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Summary A method is described for measuring the moisture content of building materials on site. It is based on a capacitance measuring technique in which the moist sample acts as a capacitor connected in parallel with a resonant LC circuit, detuning it. The degree of detuning is a measure of the capacitance, which in turn is a measure of the moisture content of the sample. The oscillator frequency was 84.5 MHz, and extensive precautions were taken to eliminate stray capacitance effects and temperature instabilities. The final device has shown itself to be suitable for on site measurements, with minimal interference with the wall, and with a small and correctable sensitivity to dissolved salts.

Development of a technique for the measurement of moisture levels in building materials

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1 Introduction

Moisture penetration due to rainfall has an important though insufficiently understood influence on the thermal performance of buildings[1]. Indeed, it has been stated[2] that this is one of the most important areas of investigation still outstanding in the study of building heat loss.

One of the difficulties encountered in trying to work in this area is the fact that no really satisfactory method exists for measuring the moisture content of common building materials such as brick. This is particularly the case if one is interested in continuous monitoring to examine the dynamic behaviour of a building structure.

Any moisture measurement technique adopted must have certain characteristics:

(1) It must be able to measure continuously and without interference for extended time periods.

(2) It must be stable for long time periods.

(3) It must, as far as possible, be sensitive only to changes in moisture content and not to variations in the salt concentration or temperature.

(4) It must be non-destructive. To require the continual removal of samples for testing would change the character of the test section, so that conclusions drawn from these data would be meaningless.

(5) If possible, any probe used must behave in the same manner as the test wall.

(6) It should preferably be capable of providing information on the distribution of moisture content through the wall. Such information on moisture profiles is needed for detailed analyses of thermal behaviour.

Many methods of moisture measurement exist but none is entirely satisfactory[3]. To find a technique which satisfies all of the above criteria is clearly impossible, and each method represents a compromise between the different requirements.

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The simplest, and therefore probably the most reliable, methods involve weighing a sample of the wet material and comparing this with the original dry weight. This method is independent of such secondary factors as salt concentration and temperature, and it provides a direct measure of the weight of water in the sample. Such techniques typically require the removal of a sample (frequently by drilling as described by Newman[4]), then weighing, drying and reweighing it. Obviously this method is time consuming and is usually destructive. It is therefore not useful for a long term analysis of moisture content. A less destructive variation on this technique is to use a predrilled 'independent core'[5] which is removed and dried as before and then replaced to be sampled again at a later date. This technique is limited in that the time taken for a core to reach equilibrium moisture content with its surroundings, coupled with the time taken subsequently for the weighing, drying and reweighing procedures mean that instantaneous, or even frequent, measurements are impossible.

Air permeability at low pressure differentials can be used to provide moisture content information which is insensitive to salt concentration[6]. However, this system uses a very small sampling area and further, each sensor/brick combination must be calibrated independently. Also, it suffers from a number of other practical difficulties because of the variability of composition of typical building bricks.

These methods of determining moisture content are admirable in their way, particularly because they are independent of the salt concentration. In particular, the weighing methods yield a direct measurement of moisture content. However, they do not provide an electrical output, which makes them unsuitable for automatic data logging applications. With the increasing importance of automatic data logging, continuously monitoring equipment with an electrical output is essential.

Several techniques have been developed which produce an electrical signal which varies with the moisture content of a sample. These include methods which depend on resistance changes, microwave attenuation, neutron scattering, gamma ray attenuation and changes in the dielectric constant of the medium.

The change in electrical resistance of a material during wetting is significant and thus provides a possible method of moisture measurement[7]. Unfortunately, however, a number of difficulties arise through the influence of one or more of: variability of the contact resistance, polarisation at the electrodes, and the effects of dissolved salts. The contact resistance and the polarisation problems can be overcome by using AC potential probe techniques, though at the expense of an extensive in situ calibration effort. The dissolved salts, however, remain as a serious problem. Hancox and Walker[8] showed that brine solutions with concentrations of about 0.1 m and above can lead to serious errors, while Laws and Sharpe[9] indicate that salt solutions with a conductivity at least as high as that of 0.1 m sodium chloride are to be expected as most building materials contain a high concentration of soluble salts. These are usually sulphates of calcium, sodium, potassium and magnesium, though chlorides and nitrates are also found. Indeed the pore liquid is usually saturated in the sparingly soluble calcium sulphate.

The electrical resistivity is also very temperature sensitive. Smith-Rose[7] showed that for a soil sample, conductivity changed by 2.0 to 2.5 per cent per K. Thus, temperature compensation is essential.

Resistance gauges have been commonly used in the measurement of moisture content[9]. These comprise electrodes embedded in a porous material (usually plaster of paris) which is then placed within the test material. The resistivity of the sample material is then measured and related to the test sample conditions once the two materials reach equilibrium. Such a system has the advantage of facilitating laboratory calibration; however, there are problems of matching the pore size of the plaster to that of the brick, of ensuring that an equilibrium state has been reached, and of the stability of the plaster itself. Indeed, such gauges were found by some workers to give reproducible readings for only three months[10]. Other gauge materials are considered more suitable for long term measurement, though these are more sensitive to the presence of salts.

Microwaves also provide a possible solution as their attenuation by a sample is related to its moisture content. These techniques have the advantages[11] that (a) Conduction losses decrease as the frequency rises (indeed dissolved salts have no effect on conduction at frequencies above about 10,000 MHz[12], (b) No physical contact is needed between the test sample and the measurement equipment, and (c) Measurements can be made relatively quickly.

According to Powley[13], microwave moisture measurement techniques are temperature independent (though, see Kraszewski[14]), and accurate over a wide range of moisture levels. The two basic techniques are transmission through the sample, and reflection off the surface of the sample. The transmission technique suffers from the disadvantage of requiring access to both sides of the test wall. The reflection technique has been found to be seriously influenced by surface roughness.

Microwave attenuation techniques are unable to provide information regarding moisture profiles, and, indeed, the absorption of microwaves by the water can affect the drying rate of the sample if used for long periods. Further, the scattering of the microwaves by the brick/mortar joints can have a serious effect on system accuracy with, under the worst conditions, no better than ± 30 per cent of the mean value being claimed[12].

Another approach uses the attenuation of a beam of neutrons or gamma rays, and both are used for soil moisture determination[15]. Of the two, neutron probes are to be preferred because of the large interaction between neutrons and the hydrogen contained in the water. Gamma rays have a much smaller interaction. The use of radiation attenuation methods implies having a source on one side of the wall and a detector on the other. This creates accessibility problems which detract from the usefulness of the approach.

Changes in thermal conductivity associated with moisture content variations have also been used to determine the moisture level[16]. A heating wire is inserted into the test material and constant heat input established. The change in temperature with time then gives the thermal conductivity of the sample and hence, from a previously determined relationship, the moisture content. This method is independent of salt concentration, but the effect of a heating wire on the moisture distribution within the test material must be considered significant if the technique is used for long periods. Also, the underlying principle of this technique may not be wholly suited to many experimental situations since the assumption of a known and fixed relationship between thermal conductivity and moisture content may not be valid.

Robertson *et al*[17] describe a low resolution nuclear magnetic resonance based approach that shows promise in some applications. They report its use in measuring the liquid content of a coal sample. However, the technique is complex and the equipment not suited to in situ operation on a building site.

Finally, it is possible to make use of the difference between the dielectric constants of water and brick to indicate the presence of moisture since the capacitance of a capacitor made from a porous material will vary directly with the moisture content. Indeed this is the electrical parameter least affected by salt concentration and temperature[18]. Thus, this technique overcomes several of the problems associated with the conductance/resistance methods discussed previously.

Since water has a dielectric constant of about 80 and most dry building materials[9] have a dielectric constant below about 5 the effect of water on the dielectric constant of the test material will be marked. Dissolved salts will have some influence, but this can be minimised by careful selection of the operating parameters.

Capacitance based moisture measurement techniques can be considered largely independent of temperature for most purposes[7]. Indeed changes of about 0.5 per cent per K appear typical for the dielectric constant of water in the range of about 1 to 20°C. Thus, the effect of temperature on accuracy is small and, in any event, can be corrected for.

The dielectric properties of a material can be described in terms of a complex dielectric constant (E)[9]

$$E = E' - jE'' \qquad (5.1)$$

where the real part, E', represents energy storage, and the imaginary part, E'', represents all dissipative effects including ohmic conductivity caused by migrating charge carriers and frictional loss in dipole orientation. E'' is known as the relative loss factor and E''/E' as the loss tangent (tan d).

Podkin[19] states that E' is determined by the moisture content of the material, while E'' is determined by 'chemical and biological' factors. In the area of interest to us, E'' is related to salt concentration and hence to the conductance of the dielectric. In order to eliminate the effect of salts on measurements the effect of conductance on the complex dielectric must be eliminated. Hancox and Walker[8] show that E″ = 2d/f, where f is the frequency (Hz) and d is the conductivity. Thus measurements can be made less dependent on conductivity by increasing f. Alternatively, a phase sensitive detector can be used to separate the real and imaginary components[20] and produce a value for E′ directly.

Millard[20] claims that conductance effects (and therefore E″) become negligible at about 100 MHz, while Watson[12] states that frequencies greatly in excess of this are necessary to totally eliminate conductance effects. At 10,000 MHz the conductivity of water seems to be largely independent of salt concentration and the loss tangent (tan d) is comparatively low[12,21].

Generally, capacitance moisture gauges operate at frequencies of 30 MHz or less. All of these techniques are strongly influenced by the conductivity, so that both a capacitance and a conductance measurement are needed to correct for the effect of soluble salts.

Capacitance techniques require the insertion of a calibrated probe into the wall. If the probe has the same brick/mortar composition as the wall, then it may be deemed to duplicate its behaviour. Further, the physical dimensions of such a probe may be altered arbitrarily to provide large or small scale measurements as desired. Obtaining information regarding moisture profiles is not straightforward, though, with a series of probes, this can be obtained with some difficulty. Since the probe can be of the same material as the wall, it can be left *in situ,* and will, in most respects, accurately represent the characteristics of the wall itself.

Several different techniques have been used for capacitance measurement. These include measuring the discharge time of a relaxation circuit[22], parametric modulation[23], and VHF admittance bridge techniques[9,24].

All in all, capacitance methods seem to offer the greatest opportunity for developing a moisture measurement system which is flexible in application and which can best represent the behaviour of a real wall under actual conditions. It was therefore decided to follow this route in the present work. The measurement system adopted was based on a high Q-factor tuned oscillator circuit, and the frequency range adopted was 78-95 MHz, which is close to Millard's recommended value of 100 MHz.

2 Resonant circuit capacitance meter

The resonance frequency of a parallel LC oscillator is given by f = 1/(2 LC) and the sharpness of the resonance curve is specified by the Q-factor which is defined either as Q = f0/(f1 — f2) or as Q = (1/R)√(L/C). Thus, the output at a given frequency will vary with the capacitance in the circuit, and the magnitude of the change can be varied by changing the Q-value. Alternatively, the frequency of the resonance will vary with changes in the capacitance and the ease with which the resonance is detected will depend on the Q-value.

As a measurement tool, the resonant circuit technique has several desirable features:—

(1) flexibility in selecting the operating frequency,

(2) variable sensitivity is possible with this system for a fixed external capacitance change,

(3) circuit simplicity is advantageous in minimising stray capacitance and contact resistance effects which can be significant at such high frequencies. Thus, this technique provides an attractive method of capacitance measurement at high frequencies.

A high impedance signal generator is necessary to ensure that the fall in circuit impedance as the capacitance changes is not compensated by a rise in supply current. If this were to happen, the output voltage from the circuit would remain constant, and current measurement would be necessary. This would involve monitoring the voltage drop across a resistance, and the increased circuit resistance would reduce the Q-factor. With a high impedance supply the current remains approximately constant, so that the output voltage varies with the test circuit impedance without affecting the sensitivity of the system.

An early version of the circuit consisted of a four turn inductor of 9.5 mm internal diameter made from 16 swg copper wire, a 22 pF ceramic capacitor and a one ohm wire wound resistor (Fig. 1). A germanium diode was selected for rectification of the output signal because of its low input rectification threshold, this being particularly important because of the low output signal (about 30 mV at resonance). A 1000 pF decoupling capacitor and a 500 pF feed-through capacitor were used on the d.c. output to minimise ripple. The capacitance to be measured is connected in parallel with the 22 pF tuned circuit capacitor to alter the resonant frequency and hence d.c. output of the tuned circuit.

The effects of stray capacitance at frequencies of 80 MHz can be significant, so all leads within the tuned circuit were made as short as possible, and once the circuit was operating correctly it was placed within a screening box and potted in resin to prevent movement of the components. Thus, the circuit is totally enclosed except for the input and output leads and two rigid connectors for the connection of the external capacitance (C′).

The resonance frequency of this circuit was about 85.5 MHz, and, in order to achieve the required stability, the input signal was supplied by a crystal controlled oscillator as described later.

The calibration was carried out by systematically adding ceramic capacitors which were chosen for their quality and stability at very high frequencies.

The chief difficulty in using a crystal controlled oscillator is that the frequency must be variable over a wide range in order to match the resonant frequency of the different tuned circuits. In the present case, the range was from 80 to 90 MHz. The variation of frequency is achieved by employing a digital phase locked loop frequency synthesis technique, similar to that now in widespread use in radio equipment[25]. Such a system comprises a phase detector, loop filter, an amplifier and voltage controlled oscillator (VCO).

The phase detector receives reference and signal signals, compares their phases and frequencies, and produces a corresponding variable output error voltage. This is then

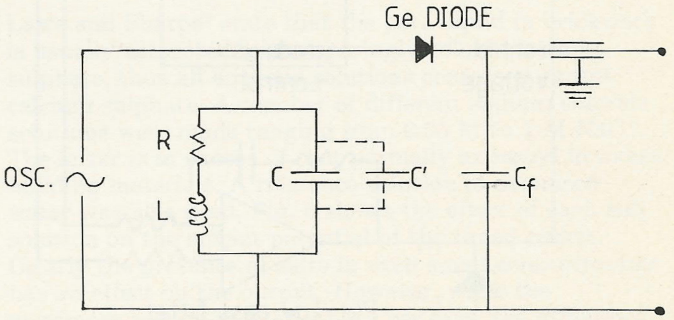

Fig. 1. Tuned RLC circuit. C′ represents the capacitance of the sample, which drives the circuit off resonance.

filtered and amplified and input to the voltage controlled oscillator in such a way that the frequency and phase differences between the two signals are reduced to zero, when the loop is said to be locked.

Since the output and reference signals are being compared to provide the error voltage a variable frequency output can be achieved by using a divide-by-N counter in the feedback loop between the VCO and the phase comparitor. Thus, instead of the phase detector locking to the output frequency of the VCO, it locks to the output of the divide-by-N counter so that the VCO frequency is N times the reference frequency.

Unfortunately, programmable counters are not available for use at frequencies above about 50 MHz, but there are several methods of matching frequency and counter characteristics. All of these have advantages and disadvantages, and the approach chosen for this application is known as dual modulus prescaling or pulse swallowing (discussed in more detail in Campbell[25]). This involves prescaling the output from the VCO using a dual modulus (divide−by−M/M + 1) prescaler, and two programmable divide−by−N counters. It is more complicated than some of the alternatives, but overcomes the worst of the problems normally encountered. In principle, using this approach with M and N both equal to 10, a frequency range of 310 kHz to 102.93 MHz is accessible with steps of 10 kHz.

The present system, shown schematically in Fig. 2, was based around the Motorola MC145152 phase locked loop frequency synthesiser integrated circuit. The voltage controlled oscillator is shown in Fig. 3. The control voltage is input to a varicap diode, altering the capacitance and hence the frequency of the oscillator. The VCO used here operated over a range of 78 to 96 MHz. Frequencies outside this range can be obtained simply, however, by changing the values of the components.

One important feature was that precise regulation of all input voltages to the oscillator was found to be necessary, and care had to be taken to ensure an adequately smoothed and stable supply. Also, the VCO had to be potted in resin to avoid vibration effects.

It was found that with this oscillator the Q-value of the test circuit was too low, and another tuned circuit was built with a higher Q-factor and potted as before. The resonance frequency was 84.5 MHz, and the capacitance response characteristic is shown in Fig. 4.

Extensive stability tests on the final circuit showed small but significant evidence that variations in the ambient temperature were affecting the tuned circuit components and altering its resonant frequency slightly. This was overcome by placing the whole system inside a temperature controlled box which uses a thermistor to switch a proportionally controlled heater (controlled by a zero voltage switch) and fan to maintain a desired temperature. This system now controls temperature to $\pm$ 0.1°C and the output signal from the resonant circuit is stable at 41.0 $\pm$ 0.1 mV.

The voltage controlled oscillator, frequency synthesiser and tuned circuit are held within separate screened boxes to minimise stray interference effects. Further, the use of the temperature controlled box means that the complete system is contained within a second screen. Only the two leads for connecting the external capacitors penetrate this screen with the result that the effects of external interference have now been largely eliminated. The external capacitors must be soldered into place to eliminate the possible effects of contact resistance and capacitance, and to avoid heat conduction to the tuned circuit components, it is important to ensure that they have an adequate heat sink.

The system has been extensively tested and has been found to measure capacitances less than 22 pF with a resolution of 0.1 pF or better.

3 Measurement of moisture content or brick sample

The system was tested initially, by using a single brick sample and cutting a series of channels 1 cm deep in one face. Into these channels totally insulated stainless steel plates were fixed using epoxy adhesive. Four plates were used to give 'brick capacitors' of 1 cm depth, 6.5 cm length and either 1, 1.5 or 2 cm width. The brick was then soaked in water and allowed to dry while the three capacitances were monitored. The drying characteristics of the three brick capacitors could be readily detected, and the change in capacitance with moisture content was obvious. However, considerable scatter was apparent in the results, particularly between runs. This scatter arose because the capacitance of a smallish sample near the surface of the brick was being monitored, while the moisture content was determined by weighing the complete brick. It is obvious that this arrangement is open to error from an uneven distribution of moisture throughout the brick. If our interest was in the

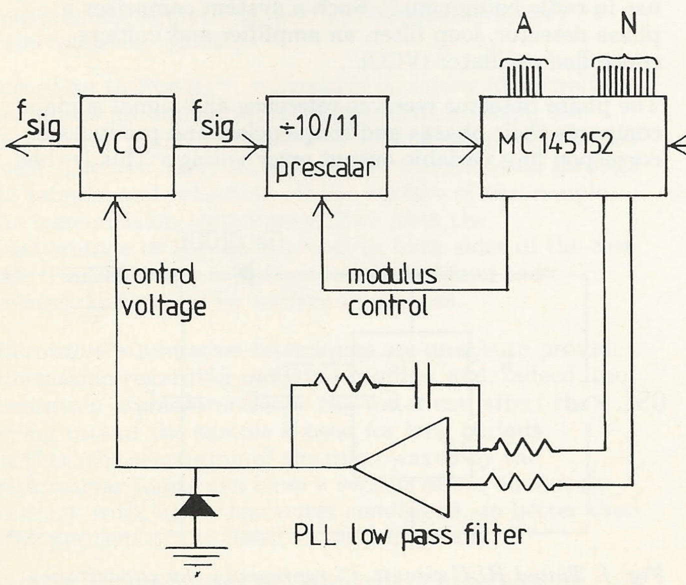

Fig. 2. Schematic diagram of frequency selector circuit.

