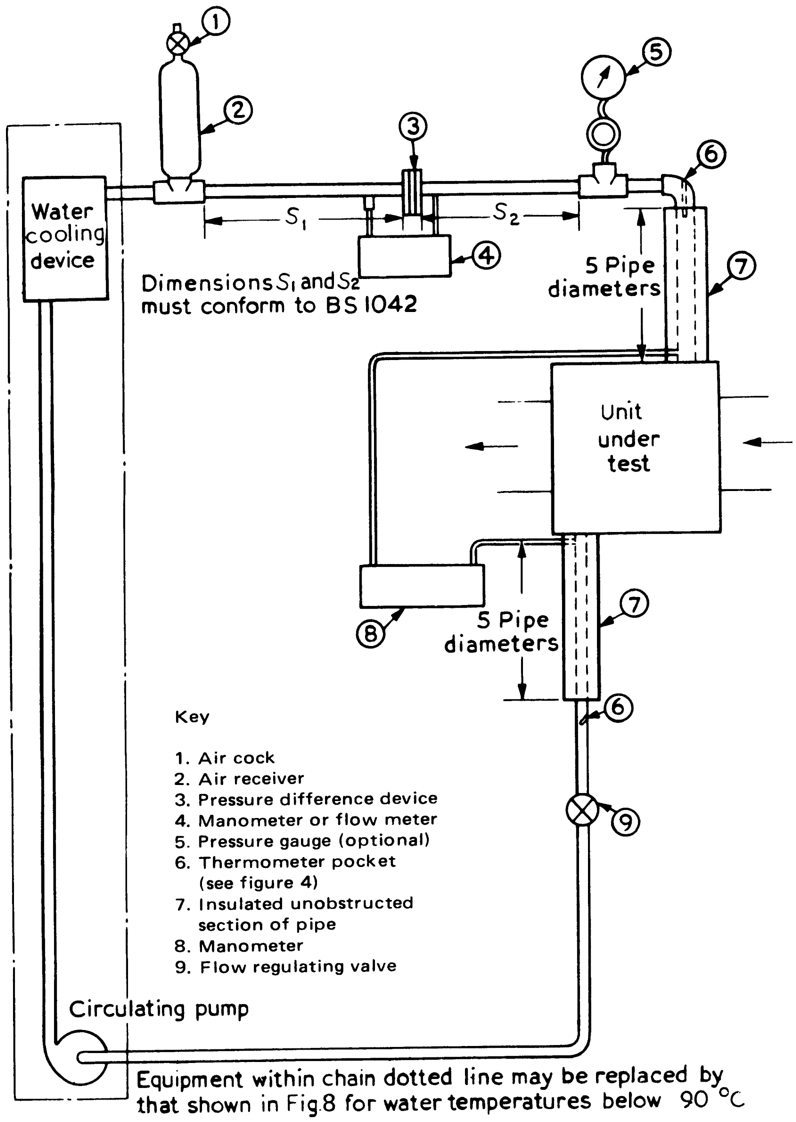
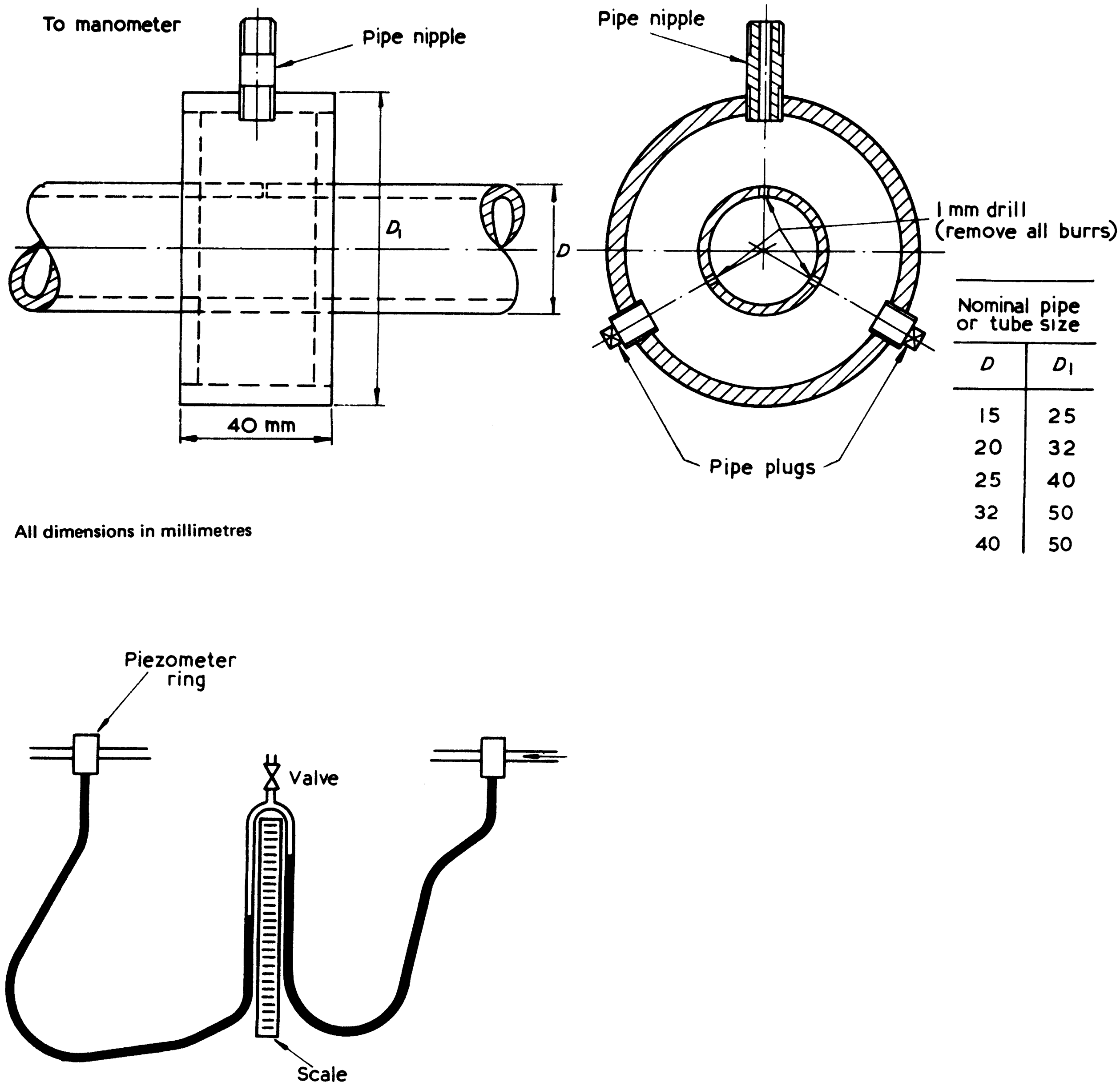


