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Summary The paper provides data for heat losses from exposed pipes, plugs, thermostat cap and base of an insulated domestic hot water cylinder (capacity 120 litres), and examines their influence, and the effect of air movement, on standing heat loss performance as prescribed by standard methods such as *BS699*. Results show that under 'still air' conditions heat losses from exposed areas are 17% of the total loss. This figure is likely to increase for higher levels of insulation, and decrease for larger cylinder sizes. The experimental data compare favourably with standard theoretical models and data given in the *CIBSE Guide*. The standing heat loss is sensitive to changes in air velocity, increasing by almost 7% in the range 0.1 to 0.3 m s^{-1}, in reasonable agreement with predictions. The pipework is responsible for the majority of the heat losses from the cylinder appendages, and is shown to be particularly sensitive to air movement.

Measurements of heat losses from an insulated domestic hot water cylinder

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List of symbols

A	Cross-sectional area of pipe (m^2)
C_s	Surface conductance of insulation (W m^{-2} K^{-1})
d_i	External diameter of pipe insulation (m)
d_p	External diameter of pipe (m)
h	Heat transfer coefficient of pipe (W K^{-1} m^{-1} run)
h_i	Heat transfer coefficient of insulated pipe (W K^{-1} m^{-1} run)
q	Total heat loss from pipe (W)
t	Temperature of pipe at distance x from cylinder (°C)
t_a	Mean ambient temperature (°C)
t_c	Cylinder temperature (°C)
t_o	Temperature of pipe at distance $x = 0$ (°C)
t_s	Surface temperature of pipe (°C)
x	Distance along pipe from cylinder (m)
Nu	Nusselt number
Gr	Grashof number
Pr	Prandtl number
P	Power input (W)
Q	Standing heat loss of cylinder (W l^{-1})
ΔQ	Reduction in standing heat loss of cylinder (W l^{-1})
Δq	Reduction in heat loss of cylinder (W)
q_1	Heat loss from pipe (W)
q_2	Heat loss from pipe connections (W)
q_i	Heat conducted into insulated pipe (W)
S	Storage capacity (l)
T_s	Absolute surface temperature of pipe (K)
T_a	Absolute mean ambient temperature (K)
V	Air velocity (m s^{-1})
ε	Radiation emissivity of pipe surface
λ	Thermal conductivity (W mK)
λ_a	Thermal conductivity of air (W m^{-1} K^{-1})
ϕ_c	Convective heat exchange per unit area (W m^{-2})
ϕ_r	Radiative heat exchange per unit area (W m^{-2})

1 Introduction

Domestic energy consumption has been considerably reduced in recent years by increased levels of insulation, and in this respect hot water cylinders have played a small but important part. Hot water cylinders are insulated either at the factory, with a rigid urethane foam for example, or provided with a flexible insulating jacket. It is recommended that factory insulated cylinders have a standing heat loss not exceeding 1 W l^{-1} of capacity when measured in accordance with the methods described in *BS699*[1], *BS1566*[2], or *BS3198*[3]. For the purposes of the test as described in *BS3198* the cylinder is tested as received and the cylinder appendages such as pipework, caps, and plugs are insulated. However in the method outlined in *BS699* and *BS1566* the cylinder is also tested as received (i.e. the bottom is uninsulated) but, as is often the case in practice, with no insulation on the feed pipe, expansion pipe, thermostat cap or any plugged unused connection holes. The standards require that the test be conducted in a draught-free environment, but are not specific about restrictions on these conditions, or the means of achieving them. This paper provides data for the heat losses from the exposed cylinder appendages and examines their influence, and the effect of air movement, on the measured performance of hot water cylinders.

