BRITISH STANDARD

BS 4856-3:
1975

Methods for

Testing and rating fan coil units, unit heaters and unit coolers —

Part 3: Thermal and volumetric performance for heating and cooling duties; with additional ducting

UDC 697.971.001.42 + 697.356.001.42

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 Supply Industry in England and Wales
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Ministry of Defence*
National Coal Board
Society of British Gas Industries
Water-tube Boilermakers' Association

The 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
Unit Heater Manufacturers' Association
Oil Companies' Materials Association

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Contents

Foreword

Part 3 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 which appear in this standard. These methods cover a range of testing conditions which encompass the complete performance envelope of the individual items of equipment.

The general approach to the test methods described has been to consider the simplest practical methods of carrying out the measurements without any unnecessary sacrifice of accuracy.

The standard relates to equipment with capacities of up to the equivalent of 75 kW.

Where reference is made to British Standards, and if no metric version is available (e.g. BS 1042 "*Methods for the measurement of fluid flow in pipes*", etc.), 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 38, 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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1 Scope

This Part of this British Standard deals with methods of carrying out thermal and volumetric tests on units intended for use with additional ducting containing fluid to air heat exchangers and incorporating their own electrically powered fan system. The units may be either for heating or for cooling applications and the latter may be with or without dehumidification under frost-free conditions. The tests are to be carried out on units in an essentially clean condition.

2 References

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

3 Definitions

For the purposes of this British Standard the following definitions apply.

3.1
fan coil unit

a fluid-to-air heat exchanger apparatus through which air is passed by its electrically powered fan system and which is intended for general commercial use in the field of generating comfort conditions. This unit may or may not contain filters

3.2
unit heater and unit cooler

a fluid-to-air heat exchanger apparatus through which air is passed by an electrically powered fan system and which is intended for general industrial use in environmental control. This unit may or may not contain filters

3.3
essentially clean conditions

those conditions applying when further cleaning of a unit does not affect the rating figures

3.4
reference air conditions

temperature 20 °C; absolute pressure 1.013 bar[1)]; relative humidity 43 %; density 1.200 kg/m^3

4 Operating conditions of units

4.1 The aerodynamic performance of a ducted unit and consequently the thermal output is to some extent dependent upon the temperature and location of the heat exchanger. If a unit is to be used for a heating duty then temperature effects will be most significant if the fan is located downstream of the heat exchanger.

The rating tests described in the standard are for heating units with the fan located upstream of the coil, and for cooling units with the fan in any position. If the fan is located downstream of the coil then for heating units the modified test series described in Appendix A shall be applied.

4.2 The primary medium shall be any of the following fluids.

a) Water.

b) Steam.

c) Heat transfer fluid (excluding volatile refrigerants).

4.3 The operating conditions of the units shall lie within the following ranges.

a) Chilled water from 2 °C to 20 °C.

b) Low temperature hot water from 40 °C to 95 °C.

c) Pressurized water up to 200 °C.

d) Heat transfer fluid from 40 °C to 200 °C.

4.4 The air volume flow rate of the unit shall lie within the range 25 × 10^{-3} m^3/s to 5 m^3/s.

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

4.5 There is no restriction on the mounting of these units, provided that they are tested whilst mounted in a way representative of their intended use.

4.6 The requirements of the methods for individual tests may be followed to determine the performance of a unit at one specified set of conditions; where the performance at conditions other than those of the actual test are to be stated, the full schedule of tests specified in the standard shall be completed.

5 Nomenclature

For the purposes of this British Standard the following nomenclature is used.

Symbol	Quantity	Unit
A_D	Area of test duct	m^2
A_i	Area of air inlet duct	m^2
A_o	Area of unit air outlet	m^2
A_s	Surface area of insulated duct	m^2
B	Bypass factor	—
c_{pa}	Specific heat capacity of air	kJ/(kg K)
c_{pf}	Specific heat capacity of primary fluid	kJ/(kg K)
c_{pw}	Specific heat capacity of water	kJ/(kg K)
H_D	Test duct hydraulic diameter	m
h_{ai}	Inlet air enthalpy	kJ/kg of dry air
h_{ao}	Outlet air enthalpy	kJ/kg of dry air
h_b	Enthalpy of saturated air at apparatus dew point	kJ/kg of dry air
k_t	Thermal conductivity of insulating material	W/(m K)
K_f	Temperature correction factor	—
L	Latent heat of evaporation	kJ/kg
l	Thickness of insulating material	mm
M_t	Net mass of condensate formed during test	kg
M_n	Net mass of condensate formed during no-load test	kg
M_f	Total mass flow of primary fluid	kg
m_a	Air mass flow rate	kg/s
m_f	Primary fluid mass flow rate	kg/s
Δp_f	Primary fluid pressure drop	Pa
Δp_{fr}	Primary fluid pressure drop at reference temperature (cooling 10 °C, heating 80 °C)	Pa
p_b	Barometric pressure	Pa
p_{br}	Reference barometric pressure (101 325 Pa)	Pa
p_f	Upstream static pressure at flow meter	Pa
Δp_F	Flow meter pressure drop	Pa
p_v	Dynamic pressure of airstream	Pa
p_{vn}	Dynamic pressure at point n	Pa
p_s	Test duct static pressure	Pa
p_d, p_h, p_B, p_i	Pressure loss corrections defined in **14.3**	Pa
p_{sc}	Static pressure rise corrected for friction, etc.	Pa

Symbol	Quantity	Unit
p_{so}	Static pressure rise corrected to unit outlet area	Pa
p_{sr}	Static pressure rise at reference conditions	Pa
Q_n	Net heat transfer	kW
Q_o	Heat losses or gains	kW
Q_g	Gross heat transfer	kW
q_a	Air volume flow rate	m^3/s
q_{ar}	Air volume flow rate at reference conditions	m^3/s
q_f	Primary fluid volume flow rate	m^3/s
q_{fr}	Primary fluid volume flow rate at reference temperature (cooling 10 °C, heating 80 °C)	m^3/s
R	Overall appliance thermal resistance	K/kW
R_a	Air side appliance thermal resistance	K/kW
R_{wm}	Water plus coil metal appliance thermal resistance	K/kW
R_m	Overall appliance thermal resistance at mid range air flow rate	K/kW
S	Sensible to total heat ratio	—
t_{ai}	Inlet air dry bulb temperature	°C
t_{ao}	Outlet air dry bulb temperature	°C
t'_{ao}	Outlet air wet bulb temperature	°C
t_{an}	Air dry bulb temperature at transfer point n	°C
t_F	Temperature at flow meter	°C
t_{fi}	Inlet primary fluid temperature	°C
t_{fo}	Outlet primary fluid temperature	°C
t_s	Saturated steam temperature	°C
t_b	Ambient air dry bulb temperature	°C
t_{ar}	Reference air dry bulb temperature at 20 °C	°C
Δt_m	Logarithmic mean temperature difference	°C
T_t	Duration of test	s
T_n	Duration of no-load test	s
Δw	Change in moisture content across coil	kg/kg of dry air
ρ_a	Air density	kg/m^3
ρ_{ar}	Reference air density (1.2 kg/m^3)	kg/m^3
ρ_f	Primary fluid density at mean fluid temperature	kg/m^3
ρ_{fr}	Primary fluid density at reference temperature (cooling 10 °C, heating 80 °C)	kg/m^3

6 Object of the test method

The object of the tests is to establish the performance of the unit over a specified range of primary and secondary fluid conditions and room temperatures. The test procedures set out in this standard are for the determination of:

a) the gross and net thermal outputs;

b) the secondary fluid flow rates;

c) the primary fluid pressure drop;

d) freedom from sweating on the unit enclosure or casework;

e) adequate condensate disposal.

7 Testing conditions

7.1 Primary fluid, liquid. The unit shall be tested at a minimum of three primary fluid inlet temperatures. The selected temperatures shall cover the range of duty over which the appliance is to be rated, that is, to within 10 % at either end of the range.

7.2 Primary fluid, steam. Gauge pressures for the tests shall be selected such that temperature intervals are not greater than 20 °C, and that a minimum of three test points are obtained. The steam shall have a minimum of 1.5 °C superheat to ensure the absence of entrained moisture, and a maximum of 3 °C superheat.

7.3 Entering air conditions. The entering air temperature shall be within the range 15 °C to 25 °C for heating and 20 °C to 30 °C for cooling.

7.4 Primary fluid flow rate. For duties quoted over a range of primary fluid flow rates, tests shall be carried out at a minimum of three primary fluid flow rates, spaced approximately equally over the required range and covering the range to within 10 % at either end.

7.5 Electrical supply conditions. The tests shall be carried out at the rated voltage and frequency of the unit which shall be stated in the test report.

7.6 Nominal rating. In the absence of a specified set of conditions, where a single nominal rating for either cooling or heating is required, the total capacity under the following conditions should be used:

Condition	Cooling	Heating
Room air temperature	21 °C dry bulb 15 °C wet bulb	18 °C dry bulb —
Primary fluid inlet temperature	5 °C	85 °C
Primary fluid outlet temperature	10 °C	70 °C

8 Instrumentation

8.1 Steam

8.1.1 *Temperature.* Temperature measurement shall comply with the requirements of BS 1041-2, BS 1041-3 and BS 1041-4. Thermometers, thermocouples and resistance thermometers shall be calibrated to a precision of 0.5 °C against a NPL thermometer (accurate to 0.25 °C) and shall be used in pockets similar to those shown in Figure 1(a), situated at the entry and outlet of the unit between 100 mm and 200 mm from the unit connections.

8.1.2 *Pressure.* Pressure gauges shall comply with clause **30** *a* of BS 1780:1960. They shall be at least 150 mm in diameter and shall have a total scale range not more than three times the measured test pressure. For test pressures below 0.3 bar[2)], a liquid manometer is desirable.

8.1.3 *Mass flow.* Mass flow shall be by mass of condensate preferably measured by direct weighing. Weighing apparatus shall be accurate to 0.2 % and collecting vessels shall not weigh more than 50 % of their normal contents. Covers shall be provided to prevent evaporation of liquid from any collecting vessels awaiting weighing.

8.2 Water or heat transfer fluid

8.2.1 *Temperature.* The measurement of temperature of the water or heat transfer fluid at entry and exit shall comply with the requirements of BS 1041. If liquid-in-glass thermometers are used, they shall be graduated for partial immersion at intervals of 0.1 °C and shall comply with the requirements of BS 593.

Thermometers, thermocouples and resistance thermometers shall be calibrated to a precision of 0.1 °C against a NPL thermometer (accurate to 0.05 °C), and shall be used in pockets similar to those shown in Figure 1(b), situated at the entry and outlet of the unit between 100 mm and 200 mm from the unit connections.

[2)] 1 bar = 10^5 N/m² = 100 kPa.

For primary liquid temperature differences of less than 5 °C it will be necessary to take special care and the differential thermocouple system shown in Figure 2 is recommended.

8.2.2 *Pressure.* The pressure drop across the unit shall be measured with an inverted U-tube manometer similar to that shown in Figure 3(a). The manometer shall be vertical and calibrated in intervals not exceeding 2 % of the measured drop.

8.2.3 *Mass flow.* Mass flow shall preferably be measured by direct weighing. Weighing apparatus shall be accurate to 0.2 % and collecting vessels shall not weigh more than 50 % of their normal contents. Covers shall be provided when necessary to prevent evaporation of liquid from any collecting vessels awaiting weighing.

Liquid flow rate may also be measured by means of any of the appropriate devices described in BS 1042-1.

8.3 Air

8.3.1 *Temperature.* Measurements of air temperature shall be made with mercury-in-glass thermometers, thermocouples or resistance thermometers, calibrated to 0.1 °C against a NPL class A thermometer. Thermometers shall be graduated at intervals not greater than 0.1 °C and calibrated for total immersion.

Temperature measuring instruments shall be shielded against radiation; suitable shields are shown in Figure 4(a). Inlet air temperatures shall be measured by thermocouples or resistance thermometers capable of being shielded as in Figure 4(b).

Where it is necessary to measure wet and dry bulb air temperatures these shall be measured by means of instruments installed in sampling tubes similar to those shown in Figure 5.

Psychrometric data shall be measured by means of the system given in Appendix D of 4194:1967 connected to a suitable sampling system as shown in Figure 6.

8.3.2 *Flow rate.* Flow meters should comply with the requirements of BS 1042-1. Where the recommendations of BS 1042-1 cannot be followed the flow meter may be calibrated in situ by the method given in BS 1042-2. It is suggested that the air flow measurement be carried out by means of the orifice plate method described in BS 1042-1 or by means of the pitot traverse method described in BS 848-1, circular airways, 24 point traverse.

8.3.3 ***Pressure***

8.3.3.1 The minimum differential pressure readings for flow measurement shall be 25 Pa for inclined U-tube manometers and micro-manometers, and 500 Pa for vertical U-tube manometers.

8.3.3.2 It shall be possible to read the manometer to an accuracy of 0.5 Pa over a pressure differential of between 25 Pa and 50 Pa and 1 % at higher differential pressures.

8.3.3.3 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. Total pressures shall be measured with the pitot-static tube described in BS 1042-2.

9 Test equipment

9.1 Steam. The steam supply and consumption test equipment shall be as detailed in **9.1.1** to **9.1.5** and arranged as shown in Figure 7.

9.1.1 The length of pipe between the steam pressure and temperature measurement positions and the unit casing shall not exceed 300 mm and shall be insulated with at least 40 mm thickness of insulating material having a thermal conductivity not exceeding 0.06 W/(m K).

9.1.2 The pipe conveying the condensate to the sight glass shall not exceed a total length of 1 m and shall be insulated as described in **9.1.1**.

9.1.3 An air cock, throttle valve, pressure gauge or manometer, a temperature measuring instrument and a separator with drain shall be fixed in the steam pipe close to the steam inlet to the unit, as a means for controlling and indicating the steam pressure and temperature. A superheater, preferably electrically heated, is also advisable.

9.1.4 An air cock, automatic or otherwise, a sight glass with water level indicator and a throttle valve shall be fixed close to the heater condensate outlet.

9.1.5 The condensate shall be collected and weighed and means shall be employed to minimize loss by evaporation. An after cooler may be employed if desired.

9.2 Water or heat transfer liquid

9.2.1 There shall be available a means for providing a continuous supply of fluid, together with a means for controlling the temperature of this fluid at any desired temperature that may be required for the test.

9.2.2 There shall be available a means for providing a constant flow of fluid such as a constant head tank or a circulation pump. A means shall also be available for controlling the fluid to any desired flow rate which may be required for the test.

9.2.3 The equipment shall be arranged generally as in Figure 8.

9.2.4 For water temperatures above 95 °C a closed pressure vessel such as a boiler operating at constant pressure should be used as the source of hot water, and an after cooler inserted before the water control valve at the discharge end. A typical arrangement is shown in Figure 9.

9.2.5 The pipework shall be arranged to give an unobstructed straight run at entry to and exit from the appliance, the pipe diameter being equal to that demanded by the unit connections and of length at least equal to five pipe diameters.

9.2.6 Hydraulic resistance side wall tappings shall be adjacent to the connection to the appliance. The tappings shall be as specified in Figure 3(b) and connected to form a piezometric ring.

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

9.2.8 When mass flow measurement is by weighing means shall be taken to minimize evaporation from vessels awaiting weighing.