Figure 9 — Alternative water supply system



Nominal pipe or tube size	
D	D_1
15	25
20	32
25	40
32	50
40	50

Figure 10 — Inverted U-tube pattern manometer and piezometer ring

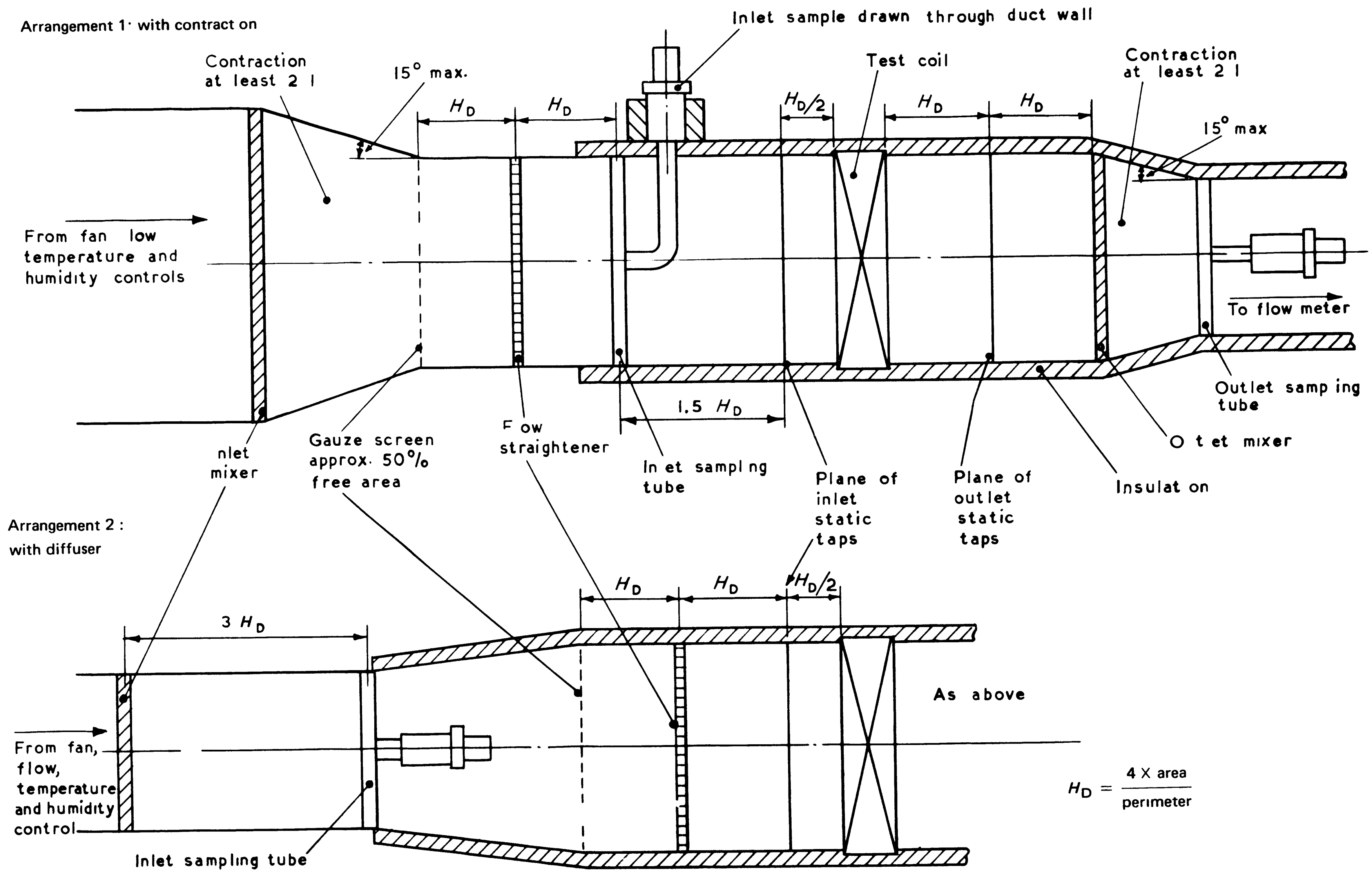