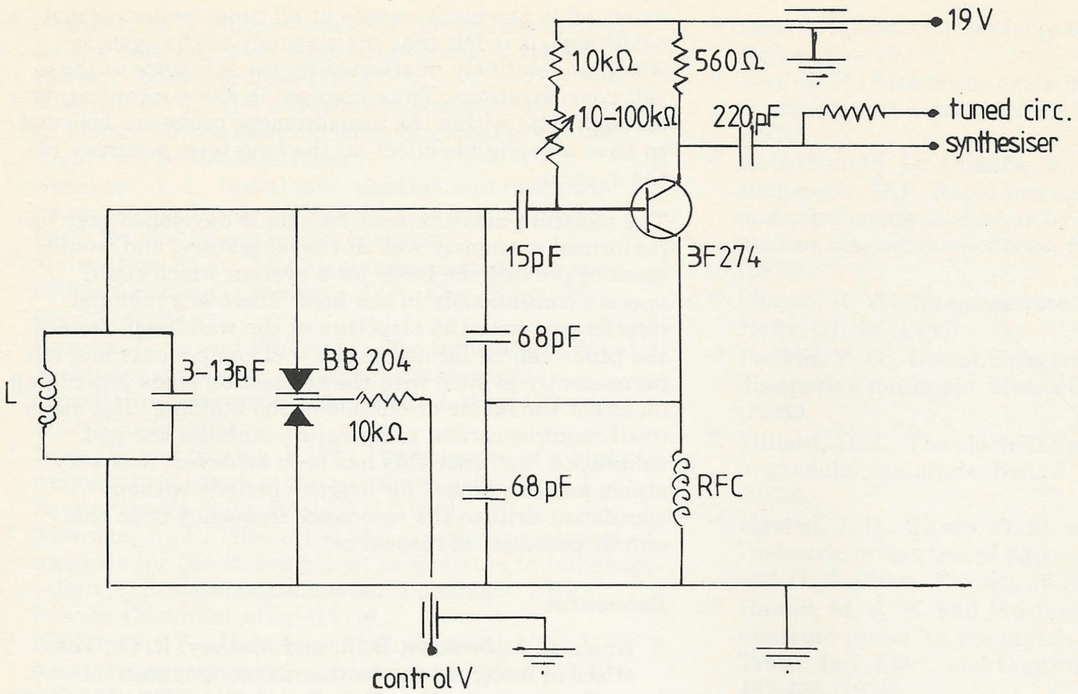

Fig. 3. Voltage controlled oscillator (VCO) circuit.

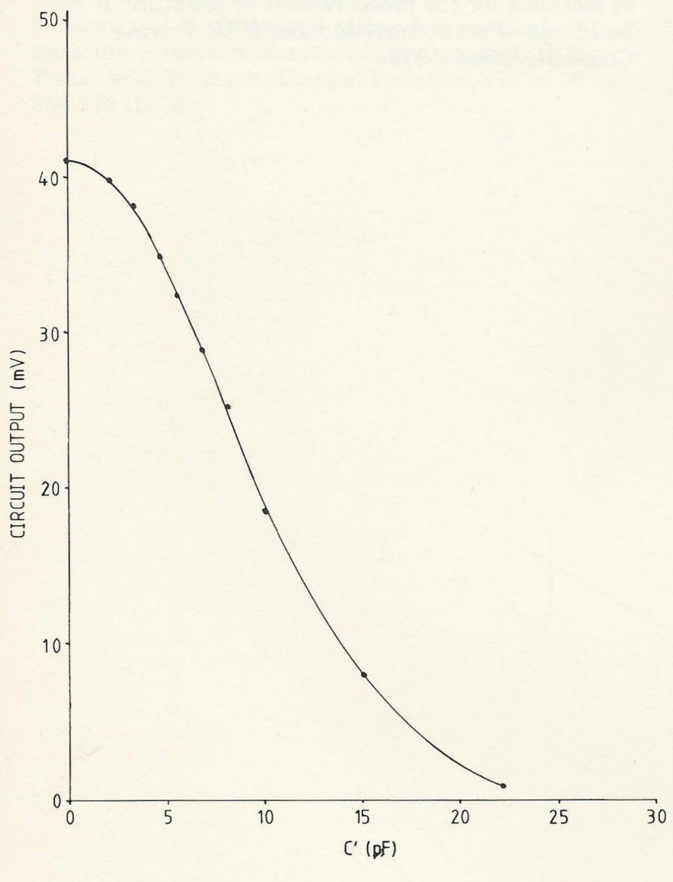

Fig. 4. Calibration curve of final circuit with oscillator frequency of 84.5 MHz, and temperature stabilisation of the components.

distribution of moisture within a wall, this would not be a problem as the capacitor would be buried in the wall. In the present instance, however, where our interest lies in proving and calibrating the system, this uncertainty is an embarrassment so a more controllable 'mini-brick' 1 cm × 2 cm × 8 cm was adopted. This allows the treatment of only the sample capacitor itself and thus the moisture content measured is wholly representative of the brick sample.

To improve the representative nature of the sample, it was formed as a brick and mortar unit of about 88 per cent brick and 12 per cent mortar to better represent a normal wall element. The sample was oven dried and weighed before testing so that its moisture content could be accurately assessed by weighing. The change in apparent capacitance with moisture content was examined for a series of drying runs.

Fig. 5 shows the variation of capacitance with moisture content for this unit during these runs. The scatter in the data is much reduced and the results of moisture content plotted against measured capacitance lie on a straight line with a correlation coefficient of 0.987. Further when considering the measurement sensitivity of about ± 0.1 pF, all points are on the line to within the experimental error.

The effect of dissolved salts on the measurement system was assessed using a series of aqueous salt solutions. This technique has been used widely for such studies (Hancox and Walker[8], Hasted *et al*[21], and Laws and Sharpe[9]. Two insulated plates with a fixed separation were placed in a number of salt solutions of different concentrations, and the change in output noted.

Laws and Sharpe[9] state that the pore liquid in brickwork is usually saturated in the sparingly soluble calcium sulphate, thus all aqueous solutions contained excess calcium sulphate. A number of different sodium chloride solutions were made ranging from 0.05 M to 1 M NaC1. The latter is in excess of that normally expected in moist building materials. A reference solution of deionised water was also used. Fig. 6 shows the effect of each salt solution on the output potential of the tuned circuit. Clearly the presence of salts in even small concentrations has an effect on the output. However, while the maximum reduction is about 3.5 per cent, the difference in output between each sodium chloride solution is minimal. Since salt concentrations of this order are to be

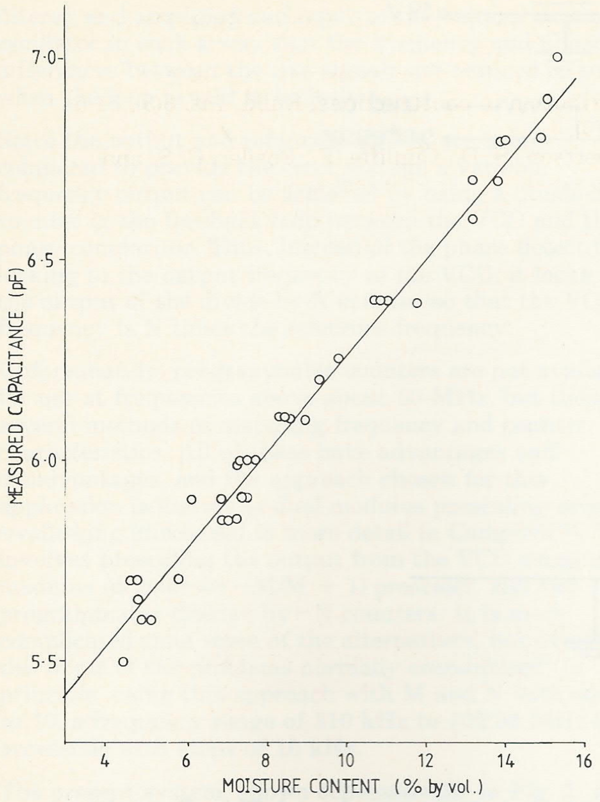

Fig. 5. Variation of measured capacitance with moisture content for brick sample. The correlation coefficient is 0.987.

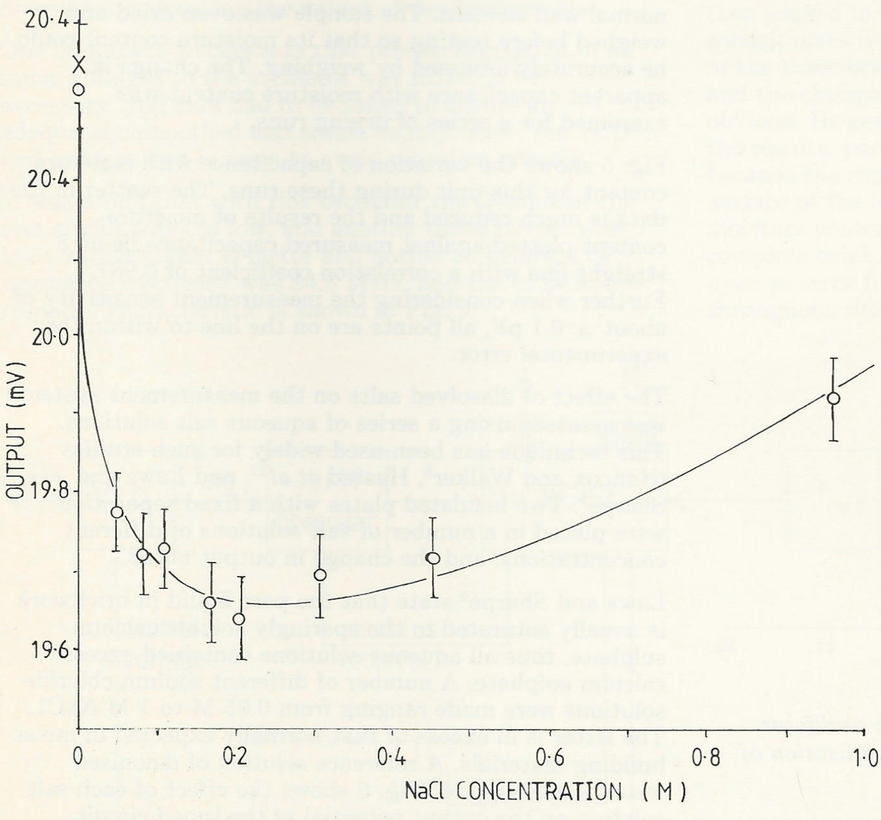

Fig. 6. Effect of dissolved salt on system output. All solutions are also saturated in $CaSO_4$, except for the one marked x, which refers to deionised water. It should be emphasised that these measurements are for a cell containing the salt solution, and not for a brick sample.

expected in the brick sample at all times under normal conditions, it is felt that the accuracy of the system would be relatively unaffected by the influence of these salt concentrations. Thus, changes in salt concentration through time within the measurement probe are believed to have a negligible effect on the long term accuracy of the system.

The moisture measurement technique developed here has performed extremely well in the laboratory, and would seem to provide the basis for a system which could operate continuously in the field. There is a minimal interference with the structure of the wall itself. Indeed, the probe can be formed of the wall components and left permanently in situ, with the connection leads protruding on either the inside or outside of the building. The meter itself requires careful temperature stabilisation and calibration, but once this has been achieved, it is very stable and can be left for lengthy periods without significant drift in the resonance frequency or in the output potential at resonance.

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Summary A comprehensive investigation into the real potential of microwave radiant energy as a low energy alternative to conventional thermal comfort systems, has been conducted over the past three years. During periods of research no detailed references were discovered. The fundamental questions discussed are: (1) Microwaves, what they are? (2) What is the truth about the associated health risks? (3) How can microwaves be used to heat buildings? (4) What are the potential advantages and benefits? Information assimilated and collated in compiling this composite reference reveals positively that 'microwaves' have the potential to be applied in the fashion portrayed. Though there remains still further experimental development to be pursued in advancing the new concept, this paper is presented to stimulate industry regarding the viability of such an idea to enhance man and his ever-changing environment. Existing conventional systems have been with us for decades or even centuries, and are essentially refinements of original schemes. The profession may well need to review its schools of thought relating to conservation and economics of engineered space heating systems. Enlightenment on a totally different method of approach is provided with unbiased explanations.

An investigation into the potential of microwave radiant energy as a source of heating for human comfort

DAVID C. SALES, MInstE, AMInstR

1 Introduction

Warmth, shelter and food are basic necessities of life and history records an increasing stream of ingenuity to devise both natural and artificial ways for harnessing food, warmth and building shelters. The simple log fire is part of universal cultural tradition. Modern central heating systems are commonplace in many of our homes today but they are closely related to methods used by the Greeks and Romans.

The idea of using microwave radiant energy for heating is not new. Research studies of Copson[6] (Peurto Rico), Adair[1] & Adams[2,3] (USA), Pound[14] (USA) and Stolwyk[15] (USA) into the practical feasibility of use have produced definite conclusions, which have confirmed the potential of microwave heating as an alternative to conventional thermal comfort systems (these are essentially energy wasting). Microwave heating offers the flexibility of individual control, saving in building space and is a pollution-free source of energy.

2 The origin and electronic principles of microwaves

2.1 General

Microwaves are a form of electromagnetic radiation—they are an integral part of the electromagnetic spectrum which includes light, infra-red and radio-waves but they have wavelengths approximately mid-way between the infra-red and the radio-waves (see Fig. 1 and Table 1). Microwaves share the properties of both neighbouring radiations and like radio-waves are used in communications, which like infra-red can be used for heating organic substances.

Microwaves have wavelengths between 1 mm and 300 mm, that is frequencies between 300 GHz and 1 GHz (1000 MHz). These divisions along the electromagnetic spectrum are rather arbitrary but in general infra-red radiation is normally produced by a hot body, while microwaves are generated electronically.

2.2 Generation and Amplification

The actual generation and amplification of microwaves is achieved by a variety of devices, each designed for a specific purpose, which in a similar way, as for radio-waves, produce transverse electric and magnetic waves.

High power microwave sources have been developed from the thermionic valve (vacuum tube). The developments took place in the early 1920s to overcome the deficiencies of the more conventional radio valves which are apparent when used to produce high powers at the higher frequencies demanded by the swiftly developing communications industry. This resulted in the magnetron, klystron and travelling wave tube. The magnetron is a self-sustaining oscillator tube capable of producing pulses of power well in excess of a megawatt, both the klystron (Fig. 2) and travelling tube are essentially amplifying devices capable of producing output powers of several kilowatts. Very low power microwave sources may be derived from either low power versions of the klystron or the maser (microwave amplification of stimulated radiant emission). In addition there are many forms of solid state microwave sources, which are very compact and in recent years a new two terminal microwave generator has been developed known as the 'Gunn Device'. It is essentially a wafer of n-type gallium arsenide. When a low dc voltage of the current polarity is connected, a dc current flows, upon which are superimposed current pulses. These current pulses occur at intervals of the order of 10^{-10} secs, and are used to indicate oscillations in a cavity of waveguide resonator. The frequency of the pulse depends on the thickness of the n-type gallium.

2.3 Transmission

In electronic apparatus concerned with the production, amplification and transmission of electrical energy at frequencies below about 1 GHz (1000 MHz), the energy is transferred from one part of the system to the other by connecting wires carrying alternating currents in much the same way, and obeying all the classical a.c. field laws.

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As a result of the work of James Clark Maxwell[13], J. J. Thompson[13], Sir Oliver Lodge[13], Lord Raleigh[13] and many others it was shown by about 1900, that electrical energy could be transmitted by waves constrained within a guiding structure or 'waveguide'. By 1936 this work had advanced to the stage where, in spite of the imperfections in the conducting or dielectric materials used in the construction of waveguides, their transmission efficiency had increased.

Since that time with further research and the rapid development of manufacturing techniques, the use of the wave guides have become quite extensive in electrical apparatus which is concerned with the generation of transfer of energy at very high frequencies.

The actual transmission of energy by waveguides is achieved by radiating the energy, in the form of electromagnetic waves, into the inside of the 'pipe like' structure by means of a transmitter aerial or coupling probe. Assuming that the waveguide structure does not introduce any loss into the operation, then this same energy is extracted from the waveguide by the use of a similar 'receiver aerial' or coupling probe. Microwave and radio-wave transmission are very similar in principle.

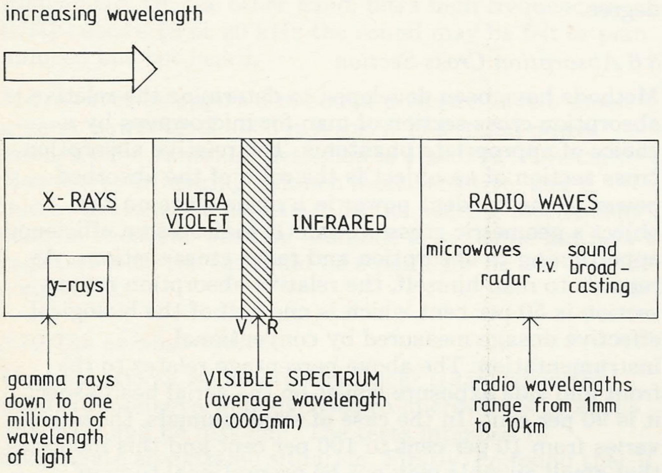

Fig. 1. The electromagnetic spectrum.

Table 1. Main sources on earth of electro-magnetic waves

Wave-band	Origin	Sources
X-Radiation	(1) High energy changes in electron structure of atoms. (2) Accelerated electrons.	X-ray tubes
Gamma Radiation	Energy changes in nuclei of atoms	Radioactive substances
Ultraviolet Radiation	Fairly high energy changes in electron structure of atoms	(1) Very hot bodies (e.g. the electric arc) (2) Electric discharge through gases, particularly mercury vapour in quartz envelopes
Visible Radiation	Energy changes in electron structure of atoms	Various lamps, flames or anything at or above red-heat
Infrared Radiation	Low energy changes in electron structure of atoms	All matter over a wide range of temperature from absolute zero upwards
Radio Waves	(1) High-frequency oscillatory electric currents (2) Very low energy changes in electron structure of atoms	

3 Radiation biology of microwaves

3.1 General

The radiation biology of microwaves is the study of the effect of microwaves on living organisms. Immediately, this phrase brings attention to the widening range of influences that microwave radiation has on man.

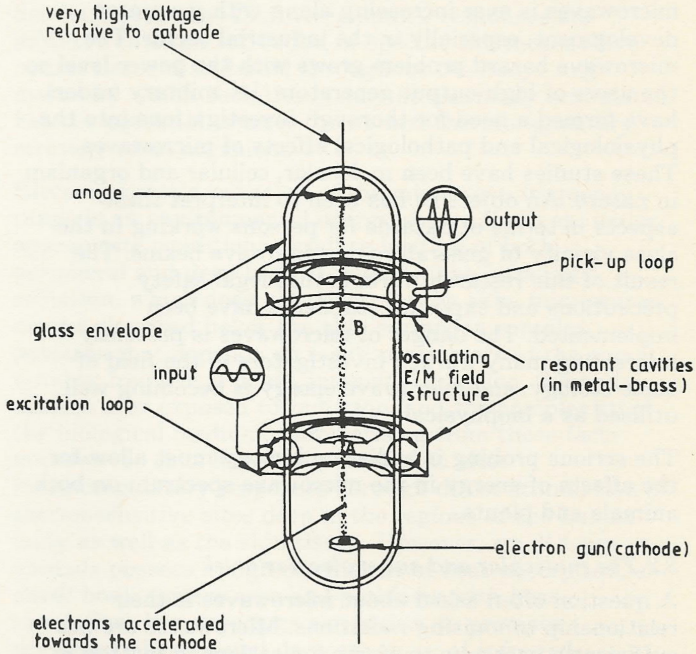

Fig. 2. A double cavity klystron[9]. This device is used to amplify the microwave signals—this replaces the valve at these high frequencies. Electrons are accelerated and retarded by alternating field in first cavity (A). When electrons reach (B) they are 'bunched' and induce large field in second cavity.

In this particular section the biology of microwaves will be found to entail the effects of this radiant energy on man and his environment.

Microwave radiation is categorised into two forms; man-made and natural. Between them organisms are exposed to microwaves almost continuously. Natural microwave radiation is generated by the sun in the 3-30 cm range, the intensity of which is a good indicator of the frequency and pattern of solar activity witnessed by astronomers on earth. Thus the atmosphere of the earth is warmed by microwaves. The intensity of man-made microwaves is ever increasing along with generator development, especially in the industrial sector. The microwave hazard problem grows with the power level so the users of high output generators, i.e. military (radar), have formed a need for thorough investigations into the physiological and pathological effects of microwaves. These studies have been molecular, cellular and organism in nature. An objective has been to interpret these aspects in terms of hazards for persons working in the close vicinity of generators or microwave beams. The result of this research work, is that legal safety precautions and exposure standards have been implemented. The danger of microwaves is presently interesting many learned investigators in the field of basic biology, and microwave energy is becoming well utilised as a biophysical tool.

The serious probing in radiation biology must allow for the effects of energy in the microwave spectrum on both animals and plants.

3.2 The molecular and sub-molecular level

A question often asked about microwaves is their relationship to ionising radiations. Microwaves are not sufficiently energetic to create ionisations in matter, in the fashion of high energy quanta associated with X and gamma rays.