2 Method of determining standing heat loss

The heat loss performance tests were carried out according to *BS699: 1984*, Appendix B1 for a direct cylinder using a top entry 3 kW immersion heater. A slight deviation from *BS699* was that the thermostatic controller was used only as a safety device, to prevent accidental overheating, and not to control the temperature of the water during test. The water temperature was maintained at a steady state using a constant power source, set to give a water temperature in the range 70 ± 2°C. The whole apparatus was assembled in an enclosed environmental chamber (3 m × 3 m × 3 m), in which the ambient temperature could be controlled accurately using a three-term Eurotherm controller, to ±0.2°C in the range 20 ± 2°C. This method of controlling the water and ambient temperatures precisely allowed a greater degree of accuracy in maintaining a constant differential temperature than could be achieved by thermostatic control. It

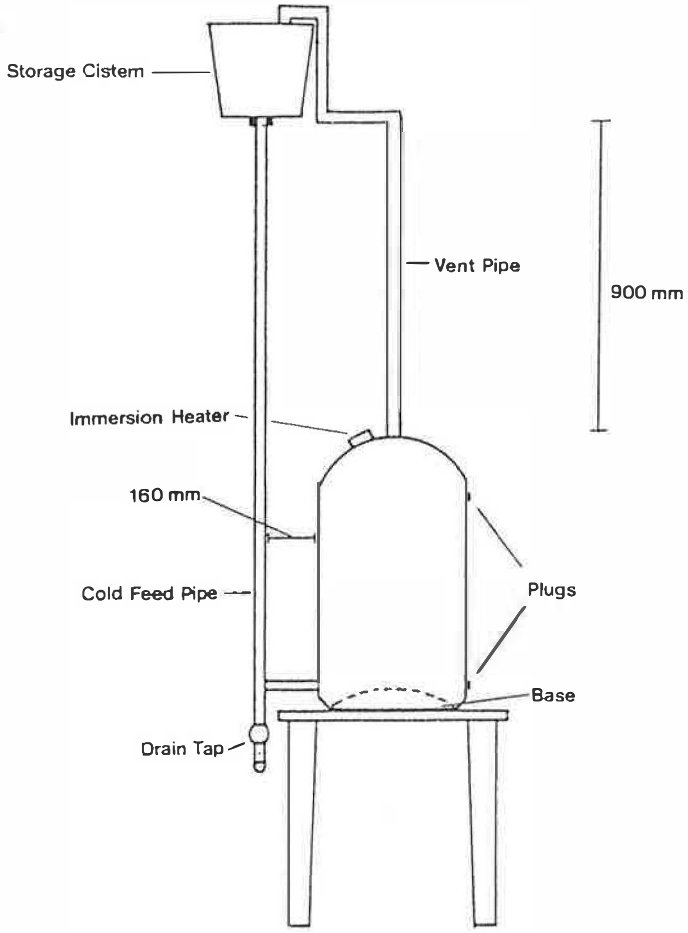

Figure 1 Apparatus for standing heat loss test

is likely that this modified technique achieves more reproducible results than the British Standard test method.

Figure 1 is a schematic diagram of the arrangement of the cold water feed and expansion pipes (22 mm copper). The walls of the environmental chamber were lined with aluminium foil (slightly tarnished) having an emissivity of about 0.2. Since very little radiation is reflected back from the surfaces of the enclosure to the relatively small cylinder appendages, the effect of the low-emissivity surfaces is considered to be negligible. The test chamber was separated from the cooling coils and air conditioner by an aluminium sheet partition to prevent forced air movement into the chamber, thereby maintaining 'still' air conditions around the cylinder. The partition also allowed sufficient heat transfer between the rooms to control the temperature of the chamber. Provision was also made to control the temperature of the test chamber directly by an air conditioning unit mounted in the room, in order also to study the effects of air movement on the cylinder.