Figure 11 — Typical test duct arrangements

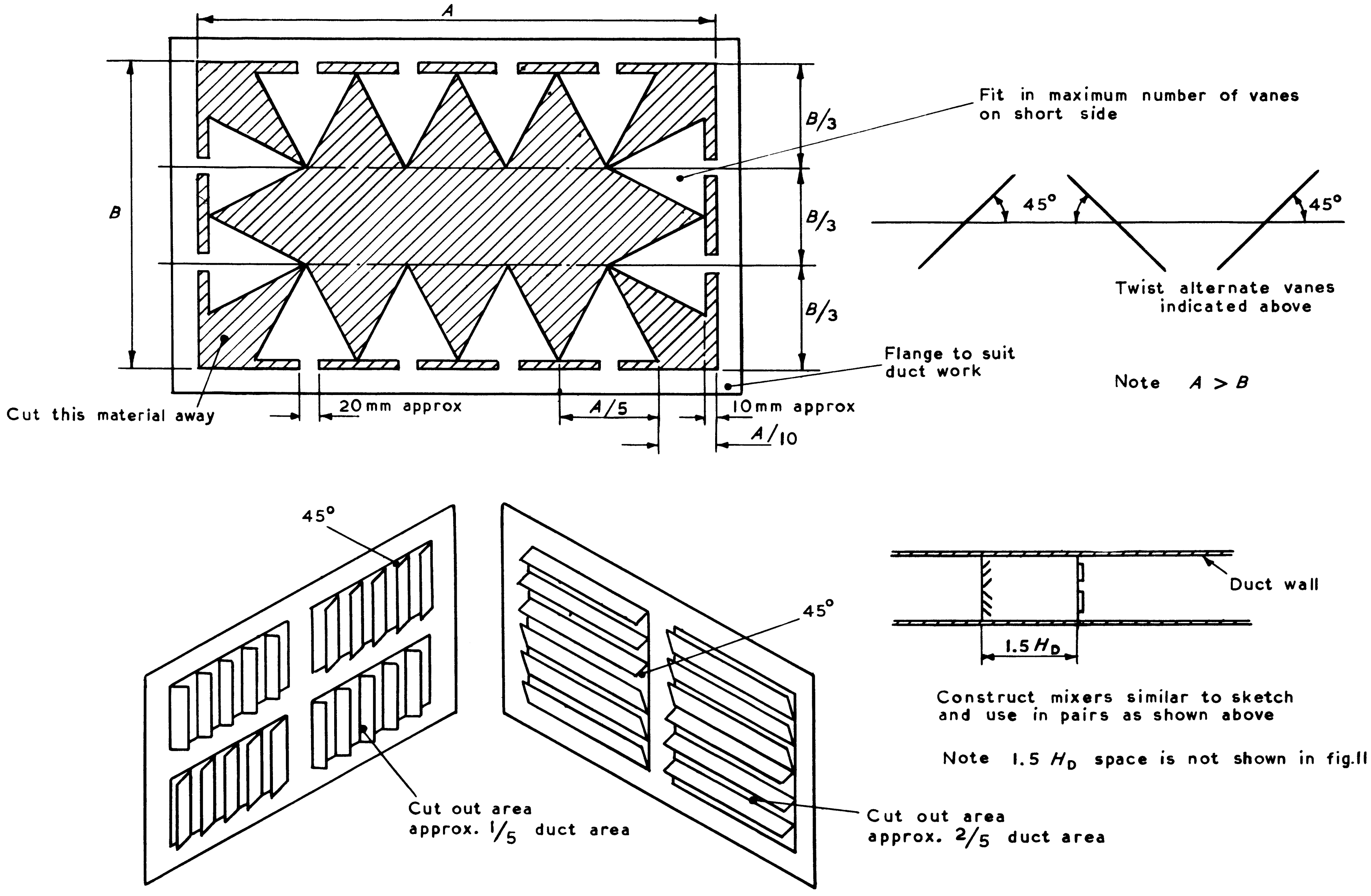


Figure 12 — Air flow mixers

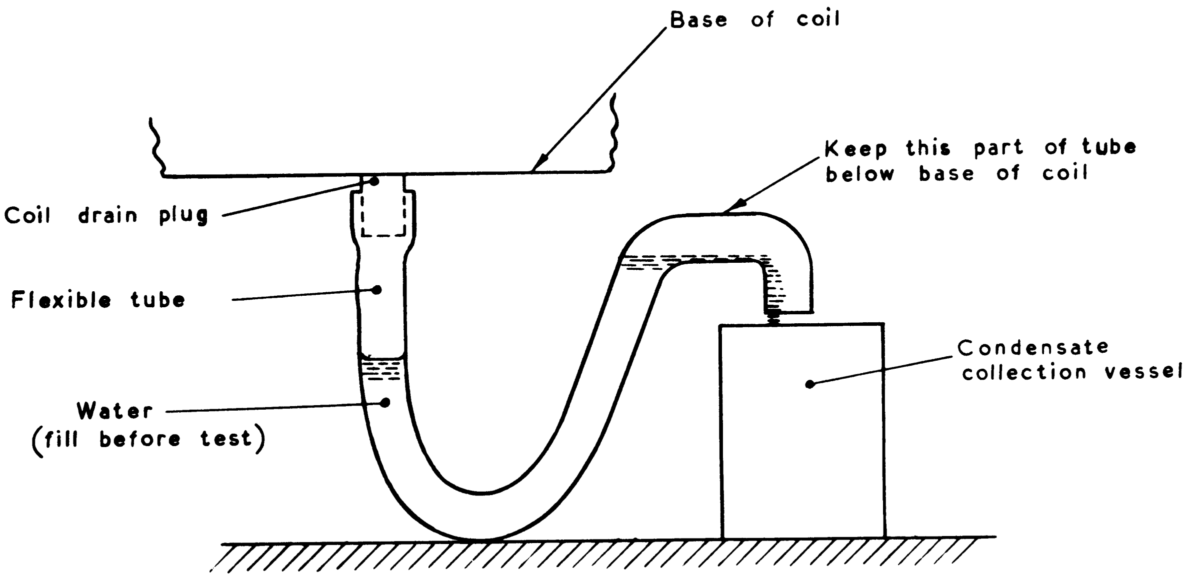


Figure 13 — Condensate drain

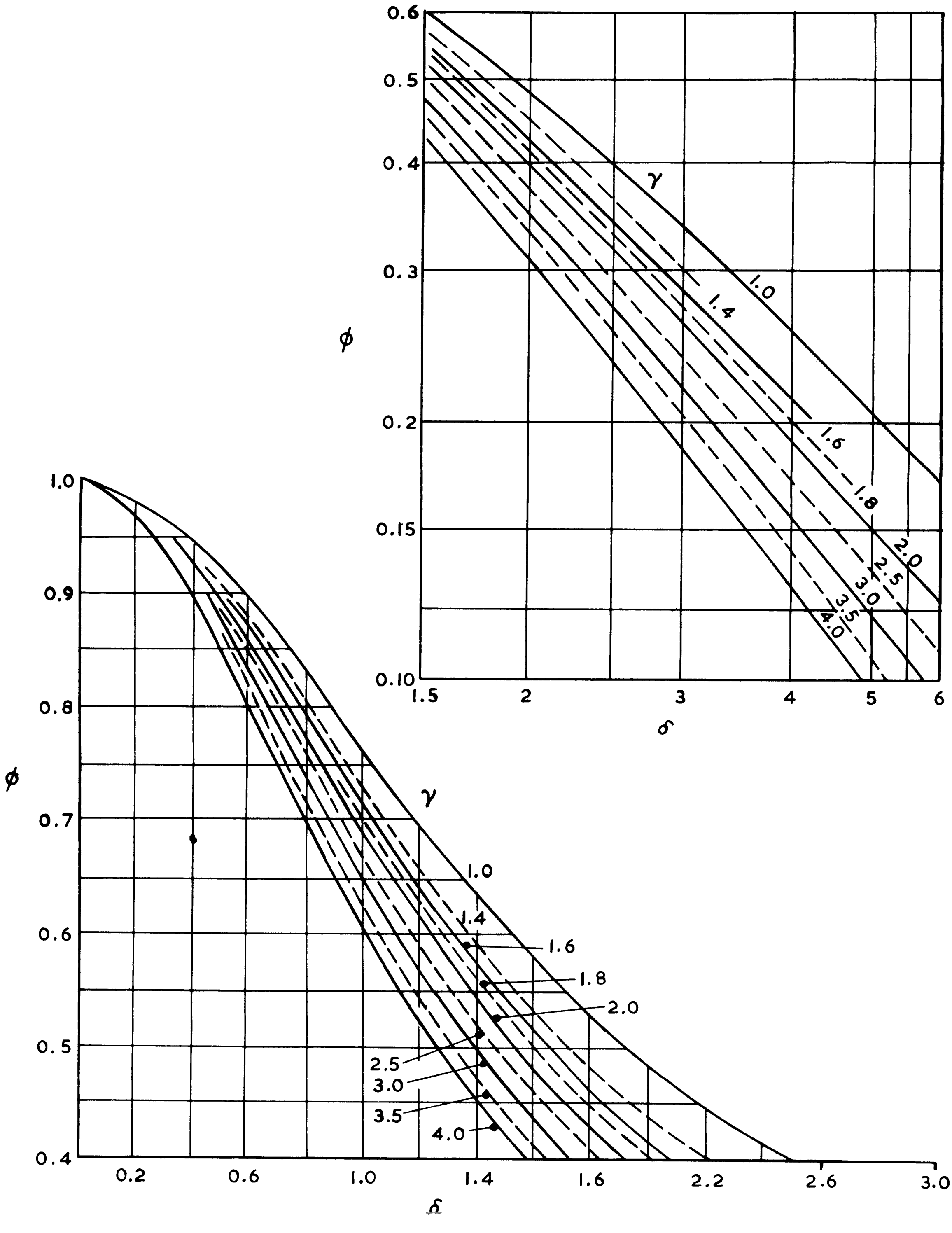


Figure 14 — Efficiency of fins of constant thickness