10 Test facility

10.1 General. A room shall be available in which air conditions can be controlled to provide any values which are required for the tests. The room shall be such as to permit the unit under test to be mounted and operated so as to represent the intended manner of use for the particular unit.

In many cases, such as when testing units on heating only, it will be possible to test units in the ordinary laboratory. In the case of the larger unit heaters or coolers it may not be possible to obtain steady state conditions and the treated air may have to be ducted away from the test environment.

Two standard test arrangements are recommended. Arrangement A is suitable for units with a mean discharge velocity in excess of 6 m/s. Arrangement B may be used for all types of unit. The two test arrangements do, however, have certain common features, as detailed below.

The test arrangement shall either be with the test duct in line with the unit outlet or with a right angle bend introduced between the unit outlet and test duct [see Figure 10(a) or Figure 10(b)]. This bend shall be fitted with a minimum of five turning vanes.

Units fitted with multiple air outlets shall be tested with a standard single air outlet (of cross-sectional area equal to the discharge area of the multiple outlets) or with the single outlet recommended by the manufacturer.

Units for use with ducted inlet and outlet shall be tested with an inlet duct of a minimum length of 4 hydraulic diameters, and of cross-sectional dimensions equal to those of the unit air inlet. This duct may be fitted with a low-loss entry.

10.2 Test arrangement A

10.2.1 The general arrangement of the test airways shall be as shown in Figure 10(a).

10.2.2 The test duct indicated in Figure 10(a) shall be circular in cross section and of a minimum length of 8 diameters. If the unit air outlet is non-circular then a transition to circular cross section shall be fitted between the unit outlet and the test duct. This transition shall be fitted downstream of any bend fitted [see Figure 10(a)], shall have a maximum included wall angle of 15°, and the outlet area shall not differ from the area of the unit's own outlet by more than 20 %.

10.2.3 The discharge end of the test duct shall be fitted with a symmetrical throttling device as a means of providing an adjustable resistance.

10.2.4 Provision shall be made for a total pressure pitot tube and temperature traverse at a point 2 test duct diameters from the discharge end of the test duct. This traverse shall be carried out with a combined temperature and total pressure probe, as shown in Figure 11. The duct wall static pressure shall be measured at the plane of the temperature and total pressure traverse.

10.2.5 The whole of the test airways shall be insulated with at least 50 mm thickness of insulating material having a thermal conductivity not exceeding 0.06 W/(m K).

10.3 Test arrangement B

10.3.1 The general arrangement of the test airways shall be as shown in Figure 10(b).

10.3.2 The test duct may be of non-circular cross section, in which case no transition shall be used between the unit outlet (or right angle bend) and the test duct; in other respects the duct shall be as described in **10.2.2** but of 5 hydraulic diameters minimum length. A ring of four static pressure taps shall be fitted 3 hydraulic diameters downstream of the flow straightener.

10.3.3 The test duct shall be fitted to a temperature mixing and sampling section; a thin orifice plate (as described in BS 1042-1) shall be installed having 10 duct diameters upstream and 5 diameters downstream with a flow straightener at each end. A booster fan as shown in Figure 10(b) shall be installed. Suitable air mixers and sampling equipment are shown in Figure 12 and Figure 6 respectively.

10.3.4 The whole of the test airway between the unit outlet and temperature sampling station shall be insulated with at least 50 mm thickness of insulating material having a thermal conductivity not exceeding 0.60 W/(m K).

11 Air temperature measurement

11.1 Unit inlet temperature

11.1.1 The unit inlet temperature (t_{ai}) shall be measured 10 mm in front of the inlet (or inlet grilles) by means of point measuring instruments [see Figure 4(b)].

11.1.2 Measuring stations shall be evenly distributed about the inlet (or inlets) as shown in Figure 13. At least four measuring stations per inlet shall be employed, and an extra measuring station shall be introduced for every degree Celsius deviation of individual readings from the mean temperature.

For example, if the measured temperatures are 19.2 °C, 20 °C, 20.6 °C, 18.5 °C, the mean of these is 19.6 °C and two extra stations will be required.

11.1.3 The additional temperature measuring stations shall be located as shown in Figure 13.

12 Test method

12.1 Steady state conditions. Carry out the tests under steady state conditions; these shall be said to exist when the following measurements have not varied by more than the specified amounts from their mean value for a minimum period of 30 min, readings being taken at 10 min intervals.

Table 1 — Steady state conditions

Quantity	Permitted variations from mean	
	Heating	Cooling
Entering air temperature		
Dry bulb	0.5 °C	0.25 °C
Wet bulb	—	0.25 °C
Entering fluid temperature		
< 100 °C	0.5 °C	0.25 °C
> 100 °C	1 °C	—
Fluid flow rate	2 %	2 %
Steam, rate of condensation	3 %	—
Supply voltage	2½ % of rated voltage	

12.2 Steam as primary fluid

12.2.1 With steam at the required pressure adjust the condensate control such that the condensate level is visible in the sight glass. Only start after a state of equilibrium has been reached. Such a state of equilibrium may be considered to exist when the conditions of **12.1** are satisfied.

12.2.2 Continue each test until three consecutive sets of readings, each taken over a 15 min period (one of which may be taken at the completion of the last 15 min of the equilibrium period) give outputs agreeing within ± 2 %.

12.2.3 Read the supply steam temperature (t_s) at the beginning and end of each test and at intervals of 15 min during the test. All readings of the supply steam temperature (t_s) shall comply with the conditions of superheat laid down in **7.2**.

12.2.4 Read the supply steam pressure at intervals of 15 min during the test; successive readings shall not vary from their mean value by more than ± 5 % absolute.

12.2.5 Collect the total condensate and weigh it on the apparatus described in **8.1.3**. Also note the mass of condensate collected in each 15 min period of the test; the condensate rate during any one of these periods shall not vary by more than ± 3 % from the mean.

12.2.6 Make a no-load test to determine the amount of steam which is condensed in the steam and condensate piping between the temperature measuring station and the condensate collecting vessel. Run this no-load test under the same conditions and following the same procedure as for testing the appliance itself. The degree of superheat shall be at least 1.5 °C and not more than 3 °C.

One of the two following methods of determining no-load condensation shall be employed.

a) Move either the supply piping or the condensate piping and the condensate seal and then joint them directly using the shortest possible connection. The condensate collected per hour will represent the total no-load correction.

b) If the piping is inflexible, substitute in place of the test unit a pipe of the same diameter and insulated in exactly the same manner as the condensate piping. Divide the condensate mass collected per hour by the total length of pipe from the inlet temperature measuring station to the condensate seal and then multiply it by the length of the permanent condensate piping. This value will then represent the total no-load correction as in **12.2.6** a).

12.3 Water or heat transfer fluid as primary fluid

12.3.1 Before starting a test the unit and all supply pipes shall be thoroughly vented by means of manual or automatic air vents.

12.3.2 Only start the test after a state of equilibrium has been reached. Such a state of equilibrium may be considered to exist when the conditions of **12.1** are satisfied.

12.3.3 Continue each test until three consecutive sets of readings, each taken over a period of 15 min, give outputs agreeing with ± 2 %.

12.3.4 During any test the variation in fluid flow rate about the mean shall not exceed 2 % and the air temperature (both wet and dry bulb temperature if cooling) shall not differ by more than the value given in Table 1 from the mean temperature recorded during the equilibrium period.

12.3.5 The following readings shall be taken.

a) Fluid flow rate measured in the manner specified in **8.2.3**.

b) Fluid temperature read at the beginning and end of the test and at intervals of 5 min. The extreme readings of each instrument shall not differ by more than the value given in Table 1. The range of fluid temperature difference readings obtained during the test shall not exceed 0.25 °C. The average of the inlet fluid temperatures and the average of the outlet fluid temperatures over the appropriate period shall be used for calculation.

c) The pressure drop across the unit under test read at the beginning and end of the test.

13 Test method (air side measurements)

Set up the unit to be tested as described in clause **10**. Carry out primary fluid measurements under the equilibrium conditions given in **12.1**. It is possible for the aerodynamic tests to occupy a longer period of time than the thermal test. Under these circumstances the thermal test shall be extended to occupy the same period of time as the aerodynamic test, with all the necessary tolerances being held for the duration of the test.

The cooling tests described here are for sensible cooling only and consequently such a test is not valid if the inlet air dew point exceeds the inlet primary fluid temperature (see clause **17**).

13.1 Test with arrangement A [see Figure 10(a)]

13.1.1 Set the desired primary fluid conditions, including the unit fan speed (if a variable speed fan is fitted) and the unit static pressure (measured at the taps described in **10.2**); commence the test when thermal equilibrium has been achieved.

13.1.2 Measure the heat transferred on both the air and primary fluid sides (see **12.2** and **12.3**).

13.1.3 Make the air side measurement of heat transfer by means of a temperature and total pressure probe (see Figure 11) traverse. Carry our this single traverse at the 24 points shown in Figure 14, and record the following data at each traverse station.

a) Local temperature t_{an} (°C).

b) Dynamic pressure (difference between total pressure measured on the pitot tube and wall static pressure measured directly on one manometer) P_{vn}(Pa).

Record the inlet air temperature t_{ai} (°C).

Measure the duct static pressure at four equally spaced intervals during the test; the mean static pressure shall be the test static pressure. Record the barometric pressure and the ambient temperature in the region of the test rig at the start and at the end of the test and employ the mean value in the analysis of the test data.

13.1.4 Check manometer zeros before and after each reading, and make any necessary adjustment to the zero.

13.1.5 Record the hydraulic pressure drop during the test.

13.2 Test with arrangement B [see Figure 10(b)]

13.2.1 Set the desired primary fluid temperature and flow rate, unit fan speed (if a variable speed fan is fitted) and test duct static pressure and start the sampling fan. When the steady state conditions specified in **12.1** have been achieved the test may be commenced.

13.2.2 Record the following data at 10 min intervals, until the completion of the tests described in **12.2** and **12.3**.

a) Orifice plate pressure drop	Δp_F (Pa).
b) Static pressure upstream of the orifice plate	p_F (Pa).
c) Air flow temperature upstream of the orifice plate	t_F (°C).
d) Air flow temperature at the sampling tubes	t_{ao} (°C).
e) Test duct static pressure	p_s (Pa).
f) Inlet air temperature	t_{ai} (°C).

Record the barometric pressure and the ambient temperature in the region of the test rig at the start and at the end of the test. Record the hydraulic pressure drop during the test.

13.2.3 Use the mean values of the test data in all calculations.

14 Calculations

The heat transfer calculations are given in this clause for heating data; cooling data will give negative values of heat transfer.

14.1 Primary fluid

14.1.1 *Steam.* Calculate the gross heat transferred (Q_g) from the following equation.

$$Q_g = \left[\frac{M_t}{T_t} - \frac{M_n}{T_n}\right] L \qquad \text{(kW)}$$

14.1.2 *Water or heat transfer fluid.* The gross heat transferred (Q_g) shall be calculated from the following equation

$$Q_g = m_f\, c_{pf}\, (t_{fi} - t_{fo}) \qquad \text{(kW)}.$$

14.2 Air side

14.2.1 ***Air mass flow rate***

Method A. Obtain the air mass flow rate from the following equation.

$$m_a = \frac{A_D}{24}\left[\frac{2\rho_{ar} p_b (t_{ar} + 273)}{p_{br}}\right]^{\frac{1}{2}} \sum_{n=1}^{n=24} \left[\frac{p_{vn}}{(t_{an} + 273)}\right]^{\frac{1}{2}} \quad \text{(kg/s)}$$

Method B. Compute the air mass flow rate from the mean values of the test data utilized in accordance with the requirements of BS 1042.

14.2.2 *Heat transferred.* Calculate the net heat transferred at the heat exchanger (Q_n) from the following equations.

Method A

$$Q_n = \left[\frac{A_D}{24}\, c_{pa} \left\{\frac{2\rho_{ar} p_b (t_{ar} + 273)}{p_{br}}\right\}^{\frac{1}{2}} \sum_{n=1}^{n=24} \left\{\frac{p_{vn}}{t_{an} + 273}\right\}^{\frac{1}{2}} (t_{ao} - t_{ai})\right] + Q_o \quad \text{(kW)}$$

where

$$Q_o = \frac{k_t \dot{A}_s}{l}(t_{ao} - t_b) \quad \text{(kW)}$$

p_b is the mean test barometric pressure (Pa).

If the test duct static pressure exceeds the mean test barometric pressure by 1 000 Pa or more this higher value shall be used in the calculation.

Method B

$$Q_n = [m_a c_{pa} (t_{ao} - t_{ai})] + Q_o \quad \text{(kW)}$$

where

$$Q_o = \frac{k_t A_s}{l}(t_{ao} - t_b) \quad \text{(kW)}$$

14.3 Unit static pressure rise. Correct the measured static pressure (p_s) for test duct friction loss, flow straightener loss, bend loss and inlet duct loss (if an inlet duct is fitted) by means of the following equations.

$$p_{sc} = p_s + p_d + p_h + p_B + p_i \qquad \text{(Pa)}$$

where

$p_d = 0.02 \dfrac{x}{H_D} \bar{p}_v$ test duct friction loss

$p_h = 0.2\ \bar{P}_v$ flow straightener loss[a]

$p_B = 0.1\ \bar{P}_v (A_D/A_o)^2$ bend loss

$p_i = 0.08\ \bar{P}_v (A_D/A_i)^2$ inlet duct loss (if an inlet duct is fitted)

[a] Rounded figure from National Engineering Laboratory Report No. 461

where

x is as defined in Figure 15 (m);

H_D is the test duct hydraulic diameter (= 4 × area/perimeter) (m);

$\bar{\rho}_v$ is the mean velocity pressure in the test duct

$$= \frac{1}{2\bar{\rho}_a}(m_a/A_D)^2 \quad \text{(Pa)}$$

where

$\bar{\rho}_a$ is the mean air density at the temperature and pressure in the test duct (kg/m³);

A_D is the cross-sectional area of the test duct (m²)

A_i is the cross-sectional area of the air inlet duct (m²);

A_o is the cross-sectional area of the unit air outlet (m²).

Then convert the static pressure rise (if necessary) to that for a test duct area equal to the unit air outlet by means of the following equation.

$$p_{so} = p_{sc} + \{m_a^2/(2 - A_D)\}(1 - A_D^2/A_o^2).$$

14.4 Heat balance. The ratio of net to gross heat transferred [ratio of air side to water (or steam) side heat transfer] will depend upon the casing heat loss (air leakage and radiation) and any heat added by the fan and motor. It is therefore to be expected that the air and water side heat transfer rates will not be equal. A consequence of this is that it is not possible to check the accuracy of test data by means of a heat balance. It is however reasonable to assume that the air side heat transfer rate is less than the water side rate. Thus a test shall not be valid if

$Q_g/Q_a \leqslant 0.975.$

15 Rating tests

The following test series will enable the performance of the unit to be determined to an accuracy of approximately ± 5 % within, and to ± 10 % beyond, the range of the test variables.

15.1 Primary fluid steam

15.1.1 The following series of tests shall be carried out in accordance with clauses **12**, **13** and **14**. The tests relate to a single fan speed. If a multispeed fan is offered then the whole test series shall be carried out at the maximum recommended speed and only series 1 repeated at the other speeds.