All temperatures measurements were made using calibrated chromel/alumel thermocouples (40 μV K^{-1}). For the cylinder temperature t_c a thermocouple, set in heat-sink compound, was attached to the cleaned copper surface at a height two-thirds of the total cylinder height. The insulation removed for this purpose was replaced after the thermocouple had been attached. Ambient temperatures were measured at three positions 900 mm away from the cylinder at heights level with the top, middle, and 150 mm from the base of the cylinder. For calculation purposes the mean ambient temperature t_a was taken to be the mean of these three values. The current through the immersion heater was determined by measuring the voltage drop across a standard resistance inserted in the power circuit. The potential across the heater terminals was measured directly, and a small correction for the voltage drop along the leads applied. All equipment, including the digital voltmeter used to measure the output from the thermocouples and heater element, was calibrated in accordance with the normal procedure of the NAMAS accredited Thermal Measurements Laboratory in the department.

The measurement procedure was to apply the appropriate power and then to allow a stabilisation period of at least 24 hours, confirmed by consistent water temperature readings, before final measurements were made over the course of each subsequent 24 hour period. During each period the temperatures remained constant to within 0.2°C, and the power readings to 0.1%. The standing heat loss Q was calculated and corrected for a 50°C temperature differential, as follows:

$$Q = \frac{50P}{(t_c - t_a)S} \tag{1}$$

The test was continued until the standard heat loss as calculated by equation 1 was within 2% (in accordance with British Standards) for at least two successive 24 hour periods, and the mean value calculated for this time. In practice, the day-to-day variation was less than 1%.

3 Results

For the purposes of this experiment two copper cylinders of the same type, grade 3 reference 7 (capacity 120 litres), and different levels of insulation, 17.4 mm (A) and 20.9 mm (B) thickness of similar urethane foam, were used. Both cylinders contained two unused secondary feed connection holes, sealed with uninsulated metal plugs. The effect of insulating the external fittings and the base of cylinder A, and the effects of air movement on both A and B were studied.

3.1 Effect of insulating exposed surfaces

The standing heat loss from cylinder A was measured, under still air conditions, and the external fittings were progressively insulated. Initially, the thermostat cap was insulated with approximately 25 mm of urethane foam (λ = 0.026 W m^{-1} K^{-1}), as were the two plugs in subsequent tests. The expansion pipe followed by the cold feed pipe were then covered with a standard pipe lagging (λ = 0.04 W m^{-1} K^{-1}) of wall thickness 12.5 mm. Finally the base of the cylinder was lined with 50 mm of insulation board (λ = 0.04 W m^{-1} K^{-1}).

Results for the cylinder in each configuration are given in Table 1. The standing heat loss from the cylinder in its normal configuration is 1.101 W l^{-1}, somewhat greater than the recommended value, but this is reduced by about 11% when all the exposed surfaces are insulated. The reduction in heat loss, as each area is insulated, is also tabulated. Although the most significant reduction is from the pipes it accounts for only about 6% of the total loss from the cylinder. The thermostat cap and the base are each responsible for less than 2% of the reduction, as are the combined effects of the plugged areas.

Table 1 Standing heat losses under 'still air' conditions from cylinder A, following progressive insulation of appendages

Insulated area		Measured			Predicted Δq
Label	Components	Q (W l^{-1})	ΔQ (W l^{-1})	Δq (W)	(W)
(1)	Cylinder only	1.101			
			0.020	2.40	<3.00
(2)	Thermostat cap + (1)	1.081			
			0.005	0.60	0.84
(3)	Top plug + (2)	1.076			
			0.011	1.32	0.84
(4)	Bottom plug + (3)	1.065			
			0.032	3.84	3.72
(5)	Expansion pipe + (4)	1.033			
			0.035	4.20	4.20
(6)	Feed pipe + (5)	0.998			
			0.016	1.92	3.12
(7)	Cylinder base + (6)	0.982			

3.2 Effect of air movement

Standing heat losses from cylinders A and B were measured under 'still' air (natural convection) conditions, and also in the presence of forced convection arising from air circulated from the air conditioning system. Although the forced air speed around the cylinder was monitored at 0.3 m s^{-1}, the speed of the air being blown out of the conditioning unit into the room was about 1 m s^{-1}. The tests were carried out according to standard procedure, with the external fittings and the base uninsulated, and the results are summarised in Table 2. Air speed was measured in close proximity to the cylinder and fittings with a hot-wire anemometer and the results given in the table are approximate values for all directions. Under 'still' air conditions the speed was not accurately measurable as it was significantly less than 0.1 m s^{-1}, which represents the best resolution of the anemometer.