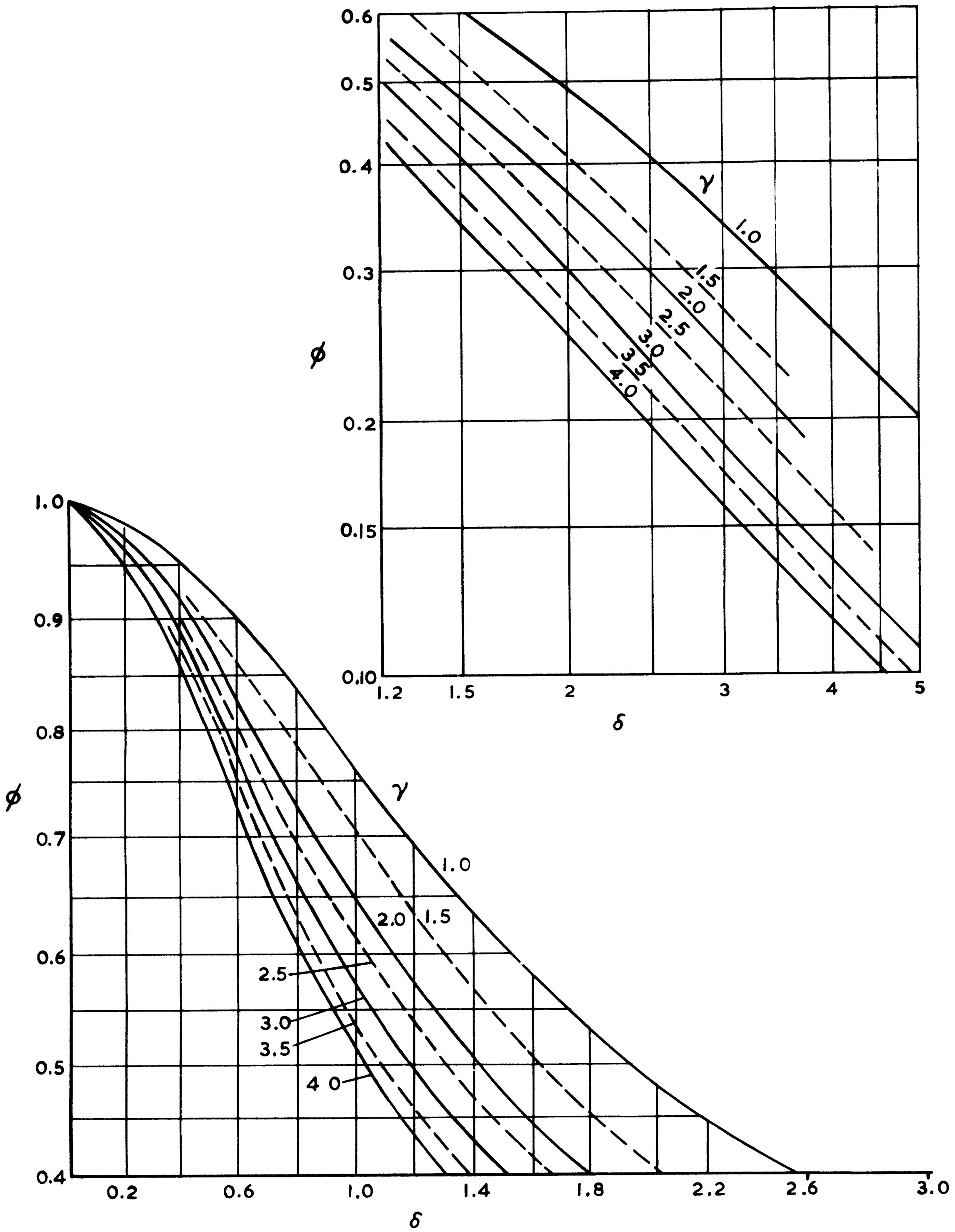


Figure 15 — Efficiency of fins of constant area for heat flow and of spiral fins

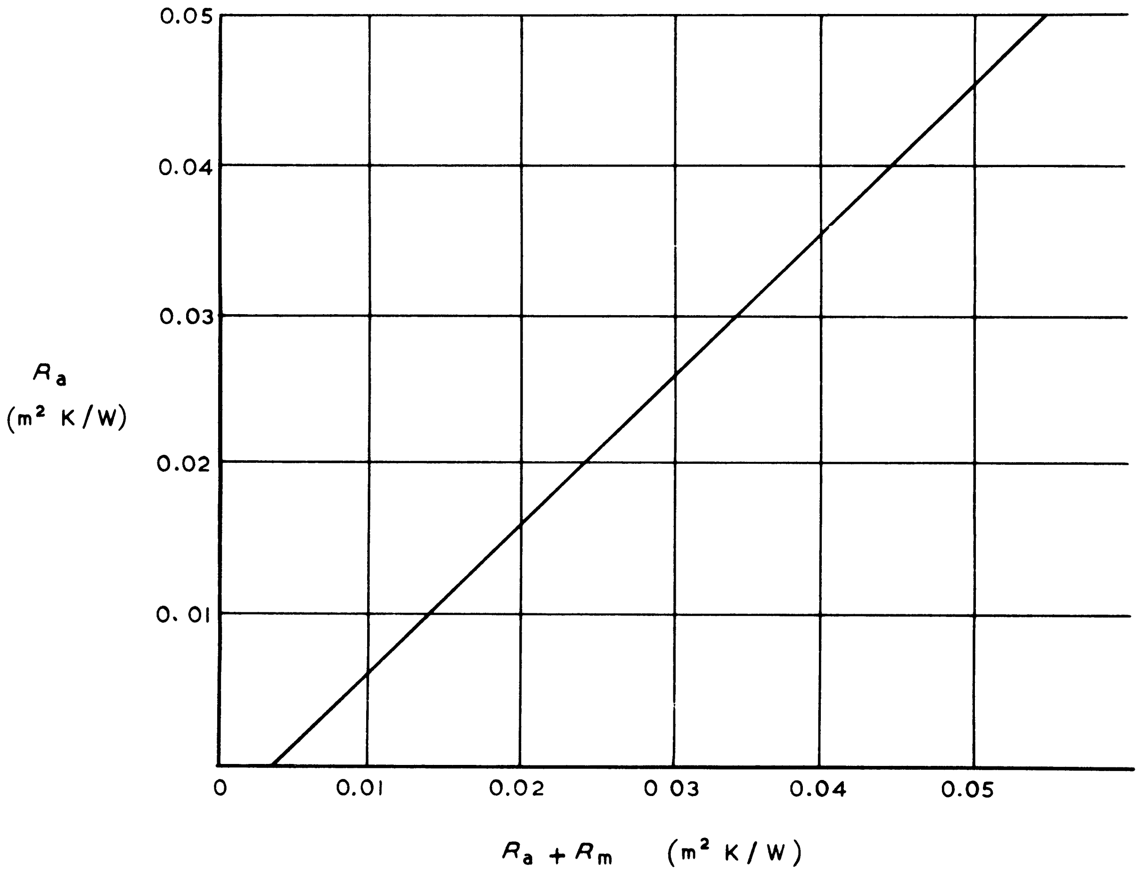


Figure 16 — Graph of R_a versus ($R_a + R_m$)