15.1.2 ***Analysis of test data***

15.1.2.1 From test series 1 plot (Figure 16) the static pressure, at reference condition (p_{sr}) against air volume flow rate (q_a)

where

$$p_{sr} = \frac{p_{so}\rho_{ar}}{\bar{\rho}_a}$$

$$q_a = \frac{m_a}{\bar{\rho}_a}$$

p_{so} is the static pressure rise converted to that for the unit outlet area (see **14.3**) (Pa);

ρ_{ar} is the reference air density (1.2 kg/m³);

$\bar{\rho}_a$ is the mean air density at the traverse station or temperature sampling position (kg/m³);

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

15.1.2.2 From test series 2 plot Q_n against $(t_s - t_{ai})$ on logarithmic graph paper and draw the best line through the test points. Compute the mean value of $q_{ar}(= \bar{q}_{ar})$ (it is possible for q_{ar} to vary slightly with the temperature although p_s is held constant) and note this value on the plot Q_n against $(t_s - t_{ai})$.

15.1.2.3 For test series 3 and 4 carry out the procedure as for test series 2.

15.1.2.4 If, for any test result, the ratio $Q_g/Q_n \leqslant 1.00$ then this ratio shall be plotted against $(\sqrt{p_{so}})/q_{ar}$ for all test results (if $0.975 < Q_g/Q_n \leqslant 1$ this shall be plotted as $Q_g/Q_n = 1$) (see Figure 18).

15.2 Primary fluid; water or heat transfer fluid (heating or cooling)

15.2.1 The series of tests given in Table 3 shall be carried out in accordance with clauses **12**, **13** and **14**. If the unit has a multispeed fan then the full series of tests shall be carried out at the maximum fan speed and only series 5 repeated at the other speeds.

15.2.2 The tests shall be carried out with the inlet air temperature in the range 15 °C to 25 °C (heating) and 5 °C to 30 °C (cooling).

15.2.3 ***Analysis of test data***

15.2.3.1 From test series 5 plot the static pressure at reference conditions (p_{sr}) against air volume flow rate (q_a)

where

$$p_{sr} = \frac{p_{so}\rho_{ar}}{\bar{\rho}_a}$$

$$q_a = \frac{m_a}{\bar{\rho}_a}$$

15.2.3.2 From test series 6 plot Q_n against $(t_{fi} - t_{ai})$ on logarithmic graph paper and draw the best straight line through the test points.

15.2.3.3 From test series 9 and 12 plot Q_n against $(t_{fi} - t_{ai})$ on the same graph as specified in **15.2.3.2** and draw the best lines through the test points parallel to the line plotted in **15.2.3.2**. (The points shall not deviate by more than ± 3 % from this line.)

Table 2 — Rating test for steam as primary fluid

Test series	Steam pressure[a]	Unit static pressure rise
1	No heat transfer	Approximately six values evenly spaced over required range[b]
2	Minimum of three to cover required temperature range to within 10 % and such that temperature intervals do not exceed 20 °C	Such that air flow rate[c] is within 10 % of minimum required
3	Minimum of three to cover required temperature range to within 10 % and such that temperature intervals do not exceed 20 °C	Such that air flow rate[c] is approximately mid-range
4	Minimum of three to cover required temperature range to within 10 % and such that temperature intervals do not exceed 20 °C	Such that air flow rate[c] is within 10 % of maximum

NOTE The inlet air temperature shall be in the range 15 °C to 25 °C

[a] There shall be a minimum of 1.5 °C and a maximum of 3 °C superheat.

[b] Test for specified duty shall comprise not less than three test points determining a short portion of the pressure flow characteristic including the specified volume.

[c] Chosen values shall be held constant during each test series.

15.2.3.4 Compute the mean fluid $(\overline{m}_f)$ and air volume $(q_{ar} = m_a/1.2)$ flow rates for the tests in series 6, 9 and 12 and note these mean values on the plot Q_n against $(t_{fi} - t_{ai})$.

15.2.3.5 Carry out stages **15.2.3.2** to **15.2.3.4** with the series of tests 7, 10, 13 and 8, 11, 14.

15.2.3.6 If, for any test, the ratio $Q_g/Q_n \leqslant 1.00$ then this ratio for all test results shall be plotted against $(\sqrt{p_{so}})/q_{ar}$ on linear graph paper (if $0.975 < Q_g/Q_n \leqslant 1$ then this shall be plotted as $Q_g/Q_n = 1$).

15.2.3.7 The primary fluid pressure drop shall be corrected for any head difference between the inlet and outlet taps and converted to the drop at

10 °C for cooling duty;

80 °C for heating duty;

by means of the following equation.

$$\Delta p_{fr} = \Delta p_f / K_f$$

where

Δp_f is the primary fluid pressure drop corrected for any head difference (Pa);

K_f is the temperature correction factor obtained from Figure 19 or Figure 20. (Water as primary heat transfer fluid only.)

Δp_{fr} shall be plotted against m_f on logarithmic graph paper.

16 Example

16.1 A complete set of test data for a single speed unit with coil is shown in Figure 16, Figure 17 and Figure 18. It is desired to check if the unit will deliver 16 kW of heat when supplied with water at 70 °C and inlet air at 10 °C. It is also desired that the water temperature drop should be 10 °C. The unit is to be connected to a duct work system with a static pressure flow characteristic reference for air of

$$\Delta p = 426\left(\tfrac{1}{2}\rho_{ar} q_{ar}^2\right)$$

where

Δp is the system pressure rise (Pa).

The calculation may be carried out as specified in **16.2** to **16.8**.

16.2 The unit pressure flow characteristic with inlet air at 10 °C may be obtained by direct application of the fan laws, i.e.

$$p_{so(10)} = \frac{p_{so}\rho_{a(10)}}{\rho_{ar}} \text{ at } q_{a(10)} = q_{ar}$$

where

$p_{so(10)}$ is the static pressure rise at 10 °C (Pa);

$\rho_{a(10)}$ is the density of air at 10 °C (kg/m^3);

$q_{a(10)}$ is the air volume flow rate at 10 °C (m^3/s)

This characteristic is then plotted on Figure 16.

16.3 The required system pressure rise is dependent upon the off unit air temperature; this may be estimated by first plotting the "cold" pressure flow characteristic in Figure 16 (i.e. $\Delta p = 213\rho_{ar}q_{ar}^2$), the intersections of the two characteristics giving an air volume flow of approximately 0.925 m^3/s. Thus it can be expected that the air temperature in the duct system will be in the region of 24 °C ($Q_n/c_{pa}m_a = t_{ao} - t_{ai}$). It is therefore necessary to plot (on Figure 16) a new system characteristic for an air temperature of 24 °C, i.e.

$$\Delta p = 213\rho_{a(24)}\left[\frac{q_{a(10)}\rho_{a(10)}}{\rho_{a(24)}}\right]^2$$

The air volume flow is corrected to that at entry conditions (air at 10 °C). The subscript numbers in parentheses refer to temperature.

Thus the unit may be expected to deliver 0.9 m^3/s (air at 10 °C) at 224 Pa (p_{so}), or 0.93 m^3/s of reference air.

Table 3 — Rating tests for water or heat transfer fluid

Test series	Flow rate for water or heat transfer fluid	Inlet temperatures[a] for water or heat transfer fluid	Unit static pressure rise
5	No heat transfer		Approximately six values spaced evenly over required range[b]
6	Within 10 % of minimum recommended	Three evenly spaced over range	Such that air flow rate is within 10 % of maximum
7	Within 2 % of that used in series 6	Three evenly spaced over range	Such that air flow rate is within 10 % of minimum
8	Within 2 % of that used in series 6	Three evenly spaced over range	Such that air flow rate is approximately mid-range
9	Within 10 % of maximum recommended	Two within 10 % of top and bottom of range	Equal to that used in series 6
10	Within 2 % of that used in series 9	Two within 10 % of top and bottom of range	Equal to that used in series 7
11	Within 2 % of that used in series 9	Two within 10 % of top and bottom of range	Equal to that used in series 8
12	Approximately mid-range	Two within 10 % of top and bottom of range	Equal to that used in series 6
13	Within 2 % of that used in series 12	Two within 10 % of top and bottom of range	Equal to that used in series 7
14	Within 2 % of that used in series 12	Two within 10 % of top and bottom of range	Equal to that used in series 8

[a] The temperature ranges shall be as specified in **4.3**.

[b] Tests for a specified duty shall comprise not less than three test points determining a short portion of the pressure flow characteristic including the specified volume.

16.4 The leakage heat may now be determined from Figure 18 at

$(\sqrt{p_{so}})/q_{ar} = 16$;

$Q_g/Q_n = 1.05$.

16.5 Assuming the unit will be suitable for the required duty the fluid mass flow rate may be computed from the following equation:

$$m_f = \frac{Q_n(Q_g/Q_n)}{c_{pf}(t_{fi} - t_{fo})} = \frac{16 \times 1.05}{4.18 \times 10} = 0.4 \text{ kg/s.}$$

16.6 From Figure 17 carry out a cross-plot of Q_n versus q_{ar} at $(t_{fi} - t_{ai}) = 60$ °C as shown in Figure 21.

16.7 From Figure 21 carry out a cross-plot Q_n versus m_f at $q_{ar} = 0.93$ as shown in Figure 22. The heat transfer at $m_f = 0.4$ kg/s may be obtained from this graph as 17 kW, which is slightly in excess of the required heat transfer, and consequently the unit will be suitable for the specified duty.

16.8 It has been assumed in the above calculations that the unit had a free air inlet. If the unit had been situated between two lengths of ducting then for the calculation of the leakage rate p_{so} should be replaced by static pressure at the unit outlet.

17 Sensible and latent cooling rating tests

17.1 General. The rating tests described in the preceding clauses are for sensible cooling only; it is however important that the performance of the unit under a latent load is measured. The measurement of performance under these conditions requires a knowledge of the coil thermal resistances, and the effect of moisture on the air side pressure drop. It is not practical or desirable to test the coil as a separate unit[3]. As a consequence it is not possible to obtain a very accurate measure of the unit's performance when actively condensing moisture on to the coil surface. The following series of tests enable the performance of the unit to be determined, for a range of operating conditions, to an accuracy of about 10 %, or to within 5 % for a fixed point test.

It should be noted that the results of these tests do not include the effects of varying amounts of moisture on the coil pressure rise because it is not practical to carry out the necessary number of tests.

The theory employed in these tests (use of the bypass factor) is only approximate.

Table 4 — Latent cooling rating test series

Test series	Water flow rate m_f (kg/s)	Air flow rate q_{ar} (m^3/s)	Sensible to total heat ratio S
A.1	any	approximately six values of static pressures shall be chosen to cover the required range	water temperature shall be as low as possible, and S shall not exceed 0.9
A.2	bottom range	mid, top and bottom range	1.0
A.3	mid and top range	mid range	1.0
A.4	bottom range	mid range	0.9 ± 5 %
A.5	bottom range	mid range	0.8 ± 5 %
A.6	bottom range	mid range	0.7 ± 5 %
A.7	bottom range	mid range	0.65 ± 5 %

NOTE 1 Inlet water temperatures should be kept as low as possible consistent with the required value of S.

NOTE 2 Inlet air conditions shall be in the range 20 °C to 30 °C and 30 % to 80 % r.h.

NOTE 3 Where mid, top and bottom of range are stated, the corresponding flow rates shall be repeated to within 2 % for all tests.

NOTE 4 It is not necessary to make an accurate measurement of the heat transferred in series A.1. This test is intended to give the approximate pressure-flow characteristic with moisture on the coil surface. However, if the variables are chosen carefully then tests from this series can be selected to cover part of the test requirements for series A.2 to A.7.

17.2 The series of tests given in Table 4 shall be made in accordance with **12.1**, **12.3** and clause **13**, with the additional provisions specified in **17.2.1** to **17.2.4**.

17.2.1 Steady state conditions shall be held for 60 min.

17.2.2 The off coil and unit inlet wet bulb temperatures shall be measured at the same time intervals as the corresponding dry bulb measurements. These measurements shall be taken by means of a psychrometer as specified in BS 4194 which shall be connected to both the inlet and outlet sampling tubes.

17.2.3 The net heat transferred shall be calculated from the following equation.

$$Q_n = m_a(h_{ai} - h_{ao}) + Q_o - (m_a \Delta W t'_{ao} c_{pw}) \text{ (kW)}$$

where

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

ΔW is the change in moisture content across coil (kg/kg dry air);

h_{ai} is the inlet enthalpy (kJ/kg dry air);

h_{ao} is the outlet air enthalpy (kJ/kg dry air);

Q_o is the sensible heat gained through the duct walls $= \dfrac{K_t A_s (t_{ao} - t_b)}{l}$ (kW)

Air enthalpies shall be determined from tables and corrected for barometric pressure.

[3] The coil-fan arrangement can be most significant factor in the performance of the system.

17.2.4 The sensible to total heat ratio shall be calculated from the following equation.

$$S = \frac{c_{pa}(t_{ai} - t_{ao} - Q_o/m_a c_{pa})}{(h_{ai} - h_{ao} - Q_o/m_a)}.$$

17.3 Rating calculations

17.3.1 Calculate the coil appliance thermal resistance and bypass factor as specified in **17.3.2** to **17.3.9**.

17.3.2 Calculate the overall appliance thermal resistance for series A.2 to A.7 from the following equation.

$$R = \frac{(t_{ai} - t_{fo}) - (t_{ao} - t_{fi})}{Q_g \log_e [(t_{ai} - t_{fo})/(t_{ao} - t_{fi})]} \quad \text{(K/kW)}$$

17.3.3 The approximate air side (R_a) and water + metal (R_{wm}) appliance thermal resistances for series A.2, A.4, A.5, A.6 and A.7 at the mid range air flow rate ($Q_{ar} = m_a/1.2$), are obtained from the following equation.

$$R_a = \frac{\Sigma(1 - S)(R_m - R)}{\Sigma(1 - S)^2}$$

$R_{wm} = R_m - R_a$

where

R_m is the overall appliance thermal resistance obtained for the mid range air flow rate from A.2;

R is the overall appliance thermal resistance from series A.4 to A.7.

NOTE The equation for R_a is a least squares fit to

$$Q = \frac{\Delta t_m}{SR_a + R_{wm}}.$$

17.3.4 R_a for the other test air flow rates is calculated by subtracting R_{wm} obtained in **17.3.3** from the value of R for the top and bottom air flow rates employed in series A.2.

17.3.5 R_{wm} for the mid and top range water flow rates employed in series A.3 may be computed from the following equation.

$R_{wm} = R - R_a$

where

R is the overall appliance thermal resistance calculated from series A.3;

R_a is the air side appliance thermal resistance at the mid range air volume flow rate.

17.3.6 The bypass factor may be calculated from the equation:

$B = \exp \{-1/(1.2\, c_{pa} q_{ar} R_a)\}$

17.3.7 Plot B against q_{ar}, plot R_a against q_{ar}, see Figure 23(a). Plot R_{wm} against m_f on logarithmic graph paper as shown in Figure 23(b).

17.3.8 From series A.1 plot static pressure p_{sr} against air volume flow rate ($m_a/\bar{\rho}_a$)

where

$$p_{sr} = \frac{p_{so}\rho_{ar}}{\rho_a}$$

This characteristic shall be used to determine air volume delivery when the coil surface is wet.

17.3.9 Plot Q_g/Q_n against $(\sqrt{p_{so}})/q_{ar}$, ($q_{ar} = m_a/1.2$), on linear graph paper; if $Q_g/Q_n < 1$ then this point shall be plotted as 1.