For both cylinders the standing heat loss increased significantly in the presence of forced air movement, the change being 5.4% for A and 7.9% for B. If it is assumed that heat loss from the cylinder appendages is the same for both cylinders then the effect of air movement on B is likely to be greater, since it is more highly insulated than A. It is possible that the differences in air speed may have been slightly different between the cases, and this may not have been detected due to the limited resolution of the anemometer.

4 Theoretical and empirical equations

The heat loss from the pipes was calculated using conventional equations for the temperature distribution in a long uninsulated rod. The cold end of each copper pipe is assumed to terminate at ambient temperature and this was verified by measurement. It can be shown that temperature t at any point x in the rod is given by

$$t - t_a = (t_o - t_a)\exp - (h/\lambda A)^{1/2}x \qquad (2)$$

where λ is the thermal conductivity of the pipe.

The total heat loss from the pipe is the same as the heat conducted into the pipe at the cylinder at $x = 0$

$$q = -\lambda A\frac{dt}{dx} = (t_o - t_a)(h\lambda A)^{1/2} \qquad (3)$$

The heat transfer coefficient for the pipe h is calculated from the convective ϕ_c and radiative ϕ_r heat exchange per unit area.

For horizontal bare pipes (*CIBSE Guide*[4]):

$$\phi_c = (\lambda_a/d_p)\,\mathrm{Nu}(t_s - t_a)$$

$$\mathrm{Nu} = 0.53(\mathrm{Gr\,Pr})^{0.25}$$

giving, under ambient conditions

$$\phi_c = 1.35[(t_s - t_a)/d_p]^{1/4}(t_s - t_a) \qquad (4)$$

The radiative component is

$$\phi_r = 5.67 \times 10^{-8}\epsilon(T_s^4 - T_a^4) \qquad (5)$$

The radiant temperature of the surrounding enclosure is assumed to be the same as the mean ambient temperature.

The heat transfer coefficient for the pipe is then given by

$$h = \frac{\pi d_p(\phi_c + \phi_r)}{t_s - t_a} \qquad (6)$$

For vertical pipes equation 6 must be multiplied by a factor of 0.8 for pipes of diameter $d_p = 22$ mm[4]. Taking t_s to be the mean surface temperature of the pipe, i.e. 45°C, $t_a =$ 20°C and the surface emissivity of oxidised copper as 0.6[4], then $\phi_c = 195.95$ W m^{-2}, $\phi_r = 97.15$ W m^{-2}, and $h =$ 0.648 W m^{-1} run K^{-1}.

The effect of changing air speed on the convective component can be determined from results for cylinders of different diameters in air (see Fishenden and Saunders[6] for summary). For velocities in the range $V = 0.03$ to 2.7 m s^{-1}, across 22 mm pipes in ambient air[6]:

$$\phi_c \propto V^{0.466} \qquad (7)$$

The heat transfer coefficient for the insulated pipes is given by[4]

$$h_i = \frac{\pi d_p}{(d_p/2\lambda_i)\ln(d_i/d_p) + (d_p/C_s d_i)} \qquad (8)$$

5 Calculations and discussion

The heat conducted into the uninsulated pipe at the cylinder is given by equation 3. The value of the product λA is an effective value for a water-filled pipe. Although the thickness of the copper walls is only 1 mm, conduction through the metal is the predominant component since the thermal conductivity of copper is 200 W m^{-1} K^{-1} compared to 0.63 W m^{-1} K^{-1} for water. The sum of products of the thermal conductivity and area for the copper walls and water is $\lambda A = 0.014$ W m^{-1} K^{-1}. Taking t_o as being the same as the cylinder wall temperature, 70°C, then $t_o - t_a = 50$°C and the heat lost from the pipe from equation 3 is $q_1 = 4.8$ W.