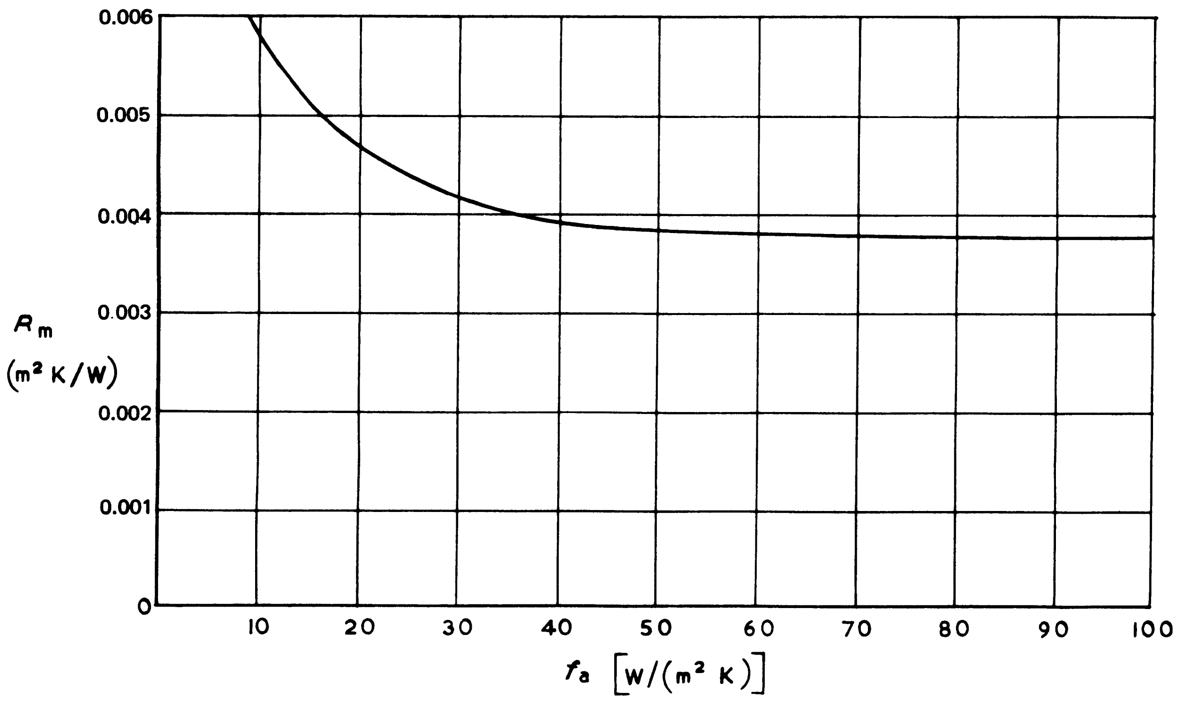


Figure 17 — Graph of R_m versus f_a

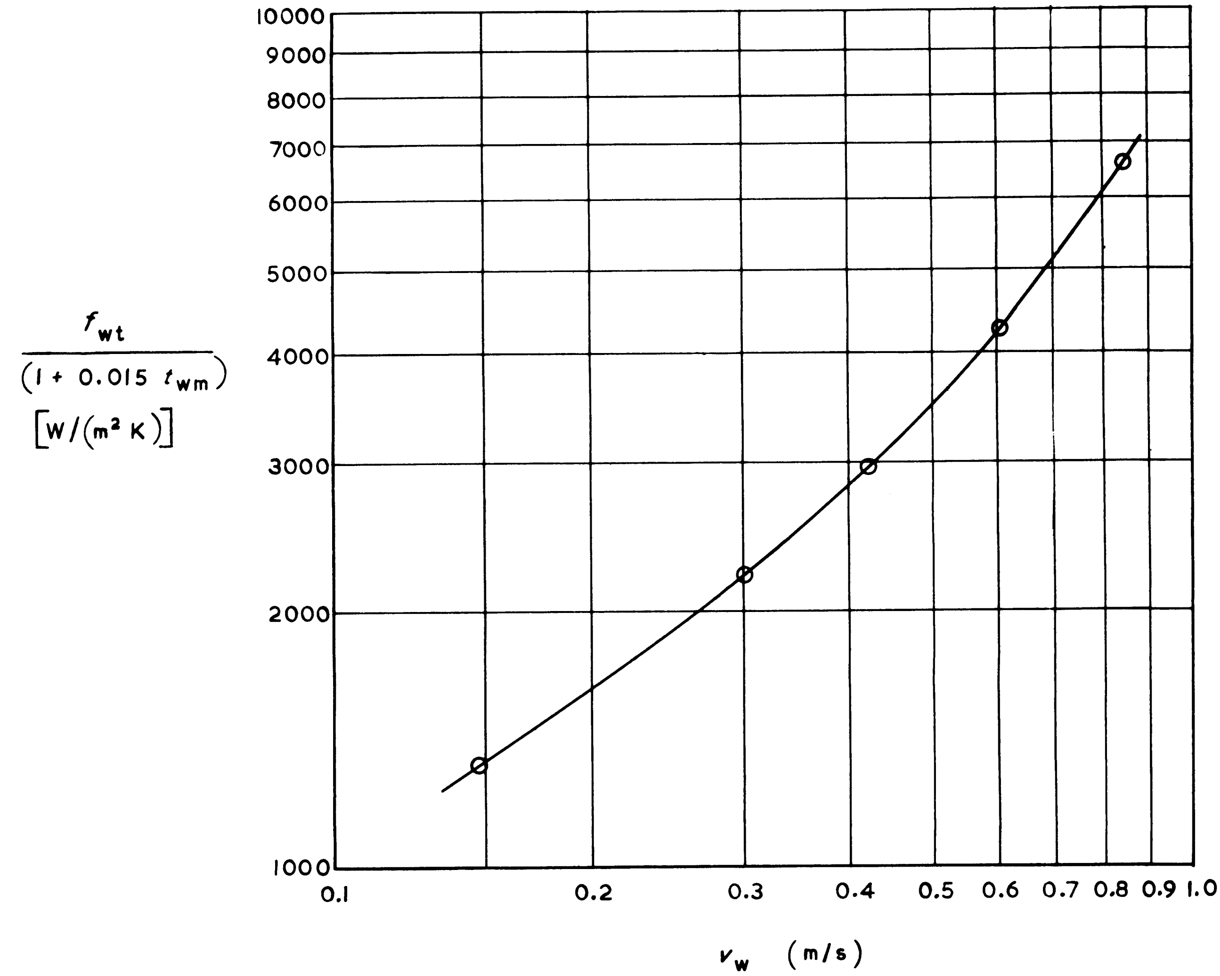


Figure 18 — Water film heat transfer coefficient for tube with turbulators