The concepts of radiation biology which deal with high energy radiations including beta, gamma rays and neutrons, cannot often be readily applied to the microwave case. It would be expected that the generation of the ionising radiations would require more power. In a microwave power flux the quanta passing through a unit area is expressed quite rightly in watts per square centimetre. In ionising radiations, the measurements are characteristic of the action and are usually called rads or roentgens.

3.3 The cellular level

Single nerve fibres produce potentials between the inside and outside of the membrane. When measured this potential difference is in the order of milli or tenths of volts and in some aquatic organisms very high voltages are produced for a short time because the series and parallel arrays of cells.

The exterior of the cell can be regarded as positive and depolarisation would mean removal of this. The membrane shows other electrical characteristics in carrying out its electro-metabolic processes, one of these being capacitance. The effect of cell membrane behaviour at various frequencies has been outlined by Schwan[6]. Copson[6] (1956) has shown that much greater absorption of microwave radiation (2,450 and 915 mc) in multi-layered tissue occurs in fatty than in muscle tissue. Thus energy is transmitted through the fat, reflects from the interface and this time it absorbs in the fatty material.

3.4 Effects on organs and organisms

Organisms are studied in a different style to molecular levels and so boundary conditions are of a great importance. Involved is the field inside the organism and wavelengths in the material composing it. First concern is the absorbed energy E_1 where:

$$E_1 = \text{Incident Energy} - (\text{Transmitted Energy} + \text{Reflected Energy}) \quad (3.1)$$

At crucial interfaces within the substance special limits exist and frequently the energy reflected from the interface will be absorbed before it leaves the outer boundaries of the organism.

3.5 Cerebral cortex

The brain and head were studied by Prinneo[6] (1961). In this region energy would be admitted through the skin fat, muscle and bone to reach the grey matter. A model was used to determine the reflection co-efficient at such interface and from such information the intensity of microwave fields may be found at any point in the region. Reflection characteristics will establish whether or not maxima in the field will be reinforced to a significant degree.

3.6 Absorption Cross-Section

Methods have been developed to determine the relative absorption cross-section of man for microwaves by a choice of appropriate phantoms. The relative absorption cross section of an object is the ratio of the absorbed power to the incident power in a plane wave on the object's geometric cross-section. It indicates an efficiency appertaining to absorption and radar cross-sections. As regards to man himself, the relative absorption cross-section is 50 per cent which is one half of the biological effective dosage measured by conventional instrumentation. The above percentage relates to the front and side exposure whilst on the aerial head aspect it is 90 per cent. In the case of small animals, the value varies from 10 per cent to 100 per cent and this indicates that small animals may not be proportional to man.

3.7 Ocular effects

Microwaves have all been inferred as the cause of eye opacities. An old and modern occupational hazard known as glass-blower's or welder's eye is certainly due to over-exposure to infra-red. Experiments with animals producing opacities show that less power is needed as the exposure is lengthened. The invitation is to conclude that a lesser penetration places the injury site on the anterior cortex. Tests carried out on real and phantom eyes (Carpenter 1961)[6] suggest a positional or resonance effect.

The microwave response of vitreous humour is such that the eye can be warmed without difficulty in a microwave field. The eye is also a structure with a poor thermal exchange but nevertheless the damage is always on the lens. This fact implies that the opacities are non-thermal in origin and the cause is positional in nature. Despite this it does not indicate whether the eye is vulnerable due to being in a beam or at focus from the bony orbit of the eye. At low frequencies (200 MHz) Addington[6] (1958) could attribute ocular damage to radiation. The reflection of such radiation from the bony encasement at the rear of the eye would place the focal point more forward, maybe out of capability of the injury.

3.8 Testicular damage

Investigators Ely and Goldman[6] (1956) informed that the testes, having a great degree of exposure are more sensitive to microwaves than is the eye. In fact, the scrotum is capable of detecting heat and is able to prewarn the body of a thermal hazard whereas the eye cannot. Again, experiments on mice have produced deteriorated testicles with little or no sperm content. Similarly, tests on dogs have demonstrated that the damage is reversible and only temporary for mild exposures.

An analysis of medical histories belonging to persons working in the vicinity of radars revealed no difference in fertility. These workers were a quality control group dealing with high power radiation appliances.

3.9 Aural effects

People exposed to microwaves have said to hear 'soundless' energy. Closely monitored tests tell us that the brain senses the interaction as 'buzzing or a knocking sound'. One deaf subject could hear microwaves but not ordinary sound. The results also made a point that the sound seemed to come from behind the person regardless of location.

Normal sound is perceived at 20-18000 Hz and at these limits the threshold of hearing requires much larger sound power density than in the intermediate frequencies. On the other hand, ultra high frequencies (UHF) above 18 or 20 kHz the sound may be felt or pain induced but not heard.

Microwaves as opposed to sound pressure have a much greater power level (0.33 mW/cm^2). The sound power intensity for minimum detection is about 10^{-18} W/cm^2.[6] However, our hearing is much more sensitive, and if microwaves can be perceived by the hearing mechanism then an excellent warning stimulus is available but how it registers with the ear could be a point for further conjecture.

3.10 The nervous system

Unusual effects were noted by Bach[6] (1958) when the heads of monkeys were exposed to microwaves at about 388 mHz and 64 mW/cm^2. The tests intended to investigate an extreme situation. Firstly, he found that if the transmitter was cycled on and off, an animal can be consequently aroused and relaxed in 20 sec cycles. Subsequently, eye movements become abnormal (involuntary), together with skin, respiration, salivation, lacrimination and blinking responses. Eventually when eye activity becomes rapid and facial grimace appears, it is a signal that general seizure is imminent. Some other animals are able to recover from these advanced reactions to microwaves. These conducted experiments on monkeys indicate that the effect is non-thermal as cooling down during off periods occurs. Also tests carried out at Russian[6] laboratories have shown that high frequency energy exposure affects the nervous response time, i.e. there is an abnormal delay between a stimulus and a conditioned reflex.

Studies involving the effects of electromagnetic radiation on the human nervous system were made by Obrosov[6] and Jasnogarodski[6] (1961) and Burham[6] (1959). Although frequencies used were much lower than those in the microwave region the results are worth a mention here. Low frequency pulses of less than 0.2 milli amp and 0.3 millisec were passed through the brain to bring about a pulsed energy sleep. The subjects quite easily fell into a deep slumber.

3.11 Hyperthermia—alterations to thermoregulatory responses

Microwave energy, coupled to living tissue, generates heat and importantly the most likely injury to humans due to microwaves will certainly be caused by hyperthermia at high exposure levels. The ability to maintain a near-constant internal body temperature is essential to the survival and optimal functioning of every organism. Each species possesses a characteristic strategy for dealing with the thermal aspects of its environment in order to maintain a characteristic body temperature.

Thermal stimuli will elicit both behavioural and autonomic thermoregulatory responses and include not only variations in the microclimate (ambient temperature, ambient vapour pressure, insulation, etc.) but also internal temperature changes. Extensive research has demonstrated that thermoregulatory responses are initiated through activation of specific thermosensitive neural tissue distributed throughout the body. Any means by which such tissue may be thermally stimulated has the potential to alter the normal thermoregulatory strategy of an organism.

Electromagnetic radiation of the microwave frequency range is an environmental energy source that can, under appropriate conditions, produce heating of both peripheral and deep body tissues. Unlike infra-red radiation, which does not penetrate the skin, microwaves can be absorbed below the skin surface in complex patterns that depends on many parameters of the signal, notably frequency, mode, intensity, duration, the body surface area exposed to radiation, and the complexity of the biological medium. Summarising from these facts, microwaves thus have the potential to alter thermoregulatory responses through direct stimulation of thermosensitive sites deep in the regions of the human body as well as the skin tissue. However, small furry animals possess a high co-efficient of heat absorption, a small body surface area and problems with heat regulation. Fur coverings can expect to promote microwave hazard and decrease those of infra-red.

Experiments with animals by Michaelson[3] (1961), Chernovetz[3] (1977), Phillips[3] (1975) and de Lorge[3] (1976), recorded elevated deep body temperatures and measured certain thermoregulatory adjustments, reduced metabolic heat production and microwave avoidance (Frey 1975)[3]. Exposure duration has received appreciably less experimental attention than the other parameters cited, except for loose designations such as acute (brief short-term) or chronic (long-term).

Hartmann[3] (1958) pointed out that long exposure to microwave radiation could induce an excessively high fever, being more dangerous if the person is operating at or near the limit of his thermal tolerance (hyperthermia).

The primary source of an organism's information about the thermal characteristics of the surrounding environment normally lies in the activity of the thermal receptors of the skin (Hardy and Oppel[3] 1937, Hensel[3] 1968, Kenshalo[3] 1968). Studies by Vendrik and Vos[3] (1958), Eijkman and Vendrik[3] (1961), Hendler and Hardy[3] (1960) and Hendler[3] (1963), demonstrated in human subjects that both 10 GHz microwaves could provide clear sensations of warmth at or near the skin surface. For a plane wave, the penetration depth in a biological absorber is a complex function of the wavelength, and its potential to stimulate peripheral thermosensitive structures depends on the wavelength, and the electrical properties of the tissue (Durney 1978)[3]. These early studies demonstrated in humans that 3 and 10 cm pulsed microwaves applied to small skin areas had to last at least 5 seconds in order for a minimal intensity to produce a thermal sensation. At shorter durations, field intensity had to be greatly increased to evoke threshold warmth. Although exposures longer than 5 seconds may not alter the magnitude of the warmth sensations, the cumulative thermal load in the body may increase with duration. This would obtain particularly for whole body exposures when the thermoregulatory system of the organism is either overtaxed or inoperative. Thus it is possible that a subthreshold microwave intensity (i.e. one that does not bring about a thermal sensation or thermoregulatory response) will if maintained long enough, eventually initiate an observable response. Mathematical simulation models have been constructed by various investigators of the subject. Such models incorporate insights from anatomy, physiology and mechanical engineering and provide us with a simplified system that reacts to internal and external thermal stimuli in essentially the same manner as the real system does.

Very early models of thermoregulation were proposed by Burton[15] and MacDonald[15] and Wyndham[15] and dealt with only heat transfer, heat production and heat loss without representing the regulatory mechanisms. Regulatory aspects were included in early models produced by Crosby[15], Stolwyk[15] and Hardy[15].

The characteristics of a model are segregated into two systems, one being the 'controlling' system. The form represents the geometry, anatomy and physiology of man and produces transfers and losses of the metabolic heat.

The regulatory or controlling system for the thermoregulation consists of the neural structures that sense local body temperature and the neural structures that integrate their thermal signals and transform them into command signals controlling the physiological actions that maintain body temperature.

In this particular model the human body is represented by six cylindrical segments, each comprising four concentric cylinders and a central blood compartment (Fig. 3). The layers represent skin, subcutaneous fat, muscle and core tissue. Each layer has a heat production characteristic of the tissue it represents. There is a conductive heat transfer between the adjacent nodes and each has a convective heat transfer via the blood flow it receives from the central blood compartment.

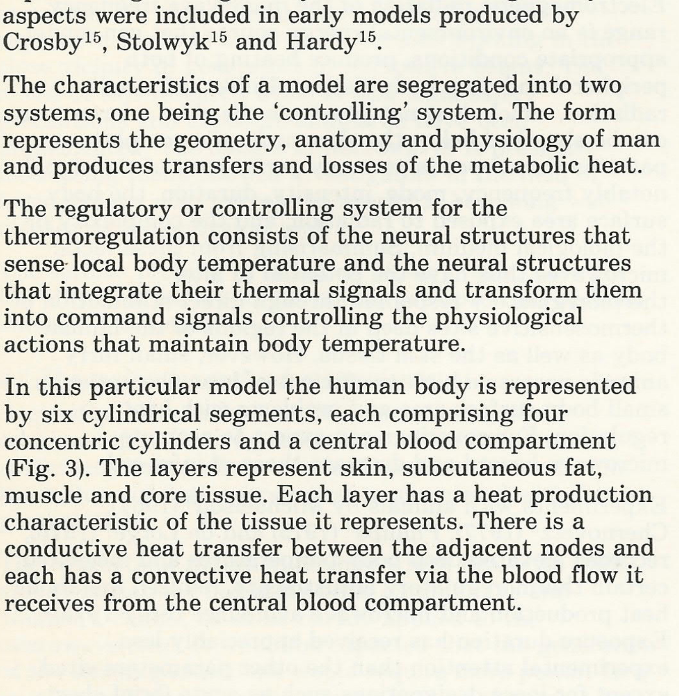

Fig. 3. Six segments of the controlled system and the arrangement of thermal nodes within one segment[15].

The controlling system, however, can be sub-divided into three parts, the integrating system, the peripheral system and the temperature sensing system. The first sub-system (integral) functions as a receiver and converts afferent signals into commands to the effector systems. The peripheral system then modifies these effector command signals in accordance with the conditions at the periphery. Lastly, the temperature sensing system consists of thermoreceptors, in the skin and in the brain.

The model recognises the fact that different skin areas contain different densities of thermal receptors and that both warm and cold receptors are present. Similarly the hypothalamus contains structures with warm and cold sensitivity. The integration of thermoreceptor signals occurs to some extent in the spinal cord but mostly in the anterior hypothalamus. The hypothalamus contains the reference system that determines the deep body temperature at which all of the systems are inactive; at this temperature the thermoregulatory system can be considered at its neutral or set point. The integrating system produces appropriate outward signals that control the systems of vasoconstriction, vasodilation, sweat secretion and shivering (Fig. 4).

These systems are scattered over various parts of the body. Sweat secretion is restricted to the skin, but the response is evenly distributed over its surface. Some areas of skin show more vasomotor response than others and shivering activity of the voluntary muscles is also non-uniform over the different muscle masses. The systems especially in the skin change their response according to local temperature.

In order to implement the model it is necessary to provide it with all the changes in physiological responses that occur within the ranges of heat stress, metabolic rate and body temperatures in which its use is anticipated. The model presented and discussed here rests on a great deal of specific experimentation specially designed to enlighten on the structure and characteristics of physiological control systems. It makes satisfactory predictions of dynamic responses to a variety of external and internal thermal stresses. Interestingly enough, examples of the fate of heat deposited in local areas are indicated in Figs. 5 and 6.

Observing from Fig. 5 the relatively high blood flow in the brain, the temperature rise due to deposition of heat at the rate of 100 watts is quite small. Fig. 6 indicates clearly that for a non-sweat escape condition all parts of the body increase in temperature at approximately the same rate.

Simulations in these figures are extensions of thermal and control system characteristics to unvalidated temperatures and should only be regarded as such.

3.12 Cumulative injury

This may occur when multiple exposures cause a weakness on a certain part of the body until damage is apparent. Eye opacities are a prime example in the occurrence of long-term injuries.

3.13 Secondary effects

A phenomenon with microwaves is that they are able to affect man indirectly via their own interactions with other materials. For instance, a man carrying a wrench in each hand formed a dipole or aerial. This circumstance was sufficient to induce a warming sensation in the body when he was surrounded by microwaves. Secondly, when the two tools were brought together and separated an electric shock was felt. However, static charges are extremely unlikely in high frequency microwave fields

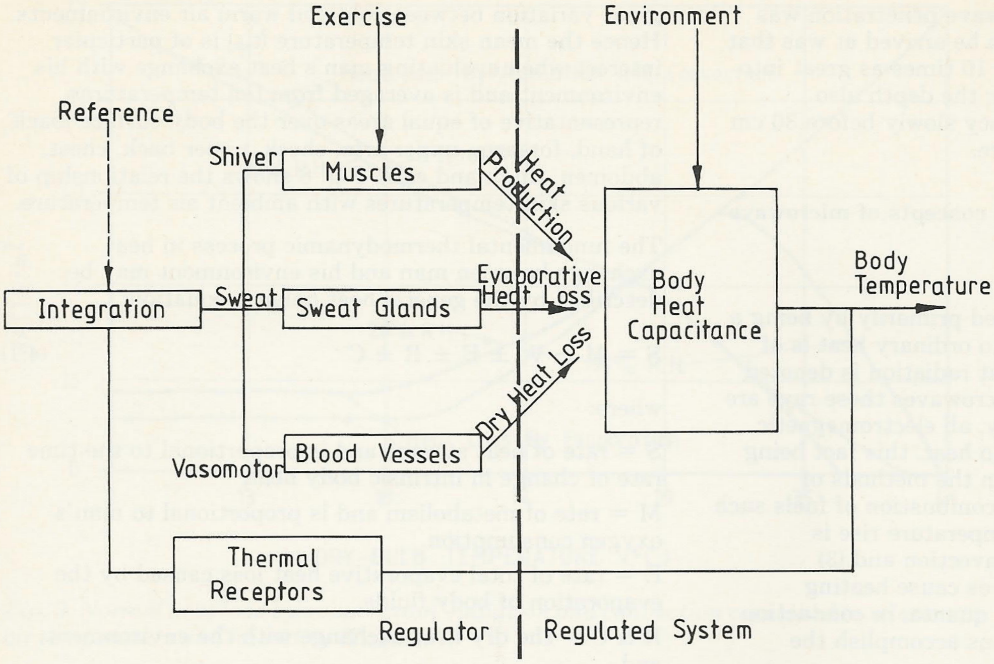

Fig. 4. Simplified block diagram of the human thermoregulatory control system[15].

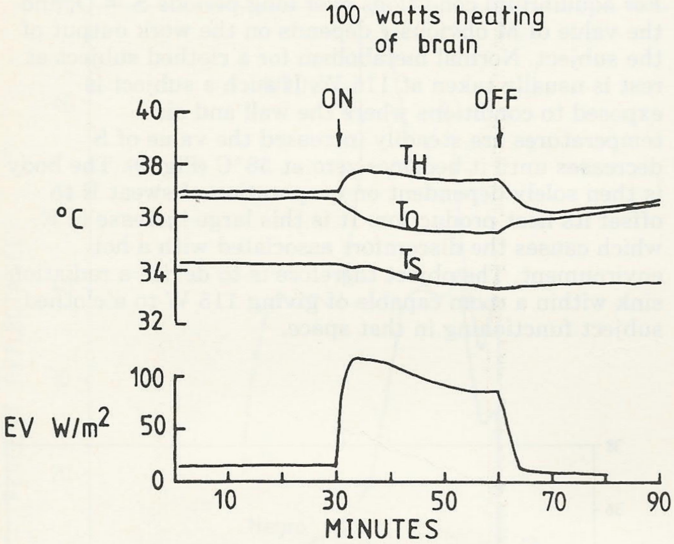

Fig. 5. Simulation of the effect of depositing 100 W of heat into the brain of a normal individual for a period of 30 minutes. Shown are the brain temp. (T_H), the esophageal temp. (T_O), the mean skin temp. (T_S), and the rate of sweat secretion (EV)[15].

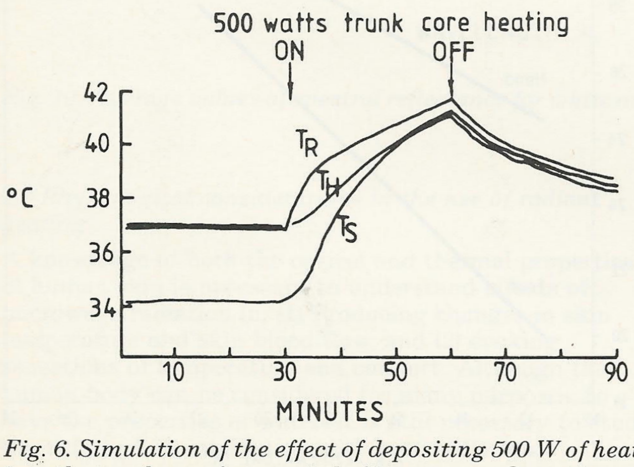

Fig. 6. Simulation of the effect of depositing 500 W of heat into the trunk core for a period of 30 minutes. Sweat is not allowed to evaporate. Shown are the trunk core temp. (T_R), brain temp. (T_H), and mean skin temp. (T_S)[15].

introduced to spaces within buildings as described in Section 4.

3.14 Perception and sensitivity

The warm sensation associated with microwave heating is dependent on local nerve supplies. Mumford[6] (1961) pointed out that those frequencies generating a feeling of warmth due to surface heating provide a warning in case of over-exposure. The body possesses insufficient temperature sensors but persons working with microwaves have experienced deeper sensations which may be unreliable warnings as sensory endings are less plentiful in deeper regions of the body. Michaelson[6] (1961) said that sensitivity is decreased at environmental temperatures over and above normal body temperature. One finding is that animals are less sensitive to microwaves at high air temperatures but are immediately sensitive to them at normal room temperature. Also a degree of desensitisation occurs within animals in as much as they are able to tolerate microwaves much better after some preliminary exposures. Sensing microwaves on the surface of the skin should be much greater than as for an equal dose of infra-red and the more visible radiation. With 3 cm microwaves there is a 20-40 per cent less elevation in skin surface temperatures. The scarcity of nerve fibres deeper into the body are more of a danger to the larger internal organs.