17.4 Method of calculation of sensible and latent duty

17.4.1 Obtain the air volume flow rate from the graph plotted as specified in **17.3.8**.

17.4.2 Q_g/Q_n may then be obtained from the graph plotted in accordance with **17.3.9**.

17.4.3 Assuming inlet conditions are given and outlet conditions are required, then from Figure 23(a) and Figure 23(b) find B, R_a and R_{wm}.

17.4.4 Assume a value of heat transferred, say that for a sensible cooling only duty.

17.4.5 Calculate h_B the enthalpy of saturated air at the apparatus dew point (t_B), as

$$h_B = h_{ai} - \frac{Q_g}{m_a(1-B)}$$

obtaining the temperature t_B from psychrometric charts or tables.

17.4.6 The outlet air dry bulb may be computed from the following equation.

$t_{ao} = t_B + B(t_{ai} - t_B)$.

17.4.7 Thus S may be obtained from the following equation.

$$S = \frac{m_a c_{pa}(t_{ai} - t_{ao})}{Q_g}$$

17.4.8 Calculate the logarithmic mean temperature difference (Δt_m) and, together with R_a and R_{wm} obtained in accordance with **17.4.3**, compute the ratio

$$X = \frac{\Delta t_m}{Q_g(SR_a + R_{wm})}.$$

If this ratio is within the limits of 0.95 to 1.05 then the assumed value of Q_g may be taken to be the correct heat transfer rate. If X is outside these limits then a minimum of two additional values of Q_g should be assumed and X plotted against Q_g. The value of Q_g at $X = 1$ may be taken as the gross total heat transfer.

17.4.9 The sensible to latent ratio of the net heat transferred may be assumed to be the same as that for the gross heat transferred, and computed from steps **17.4.5** to **17.4.7**.

17.4.10 The net total heat transfer is obtained by multiplying Q_g by the value of Q_n/Q_g obtained in **17.4.2**.

18 Enclosure sweating tests and condensate disposal tests

18.1 Enclosure sweat test conditions. The conditions which shall be used during enclosure sweat tests for all models are given in Table 5.

Table 5 — Enclosure sweat test conditions

Room air temperature	
dry bulb	27 °C
wet bulb	24 °C
Test frequency	Rated frequency[a]
Test voltage	Rated voltage[b]

[a] Units with dual rated frequencies shall be tested at each frequency.
[b] Units with dual rated voltages shall be tested at the higher voltage.

18.2 Condensate disposal test conditions. Condensate disposal tests shall be conducted under the same conditions as those specified for enclosure sweat tests in **18.1**.

18.3 Enclosure sweat test

18.3.1 Unit coolers shall meet the requirements of the following enclosure sweat test, when operating at the test conditions specified in **18.1**. The unit's controls, fans, dampers and grilles shall be set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's operating instructions.

18.3.2 *Procedure.* After establishment of the specified temperature conditions operate the unit continuously for a period of 4 h.

18.3.3 *Requirements.* During the test no condensed water shall drip, run or blow off the unit.

18.4 Condensate disposal test

18.4.1 Unit coolers shall meet the requirements of the following condensate disposal test when operating at the test conditions specified in **18.2**. The unit's controls, fans, dampers and grilles shall be set to produce the maximum tendency to sweat, provided such settings are not contrary to the manufacturer's operating instructions.

NOTE This test may be conducted concurrently with the enclosure sweat test (see **18.3**).

18.4.2 *Procedure.* After establishment of the specified temperature conditions, start the unit cooler with its condensate collection pan filled to the overflow point and operate it continuously for 4 h after the condensate level has reached equilibrium.

18.4.3 *Requirements.* During this test, the unit cooler shall have the ability to dispose of all condensate and there shall be no dripping or blowing off of water from the unit such that the building or surroundings may become wet.

Appendix A Heating units with fan installed downstream of the coil

The test requirements for a unit with fan downstream of the coil differ from those for the upstream fan because of the large effect of temperature variations on fan performance.

The unit shall be tested in accordance with clauses **12**, **13**, **14** and **15**, with the following change in the test series procedure.

The heat transfer tests shall each be conducted at a constant air mass flow rate (instead of a constant pressure). The air mass flow rates chosen shall be repeated to within 2 % for all tests at a given air mass flow rate.

The test analysis shall be as described in clause **15**.

The performance calculations shall take into account the effect of the air temperature on the fan performance by direct application of the fan laws, i.e.

At temperature t_{ao} the static pressure rise = $\frac{P_{sr}}{\rho_{ar}} \rho_a$.

The flow remains constant.

It may therefore be necessary to make iterative calculations, initially assuming an air off coil temperature, calculating the performance of the unit and consequently obtaining a new air off coil temperature.