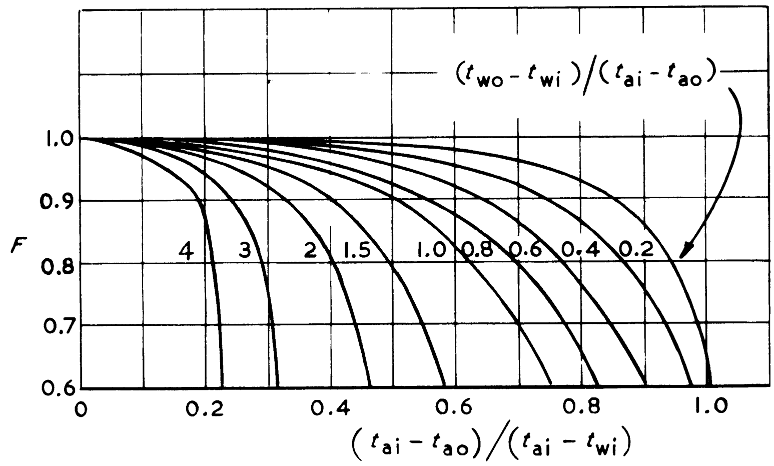


Figure 19 — Graph of *F*, factor for corrected logarithmic mean temperature

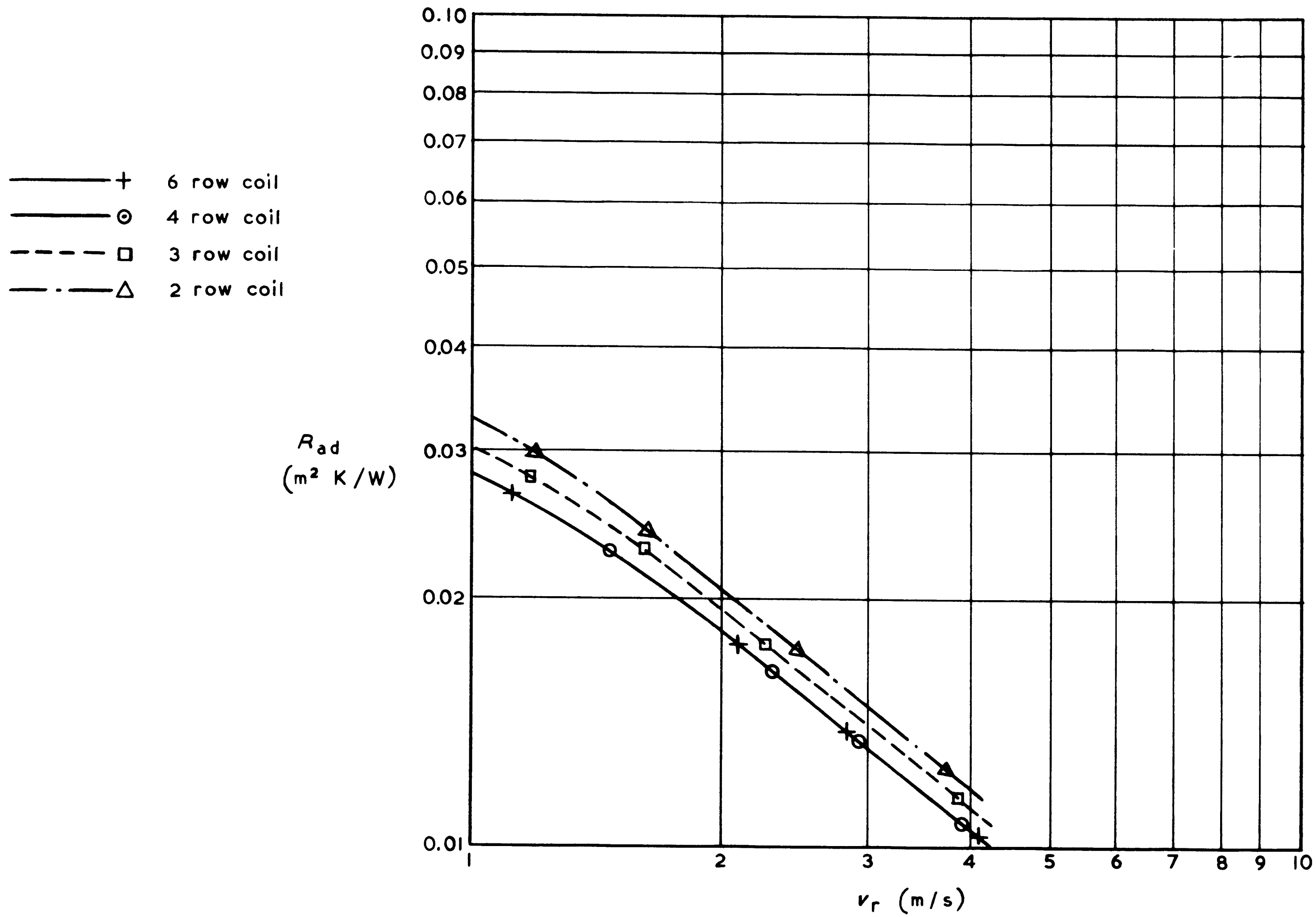
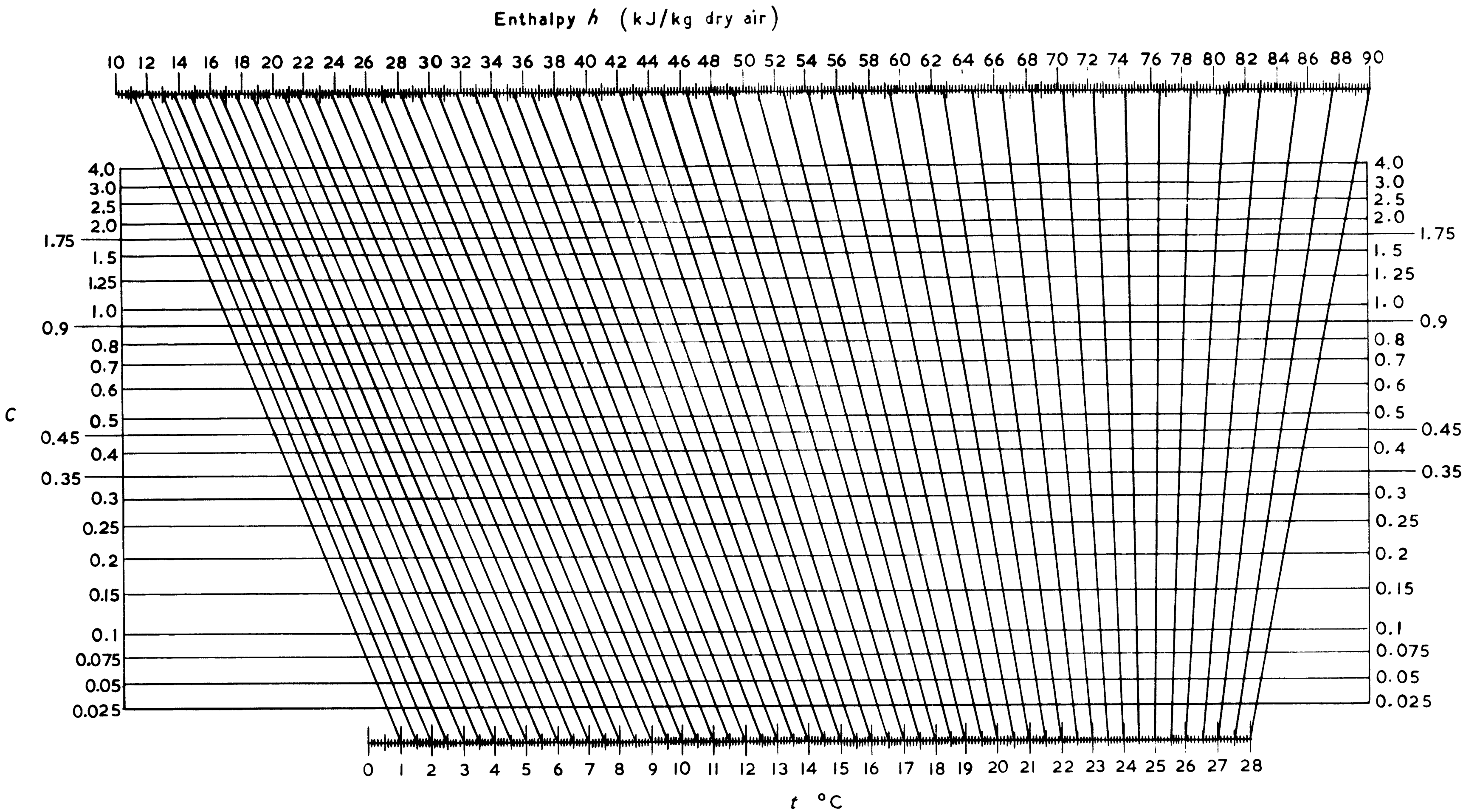