The threat is pertinent to frequency and penetration and greatest in small bodies as opposed to large ones. During experiments on animals' internal organs were strikingly higher in temperature whereas mouth and rectum temperatures remained normal. Abdominal temperatures were highest of those tested.

3.15 Microwave penetration

At sample wavelengths 1.27, 3.0 and 10.0 cm the greatest penetration will be at 10 cm[6]. For this wavelength yellow marrow has by far the deepest penetration with sebaceous tissue next. Wave penetration into other tissues such as the brain, skin and blood is more limited. Microwave energy would have decreased to 37 per cent of its initial value after about 10 cm into marrow, 2.5 cm into tallow and just under 1 cm into other tissues. Also water content of tissues will usually determine the depth of penetration to a large extent.

A comprehensive study of microwave penetration was made by Schwan[6]. The conclusion he arrived at was that the depth of penetration is about 10 times as great into muscle as opposed to fatty tissue; the depth also decreases with increasing frequency slowly before 30 cm and more rapidly above this figure.

4 The principles and application concepts of microwave radiant energy in space heating

4.1 General

Microwave heating is distinguished primarily by being a radiant process. Its relationship to ordinary heat is of some interest. The creation of heat radiation is denoted by the word emission and like microwaves these rays are destroyed by absorption. Actually, all electromagnetic radiations can be transformed into heat, this fact being provided by Herschel[13] in 1800. In the methods of applying heat to material by the combustion of fuels such as wood, coal, oil and gas, the temperature rise is obtained by (1) conduction, (2) convection and (3) radiation. While radiant heat waves cause heating through the absorption of energy quanta, in conduction and convection, molecular collisions accomplish the energy transfer (Fig. 7).

The radiant process described above causes a feeling of warmth on the human skin and this sensation is comparable with the heat emitted from a warm stove which involves mainly smaller wavelengths, although weak centimetre waves are included. Subcutaneous penetration can occur but this is very dependent on the controlled wave frequency and physiological characteristics of individuals. However, implications of deeper microwave penetration have been discussed in Section 3.

4.2 Thermal interchanges between man and his environment

The average temperature of the human body (t_b) depends on a balance between net heat produced and net heat loss to the environment. The body's principal heat source is oxidation of food elements (metabolism M). At the same time, the human body may be: (1) performing work (W); (2) losing heat by evaporation (E) body fluids; and (3) exchanging heat by radiation (R) and convection (C). During rest and exercise these processes result in an average deep body temperature (t_b) of from 36°C to 38°C[5].

There are many specific measures of internal body temperature including oral, rectal, esophageal (blood temperature) and eardrum (brain temperatures—thermoregulation). The skin surface temperature has

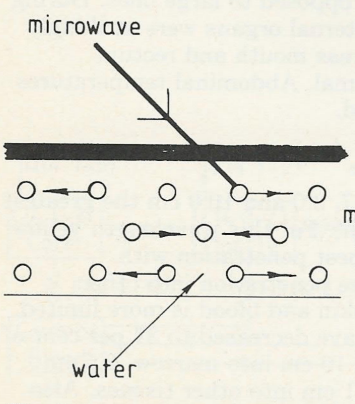

Fig. 7. Molecular cell movement producing warmth due to microwave penetration.

great variation between cold and warm air environments. Hence the mean skin temperature (t_{sk}) is of particular interest when evaluating man's heat exchange with his environment and is averaged from ten temperatures representative of equal areas over the body surface (back of hand, forearm, upper arm, cheek, upper back, chest, abdomen, thigh and calf). Fig. 8 shows the relationship of various skin temperatures with ambient air temperature.

The fundamental thermodynamic process in heat exchange between man and his environment may be described by the general heat balance equation:

$$S = M \pm W \pm E \pm R \pm C \tag{4.1}$$

where:

S = rate of heat storage and is proportional to the time rate of change in intrinsic body heat,

M = rate of metabolism and is proportional to man's oxygen consumption,

E = rate of total evaporative heat loss caused by the evaporation of body fluids,

R & C = the dry heat exchange with the environment, and

W = mechanical work accomplished.

For equilibrium conditions over long periods S = O, and the value of M obviously depends on the work output of the subject. Normal metabolism for a clothed subject at rest is usually taken at 115 W. If such a subject is exposed to conditions where the wall and air temperatures are steadily increased the value of S decreases until it becomes zero at 38°C (Fig. 9). The body is then solely dependent on evaporation of sweat E to offset its heat production. It is this large increase in E which causes the discomfort associated with a hot environment. The object therefore is to devise a radiation sink within a room capable of giving 115 W to a clothed subject functioning in that space.

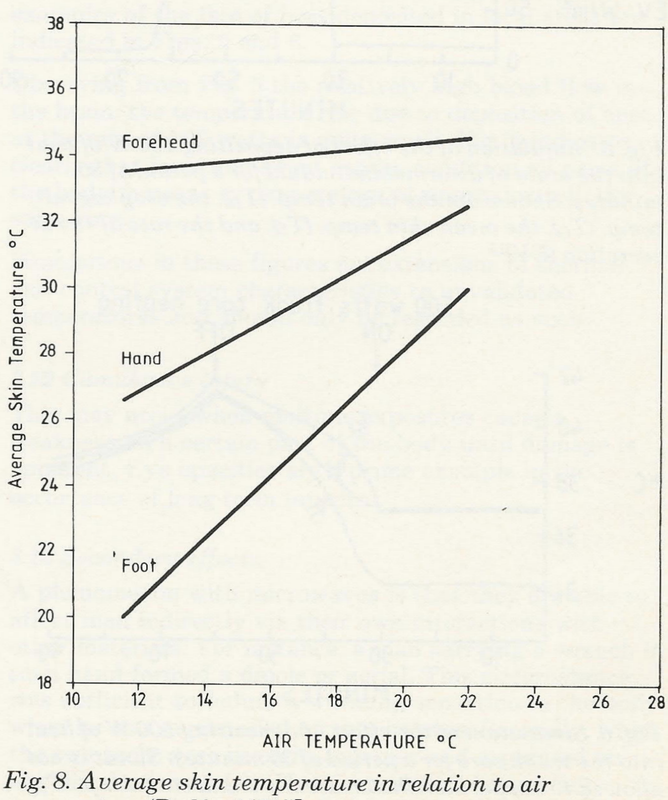

Fig. 8. Average skin temperature in relation to air temperature (Bedford '71')[7].

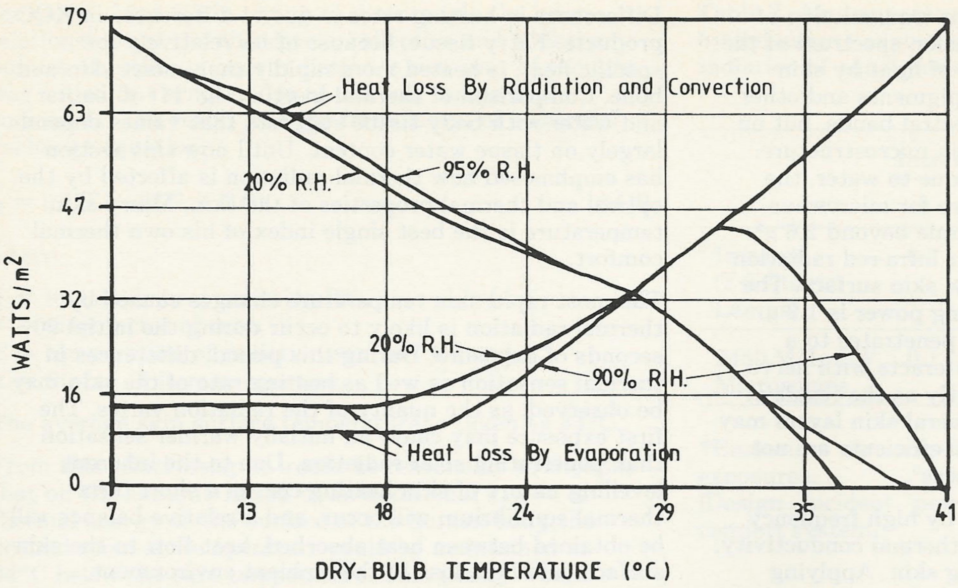

Fig. 9. Normal heat loss by evaporation, radiation and convection combined in relation to air temperature and humidity (Yaglou)[11].

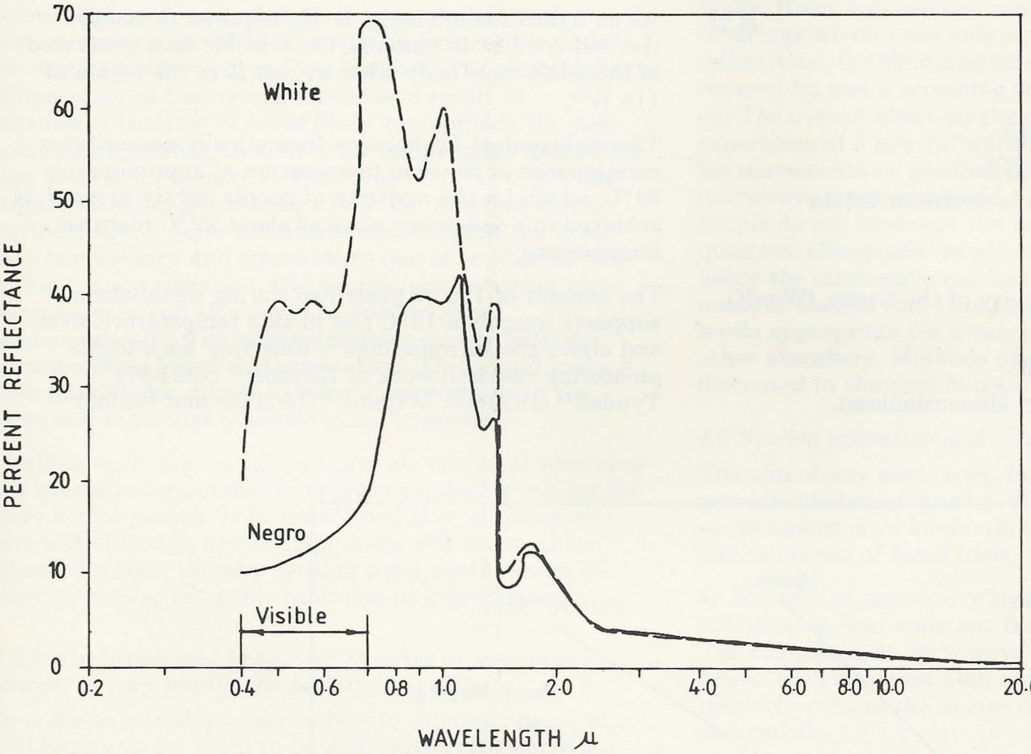

Fig. 10. Average values of spectral reflectance for white and dark-negro skin[5].

4.3 Physiological considerations in the use of radiant heating

A knowledge of both the optical and thermal properties of human skin is necessary to understand effects of microwave radiation in: (1) Producing changes in skin temperature and skin blood flow; and (2) evoking sensations of temperature and comfort. Although the human body can be considered for many purposes, to have the properties of water, it is still necessary to study the skin and its interaction with heat radiation.

Human skin has variable thickness over different parts of the body and is extremely non-homogeneous. This is an important factor in the study of those optical and thermal skin properties which determine heating by radiation. In a physiological study of thermal radiation effects, the first question is skin reflectance and transmittance in the region where skin does not fully radiate, i.e. 0.3 to 3.0 μ Fig. 10 shows reflection curves for white and dark negro skin. Skin reflectivity has been studied from the ultra-violet through to the far infra-red. From 3 to 20 μ reflectivity is low, approximately 1 per cent. Between 0.4 and 2.0 μ, reflectivity is highly variable, depending to a great extent on pigmentation as well as skin blood flow. Maximum reflectivity occurs from about 0.8 to 1.2 μ[5].

The second important optical factor is transmission of radiation into the skin. The transmission spectrum of the skin is shown in Fig. 10. Absorption of light by skin depends not only on skin and blood pigments and other substances which absorb specific spectral bands, but on degree of scattering action due to skin microstructure. The absorption bands in Fig. 10 are due to water, the principal absorber in biological tissues for microwave radiation. The skin is essentially opaque beyond 2.6 μ[5]. However, significant amounts of near infra-red radiation penetrate considerable distance below skin surface. The wavelength of the greatest penetrating power is 1.2 μ. At least 50 per cent of the radiation penetrates to a depth of 0.8 mm and thus directly interacts with nerve endings and small blood vessels. Partly as the result of scattering action and because the several skin layers may absorb differently, skin absorption co-efficients are not constant as a function of skin thickness[5].

Changes in skin temperature caused by high frequency microwave radiation also depend on thermal conductivity, density and specific heat of the living skin. Applying heat flow theory to the problem of skin heating, it can be shown that a parabolic relationship exists between exposure time and skin temperature rise for non-penetrating radiation is thus:

$$t_{sf} - t_{si} = \triangle t = 2Ia\sqrt{\pi kQC} \quad (4.2)$$

where

t_{sf} = final skin temperature (°C),

t_{si} = initial skin temperature (°C),

I = irradiation intensity from a source at kelvin temperature, (W/m^2),

t = time (seconds),

k = specific thermal conductivity of the tissue, (W/mK),

Q = density (kg/m^3),

C = specific heat (J/kgK), and

a = skin absorbance at kelvin (dimensionless).

Differences in heating rates is due to differences in KQC products. Fatty tissue, because of its relatively low specific heat, is heated more rapidly than moist skin and bone. Comparison of thermal inertia (Fig. 11) of leather and water with body tissue suggests that values depend largely on tissue water content. Until now this section has emphasised how thermal radiation is affected by the optical and thermal properties of the skin. Man's skin temperature is the best single index of his own thermal comfort.

The most rapid skin temperature changes caused by thermal radiation is likely to occur during the initial 60 seconds of exposure. During this period, differences in thermal sensation as well as heating rate of the skin may be observed, as the quality of the radiation varies. The first exposure may cause an initially warmer sensation than penetrating solar radiation. Due to the inherent levelling nature of skin heating curves a short term thermal equilibrium will occur, and a relative balance will be obtained between heat absorbed, heat flow to the skin surface, and heat loss to the ambient environment.

In utilising microwave radiant heating for human comfort, several simplifying concepts will be employed in what follows.

4.4 Levels of thermal equilibrium and resultant comfort

As an indication of energy levels required to sustain thermal comfort in humans, the sensible heat generated in the adult male body when at rest is in the region of 115 W[14].

Thermal comfort for humans theoretically necessitates maintenance of the skin temperature of approximately 34°C, which for the majority of people lightly dressed, is achieved in a sedentary state at about 22°C room air temperature[14].

The amount of 115 W generated during metabolism supports roughly a 12°C rise in skin temperature, over and above the surroundings[14]. Referring back to the pioneering research work of Kirchoff[13] (1824-87), Tyndall[13] (1820-93), Magnus[13] (1802-70) and Balfour

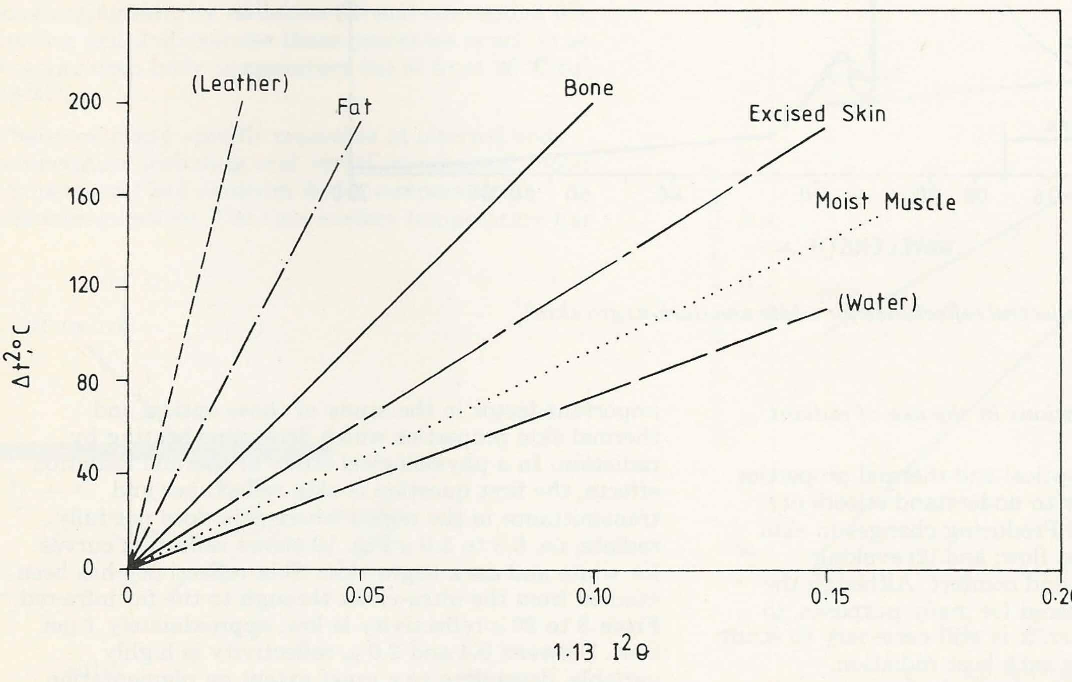

Fig. 11. Comparison of thermal inertia of fat, bone, moist muscle and excised skin with that of leather and water[5].

Stewart[13] (1828-87) in trying to disentangle the interrelationships of the absorptive, reflective and emissive powers of bodies of heat radiation, a new unit of thought was added to Carnot's idea of uniform temperature enclosures. The result was the formulation of a thermal exchange equation:

$$q = kF[T - (273 + 34)] \qquad (4.3)$$

where:

q = metabolic heat emission (W),
F = view factor of man,
T = temperature of radiation source (°C), and
k = Stefan Boltzman constant.

The average skin surface temperature is taken as 34°C.

From this relationship it would be reasonable to theorise that on the administration of an additional 115 W of thermal energy to the surface of the human body the desired room air temperature could be maintained at 24°C below the skin temperature. This theory would result in a 10°C comfortable room air temperature but at this figure cold touch and condensation could be a problem.

Therefore an approximate minimum of 13°C should be adopted as an acceptable winter internal air temperature before the occurrence of condensation. A design air temperature target of 15°C would be permissible under the required criteria of radiant comfort. The mean radiant temperature of the room surfaces as a result of microwave heating, is never likely to approach the dew-point corresponding to a 13°C air temperature and 80 per cent saturation humidity. The value of 13°C would also require something less than 115 W of radiation input. Furthermore, a quite low air radiant temperature is all that is necessary and equitable to one at or near 22°C without space radiation. Old people require higher air temperatures and this could be comfortably accommodated by microwave radiant energy with its corresponding lower air temperature. This would make it then possible for others to remain comfortable in the same and otherwise overheated environment.

Clothing ensemble design can provide means of screening out excess radiation and to depress comfort levels for the more active person. It is established that microwaves penetrate through dry clothing more effectively than infra-red waves, thereby causing a net containment of warmth similar to visible radiation in greenhouses.

4.5 Distribution and behaviour theories of microwave radiant energy within the building envelope

In order to introduce microwaves to internal spaces of buildings and for them to be distributed efficiently, a forethought to the building structure is essential. Positions of microwave beamers could be mounted in weatherproofed sites as an integral part of external walls and therefore maintained simply and easily from the outside. Alternatively, wave generators and components may be positioned in ceiling voids thereby being concealed to aesthetic satisfaction. At the same instance consideration to interior finishes and surface configurations must form part of the overall design plan.

Microwaves can be more effectively reflected than can infra-red, only if the internal structural surfaces are finished with a good metallic conductor. Therefore, contrary to the intentions of conventional space heating, the walls nor the room air need be significantly heated, adhering only to the 13°C minimum in order to prevent condensation. Table 2 shows the likely increase in room air temperature due to the presence of microwaves.