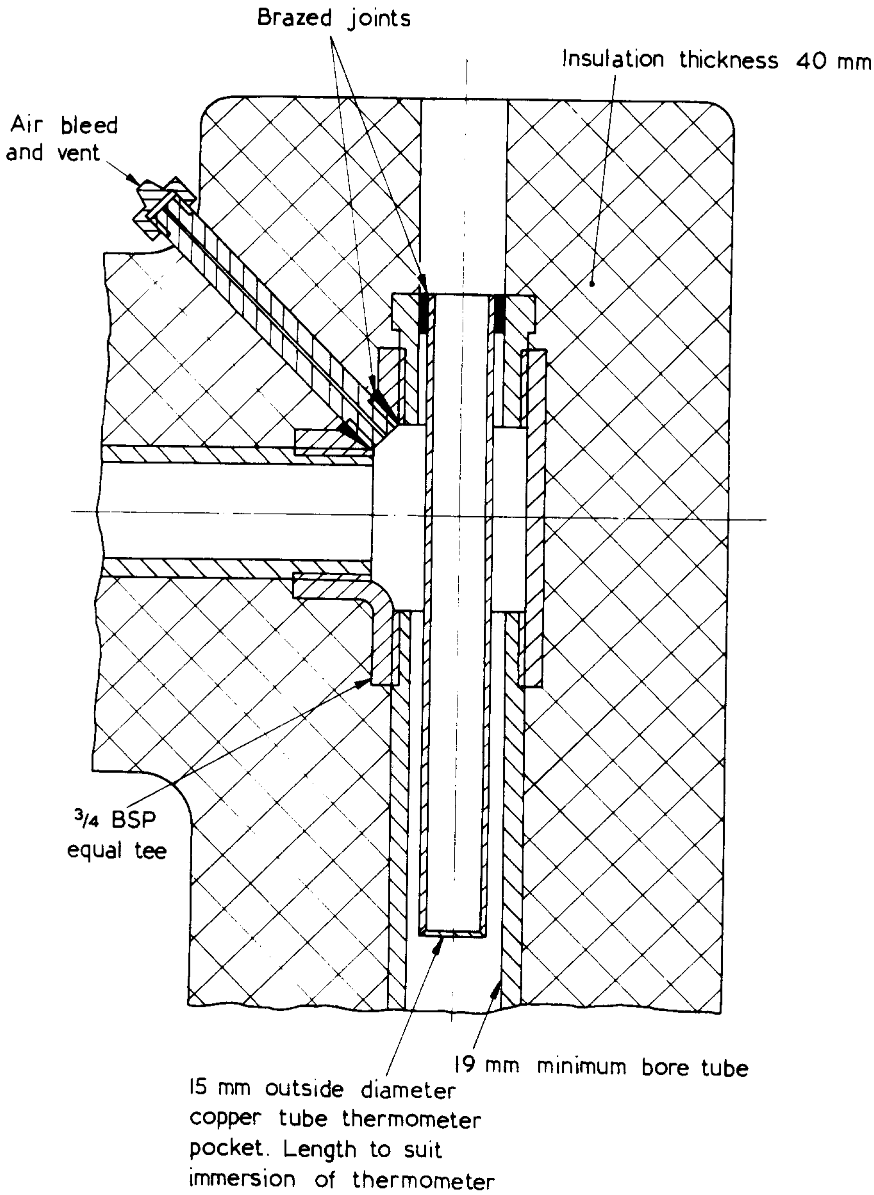


Figure 1(a) — Thermometer pocket for steam test

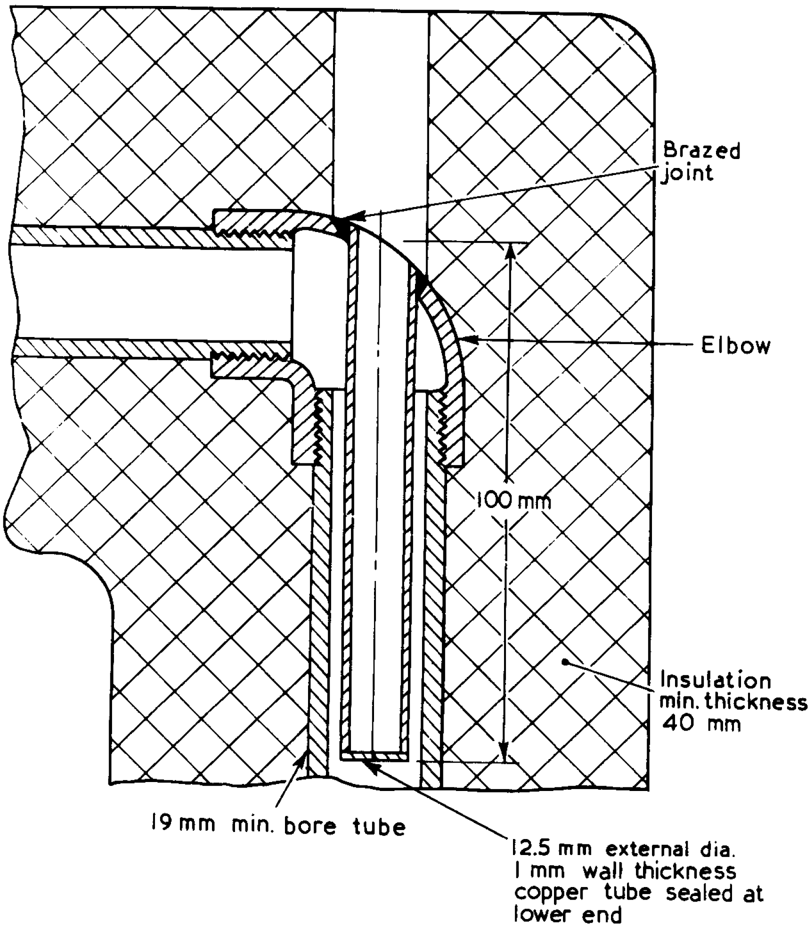


Figure 1(b) — Thermometer pocket for hot water test

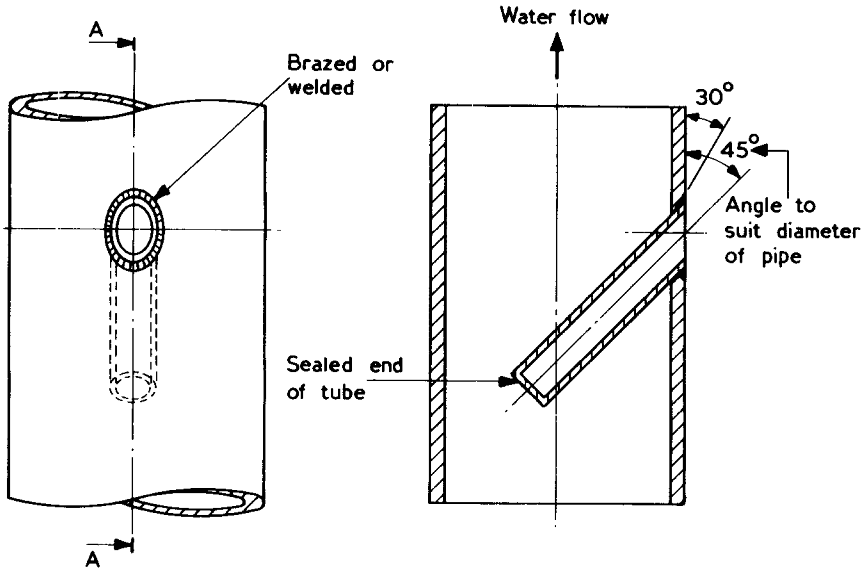


Figure 1(c) — Thermometer pocket for use in vertical pipes over 75 mm diameter

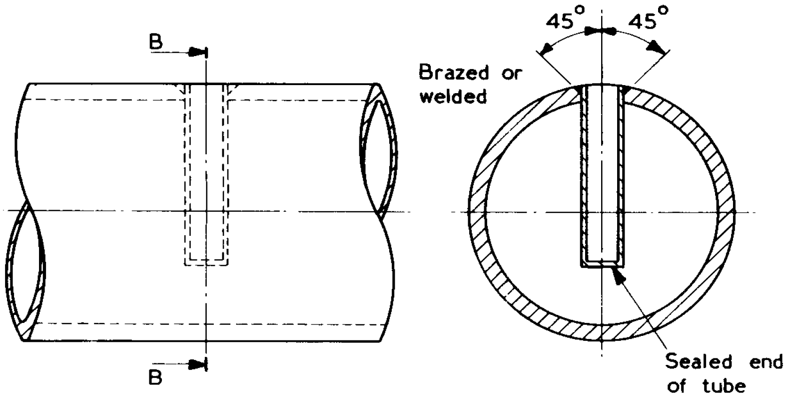


Figure 1(d) — Thermometer pocket for use in horizontal pipes over 75 mm diameter

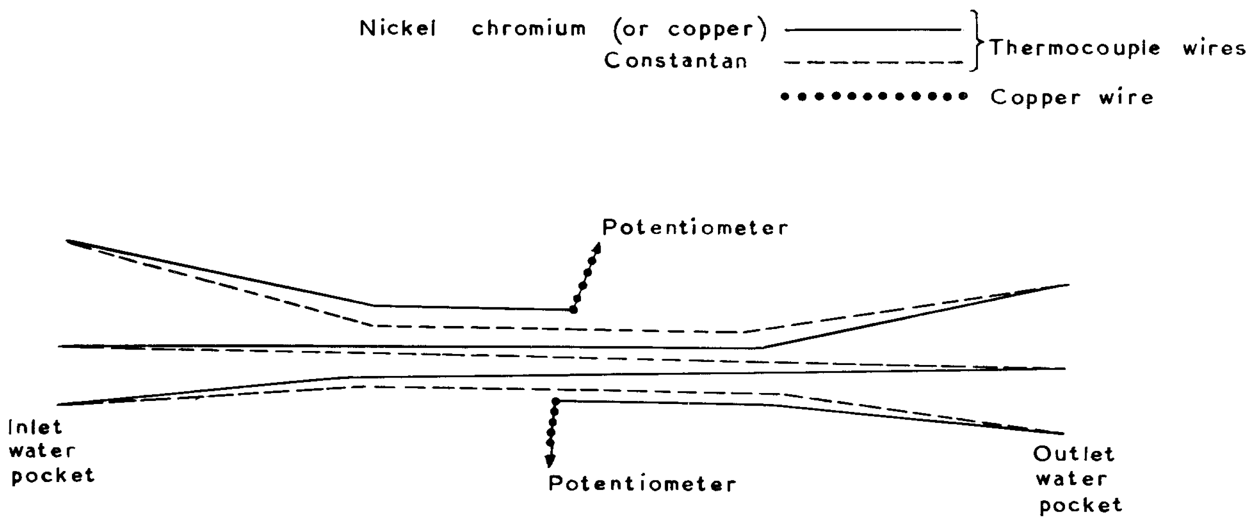


Figure 2 — Multi-junction differential thermocouple

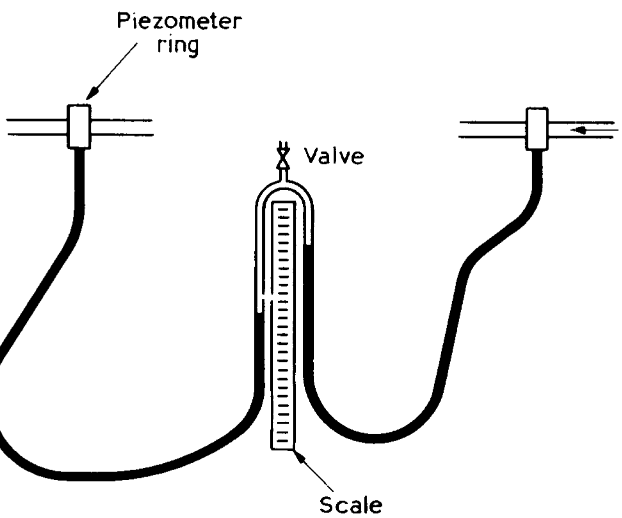
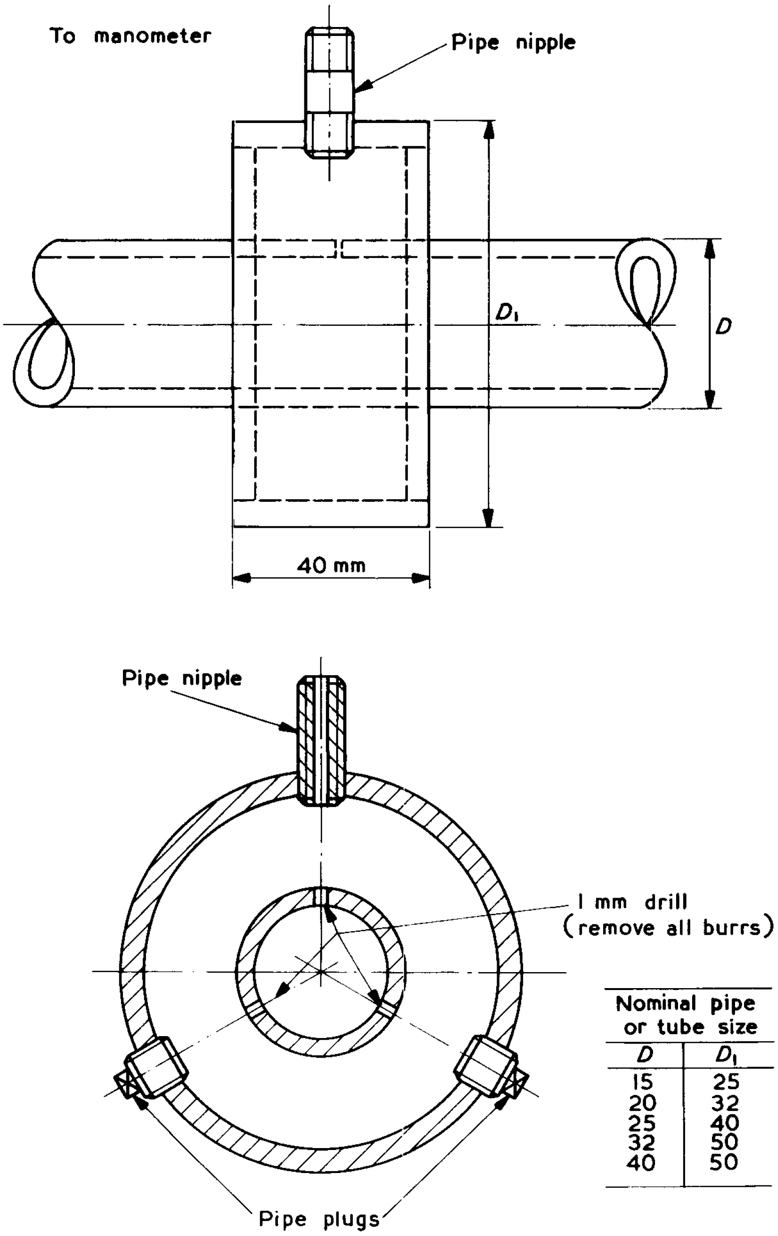