Figure 20 — Graph of R_{ad} as a function of velocity (v_r)



For absolute pressure range 995 mbar to 1025 mbar.

Figure 21 — Surface temperature chart

To obtain *C* scale

$$\frac{X}{Y} = C\left(\frac{y}{x}\right)$$

where x = length representing 1 kJ/kg
and y = length represen ing 1 °C

e.g. if x = 3 mm
y = 6 mm

for C = 0.5

$$\frac{X}{Y} = 0.5\left(\frac{6}{3}\right) = 1$$

$X + Y$ = 130 mm ∴ X = 65 mm Y = 65 mm

for C = 1.0

$$\frac{X}{Y} = 1\left(\frac{6}{3}\right) = 2$$

$X + Y$ = 130 mm .. $Y = \frac{130}{3}$ = 43.3 mm

and X = 86.6 mm

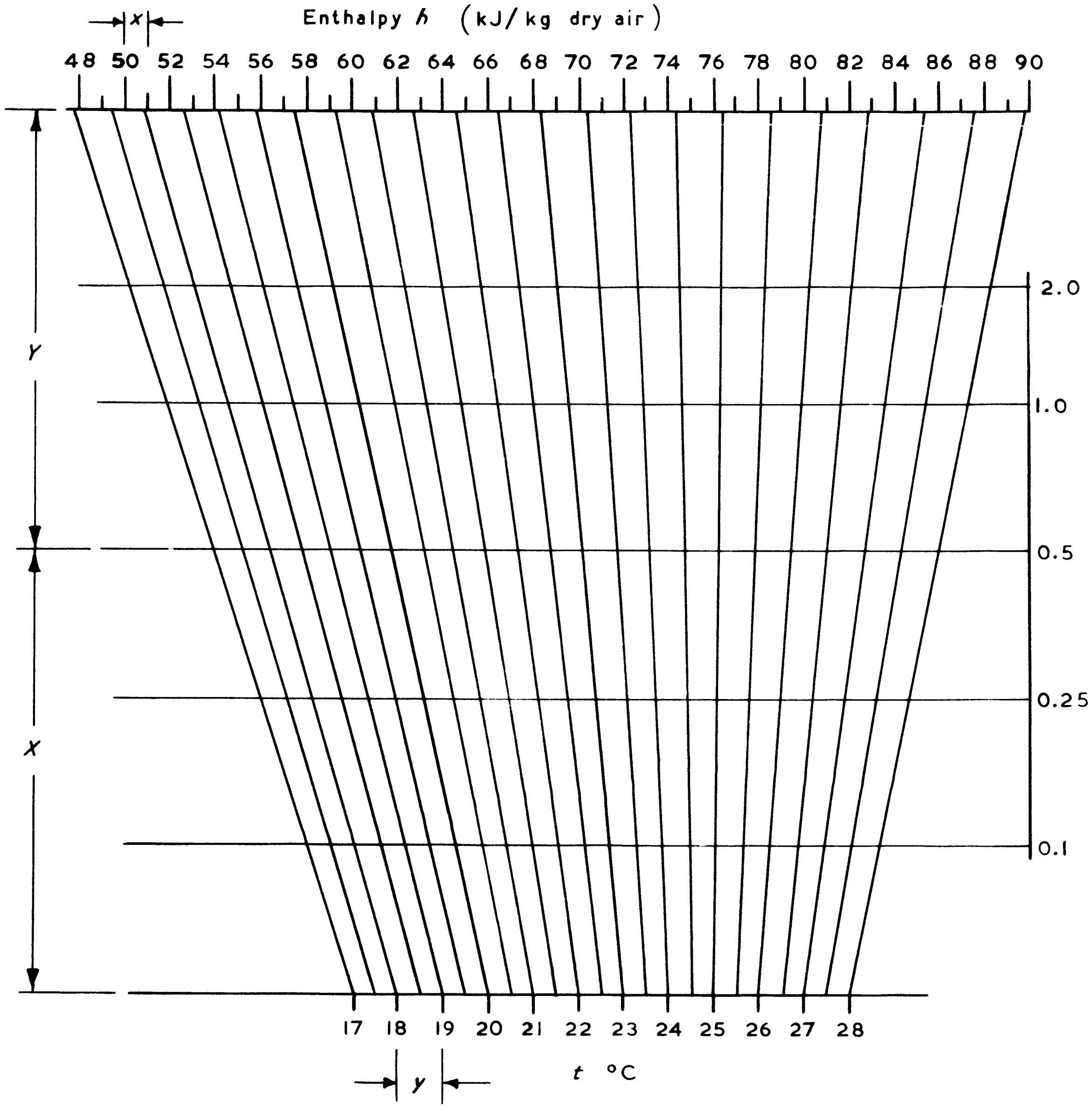


Figure 22 — Surface temperature chart; determination of *C* scale

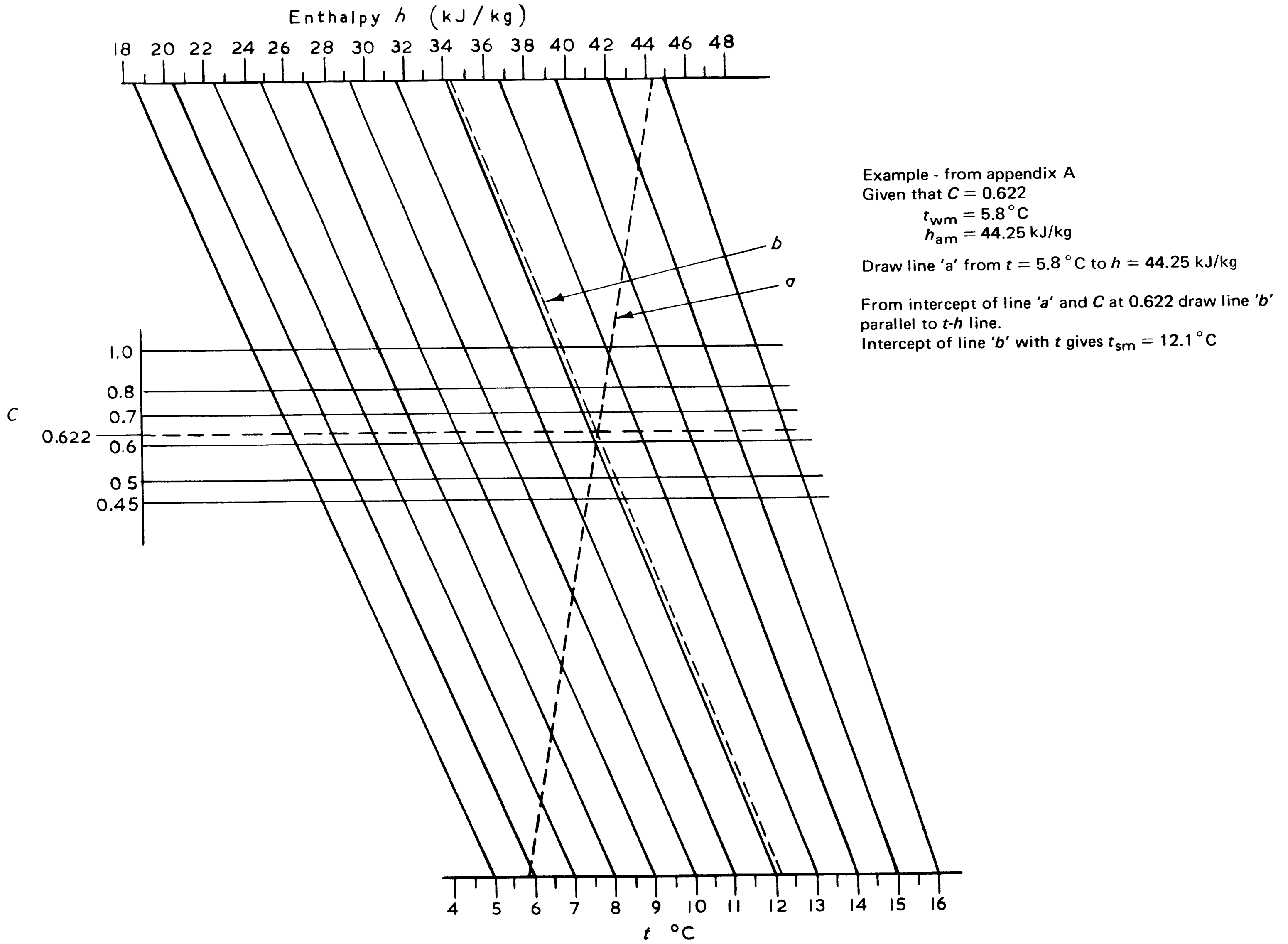
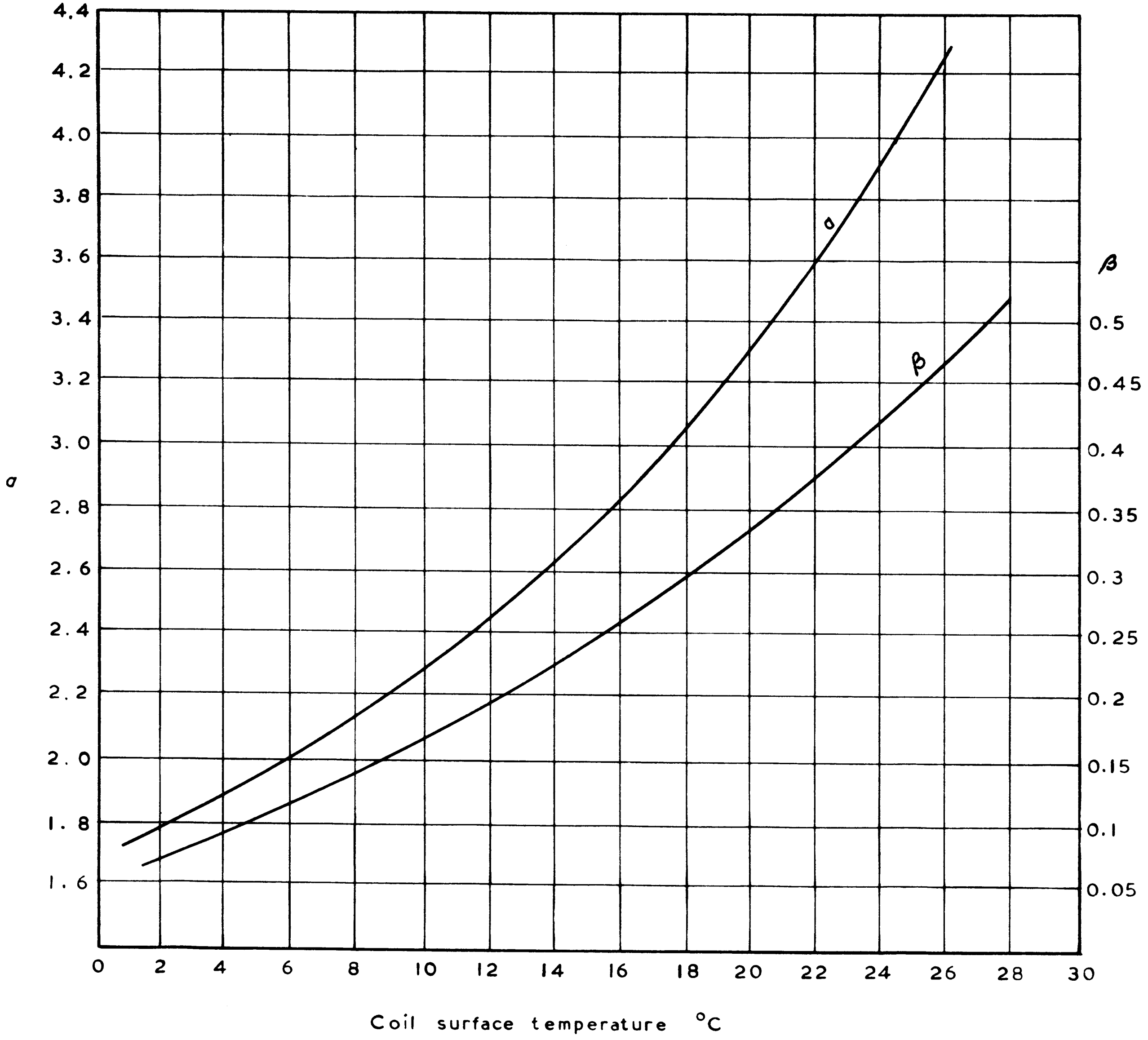


Figure 23 — Use of surface temperature chart



$$Z = \alpha + \frac{\beta \{1013 - \rho_b\}}{50}$$

ρ_b = Atmospheric pressure (mbar)

Figure 24 — Graph for determination of *Z* in relation to coil surface temperature