Table 2 The effect of air speed on the standing heat loss from cylinders A and B with uninsulated appendages

Cylinder	Air speed m s^{-1}	Standing heat loss (W l^{-1})
A	<0.1	1.101
	0.3	1.160
B	<0.1	0.961
	0.3	1.037

This value is unlikely to account for all the heat lost at this point as there are connection fittings approximately 33 mm in diameter and 35 mm long between the pipe and cylinder. The effective area of these connections is about 0.004 m^2. Taking $t_s = t_o = 70°C$ at this point the value of the conductance $h/\pi d_p$ calculated from equation 6 is 12.8 W m^{-2} K^{-1} for the horizontal cold feed pipe fittings, and 10.2 W m^{-2} K^{-1} for those on the vertical expansion pipe. Heat losses from the connections are therefore $q_2 =$ 2.52 W from the cold feed and $q_2 = 2.04$ W from the expansion pipe.

The heat conducted into the insulated pipe can be calculated from equations 3 and 8; for $d_i = 47$ mm, $\lambda_i =$ 0.04 W m^{-1} K^{-1} and $C_s = 10.0$ W m^{-2} K^{-1} giving $q_1 =$ 3.12 W. The reduction in heat loss Δq resulting from insulating the pipes is simply $q = q_1 + q_2 - q_i$.

As shown in Table 1, the agreement between this calculation method and the measurements is surprisingly good considering that the analysis of results is very indirect, being derived from small differences. Although care was taken to ensure that the added insulation was defined as accurately as possible, an uncertainty of up to 10% could reasonably be applied to q_i to allow for imperfect lagging. It should also be noted that variations in the surface temperature of the copper cylinder have not been allowed for, and it is assumed that the values of T_o at both outlets is the same as that measured at a height two-thirds of the total cylinder height, i.e. 70°C.

The diameter of the uninsulated plug areas is 45 mm and the conductance from a high-emissivity vertical surface is given by Reference 4 as 11.8 W m^{-2} K^{-1}, reducing to 0.9 W m^{-2} K^{-1} when insulated. Consequently the change in heat loss Δq for a 50°C temperature differential is 0.84 W, which also corresponds closely to measured values.

Estimation of heat losses from the thermostat cap is a complex problem because the cap is ventilated and there are electrical terminals inside. An approximate value is given by considering only the area at the base of the cap, 80 mm in diameter, and a heat emission of 12.8 W m^{-2} K^{-1} for a horizontal surface looking up[4]. This would represent the maximum possible reduction in heat loss, after insulating, which is calculated to be $\Delta q = 3.0$ W.

The heat loss from the four electrical leads, each comprising 50 strands 0.25 mm diameter copper wire sleeved in 4.2 mm PVC sheathing, is estimated to be only 0.48 W.

The base of the cylinder represents the largest uninsulated area. However, the heat losses are restricted as the rim of the concave surface is mounted on a 50 mm thick timber plinth, resulting in an airspace. The resistance of this combined with that of the plinth and the surface resistance beneath gives an estimated conductance of 1.1 W m^{-2} K^{-1}[5]. With insulation applied to the underside the conductance reduces to 0.5 W m^{-2} K^{-1} giving, for a total area of 0.105 m^2, a change in heat loss of 3.12 W. The measured value $q =$ 1.92 W suggests that the surface temperature at the base of the cylinder is somewhat lower than 70°C. The temperature distribution in the water is such that the bottom is the coldest part of the cylinder. Furthermore, applying insulation to the bottom can be expected to change local temperatures since that part of the cylinder below the heating element derives its heat by conduction through the water and copper surfaces.