Table 2. Air temperature increment (°C) produced by ten-minute exposures to infrared or microwave radiation of equal power density*[3]

	Incident Power Density (mW/cm^2)					
	1	5	10†	15	20	25
Infrared (T-3 Quartz Lamps)	0.2	0.8	1.4	1.9	2.6	3.1
2450 MHz CW Microwaves	0.1	0.6	1.3	1.9	2.5	3.1

*Enclosure air was 35°C at beginning of all exposures.
†Design incident power density for microwave comfort system.

The heated space in this form would act as a resonant cavity holding a large number of wave forms which would tend to overlap slightly in frequency. The room would be relatively filled evenly with microwave radiant energy. Walls, floors and ceilings could be decorated with metallic coverings which have only a marginal effect on wave reflectivity. Building apertures such as doors might be covered by metal screening to avoid energy losses and so on. The overall effect on the occupant would be that experience of a general warming sensation irrespective of his movements or position in the space. Obviously, microwave emitters should be deployed so that objects or people do not intercept the radiation directly at close quarters. Occupants' needs would vary considerably, hence the microwave comfort system must incorporate a mode of control that is capable of maintaining energy levels appropriate for a varying number of occupants or other absorbers. Methods of system modulation are discussed in the next sub-section.

4.6 System operation and control modes

The aim of any engineered building environment is to provide a balanced quality of conditions within the environment, at a minimum total cost and with the minimum use of fossil fuels.

In the case of microwave radiant energy a fine tuning of radiation at near constant frequencies has to be achieved. The energising of microwave generators must be closely linked with occupant skin temperature in order to precisely control the degree of thermal comfort demanded.

As varying frequencies of microwave radiation cause varying penetration depths of the human body it is critical to emit radiation at a safe design frequency. Pulsed emission (intermittent on/off mode) at this frequency solves the problem of fine radiation balances between supply and demand of the engineered comfort system.

A temperature sensing probe, possibly in the form of a silicon chip, may be placed as an item of jewellery or clothing on the occupant. The location of this 'thermostat' preferably should be at the foot region as this is the most accessible thermosensitive site on the human body. Computation of skin temperature value would be the function for this device which in turn would transmit an electronic signal direct to the microwave emitter. Immediately, the emitter would alter its quantity of radiation emission by on/off action (Fig. 12).

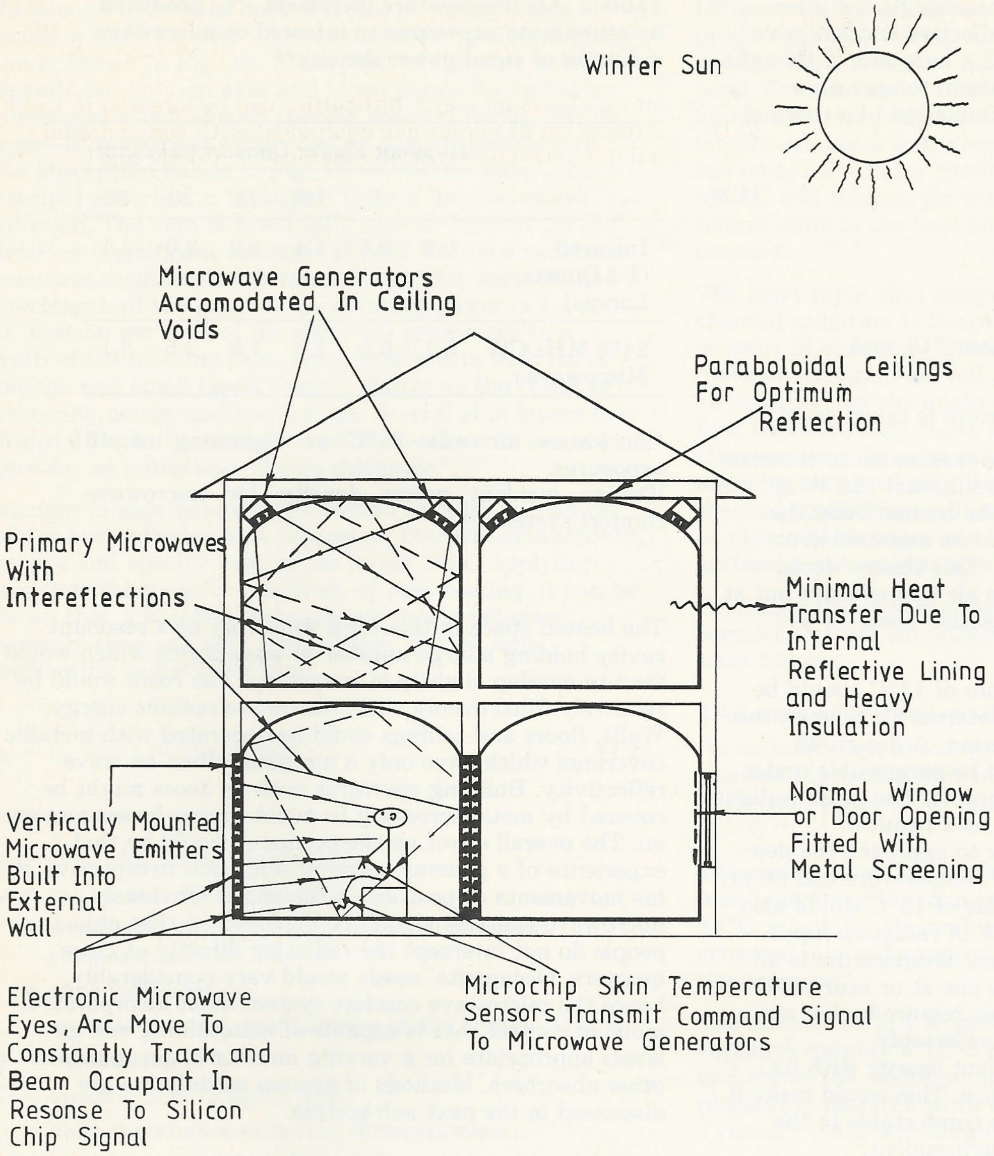

Fig. 12. Integrated system concept for microwave comfort heating.

Bursts of mean frequency radiation maintains the desired temperature of the skin (34°C). Various stages of the emitter can be operated as the space heating demand fluctuates.

5 Consideration of potential health hazards and exposure standards

In Section 3 health hazards posed by microwave exposure have been ascertained, and these dangers mainly relate to the industrial sector. Strictly speaking detriment caused by microwave radiation, such as eye opacities, has revealed its potential as occurring at the lower frequency of approximately 200 MHz. According to research carried out by the author the application of microwaves to surface warming of the skin is achieved at a mean frequency of 0.5 cm (250 GHz) or less, which is considerably higher.

With pulsed radiation at variable time intervals, it is highly unlikely that anything serious would occur due to time relaxation recovery in between. Indirect effects remain uncertain at ultra-high frequencies and thus require more field trials to assess their potential. On this basis it is only possible to comment on levels of permitted exposure and relate these to the resultant exposure likely to be experienced in microwave space heating enclosures.

The present radiation protection standards in Western countries permits field strengths of 10 mW/m^2 of body surface area, without reference to wavelength[14]. The power incident on a human being in a field of that intensity is in the order of 100 W, estimating the effective body area to be 1m^2.[14] The portion of flux absorbed by the skin is said to be approximately 0.35, corresponding to a frequency of 2.45 GHz, as used in microwave ovens. In addition, there is evidence that it increases to 0.8 at 8 GHz and to 0.95 in the range 40 to 90 GHz[14]. These exposure values certainly indicate that the degree of exposure likely to occur with our microwave comfort system is acceptable.

The legal permitted exposure standards were implemented on the basis that microwaves are harmful. The general opinion is that further experiments are necessary in order to determine parameters required in practice. However, a check guideline is that for short wavelengths the penetration depth in tissue with a sizeable water content is proportional to the square of the wavelength.

Hence, the approximate wavelength for cutaneous heating taking an assumed average depth of penetration as 1.5 mm into skin strata is:

$$d = \lambda^2 \tag{5.1}$$

where:

d = depth of penetration (cm),
λ = wavelength of microwaves (cm).

$0.15 \text{ cm} = \lambda^2$

transposing for λ this now becomes

$$\lambda = \sqrt{0.15}$$

Therefore, $\lambda = 0.387$ cm, which is not too far distant from the 0.5 cm wavelength region.

The evaluated wavelength above would be that which is desired to achieve sufficient skin penetration. With this principle of warming the skin, it seems probable that there is little difference between this and the similar warming sensation provided by the more conventional infra-red waves.

However, the figure of 10 mW/cm^2 was a value ascertained by experts some 30 years back when the matter of exposure to microwaves became important. The guidelines was a calculated power intensity that could effectively commence to raise the temperature of a person against his regulatory mechanism for temperature. Since people differ so greatly in this aspect and the working environment, the limit of 10 mW/cm^2 is offered only as a voluntary approximation.

Lastly, an area not covered so far is the influence microwaves may have on cardiac pacemakers. People fitted with these radio-isotope powered devices face a danger not only from microwaves, but all forms of electromagnetic interference. Certain types of pacemaker are protectively resistant to interference, those that are encased in a metal capsule, and therefore the internal circuitry would be shielded from external fields. Others like the ones covered in plastic are vulnerable. Also portable external type pacemakers, often for immobile persons are susceptible. Measures for complete safeguard against microwave interference are undoubtedly within the knowledge of pacemaker designers.

6 Discussion and conclusions

The majority of research work carried out on the radiation biology of microwave radiant energy has been real or simulated on animals rather than man himself. To extrapolate from these results for man would be an inaccurate and unreliable step and could only lead to more hazards.

Fear of this proposed form of space heating for human comfort may come from health endangerment associated with microwave ovens used in the cooking of foodstuffs. Similarly that application utilises the same method of heating the desired object, with the development of little heat in the structure or air of the enclosure. Microwave ovens produce wavelengths in the region of 12 cm which penetrate roughly 100 times deeper than 1 cm waves.

Injuries caused by microwaves are long term occurring at the longer band of wavelengths, like those of microwave ovens. Hyperthermia is avoided by the fact that the microwave system is thermally and precisely regulated.

The manufacturing technology of microwave generators is rapidly accelerating as can be seen from this research paper. Individual components are extremely flexible and on this basis no major difficulties can be foreseen in the integration of microwave equipment with conventional building structures, at all magnitudes. Costly and obtrusive plant rooms can be expected to reduce or in some cases disappear altogether, apart from the massive impact on the general market of heating apparatus.

Microwave space heating would benefit dwelling occupants by directly heating them instead of indirectly via structural elements and surrounding air. Use of this type of radiant energy would significantly contribute to the national energy conservation objectives presently being advocated by government departments. Also it would assist in alleviating the more crucial world energy crisis. All these advantages are attributable to the microwave plant being capable of operating sufficiently at lower than normal environmental temperatures. Heat losses through structural elements would be reduced accordingly. In some climates space heating may entirely be eliminated and in others heating of buildings may be reduced enough to just temper the occupied space thus avoiding thermal shock and cold surfaces.

For every benefit for man there may be a parallel hazard. Perhaps this investigation will prompt further and much needed research into the subject with a view of engineering our first microwave heated building.

APPENDIX

Telecommunications for buildings using microwaves

The increase in demand of communication systems has caused a rapid development of microwave relay systems, to alleviate the load on existing cable and radio communication systems.

Microwave transmission equipment could be readily designed to replace conventional transmitters/receivers in buildings, both for internal and external communications.

However, microwave signals are only capable of being transmitted a maximum distance of 500 km due to their diffraction and diffusion characteristics[9].

Microwave heating possibilities for domestic hot water

The potential of microwave heaters generating domestic hot water can only be assessed presently from an energy point of view. Equipment adaptability must be a subject for manufacturers but again, the flexibility of individual components always point towards certain feasibility.

The power required to heat a fixed quantity of domestic hot water from 10°C to 65°C storage temperature in 2 hours is 300 W (Fig. A1). This is really no different to that required by conventional immersion heaters.

However, if we shorten the regeneration time to 1 hour the required power using microwaves reads off as 6000 W whereas the equivalent conventionally is 6420 W. Obviously, it is clear that microwaves prove themselves energy wise at the higher power intensities and the benefits increasing as the regeneration time decreases or is quickened.

Concluding from this characteristic of microwave water heating, economic advantages can be obtained at short heat-up periods, especially where building use causes frequent and heavy demands on the domestic hot water storage and supply system/s. Hospitals and process industries are examples of such consistent loads.

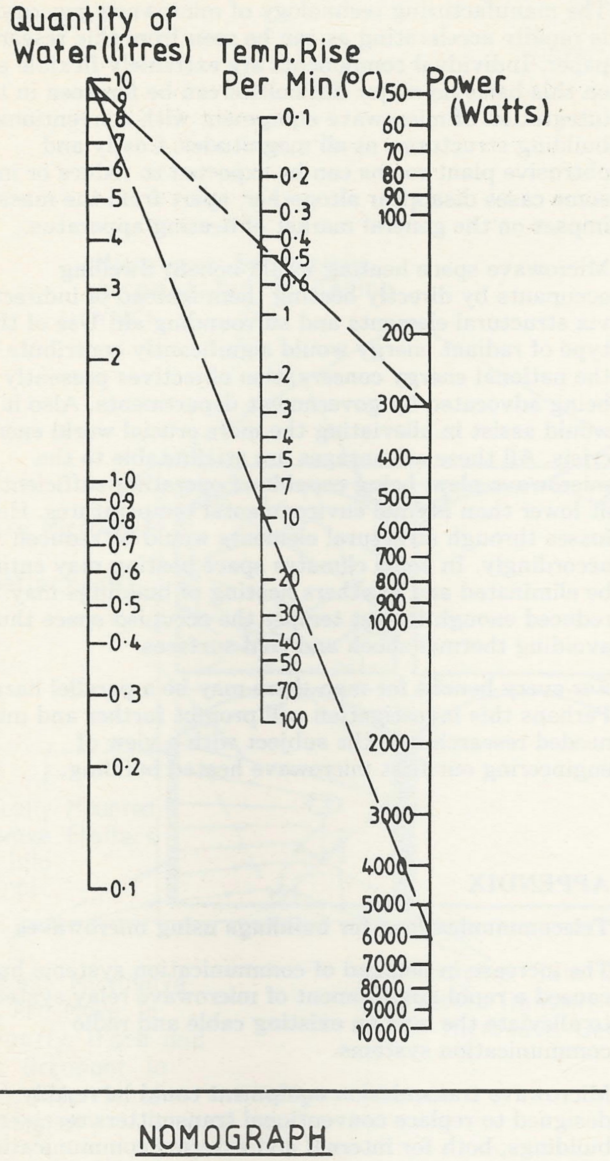

Fig. A1. Heating of water by microwaves. Power required for temperature rise per minute[6].

Acknowledgements

Many experts have so far carried out enormous amounts of work in the vast field of electromagnetic radiation. Microwaves form one part of that field together with the scientists, physicists and engineers who try to study about their behaviour, and as this paper hopes to reveal, the different beneficial uses.

It is because of this deep feeling that acknowledgement is made to Prof. D. Copson[6] at the University of Peurto Rico for the use of his excellent work on microwave heating. Also to Marshall Cavendish Ltd.[9] for their encyclopaedia, 'How It Works', and to R. V. Pound[14] of Harvard University for use of his interesting report. Similar credits are extended to the past and present authors of works as listed in the references.

Finally, special thanks are due to Dr. Derek J. Croome[7] of the University of Bath, for his enthusiasm, interest and technical contribution to the paper, especially on the aspect of applications and control modes.

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Summary In museums and galleries the preservation ot the artifacts is of paramount importance and air conditioning is demanded to give the stable conditions required. Though ideal temperatures and humidities for materials vary, a compromise is possible within the tolerance of most items. Humidity change is the primary consideration and psychrometric data is analysed to find the ways to reduce the energy changes in treated air to produce stable room conditions. While plant elements can be of conventional design, a new control philosophy based on accurate relative humidity sensors is developed. When enthalpy control is not limiting fresh air to minimum, humidity control loops adjust chiller coil or fresh air ratio or steam injection while a temperature control loop copes with the heat gains or losses. Means of adjusting the minimum fresh air to match population are explored. By reducing the energy changes needed to meet supply conditions economical operation is achieved.

A control philosophy for the economical air conditioning of museums and galleries

A. READING, CEng, MIEE, MIMechE

1 Introduction

Full air conditioning to stringent standards is being increasingly demanded by conservators in museums and art galleries. The difficulty of satisfying these demands is often made more onerous as many of the buildings are old and of historic and architectural interest in themselves. Indeed the buildings which house national collections in London are all Grade I or Grade II listed buildings. Attempts to insert air conditioning create problems of space and aesthetics to add to those of normal engineering. Generally, but particularly in art galleries, 'all air' systems must be used to prevent the chance of failure causing water damage to the collection. This article explores the physics of 'all air' systems required by museums and suggest an approach leading to solutions which are as economical in energy as practicable. Though centred on the needs of museums the discussions are equally valid for many other systems.

2 Criteria and constraints

All artifacts have a range of temperatures and humidities which they can tolerate without appreciable deterioration over long periods, provided there are no rapid changes in either parameter. The actual values of these ranges vary for a mixed collection so that no answer can be perfect. The humidity requirements are the key consideration, but with mixed collections the ideal values of humidity can vary over the range of 20 to 60 per cent RH so a compromise must be accepted except where items are so valuable that they are treated separately. There is a small span of humidity around 50 per cent RH which is within the tolerance band of most artifacts and a concensus is appearing that the standard should be 50 per cent RH $\pm$ 4 per cent. For art galleries and where paintings are major items a higher level of 54 per cent RH with the same tolerance is required as in the National Gallery, London. Temperature presents a different problem. Most artifacts would benefit from stable temperatures at a lower level than people are prepared to tolerate. In particular consideration has to be given to warding staff who must remain with the collections for long periods without activities to generate warmth. The collections need stability of temperature so a value of 20°C $\pm$ 1°C is chosen but as a concession to economy some conservators are prepared to accept a mean temperature adjusted gradually between 19°C in winter to 21°C or 22°C in summer.

Museums and galleries suffer from other factors which affect air conditioning design as these buildings often have large lighting loads and highly variable humidity gains from visitors. As will be shown later air infiltration, which is always difficult to quantify and control, is a significant factor in design. Apart from the infiltration through the fabric, visitors enter and leave through doors moving considerable volumes of air in the process. These doors are often large to allow the entry and removal of artifacts so that the conditioned space cannot be pressurised effectively as the pressure differential would make the doors unmanageable. Pressure relief dampers to alleviate this problem make the differential pressure too low to prevent substantial interchange of air between the controlled and outer space. An air lock between the spaces is the best solution and is essential where the controlled space is entered directly from outside the building. Unfortunately in many museum buildings internal air locks are not physically or aesthetically possible and, though minimised, air infiltration has to be accepted.