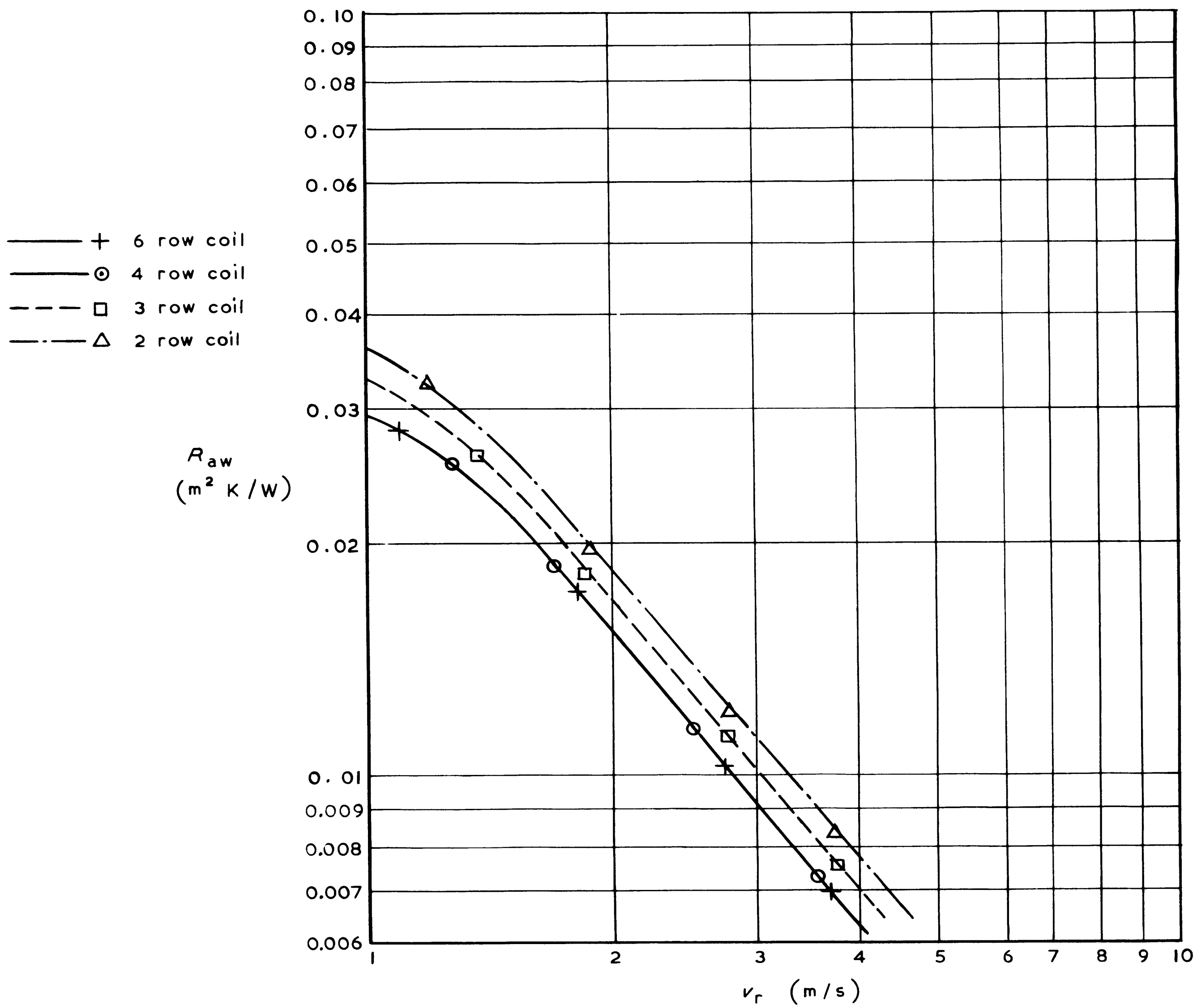


Figure 25 — Graph of R_{aw} as a function of velocity (v_r)

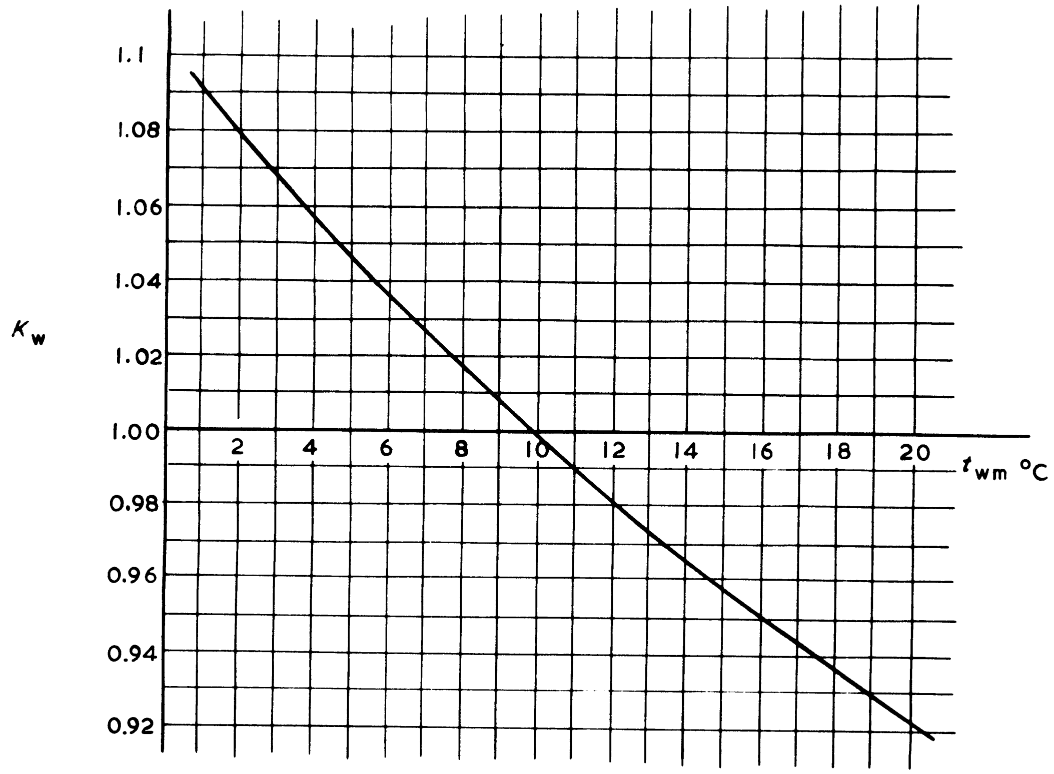


Figure 26 — Hydraulic pressure drop correction factor

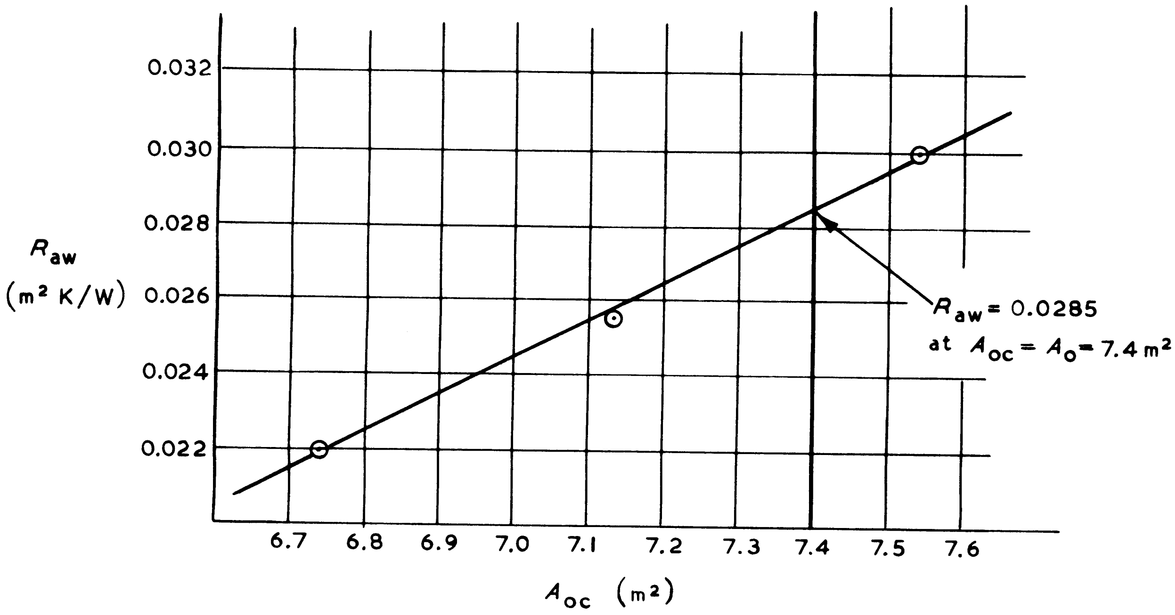


Figure 27 — Graph of A_{oc} versus R_{aw}

Publications referred to

This standard makes reference to the following British Standards:

BS 593, *Laboratory thermometers.*

BS 1041, *Code for temperature measurement.*

BS 1042, *Code for flow measurement.*

BS 3208, *Methods of test and rating for hot-water air-heater batteries.*

BS 4194, *Design requirements and testing of controlled-atmosphere laboratories.*

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