Figure 3(a) — Inverted U-tube manometer



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

Figure 3(b) — Piezometer ring

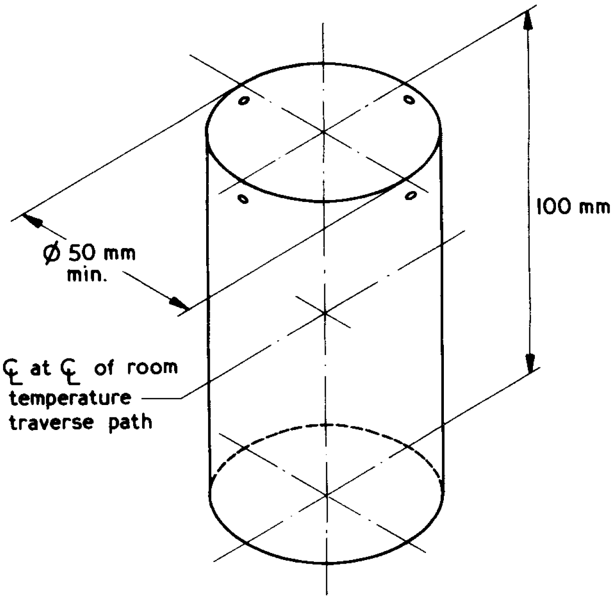


Figure 4(a) — Thermometer shield

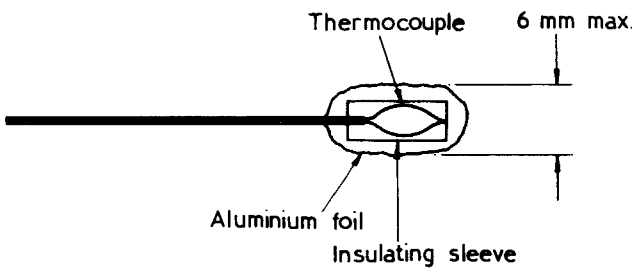


Figure 4(b) — Thermocouple shield

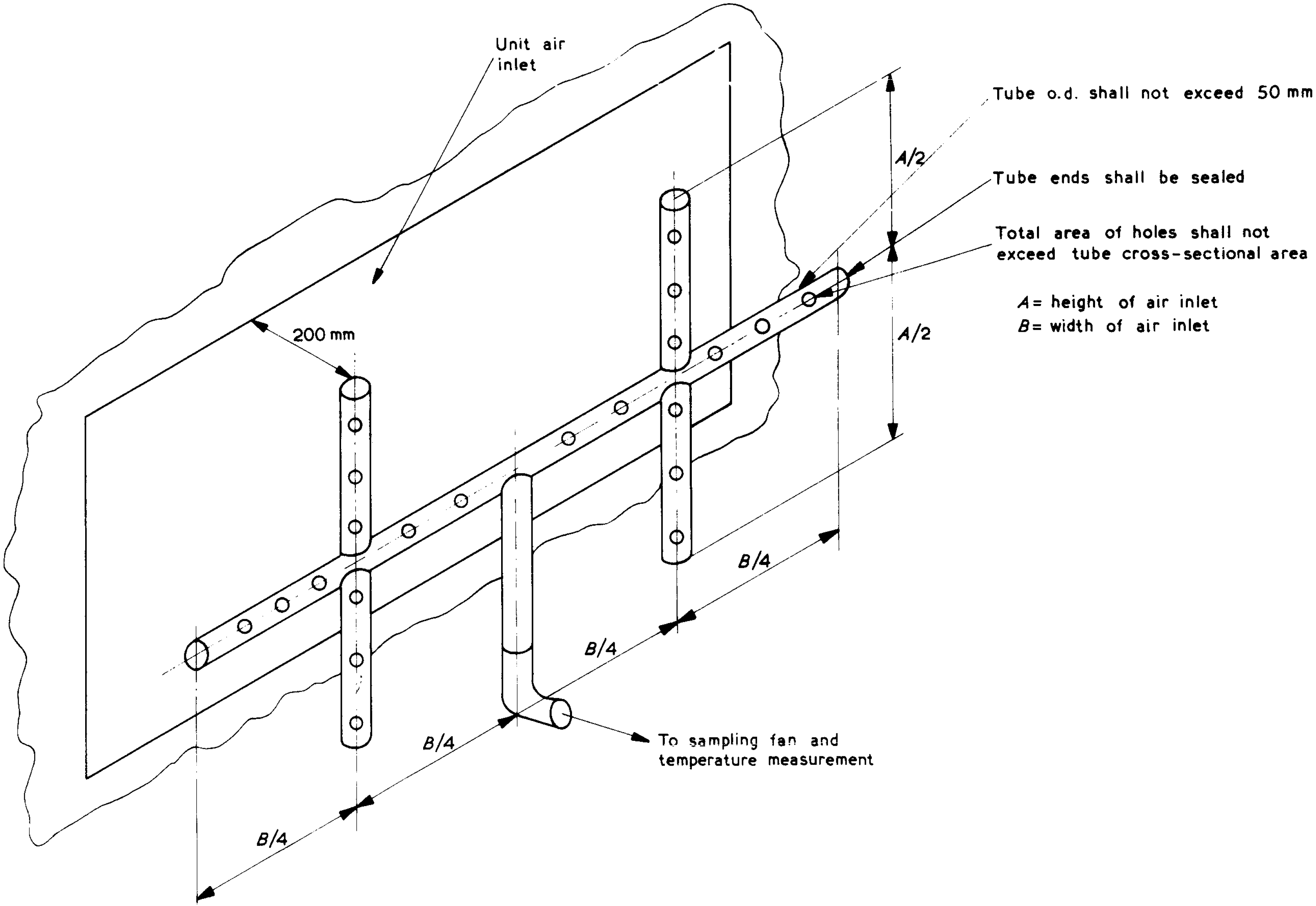


Figure 5 — Inlet air sampling tube

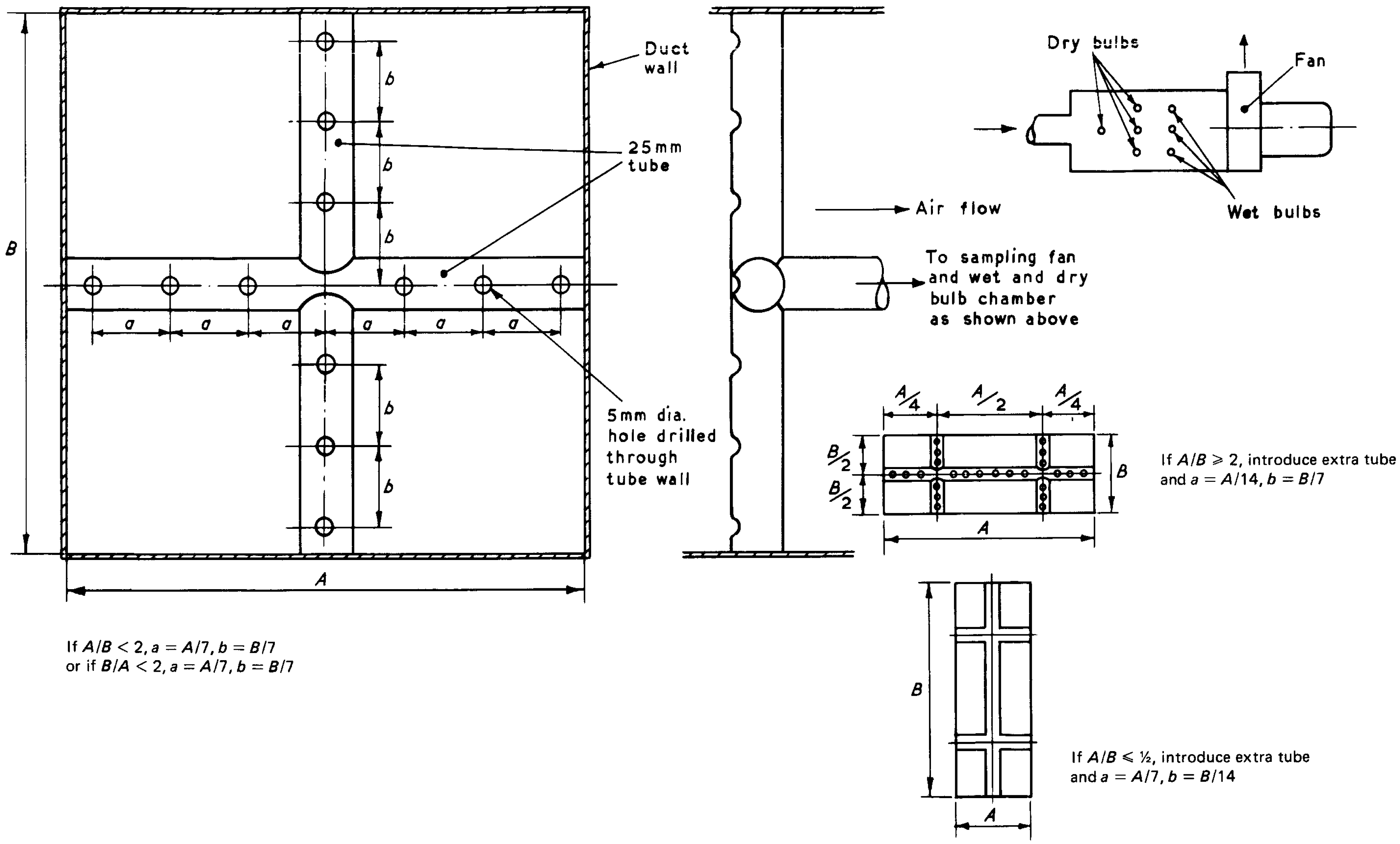


Figure 6 — Outlet air sampling tube

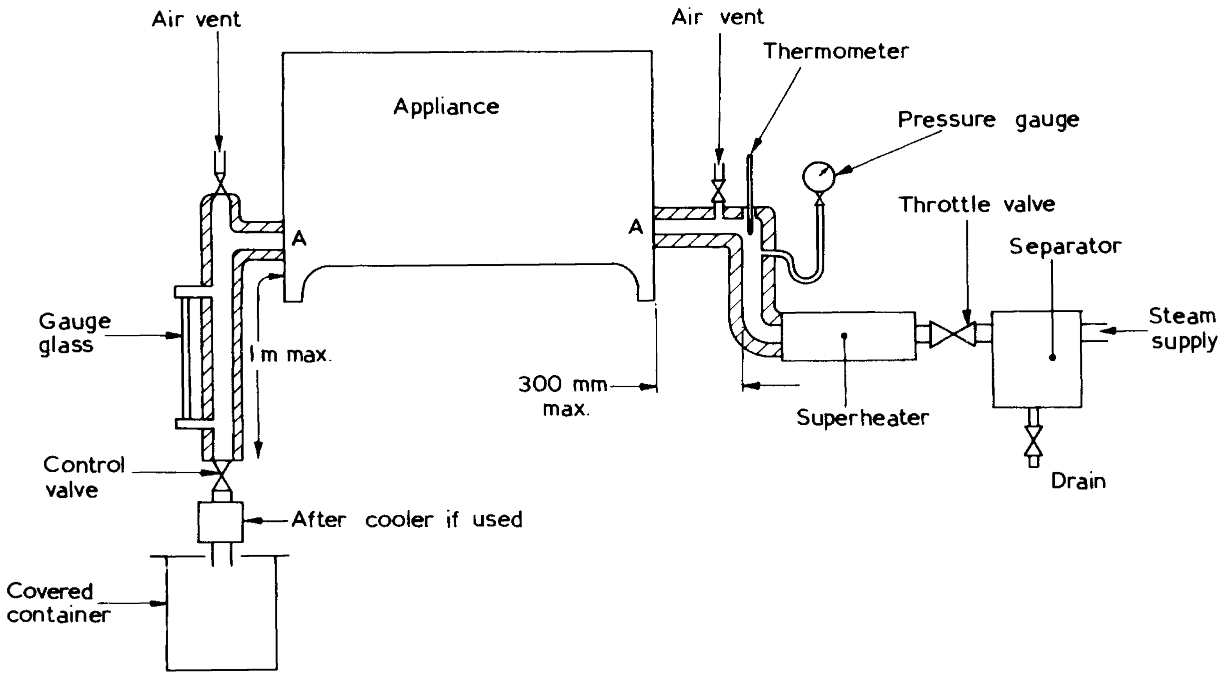
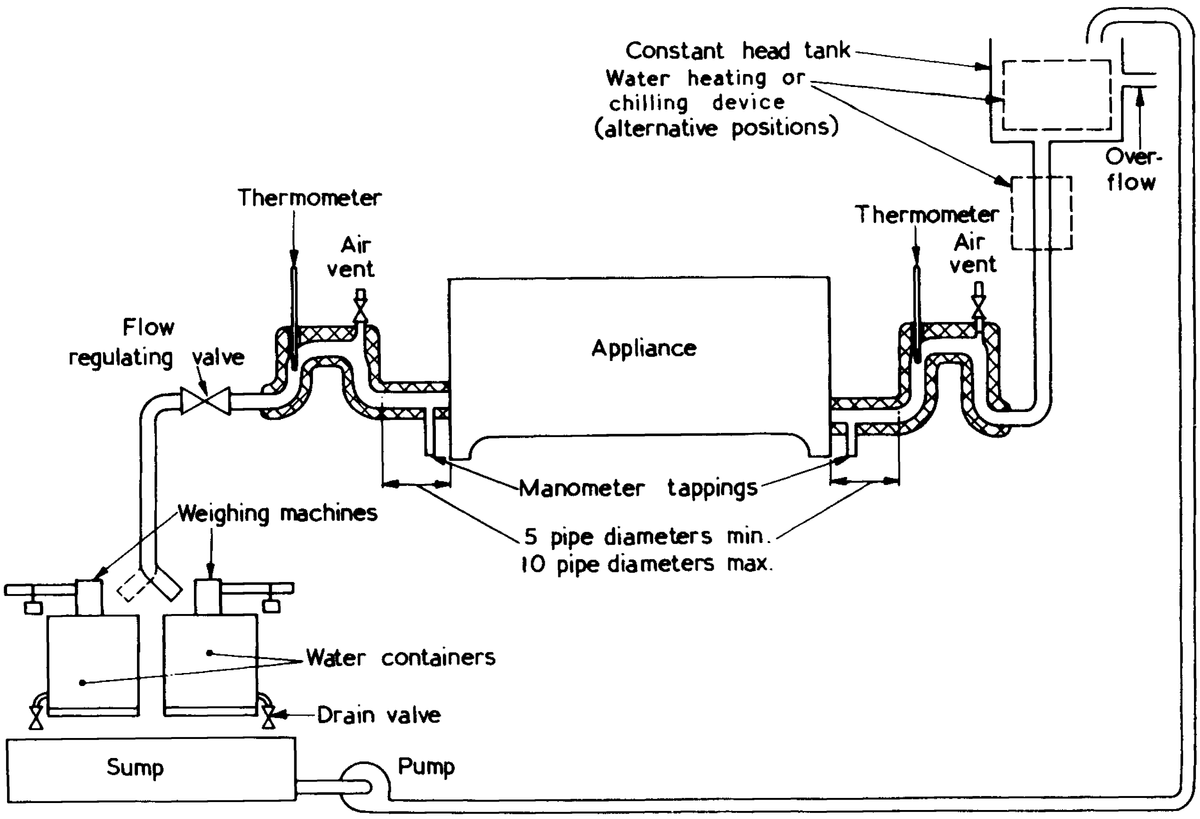
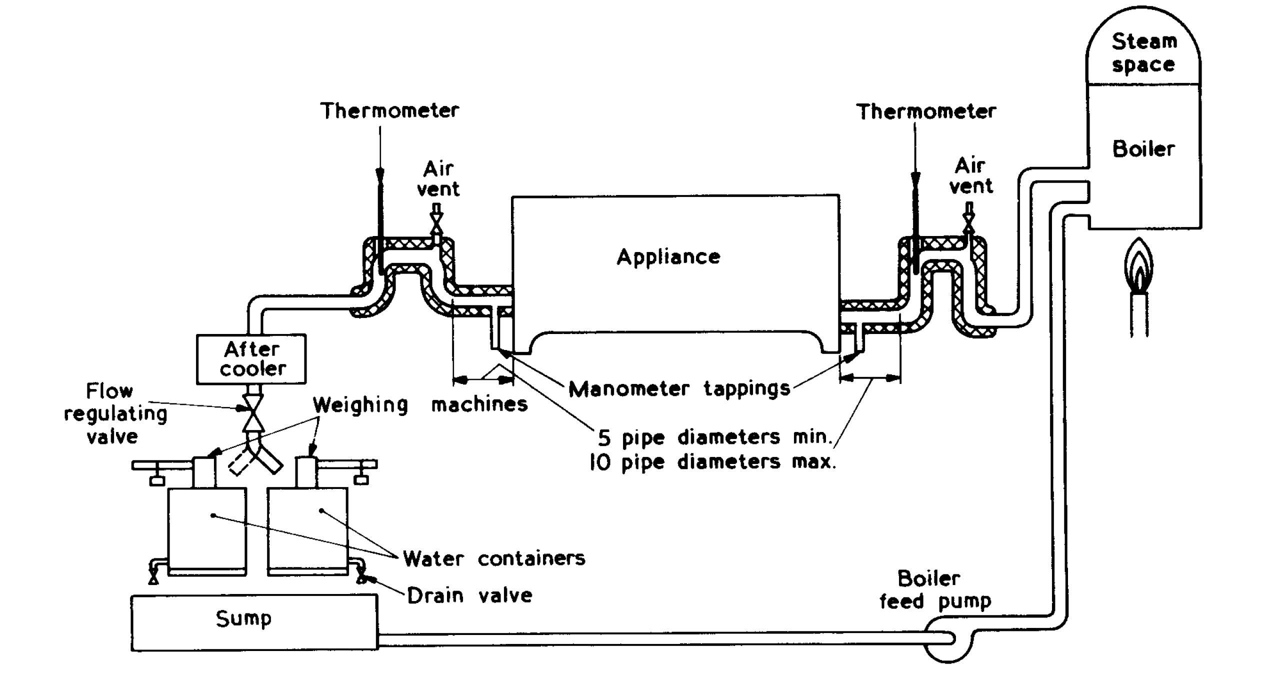


Figure 7 — Steam test equipment



NOTE Water supply arrangement shown is applicable to a maximum water temperature of 95 °C. For higher temperature, supply shall be from a boiler or other pressure vessel, and an aftercooler is necessary prior to flow regulating valve.

Figure 8 — Water test equipment



NOTE For heat transfer fluids it will be necessary to modify the test equipment by the insertion of a heat exchanger between the boiler and heat transfer fluid circuits.

Figure 9 — Hot water test equipment (maximum water temperature above 95 °C)