3 Early analysis

Accepting these requirements and limitations this analysis shows that a practical solution is possible. The range of outside conditions experienced in southern England is covered by shaded areas on the psychrometric chart in Fig. 1 where R represents the room conditions of 20°C and 54 per cent RH. In the area A, above the enthalpy line E, the external air has more energy than the room air so it is best to use as little of it as possible. Area B covers external conditions where the energy is

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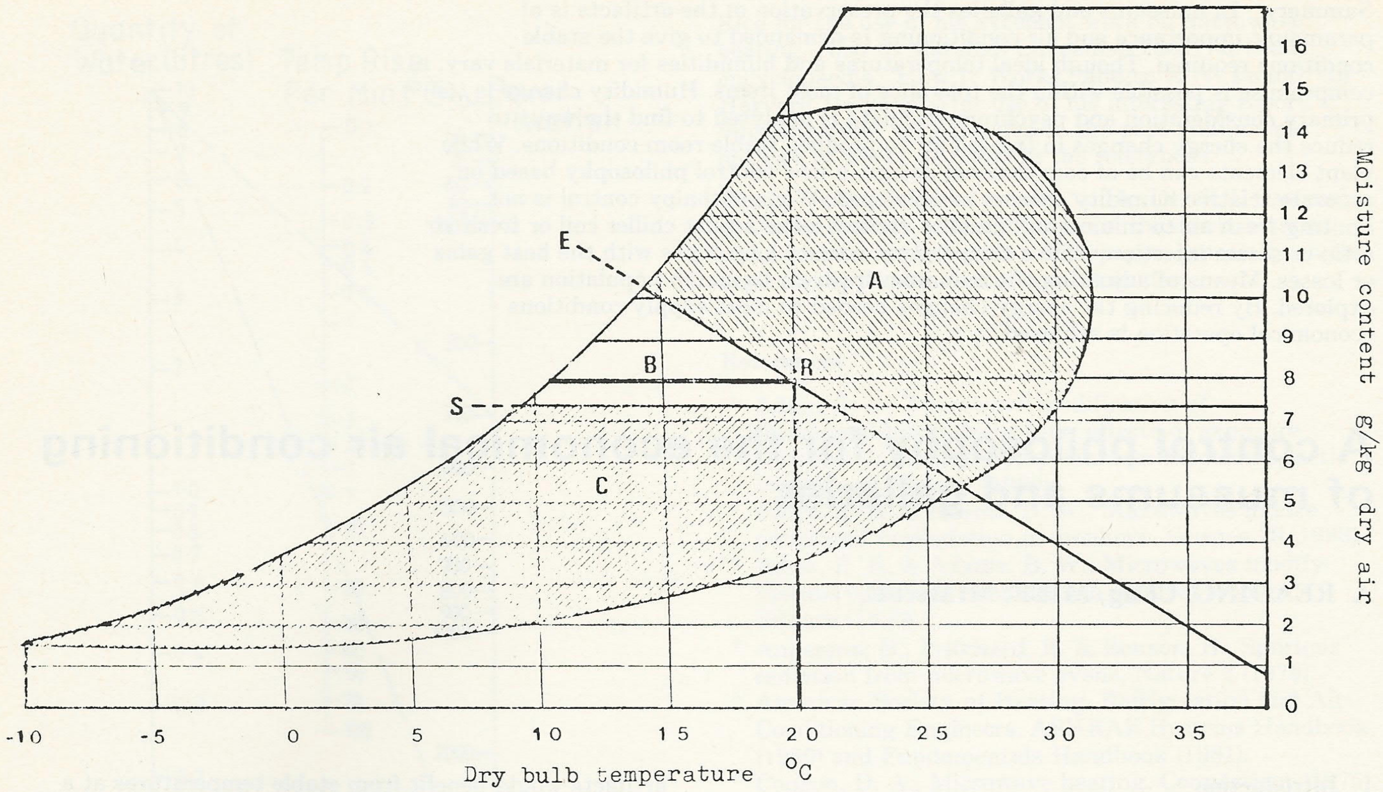


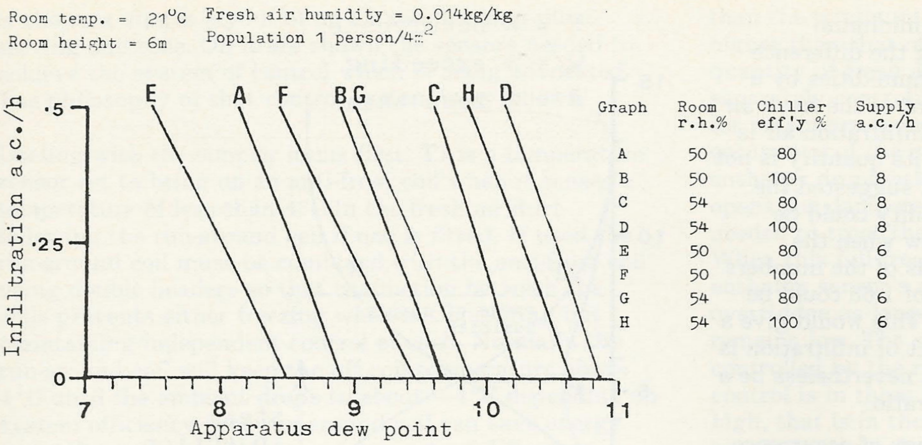
Fig. 1. Psychrometric chart.

less than in room air but the humidity is higher than the required supply conditions. Any mixture of this air with room air will have a higher energy content than the fresh air. As the air must be chilled to remove moisture, it is more economical to treat the fresh air and so full fresh air should be used in these conditions. In the area C the balance between fresh and recirculated air is variable depending on temperature and humidity and a solution which gives minimum energy consumption is complex. While computer modelling could provide a solution, what is needed is an answer which leads to plant which can be readily operated and maintained. Any system of control which produces a mixed air condition in the area B will use extra energy to chill the mixed air to dew point and then need reheat so it will not be the most economic.

Investigation of the psychrometric conditions in the area C shows that there is a small area near the line S in which proportioning the fresh air using temperature control is the most efficient solution. A further small area where temperatures are above the room temperature gives better results with proportioning controlled by moisture balance. For the remainder there is no difference in energy terms of using either control. If temperature control is used however there are conditions near the line in which the mixing under temperature control would put the mixed condition into area B and avoiding this condition would be a complication in the control system. A control which proportions the fresh and recirculated air so that the humidity is on or below the line S, the supply humidity, will be a better compromise. The limitation on keeping close to the line is the requirement of fresh air for the occupants. This requirement changes with occupancy and there are marked changes in level when the building closes to the public and at night when only security patrols are present. The position of the line S moves up to approach the level of room absolute humidity when fresh air needs and humidity gains are minimal. The possibilities of controlling the minimum fresh air level are investigated later.

To apply these considerations to plant design some assumptions are made to make meaningful calculations possible. The required supply humidity at any moment is influenced by the required room humidity and moisture gains from occupants and infiltration of fresh air. The effects are explored using room humidities of 50 and 54 per cent RH. The density of occupation is varied from the assumed maximum of 4 m^2 per person to 8, 10 and 12 m^2 per person for room heights of 5 and 6 m. Infiltration rates of 0.5, 0.25 and 0 air changes per hour are tested for supply air rates of 6 and 8 ach. Supply air changes of 6 and 8 are chosen as being towards the lower end of the scale of practical values. Heavyweight construction with double glazing and sunscreening above laylights in top lit galleries help to reduce fabric gains and losses so low values may be usable with advantages for plant and duct sizes. There is, however, a danger of giving the feeling of a 'dead' atmosphere which will be unpopular. The effects of the efficiency of the chiller coil on the apparatus dew point are investigated for efficiencies of 100 and 80 per cent. For the calculations a fresh air humidity of 0.014 kg/kg is assumed as a value rarely exceeded in this country and because this only occurs in summer the room temperature is taken as 21°C. Fresh air of 0.0095 m^3/s per person is provided as a reasonable quantity for odour control. To aid the application of this approach to other situations the tabulated results are set out in Appendix 1 and the formulae on which they are based in Appendix 2. It should be noted that in this analysis the infiltration air is not treated as a simple load on the system but as an uncontrolled fresh air supply.

The graph Fig. 2 shows the relationship between apparatus dew point, infiltration rates and the other factors and illustrates the importance of reducing the air infiltration and getting high efficiency chiller coils to keep the apparatus dew point at a manageable value. Against the advantages of high efficiency coils in dew point control must be set the disadvantage of higher fan loads so a balance must be made between the two. In



Graph	Room r.h.%	Chiller eff'y %	Supply a.c./h
A	50	80	8
B	50	100	8
C	54	80	8
D	54	100	8
E	50	80	6
F	50	100	6
G	54	80	6
H	54	100	6

Fig. 2. Effect of infiltration on dew point.

Fig. 3 the graphs show the effect of population density on the required supply dew point. In the calculations for Fig. 3 it was assumed that the fresh air was adjusted to the minimum for the given population density. This assumption affects apparatus dew point where the efficiency of the chiller coil is somewhat less than 100 per cent. In the worst case changing the fresh air ratio from 0 to 25 per cent with a coil efficiency of 80 per cent means that the dew point of the apparatus and probably the temperature of the chilled water supply must be reduced by approximately 1°C. The effect of population density on the fresh air ratio is shown in Fig. 4. This implies that major economies in treating fresh air can be made if it can be controlled in relation to the number of people present. If a fixed ratio is adopted while the public is present it must be set to the maximum needed for the most crowded occasion and will involve the treatment of excessive quantities of fresh air for most of the time. A scheme to measure, even approximately, the numbers present could produce considerable energy savings.

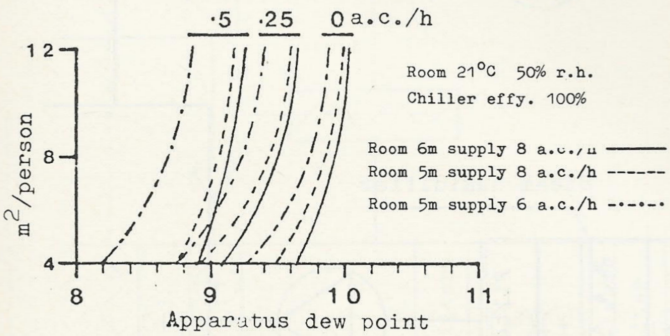


Fig. 3. Effect of population on dew point.

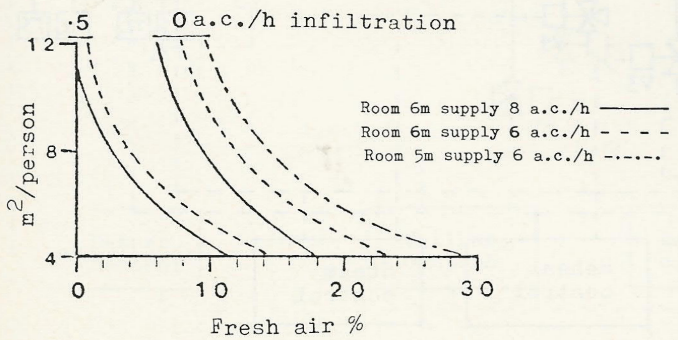


Fig. 4. Effect of population on fresh air ratio.

Various methods have been suggested for measuring the population of the air conditioned space, including turnstiles, infra-red counters and carbon dioxide monitoring. All have some disadvantages but CO_2 monitoring can be effective[2] and merits consideration. There is another alternative. People in a stable temperature and moderate humidity emit moisture and require fresh air at predictable rates. Both moisture emission and the need for fresh air increase and decrease together as the degree of activity changes. Where there is no infiltration, the people are the sole source of moisture and the air conditioning system can itself monitor the population. The difference in absolute humidity between the room condition and the supply air is thus a measure of the fresh air requirement. In Appendix 3 the relationship between the supply absolute humidity and the minimum fresh air ratio is derived and the effect of population on that ratio is illustrated in Fig. 5. The difference between supply and room absolute humidity is thus a possible means of controlling the minimum fresh air ratio and hence increasing the range over which fresh air can be used for free cooling. Infiltration will introduce an error in this relationship but since it brings fresh air with it the error is towards excess fresh air which is acceptable in reasonable quantities. Moisture migration through walls could be a factor in the equation but with the generally massive construction the rate of change will be small and have only a smoothing effect on changes.

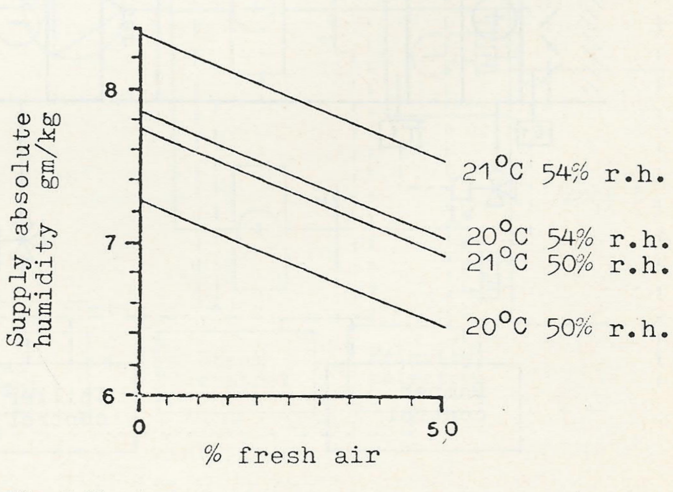


Fig. 5. Fresh air requirement for people.

The percentage of fresh air to meet the minimum requirements can be derived by dividing the difference between the room and supply absolute humidities by a factor of 1.63. This does not take account of the fresh air which infiltrates as is normal practice. Infiltration air is just as good to breathe but calculating its quantity is not usually practicable. In the type of plant suggested the effect of infiltration on the supply humidity could be measured by comparison of fresh air flow when the dampers were at minimum and the needs of the numbers of people present. From this the factor of 1.63 could be adjusted for the particular installation. This would give a better approximation but since the effect of infiltration is variable it would be imperfect. It would nevertheless be a great improvement on a fixed fresh air ratio.

Data published[1], which give the frequency of occurrence of outside air conditions at Heathrow, are useful in assessing how the load on a system will vary. Analysis of the data shows that dew point control of supply air humidity is only needed for 15 to 30 per cent of the daytime hours depending on the population density and fresh air ratio in the conditioned space. In the night hours if the fresh air is reduced to near zero the value will be about 15 per cent. Under lower humidity conditions when the supply absolute humidity will be controlled by adjusting the fresh/recirculated air ratio a limit will be reached when fresh air has been reduced to a minimum. For fresh air humidities below this level the supply humidity will be brought up to the required value by steam injection. The lower line on Fig. 6 shows that for a room temperature of 19°C and 50 per cent RH, with 8 ac/h and 0.25 ac/h of infiltration, steam injection will be needed for approximately 50 per cent of the year.

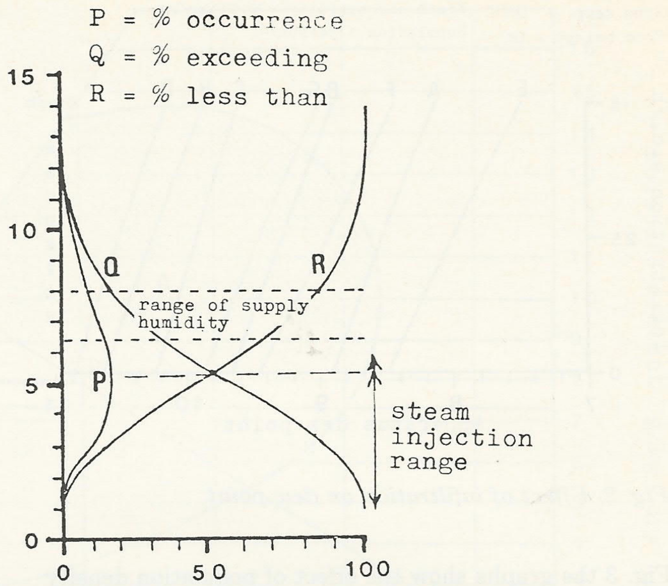


Fig. 6. Occurrence of humidity conditions. Supply absolute humidity in gm/kg is plotted against per cent occurrence.

4 Application to plant control

It may be argued that the calculations and analysis above are fine in theory but could not work in practice and until recently the argument would have had merit. The important change that has made it possible to produce a plant to exploit this theoretical analysis is the availability of accurate and reliable sensors[3] to measure the humidity and give suitable control signals. In Fig. 7

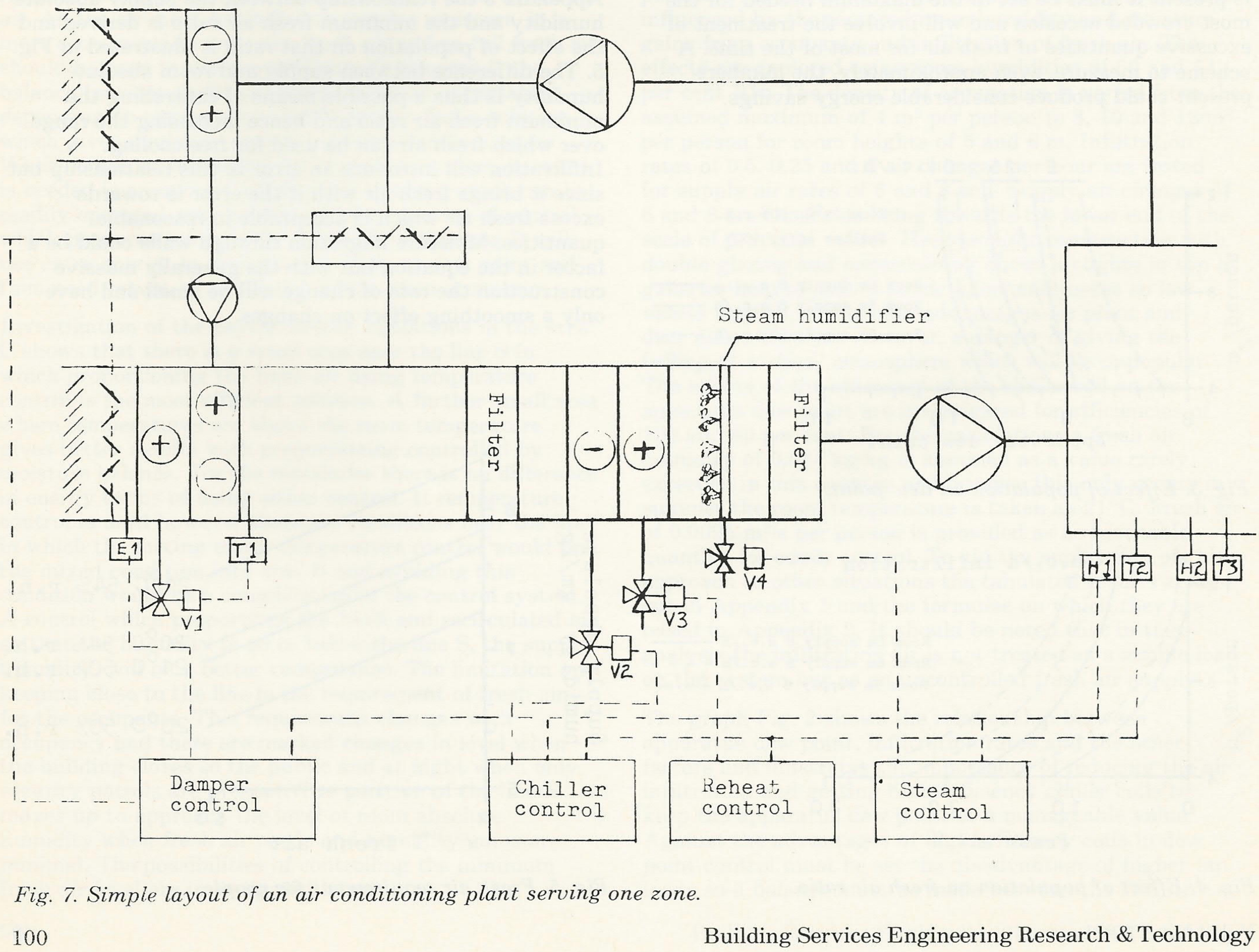


Fig. 7. Simple layout of an air conditioning plant serving one zone.

is shown a simple layout of an air conditioning plant serving one zone. On it are shown the sensors needed to achieve the system of control which is being advocated. The philosophy of that control system is as follows.

Dealing with the simpler items first, T1 is a temperature sensor set to bring on an anti-frost coil when it senses a temperature of less than 4°C in the fresh air duct following the run-around coil if one is fitted. If used the run-around coil must be combined with the anti-frost coil using double headers so that conduction between the coils prevents either freezing when not operating but maintaining independent control of each. Normally the run-around coil will keep the off coil temperature above 4°C until the ambient drops to about —4°C depending on system efficiency. The run-around coil can save energy when there are a few degrees temperature difference between exhaust and ambient air but in this system the use of the run-around coil could be counter-productive when the fresh air was being used for free cooling. It could be controlled to come into action when the heating coil was operating and the chiller was not and also when the outside temperature exceeded the room temperature by a few degrees. With these limitations the economic viability of the run-around coil must be carefully considered. In gallery installations the air should be purified by a carbon filter downstream of T1 to remove sulphur dioxide, etc. from the incoming air and a further filter should do the same for the mixed air stream. Often physical limitations prevent the ideal being achieved and a carbon filter is only put in the mixed air stream where it is preceded and followed by particulate filters.

The fresh and recirculating air are proportioned by damper control in which an enthalpy sensor E1, set at the required room enthalpy, prevents the entry of more than the minimum fresh air if the fresh air enthalpy is higher than that of the room. The value of this minimum quantity of fresh air may be fixed in a simple system or separately controlled with the option of night set back or, for maximum economy as discussed above, be related to occupancy of the conditioned space. When the external enthalpy drops below that of the room, the controller opens the dampers to full fresh air to reduce the energy needed to treat the air to the required supply conditions. When this full fresh air condition has been reached the enthalpy sensor's control over the dampers can be overridden as long as the enthalpy of the fresh air remains low. The supply air humidity is entirely controlled by the room humidity detector H1. This control is in three steps. When the external humidity is high, that is in the region above line S in Fig. 1 the sensor controls the dew point of the chiller coil through V2. As the fresh air dew point falls the chiller coil will be shut off. At that point there are two alternatives. If fresh air enthalpy is high that is in the area A below line S then the fresh air will be held to minimum by the enthalpy control and the humidity sensor will make up the deficiency in humidity by controlling steam injection. When the conditions fall into area C the dampers will have opened to full fresh air and the room humidity sensor will take control of the dampers to adjust the mixed air humidity to the value S required in the supply air. At some point further reductions in fresh air humidity will mean that the dampers will close under the control of the humidity sensor to the minimum fresh air condition. At this point the humidity sensor would change over to control of the steam injection to keep the required supply humidity. These change-over points can be readily detected by position switches to detect when V2 is closed and when the dampers are at minimum position.