The actual heat loss from the insulated cylinder alone, after allowing for the losses from the insulation on the fittings as well as those from the electrical leads, is 109.8 W. Figure 2 gives the dimensions of the copper cylinder and after allowance is made for the area of the fittings, and the base, the external surface area of the insulated tank is estimated to be 1.45 m^2. For a 50°C temperature differential, and given that the outside surface heat transfer coefficient is 10.0 W m^{-2} K^{-1}[4], the thermal conductivity of the foam (thickness 17.4 mm) is estimated to be 0.031 W m^{-1} K^{-1}. Published values[5] suggest that 0.028 W m^{-1} K^{-1} is likely to be the maximum value for aged urethane foams at an elevated mean temperature of 45°C. Although elevated temperatures can accelerate the ageing process, it is unlikely that the foam was fully aged as it was less than 6 months old. A similar calculation carried out for cylinder B with a foam thickness of 20.9 mm, by assuming the same appendage heat losses as those obtained for cylinder A, yields a conductivity of 0.030 W m^{-1} K^{-1}. The calculation method gives consistent values for the conductivity of the foam, which is nominally the same on both cylinders. However, the analysis is undermined by the assumption that all parts of the cylinder are at 70°C, i.e. only one measurement position is used which is assumed to be representative of the whole. For a more reliable measure of the conductivity of the urethane insulation an accurate determination of the temperature distribution, thickness and area is needed. It should also be noted that there are difficulties in defining a

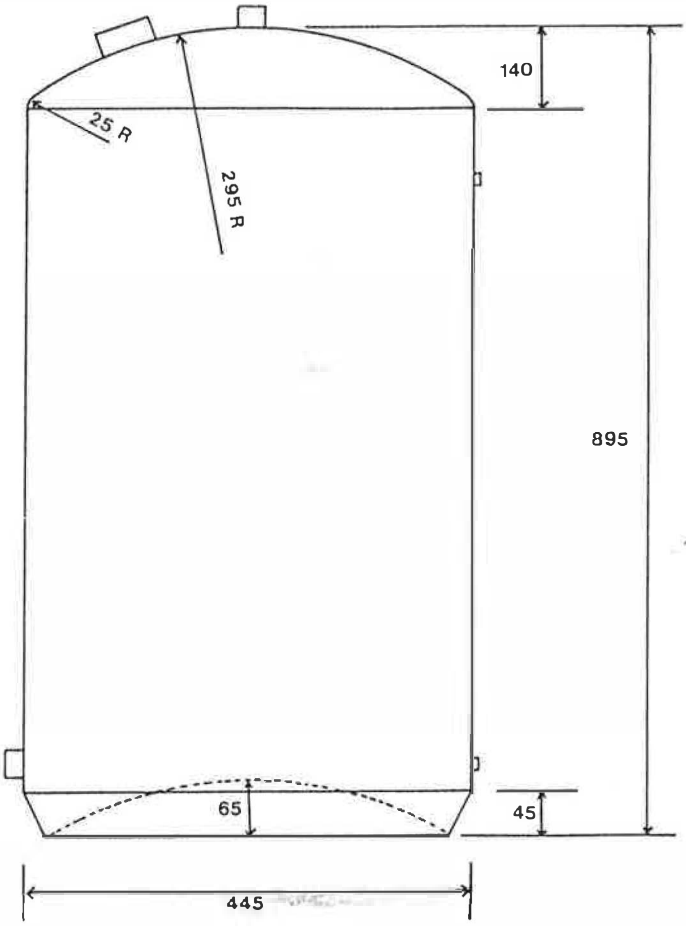

Figure 2 Dimensions (mm) of copper cylinder

Table 3 Comparison between 'measured' heat losses under 'still air' conditions and those predicted at an air speed of 0.3 m s^{-1} for cylinder A (total measured heat loss at 0.3 m s^{-1} = 139.2 W)

Location	Heat loss (W)	
	'Measured' in 'still air'	Predicted at air speed 0.3 m s^{-1}
Insulated cylinder	109.8	110.8
Thermostat cap	2.6	2.9
Top plug	0.7	0.7
Bottom plug	1.5	1.6
Expansion pipe	6.8	8.8
Feed pipe	7.3	9.2
Cylinder base	2.9	2.9
Electrical leads	0.5	0.5
Totals	132.1	137.4

true mean thickness, as the depth of foam on the cylinder can vary by up to 25%.