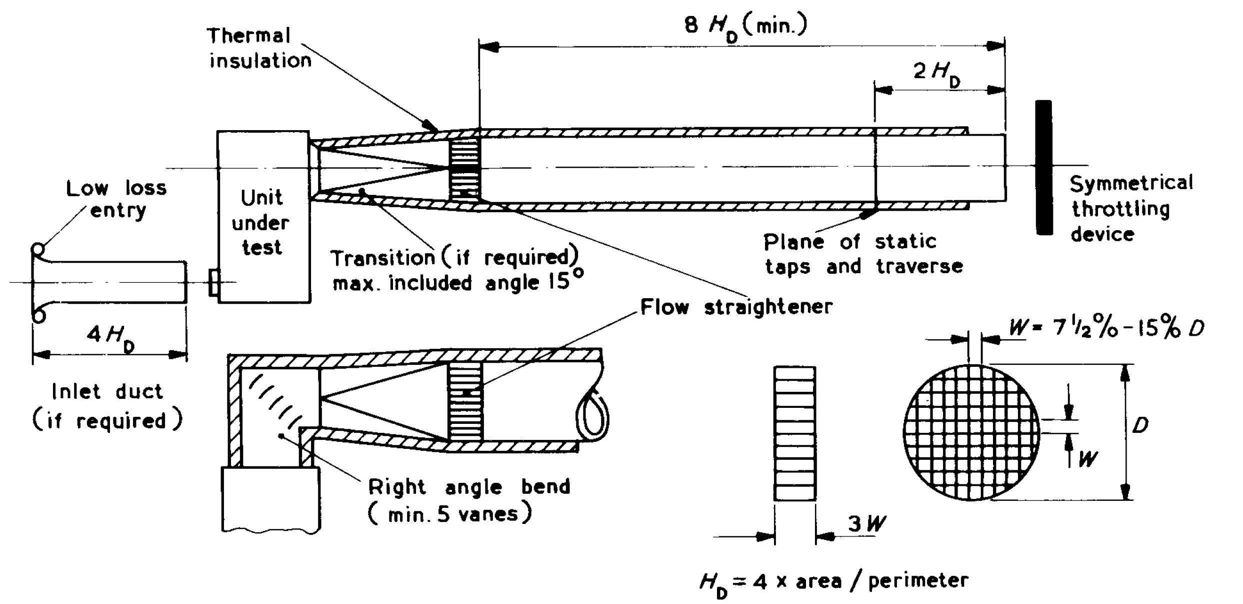


Figure 10(a) — Arrangement A of test apparatus

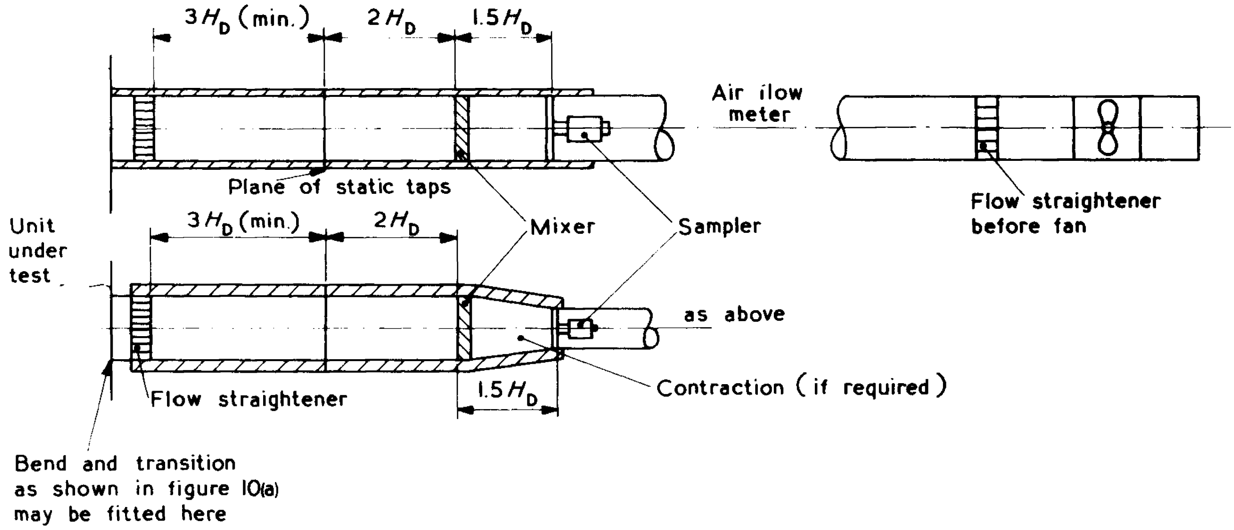


Figure 10(b) — Arrangement B of test apparatus

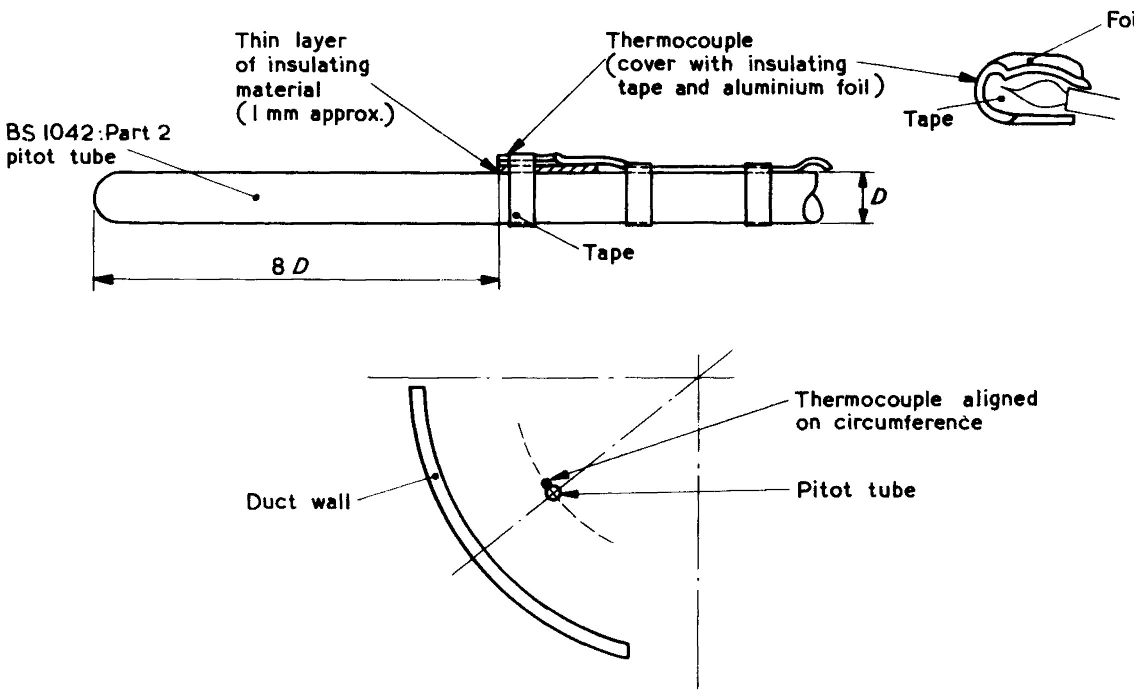
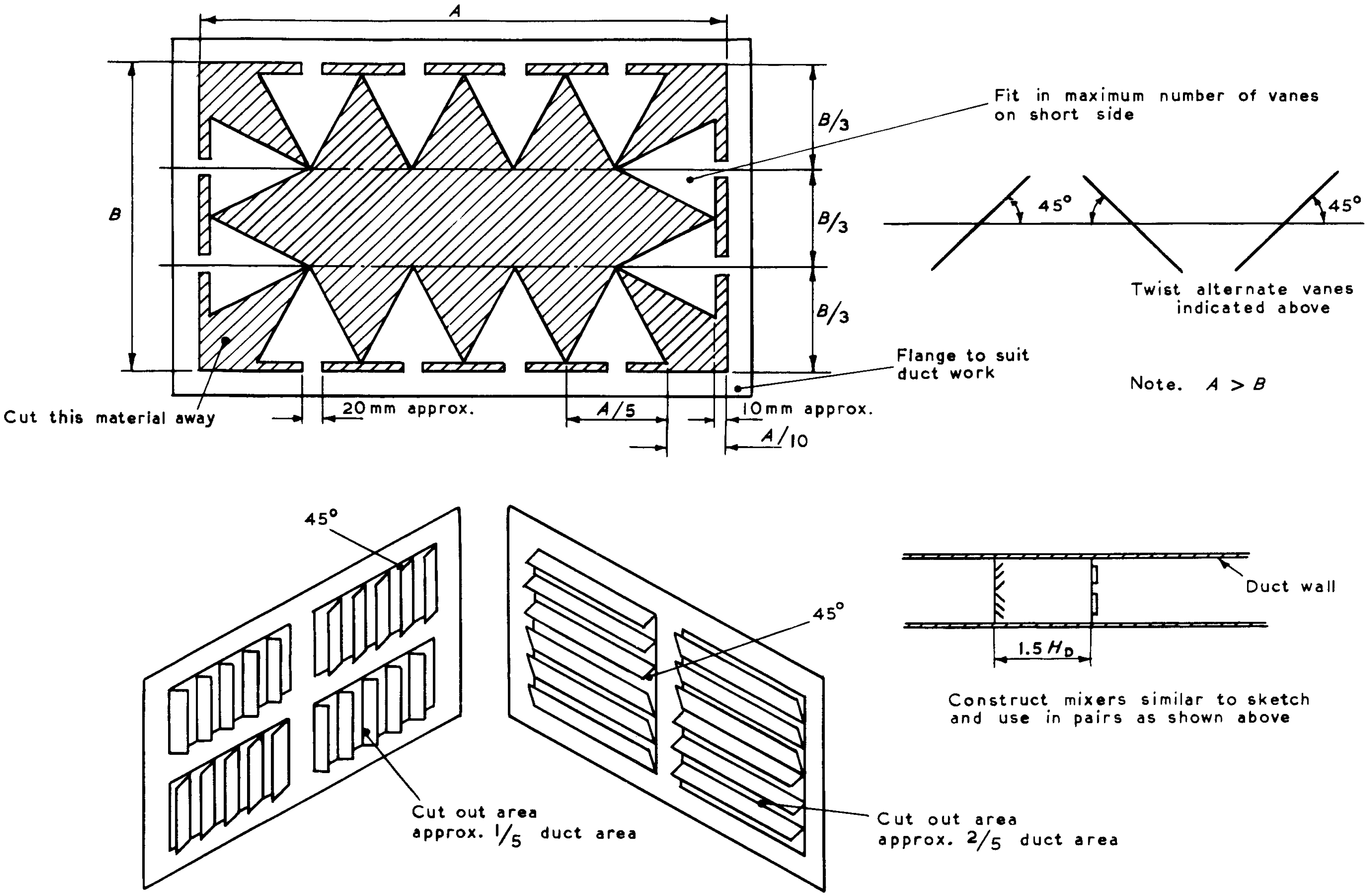


Figure 11 — Temperature and total pressure probe



NOTE 1.5 H_D space is not shown in Figure 10(a). H_D is hydraulic diameter.