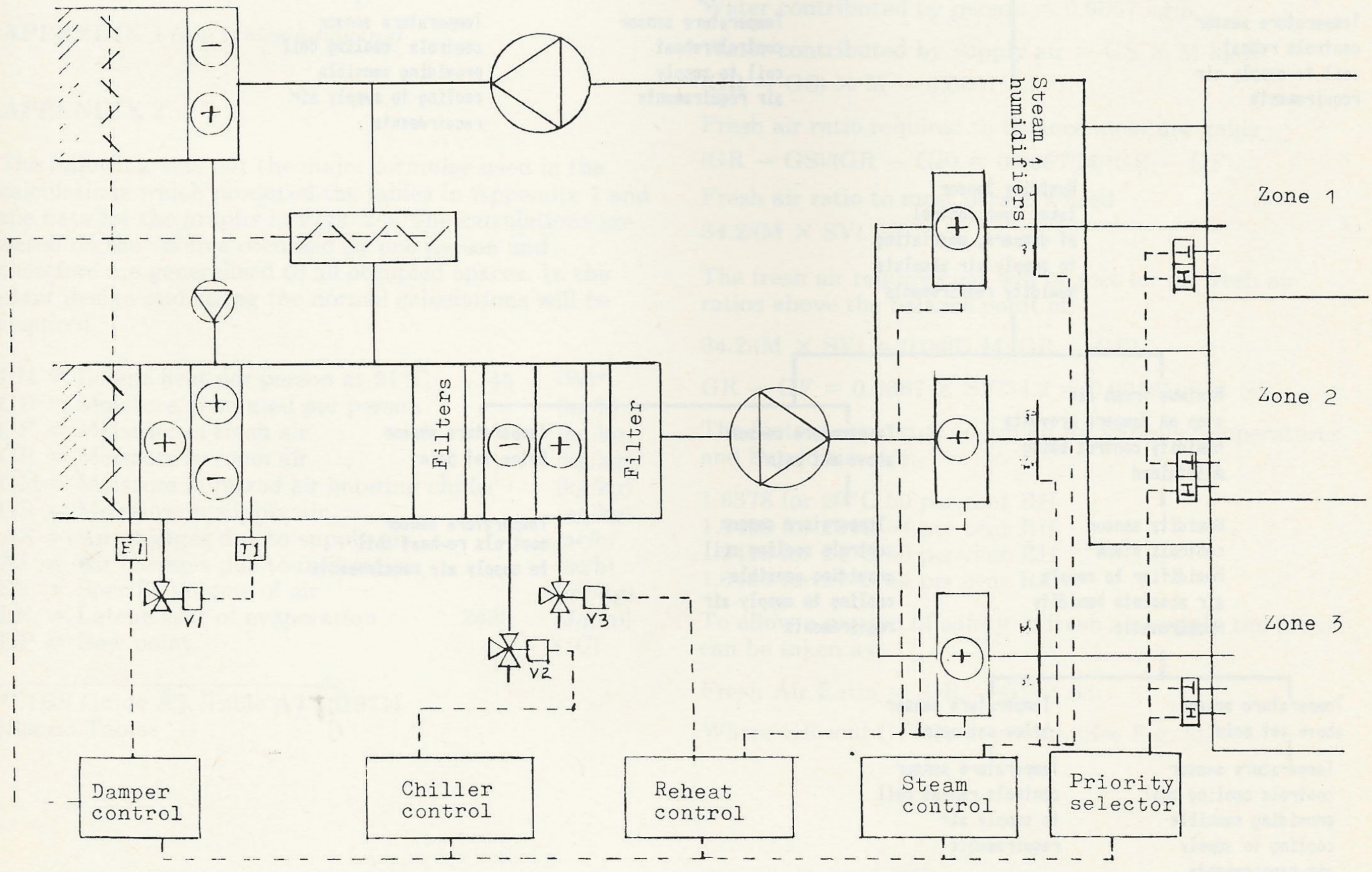


Fig. 8. Controlling several rooms from one plant.

Temperature is controlled by the room temperature sensor T2 which controls the reheat coil to adjust supply conditions until the mixed air temperature rises to a level where the reheat coil is turned off. When this is detected by a position switch on V3, the temperature sensor transfers its control to the chiller coil. An interlock will be required so that the opening of V2 under control of the temperature sensor does not affect the humidity control which at that time will be controlling either the dampers or steam injection. It could be upset because at that stage it would be expecting the valve V2 to be closed. Simple interlocks will avoid conflict of control.

Two sensors H2 and T3 are shown. These are important safeguards for the system as they are connected to an independent monitoring and alarm system which allows records to be stored of the actual conditions to which the artifacts are subjected over long periods. This is very useful information for conservation and at the same time can give alarm at a manned location when necessary either directly or as an element in a building management system. A further refinement is to use three sensors and the computer to operate a best of three voting system for both monitoring and control.

In museums and galleries it is often necessary to condition several rooms from one plant. These rooms however may have different heat and humidity gains due to different orientation or occupancy. The plant shown in Fig. 8 demonstrates how the control philosophy

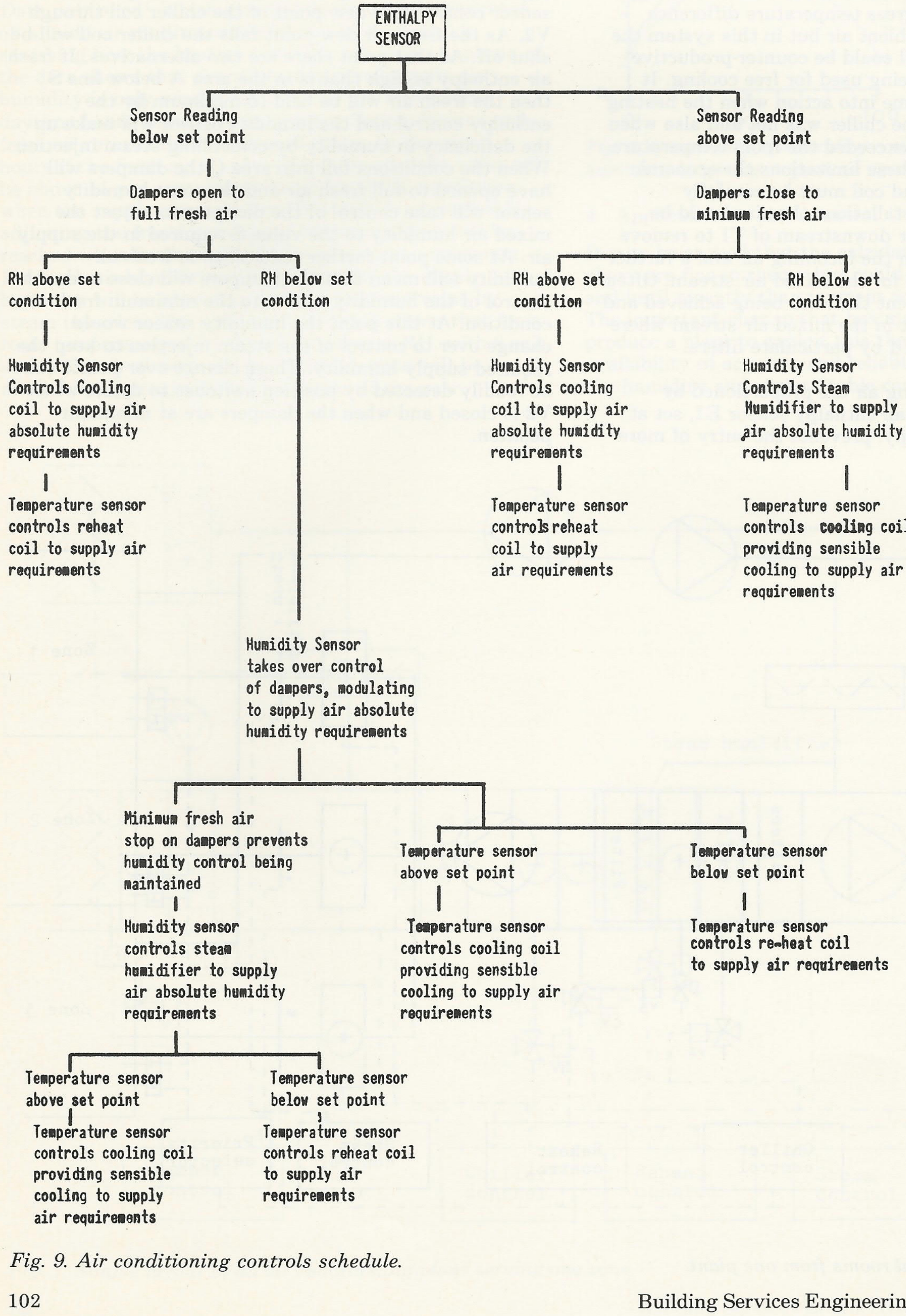


Fig. 9. Air conditioning controls schedule.

advocated can be applied in these circumstances. Basically the plant is similar to the simple plant with no steam injection and in each of the branches a small additional reheat coil and steam injection are added. Each zone has its own humidity and temperature sensors. The control system also is similar to the simple system in that the zone humidity sensor which is demanding the lowest humidity is in control of the main plant and the others control steam injection into their zone to make up the deficit. In temperature control the sensor requiring the lowest supply temperature controls the main plant and the difference in the other zones is made up by reheat in the duct feeding that zone with control by the local temperature sensor. In this case the reheat batteries in the ducts will only have to cope with the differences between the various zones' heat losses. This may make electric heaters in the ducts a practical proposition.

The system advocated is unorthodox, particularly in its advocacy of control by humidity sensors, but is considered to offer advantages which are well worth developing both for the accuracy of control and the reduction of energy requirements. Even where all the properties of this system cannot be applied to a particular installation the use of the design philosophy can be advantageous.

References

[1] Collingbourne, R. H. and Legg, R. C., The frequency of occurrence of hourly values of outside air conditions in the United Kingdom, Heat. and Vent. Eng., **55**, 6-13 (July/August 1981).

[2] Warren, B. F., Energy saving in buildings by control of ventilation as a function of indoor carbon dioxide concentration, BSER&T, **3**, (1), 4-12 (1982).

[3] Example Novasina Series 83 sensors from Humitec.

APPENDIX 1 *(see tables following)*

APPENDIX 2

The following sets out the major formulae used in the calculations which produced the tables in Appendix 1 and the data for the graphs in Figs. 2-5. The calculations are based on the volume occupied by one person and therefore are generalised to all occupied spaces. In the plant design and sizing the normal calculations will be required.

LH	= Latent heat per person at 21°C	45	(W)*
GP	= Moisture generated per person		(kg/h)
GF	= Moisture in fresh air		(kg/kg)
GR	= Moisture in room air		(kg/kg)
GM	= Moisture in mixed air entering chiller		(kg/kg)
GS	= Moisture in supply air		(kg/kg)
AS	= Air changes due to supply air		(ac/h)
AI	= Air changes due to infiltration		(ac/h)
SV	= Specific volume of air		(m^3/kg)
LE	= Latent heat of evaporation	2430	(J/gm)†
DP	= Dew point		(°C)
MF	= Mass flow through space occupied per person		(kg/h)
FA	= Fresh air requirement per person		(m^3/h)
FR	= Fresh air as percentage of supply		(per cent)
H	= Room height		(m)
A	= Area occupied by one person		(m^2)
G	= Moisture in Supply air less saturation moisture content of air at 0°C		(gm/kg)

*CIBS Guide A7, Table A7.1 (1971)
†Steam Tables

GP = (LH/LE) × (3600/1000) = 0.667 kg/h at 21°C

MF = AS × (A × H)/SV

GS = (GR × (AS + AI) − (GF × A1))/AS − GP/MF

FA = 0.0095 × 3600 = 34.2 m^3/h

FR = 100 × (34.2 − (A × H × AI))/(A × H × (AI + AS))

GM = (FR/100) × (GF − GR) + GR

Over the range of interest in this work a formula was found by fitting a curve to the dew points given in the CIBS Guide. This formula has an accuracy better than 2 per cent and mostly better than 1 per cent.

$$G = GS \times 1000 - 3.789$$

$$DP = 0.13 + 3.51 \times G - 0.445 \times G^{1.562}$$

APPENDIX 3

Calculation of Minimum Fresh Air Ratio

It is assumed that there is no infiltration.

Mass of air occupied by one person during one hour, M = AS + A + H/SV kg/h.

Mass of water in air = GR × M.

Water contributed by person = 0.0667 kg/h.

Water contributed by supply air = GS × M kg/h.

(GR − GS) × M = 0.0667.

Fresh air ratio required to balance moisture gains (GR − GS)/(GR − GF) = 0.0667/M/(GR − GF).

Fresh air ratio to meet person's need 34.2/(M × SV).

The fresh air requirement will be met for all fresh air ratios above the balance point of

34.2/(M × SV) = 0.0667/M/(GR − GF)

GR − GF = 0.0667 × SV/34.2 = 0.0019503 × SV

The factors converted to gm/kg for various temperatures and humidities are:

1.6378 for 20°C 50 per cent RH
1.6496 for 20°C 54 per cent RH
1.6446 for 21°C 50 per cent RH
1.6464 for 21°C 54 per cent RH

To allow a margin of safety in fresh air supply the factor can be taken as:

Fresh Air Ratio = (GR − GF)/1.63

Where GR and GF are in gm/kg. (See Fig. 5).

Room height 5 metres RH 50 per cent

Area/ Person m²	Chiller Effy per cent	Air Change Supply Ach	Air Change Infilt Ach	Fresh Air per cent	Supply Humid. gm/kg	Dew Point °C
4	100	8	0.50	14.24	7.0210	8.69
4	80	8	0.50	14.24	7.0210	7.79
4	100	8	0.25	17.70	7.2160	9.11
4	80	8	0.25	17.70	7.2160	8.22
4	100	8	0.00	21.37	7.4111	9.52
4	80	8	0.00	21.37	7.4111	8.63
4	100	6	0.50	18.62	6.7754	8.15
4	80	6	0.50	18.62	6.7754	6.93
4	100	6	0.25	23.36	7.0354	8.72
4	80	6	0.25	23.36	7.0354	7.51
4	100	6	0.00	28.50	7.2955	9.28
4	80	6	0.00	28.50	7.2955	8.06
8	100	8	0.50	4.18	7.1943	9.07
8	80	8	0.50	4.18	7.1943	8.62
8	100	8	0.25	7.33	7.3894	9.48
8	80	8	0.25	7.33	7.3894	9.04
8	100	8	0.00	10.69	7.5844	9.88
8	80	8	0.00	10.69	7.5844	9.44
8	100	6	0.50	5.46	7.0065	8.66
8	80	6	0.50	5.46	7.0065	8.06
8	100	6	0.25	9.68	7.2666	9.22
8	80	6	0.25	9.68	7.2666	8.63
8	100	6	0.00	14.25	7.5266	9.76
8	80	6	0.00	14.25	7.5266	9.18
10	100	8	0.50	2.16	7.2289	9.14
10	80	8	0.50	2.16	7.2289	8.78
10	100	8	0.25	5.26	7.4240	9.55
10	80	8	0.25	5.26	7.4240	9.20
10	100	8	0.00	8.55	7.6191	9.95
10	80	8	0.00	8.55	7.6191	9.60
10	100	6	0.50	2.83	7.0527	8.76
10	80	6	0.50	2.83	7.0527	8.28
10	100	6	0.25	6.94	7.3128	9.32
10	80	6	0.25	6.94	7.3128	8.85
10	100	6	0.00	11.40	7.5729	9.85
10	80	6	0.00	11.40	7.5729	9.39
12	100	8	0.50	0.82	7.2521	9.19
12	80	8	0.50	0.82	7.2521	8.89
12	100	8	0.25	3.88	7.4471	9.60
12	80	8	0.25	3.88	7.4471	9.31
12	100	8	0.00	7.12	7.6422	10.00
12	80	8	0.00	7.12	7.6422	9.71
12	100	6	0.50	1.08	7.0835	8.83
12	80	6	0.50	1.08	7.0835	8.43
12	100	6	0.25	5.12	7.3436	9.38
12	80	6	0.25	5.12	7.3436	8.99
12	100	6	0.00	9.50	7.6037	9.92
12	80	6	0.00	9.50	7.6037	9.53

Room height 5 metres RH 54 per cent

Area/ Person m²	Chiller Effy per cent	Air Change Supply Ach	Air Change Infilt Ach	Fresh Air per cent	Supply Humid. gm/kg	Dew Point °C
4	100	8	0.50	14.24	7.6893	10.09
4	80	8	0.50	14.24	7.6893	9.32
4	100	8	0.25	17.70	7.8647	10.44
4	80	8	0.25	17.70	7.8647	9.68
4	100	8	0.00	21.37	8.0401	10.78
4	80	8	0.00	21.37	8.0401	10.02
4	100	6	0.50	18.62	7.4567	9.62
4	80	6	0.50	18.62	7.4567	8.57
4	100	6	0.25	23.36	7.6906	10.09
4	80	6	0.25	23.36	7.6906	9.05
4	100	6	0.00	28.50	7.9245	10.56
4	80	6	0.00	28.50	7.9245	9.52
8	100	8	0.50	4.18	7.8629	10.44
8	80	8	0.50	4.18	7.8629	10.06
8	100	8	0.25	7.33	8.0383	10.78
8	80	8	0.25	7.33	8.0383	10.41
8	100	8	0.00	10.69	8.2137	11.12
8	80	8	0.00	10.69	8.2137	10.75
8	100	6	0.50	5.46	7.6881	10.09
8	80	6	0.50	5.46	7.6881	9.57
8	100	6	0.25	9.68	7.9219	10.55
8	80	6	0.25	9.68	7.9219	10.05
8	100	6	0.00	14.25	8.1558	11.01
8	80	6	0.00	14.25	8.1558	10.51
10	100	8	0.50	2.16	7.8976	10.51
10	80	8	0.50	2.16	7.8976	10.20
10	100	8	0.25	5.26	8.0730	10.85
10	80	8	0.25	5.26	8.0730	10.55
10	100	8	0.00	8.55	8.2484	11.18
10	80	8	0.00	8.55	8.2484	10.89
10	100	6	0.50	2.83	7.7344	10.18
10	80	6	0.50	2.83	7.7344	9.77
10	100	6	0.25	6.94	7.9682	10.64
10	80	6	0.25	6.94	7.9682	10.24
10	100	6	0.00	11.40	8.2021	11.10
10	80	6	0.00	11.40	8.2021	10.70
12	100	8	0.50	0.82	7.9207	10.55
12	80	8	0.50	0.82	7.9207	10.30
12	100	8	0.25	3.88	8.0961	10.89
12	80	8	0.25	3.88	8.0961	10.65
12	100	8	0.00	7.12	8.2715	11.23
12	80	8	0.00	7.12	8.2715	10.98
12	100	6	0.50	1.08	7.7652	10.24
12	80	6	0.50	1.08	7.7652	9.90
12	100	6	0.25	5.12	7.9991	10.70
12	80	6	0.25	5.12	7.9991	10.37
12	100	6	0.00	9.50	8.2329	11.16
12	80	6	0.00	9.50	8.2329	10.83