5.1 Heat losses from cylinder appendages and effect of air movement

The heat losses in still air from the appendages of cylinder A can be determined indirectly using the measured value of Δq from Table 1, and adjusting for the estimated heat loss when insulated. These are given in Table 3 as 'measured' values. The error associated with the added insulation, when compared with the total loss in each area, is sufficiently small to give credence to the results. In general the losses from the pipework and plugs are confirmed by prediction, and might reasonably be applied to any size of cylinder operating at 70°C. It should be stressed that the effect of air movement needs to be considered.

Tables 13, 16 and 19 in the *CIBSE Guide*[4] give correction factors for surface heat transfer in moving air, and these suggest that the following correction factors may be applied for an air speed of 0.3 m s^{-1}:

(*a*) 1.18 to convective heat transfer from plane surfaces

(*b*) 1.06 to the outside surface heat transfer coefficient of insulated surfaces.

The factor 1.03 assigned to pipes appears to grossly underestimate the effect, as equation 7 indicates that a change in V from 0.1 to 0.3 m s^{-1} would increase the heat transfer coefficient h by a factor 1.45. This larger value has been used along with the above factors to predict the heat losses at 0.3 m s^{-1} given in Table 3.

The predicted increase of 4% in the total heat loss is somewhat less than the 5.4% measured for cylinder A, and the effect of air moving over the pipework appears to be responsible for most of the increase. Somewhat larger correction factors would need to be invoked to account fully for the effect of air movement, especially in the case of cylinder B for which the heat loss increased by 7.9%. The pipes are clearly sensitive to air speed, and the cylinder results could be more readily explained if there had been a slightly larger change than the threefold increase in V, used above. This would be a reasonable assumption to make, considering the large measurement errors associated with the anemometer at low air speeds.

6 Conclusion

It has been shown that the total heat losses from the uninsulated pipes, thermostat, plugs and base of an insulated direct hot water cylinder, tested to *BS699*, are significant, and can in general be predicted fairly accurately using published data (e.g. *CIBSE Guide*[4]). These losses were found to be 17% of the measured standing heat loss (1.101 W l^{-1}) from an insulated cylinder of water capacity 120 l, and the majority of the loss was from the pipework.

Results show that the standing heat loss is sensitive to air movement around the cylinder and associated fittings, and of these the pipework is predicted to be most sensitive. For an air speed of 0.3 m s^{-1} the standing heat loss was found to be about 7% greater than that under 'still' conditions, where the speed was approximately 0.1 m s^{-1}. After allowing for uncertainty in the measurement of low air speeds, the changes can be accounted for by published empirical formulae and correction factors for air movement.

The losses from the pipework and plugs agree well with prediction, and might reasonably be applied to any size of cylinder operating at 70°C. As a proportion of the total, the losses would be less in larger cylinders. It should be stressed that the effect of air movement needs to be given careful consideration, especially in the case of highly insulated cylinders where the relative effect of heat losses from the cylinder appendages is greater. The standard requirement for conditions to be draught free is vindicated, but there is a case for specifying this more closely, as the air speed in nominally 'still' environments of different test laboratories may differ. Since the exposed areas are a required part of the standard test, it is also important that these are carefully defined and do not extend beyond the minimum necessary.

References

1 *Copper direct cylinders for domestic purposes* BS699: 1984 (Milton Keynes: British Standards Institution) (1984)

2 *Copper indirect cylinders for domestic purposes* BS1566: 1984 (Milton Keynes: British Standards Institution) (1984)

3 *Copper hot water storage combination units for domestic purposes* BS3198: 1981 (Milton Keynes: British Standards Institution) (1981)

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6 Fishenden M and Saunders O A *An Introduction to Heat Transfer* (Oxford: Oxford University Press) (1950)