Figure 12 — Air flow mixers

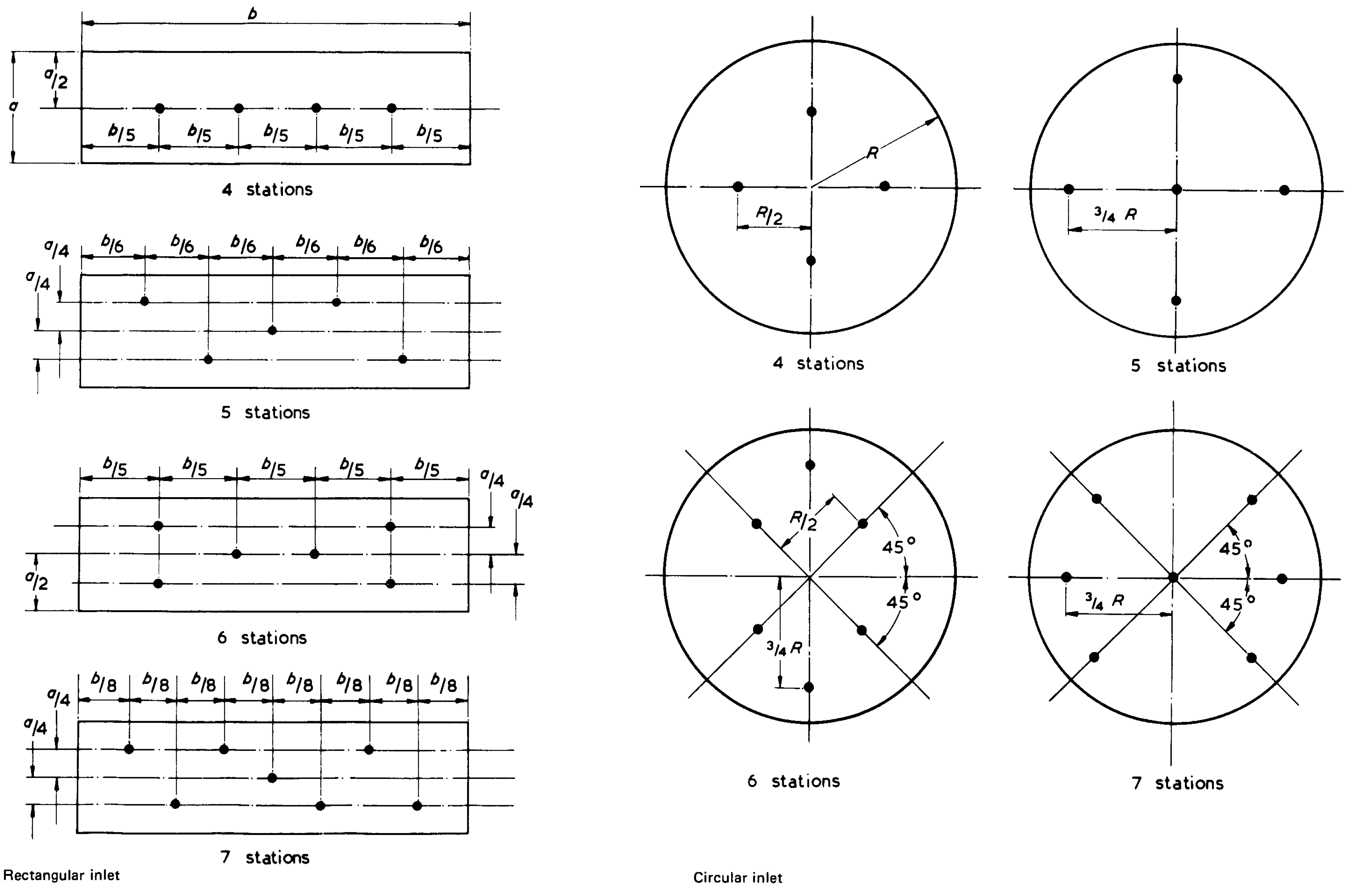


Figure 13 — Inlet temperature measurement

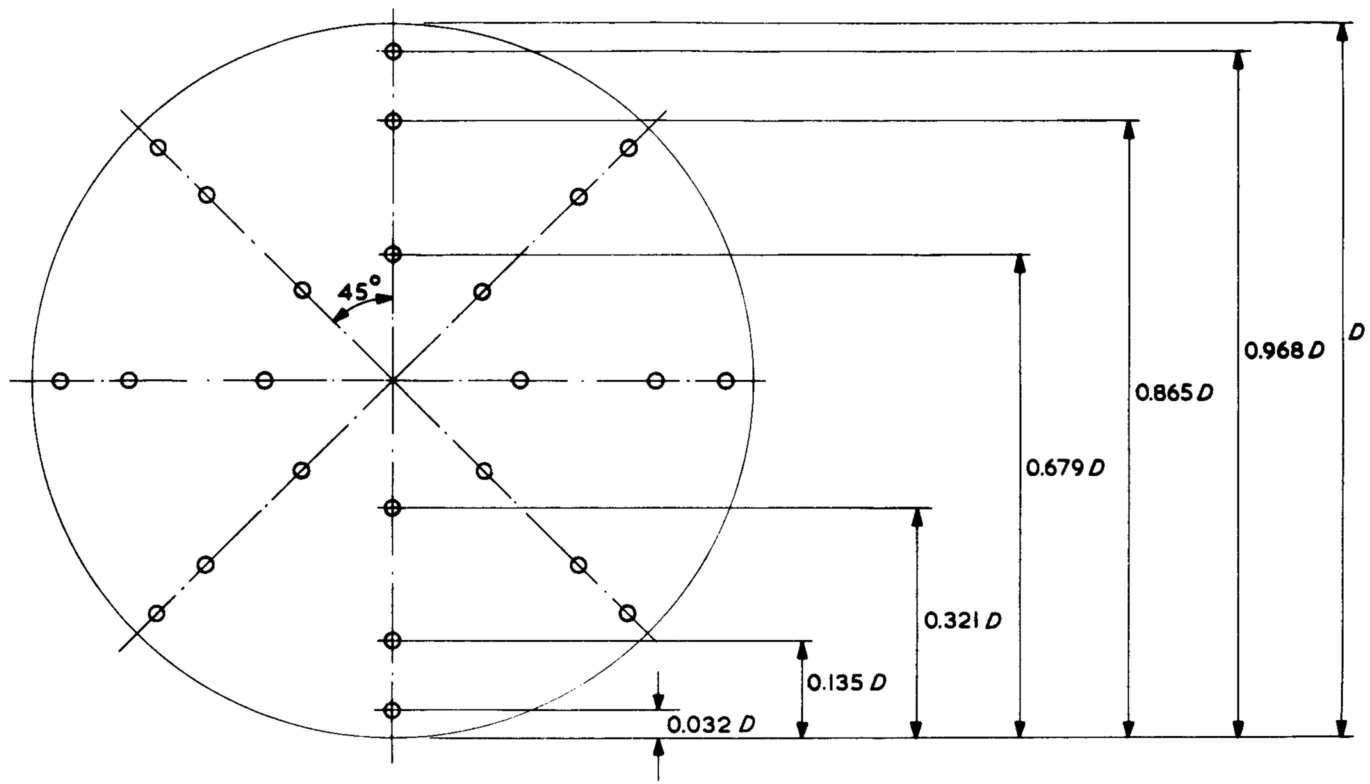


Figure 14 — Traverse stations

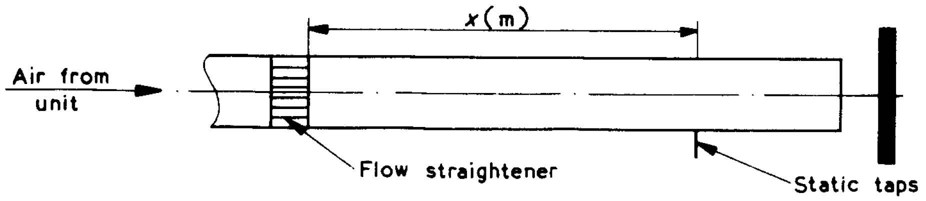


Figure 15 — Definition of x

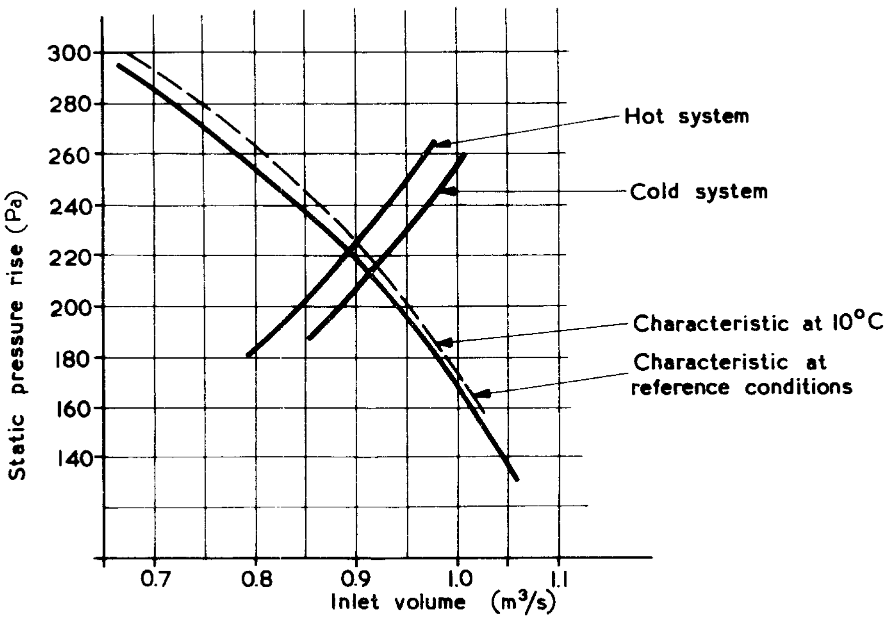


Figure 16 — Pressure-flow characteristics

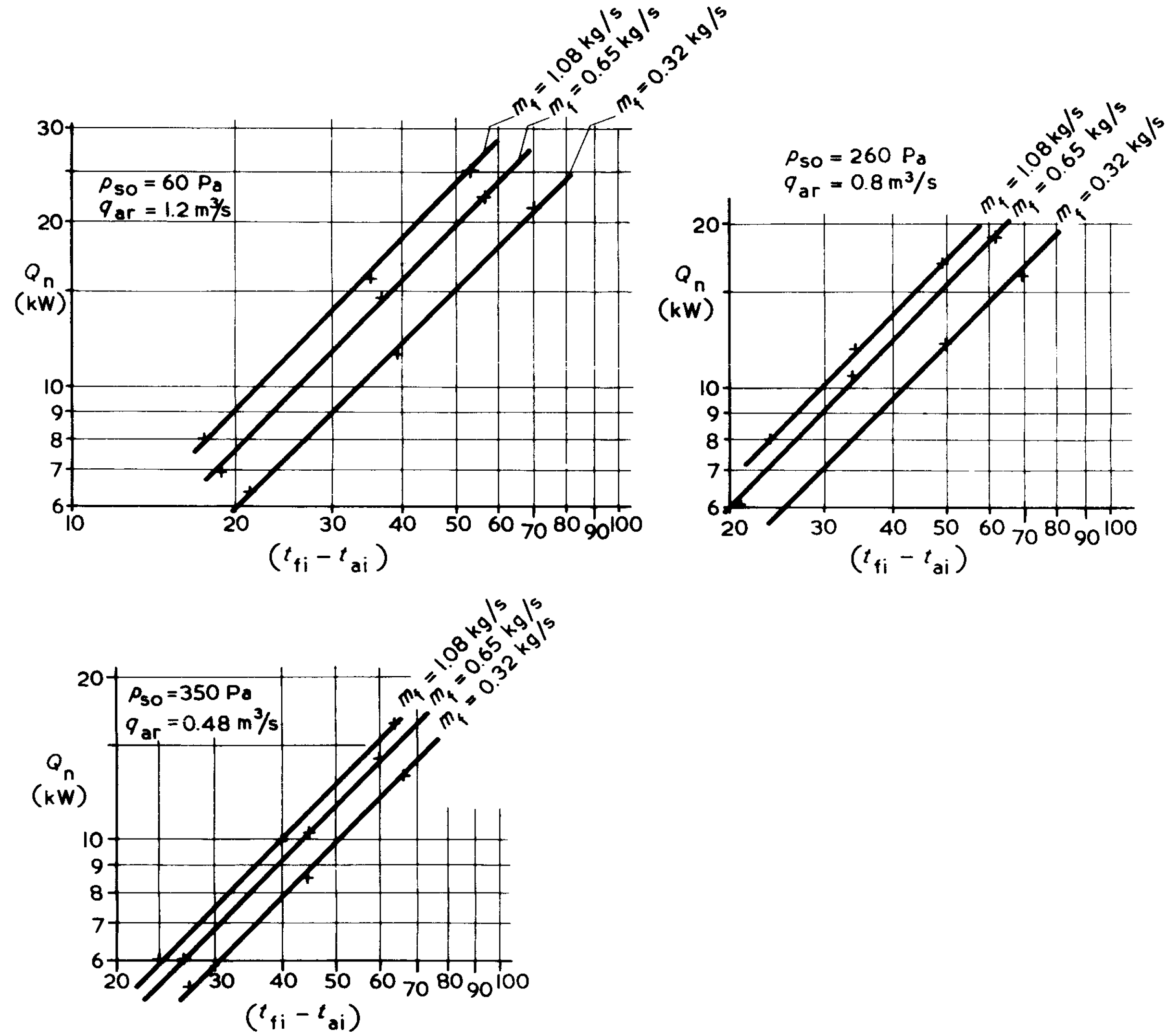


Figure 17 — Heat transfer

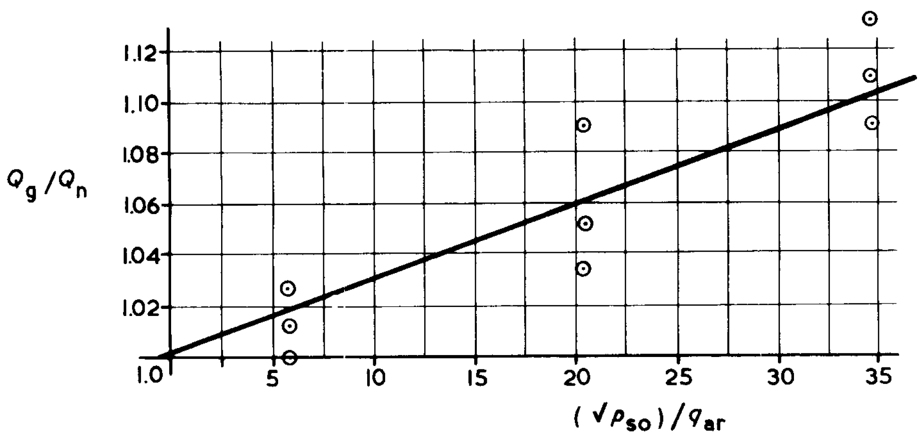


Figure 18 — Thermal efficiency

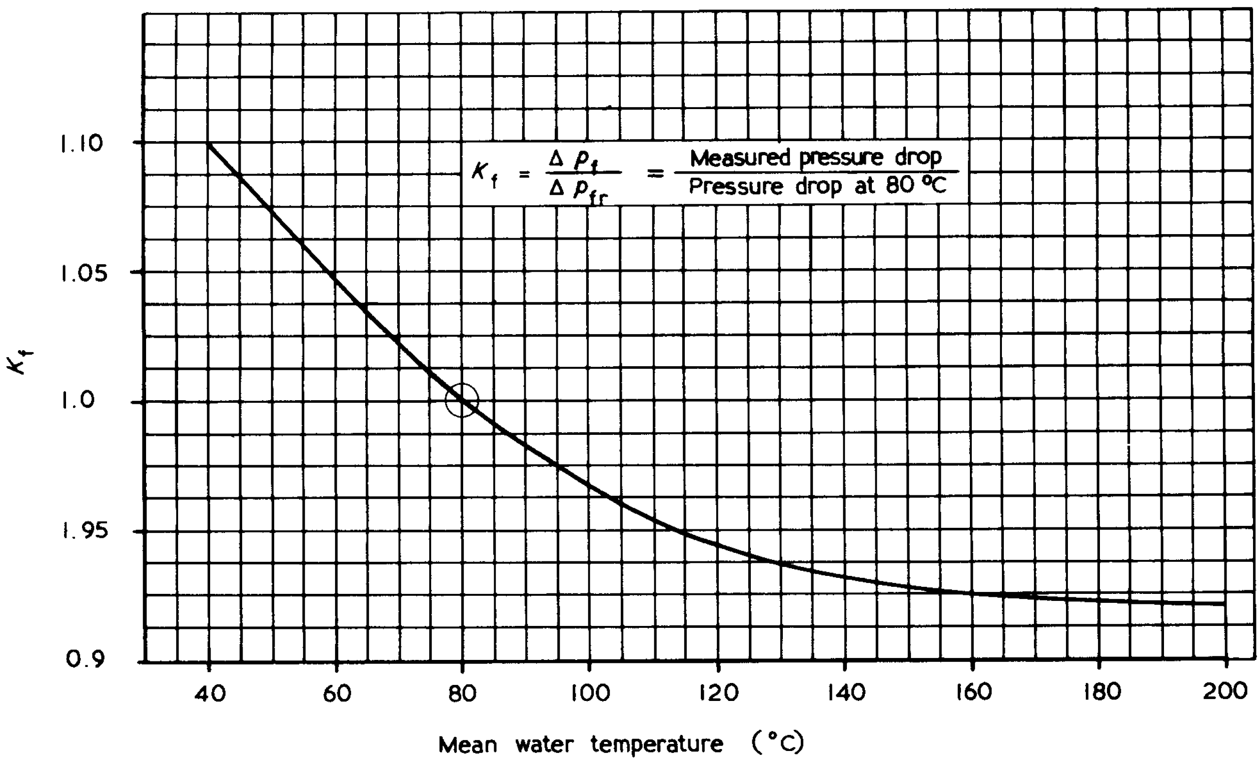


Figure 19 — Water temperature correction factor heating

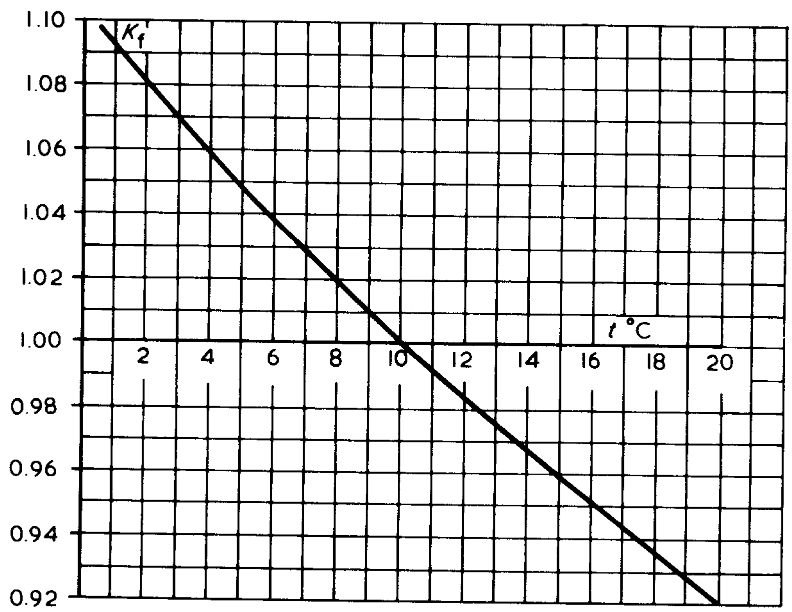


Figure 20 — Water temperature correction factor cooling

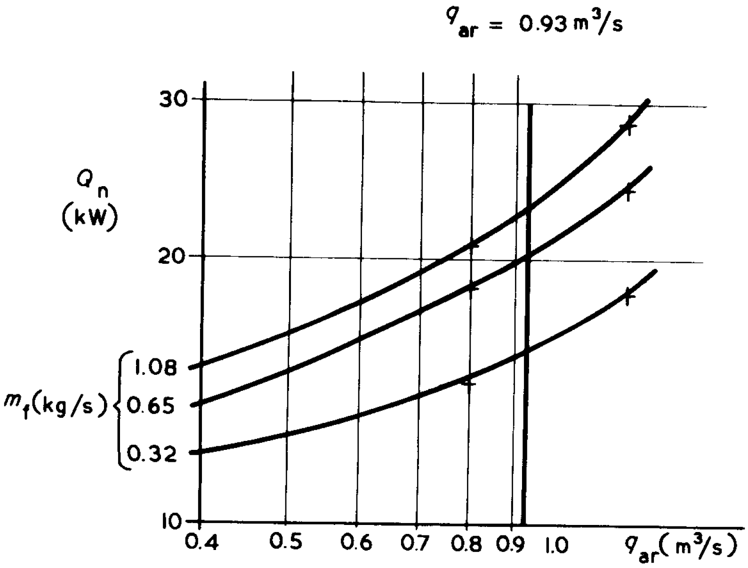


Figure 21 — Graph of Q_n versus q_{ar} at $(t_{fi} - t_{ai})$ = 60 °C

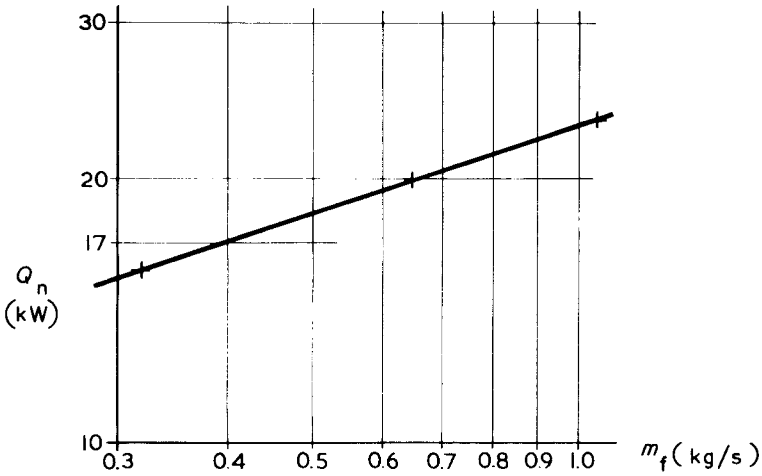


Figure 22 — Graph of Q_n versus m_f at q_{ar} = 0.93 m³/s

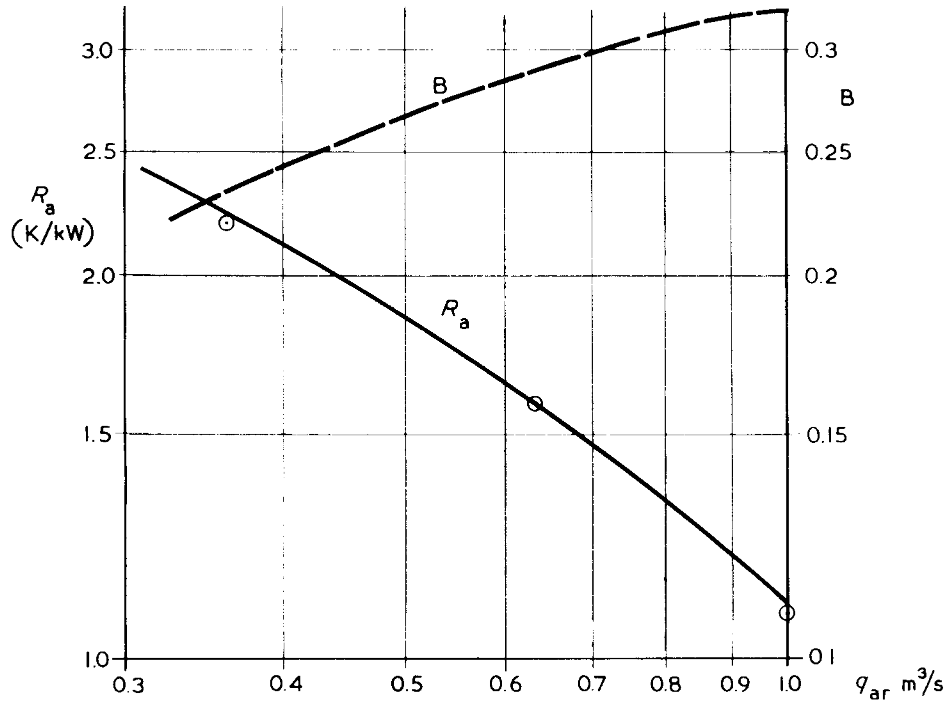


Figure 23(a) — Graph of R_a and B versus q_{ar}

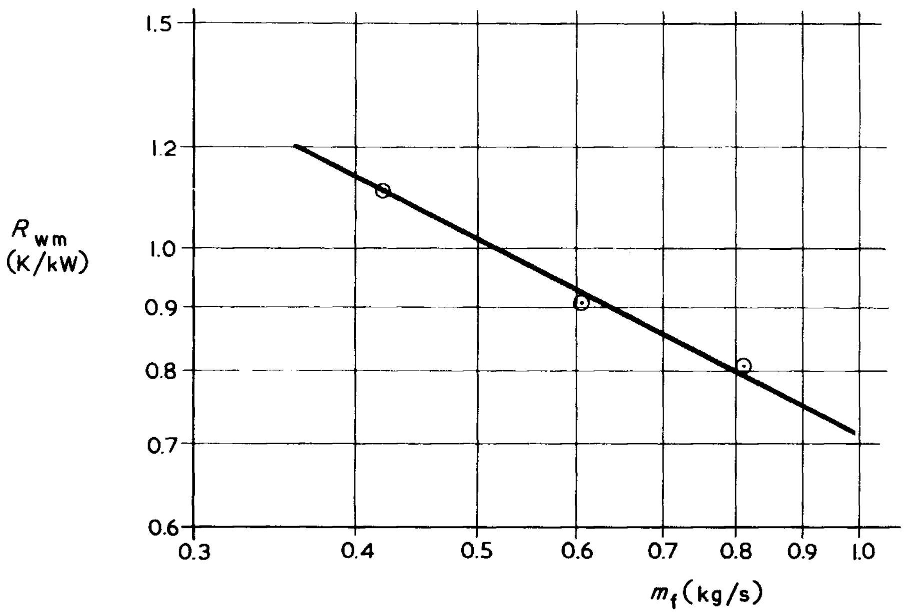


Figure 23(b) — Graph of R_{wm} versus m_f

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Publications referred to

This standard makes reference to the following British Standards:
BS 593, *Laboratory thermometers.*
BS 848, *Methods of testing fans for general purposes, including mine fans.*
BS 848-2, *Fan noise testing.*
BS 1041, *Code for temperature measurement.*
BS 1041-2, *Expansion thermometers.*
BS 1041-3, *Industrial resistance thermometry.*
BS 1041-4, *Thermocouples.*
BS 1042, *Methods for the measurement of fluid flow in pipes.*
BS 1042-1, *Orifice plates, nozzles and venturi tubes.*
BS 1042-2, *Pitot tubes.*
BS 1780, *Bourdon tube pressure and vacuum gauges.*
BS 4194, *Design requirements and testing of controlled-atmosphere laboratories.*

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