Room height 6 metres RH 50 per cent

Area/ Person m^2	Chiller Effy per cent	Air Change Supply Ach	Air Change Infilt Ach	Fresh Air per cent	Supply Humid. gm/kg	Dew Point °C
4	100	8	0.50	10.88	7.0787	8.82
4	80	8	0.50	10.88	7.0787	8.07
4	100	8	0.25	14.24	7.2738	9.23
4	80	8	0.25	14.24	7.2738	8.50
4	100	8	0.00	17.81	7.4689	9.64
4	80	8	0.00	17.81	7.4689	8.90
4	100	6	0.50	14.23	6.8524	8.33
4	80	6	0.50	14.23	6.8524	7.31
4	100	6	0.25	18.80	7.1125	8.89
4	80	6	0.25	18.80	7.1125	7.89
4	100	6	0.00	23.75	7.3726	9.44
4	80	6	0.00	23.75	7.3726	8.44
8	100	8	0.50	2.50	7.2232	9.13
8	80	8	0.50	2.50	7.2232	8.76
8	100	8	0.25	5.61	7.4182	9.54
8	80	8	0.25	5.61	7.4182	9.17
8	100	8	0.00	8.91	7.6133	9.94
8	80	8	0.00	8.91	7.6133	9.58
8	100	6	0.50	3.27	7.0450	8.75
8	80	6	0.50	3.27	7.0450	8.24
8	100	6	0.25	7.40	7.3051	9.30
8	80	6	0.25	7.40	7.3051	8.81
8	100	6	0.00	11.87	7.5652	9.84
8	80	6	0.00	11.87	7.5652	9.36
10	100	8	0.50	0.82	7.2521	9.19
10	80	8	0.50	0.82	7.2521	8.89
10	100	8	0.25	3.88	7.4471	9.60
10	80	8	0.25	3.88	7.4471	9.31
10	100	8	0.00	7.12	7.6422	10.00
10	80	8	0.00	7.12	7.6422	9.71
10	100	6	0.50	1.08	7.0835	8.83
10	80	6	0.50	1.08	7.0835	8.43
10	100	6	0.25	5.12	7.3436	9.38
10	80	6	0.25	5.12	7.3436	8.99
10	100	6	0.00	9.50	7.6037	9.92
10	80	6	0.00	9.50	7.6037	9.53
12	100	8	0.50	−0.29	7.2713	9.23
12	80	8	0.50	−0.29	7.2713	8.98
12	100	8	0.25	2.73	7.4664	9.64
12	80	8	0.25	2.73	7.4664	9.40
12	100	8	0.00	5.94	7.6615	10.03
12	80	8	0.00	5.94	7.6615	9.80
12	100	6	0.50	−0.38	7.1092	8.88
12	80	6	0.50	−0.38	7.1092	8.55
12	100	6	0.25	3.60	7.3693	9.43
12	80	6	0.25	3.60	7.3693	9.11
12	100	6	0.00	7.92	7.6294	9.97
12	80	6	0.00	7.92	7.6294	9.65

Room height 6 metres RH 54 per cent

Area/ Person m^2	Chiller Effy per cent	Air Change Supply Ach	Air Change Infilt Ach	Fresh Air per cent	Supply Humid. gm/kg	Dew Point °C
4	100	8	0.50	10.88	7.7472	10.21
4	80	8	0.50	10.88	7.7472	9.57
4	100	8	0.25	14.24	7.9226	10.56
4	80	8	0.25	14.24	7.9226	9.92
4	100	8	0.00	17.81	8.0980	10.90
4	80	8	0.00	17.81	8.0980	10.26
4	100	6	0.50	14.23	7.5338	9.77
4	80	6	0.50	14.23	7.5338	8.91
4	100	6	0.25	18.80	7.7677	10.25
4	80	6	0.25	18.80	7.7677	9.39
4	100	6	0.00	23.75	8.0016	10.71
4	80	6	0.00	23.75	8.0016	9.85
8	100	8	0.50	2.50	7.8918	10.49
8	80	8	0.50	2.50	7.8918	10.18
8	100	8	0.25	5.61	8.0672	10.84
8	80	8	0.25	5.61	8.0672	10.53
8	100	8	0.00	8.91	8.2426	11.17
8	80	8	0.00	8.91	8.2426	10.86
8	100	6	0.50	3.27	7.7266	10.17
8	80	6	0.50	3.27	7.7266	9.74
8	100	6	0.25	7.40	7.9605	10.63
8	80	6	0.25	7.40	7.9605	10.21
8	100	6	0.00	11.87	8.1944	11.08
8	80	6	0.00	11.87	8.1944	10.67
10	100	8	0.50	0.82	7.9207	10.55
10	80	8	0.50	0.82	7.9207	10.30
10	100	8	0.25	3.88	8.0961	10.89
10	80	8	0.25	3.88	8.0961	10.65
10	100	8	0.00	7.12	8.2715	11.23
10	80	8	0.00	7.12	8.2715	10.98
10	100	6	0.50	1.08	7.7652	10.24
10	80	6	0.50	1.08	7.7652	9.90
10	100	6	0.25	5.12	7.9991	10.70
10	80	6	0.25	5.12	7.9991	10.37
10	100	6	0.00	9.50	8.2329	11.16
10	80	6	0.00	9.50	8.2329	10.83
12	100	8	0.50	−0.29	7.9400	10.59
12	80	8	0.50	−0.29	7.9400	10.38
12	100	8	0.25	2.73	8.1154	10.93
12	80	8	0.25	2.73	8.1154	10.72
12	100	8	0.00	5.94	8.2908	11.27
12	80	8	0.00	5.94	8.2908	11.06
12	100	6	0.50	−0.38	7.7909	10.29
12	80	6	0.50	−0.38	7.7909	10.01
12	100	6	0.25	3.60	8.0248	10.75
12	80	6	0.25	3.60	8.0248	10.48
12	100	6	0.00	7.92	8.2586	11.20
12	80	6	0.00	7.92	8.2586	10.93

Summary Improved data correlations have been derived for buoyancy-driven convective heat transfer from the internal surfaces of naturally-ventilated buildings. They cover the full range of laminar, transitional and turbulent airflows, and are based on the mathematical model of Churchill & Usagi (1972). The new correlating equations are presented in a convenient form for incorporating into modern computer programs which simulate the dynamic thermal performance of buildings. They compare favourably with the available exerimental data for isolated surfaces, and are shown to be an improvement on the 'standard' correlations recommended in the CIBS Guide. The factors which affect the accuracy of such data correlations, when used for the energy-conscious design of 'real' buildings, are briefly discussed.

Improved data correlations for buoyancy-driven convection in rooms

F. ALAMDARI, MSc and G. P. HAMMOND, MSc, CEng, MIMechE, MInstR

List of symbols

A, B, C	empirical coefficients in correlating Equations (3) and (5)	
a, b	empirical coefficients in correlating Equation (7)	
A_s	area of heat transfer surface	m^2
C_p	fluid specific heat at constant pressure	J/kgK
g	gravitational acceleration	m/s^2
Gr	Grashof number ($\equiv \rho^2 g \beta \triangle T L^3/\mu^2$)	
h_c	surface-averaged convective heat transfer coefficient	$W/m^2 K$
k_f	fluid thermal conductivity	W/mK
L	characteristic length of heat transfer surface	m
m,n,p,q	exponent in correlating Equations (3), (5) & (7)	
Nu	Nusselt number ($\equiv h_c L/k_f$)	
Pr	Prandtl number ($\equiv Cp\mu/k_f$)	
P_s	perimeter of heat transfer surface	m
q_c	convective heat flux	W/m^2
Ra	Rayleigh number ($\equiv$ Gr Pr)	
T_a	temperature of the quiescent (ambient) air	K
T_f	'film temperature'	K
T_w	'wall' (surface) temperature	K
β	coefficient of cubic expansion ($= T_f^{-1}$ for gases)	K^{-1}
$\triangle T$	($\equiv T_w - T_a$)	K
μ	fluid dynamic viscosity	kg/m s
ρ	fluid density	kg/m^3

1 Introduction

In the last decade it has become increasingly recognised that in order to develop realistic methods for the energy-conscious design of buildings it is necessary to model the dynamic thermal response of the system. These dynamic models normally require computational solution (as noted in the reviews by Clarke[1] and Day[2]; the latter summarising the efforts of the UK Science and Engineering Research Council to stimulate research in this area), in contrast to the manual-calculation methods used with the traditional *steady-state* procedures. Unfortunately, the accuracy of the new generation of building thermal models is presently limited by uncertainties in the input data, particularly for air infiltration and convective heat transfer rates (see, for example, the results of the International Energy Agency (IEA) study reported by Irving[3]). The aim of the present study was therefore to develop improved methods for calculating buoyancy-driven convective heat transfer within naturally-ventilated buildings.

The convective heat flux from a surface may be written, from 'Newton's Law of Cooling', in the form:

$$q_c = h_c \triangle T \tag{1}$$

The surface-averaged convective heat transfer coefficient, h_c, may then be combined with the corresponding radiative coefficient[4] to yield an internal 'surface resistance' for the building elements. In the case of buoyancy-driven (sometimes, but misleadingly, termed 'free' or 'natural') convection, h_c is itself a function of the temperature difference, $\triangle T$, as well as the length of the surface and the physical properties of the convected fluid. Dimensional analysis may be employed to correlate experimental data reflecting this dependence in terms of dimensionless parameters (defined above in the List of Symbols):

$$Nu = Nu\ (Gr, Pr) \tag{2}$$

The Nusselt number relation, for fluids with moderate Prandtl numbers ($0.7 < Pr < 70$), may be closely represented by a 'power-law' of the form:

$$Nu = C\ Ra^n \tag{3}$$

The exponent, n, is found to be about 1/4 for low Rayleigh numbers typically in the range $10^4 < Ra < 10^6$, which correspond to laminar flow induced by short surface lengths and/or small temperature differences. Conversely, for $Ra > 10^{10}$ transition to turbulent flow occurs, and n asymptotes to a value of about 1/3 (except for stable-stratification near horizontal surfaces where diffusion on a molecular scale persists up to quite high

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Rayleigh numbers, and n has a value of about 1/5; see Section 2.3 below). Thus, conventional practice is to calculate h_c via data correlations of the form of Equation (3), using an exponent of 1/4 for Rayleigh numbers less than about 10^8 and 1/3 otherwise. This 'two-part' correlaion, implying an abrupt transition, has been adopted in the CIBS Guide[4]. It was also used by the present authors[5] to model off-cycle, buoyancy-driven heat transfer in their 'intermediate-level' calculation procedure for warm-air heated rooms. When used with dynamic thermal models the two-part correlation Equation (3) is usually close to its intersection point, due to the relatively large size of typical building elements: ceilings, floors, walls and windows. This point corresponds to the location of maximum error in the calculated heat transfer coefficient, and the sudden change in the exponent there may also give rise to numerical instability in building thermal models.* In Section 2 below, a more elaborate set of correlating equations is developed that cover the full range of laminar, transitional and turbulent flows, and avoid the disadvantages of the simpler, 'two part model'. These improved data correlations are presented in a dimensional form that permit the direct calculation of convection coefficients appropriate to buoyancy-induced air movement at conditions typical of those found in the built environment.

2 Development and validation of the improved correlating equations

The problem of obtaining a correlating equation which will fit both laminar and turbulent buoyancy-driven convection is one of a class of problems involving transfer processes in which solutions are known for asymptotically large and small values of an independent variable, namely:

$$y \rightarrow A x^p \text{ as } x \rightarrow 0$$
$$y \rightarrow B x^q \text{ as } x \rightarrow \infty$$

where x and y are the independent and dependent variables respectively. Fortunately, Churchill & Usagi[6] have developed a general solution for this class of problems in the form:

$$y = \{ (Ax^p)^m + (Bx^q)^m \}^{1/m} \tag{4}$$

where $m > 0$ if $p < q$, and vice versa. They originally applied this formulation to laminar buoyancy-driven convection with $x = Pr$, and to laminar, mixed (combined buoyancy-driven and forced) convection. In the present case of laminar/turbulent buoyancy-driven convection a solution may be obtained in the form of Equation (4) as:

$$Nu = \{ (A\, Ra^p)^m + (B\, Ra^q)^m \}^{1/m} \tag{5}$$

*J. A. Clarke, private communication, 1981.

An examination of experimental data for vertical and horizontal surfaces (see Sections 2.1 to 2.2 below) suggests an optimal value of the exponent m of 6. The empirical coefficients in Equation (5) were chosen as mean values from the range employed in the 'standard' data correlations for the asymptotic states published in the literature, and these are summarised in Table 1. This equation, together with the latter coefficients, is valid over an extensive range ($10^4 < Ra < 10^{12}$) which encompasses all the conditions of practical significance for the built environment. The physical properties of the convected fluid, air in the case of buildings, may be obtained from standard tables such as those given by Mayhew & Rogers[7]. These properties are primarily dependent on temperature, and their values at the so-called 'film temperature', namely:

$$T_f = (T_a + T_w)/2 \tag{6}$$

are normally employed.

In the context of the built environment the physical properties of air do not vary greatly. It is therefore possible, and obviously convenient, to simplify the 'exact' correlating Equation (5) so that the convection coefficient is recovered in a dimensional form. Adopting a median film temperature applicable to naturally-ventilated buildings (say 27°C) this yields:

$$h_c = \left[\left\{ a \left(\frac{\Delta T}{L} \right)^p \right\}^m + \left\{ b (\Delta T)^q \right\}^m \right]^{1/m} \tag{7}$$

where the empirical coefficients, a and b, for both vertical and horizontal surfaces are again given in Table 1. The temperature variations experienced in buildings imply that the convection coefficients obtained from Equations (5) and (7) are unlikely to differ by more than ± 4 per cent. This is small in comparison with the experimental uncertainty in buoyancy-driven convection data which is typically ± 20 per cent.

The experimental data on buoyancy-driven convection reported in the literature were often obtained using fluids, surface lengths and temperature differences which are unlike those found in buildings. It is therefore desirable to check the empirical coefficients in correlating equations from data more appropriate to this case. Thus, the present data correlations, Equations (5) and (7), were validated by comparison with heat transfer measurements from relatively large area surfaces, mainly in air. Only when data for air were found to be too sparse, were selected measurements for water employed. Details of the comparisons for vertical and horizontal surfaces are discussed in Sections 2.1 to 2.3 below. An idealised representation of the resulting buoyancy-induced flow patterns near isolated surfaces are shown in Fig. 1. These diagrams are used below to distinguish the physical nature of each flow pattern. The effect of the room element and flow interactions present in 'real' buildings are briefly discussed in Section 3.

Table 1. Empirical coefficients in the data correlations for Nu and h_c (Equations (5) and (7)).

Flow and surface orientation	A	B	a	b	p	q	m
Buoyancy-driven convection over vertical surfaces	0.58	0.11	1.50	1.23	1/4	1/3	6
Buoyancy-driven flow on horizontal surfaces	0.54	0.14	1.40	1.63	1/4	1/3	6

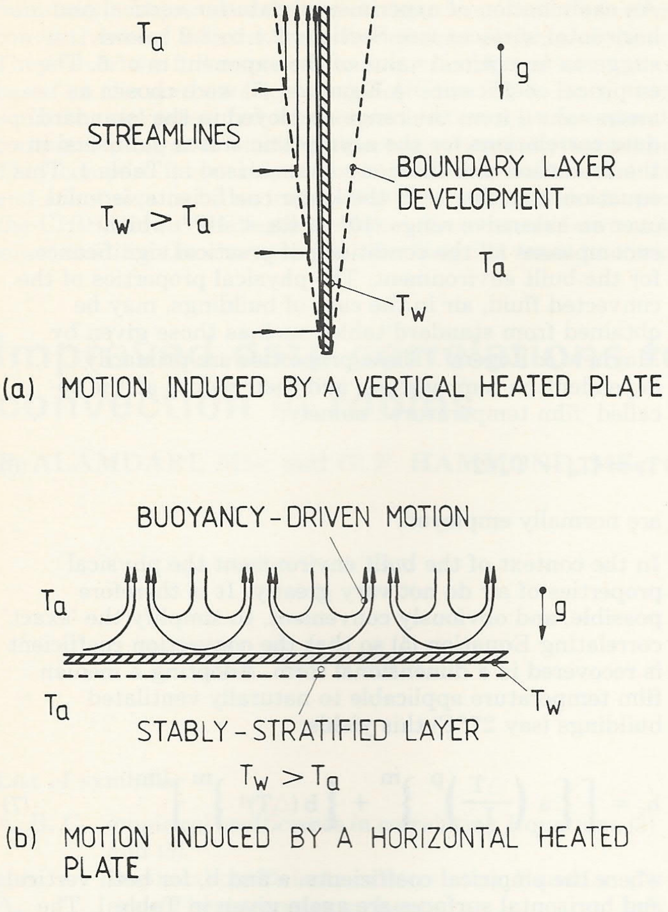

Fig. 1. Idealised buoyancy-driven convection near flat surfaces, motion induced (a) by a vertical heated plate, and (b) by a horizontal heated plate.

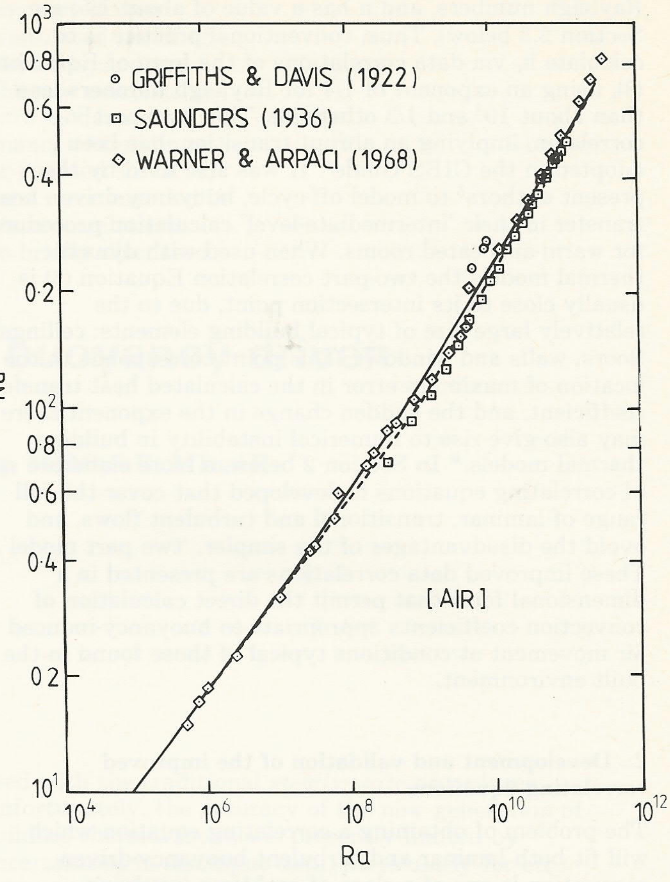

Fig. 2. Nusselt versus Rayleigh number relation for buoyancy-driven convection near vertical surfaces. (Solid line: present correlation. Broken line: asymptotic laminar and turbulent behaviour).

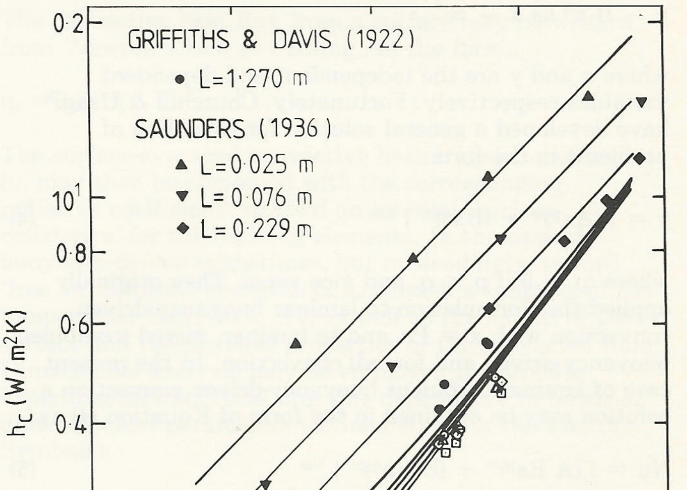

Fig. 3. Heat transfer coefficient for buoyancy-induced air flow near vertical surfaces.

2.1 Buoyancy-driven convection near vertical surfaces

The flow pattern induced by a hot, vertical plate results from the heated, lower density fluid adjacent to the surface rising upward (see Fig. 1a). Conversely, a cold, vertical plate, where $T_w < T_a$, induces a downward motion with a heat flux of similar magnitude, but opposite sign. 'Window downdraught' is an obvious example of the latter motion in the context of buildings. The characteristic length scale, L, associated with these vertical surface flows is the height of the plate. Nusselt number data for air[8-10], scaled on this basis, are compared with the present correlating Equation (5) in Fig. 2. It is evident that the scatter of this data is small, and that the improved data correlation exhibits a good fit throughout the laminar, transitional and turbulent regimes. Not surprisingly, therefore, the recovered data for h_c, shown in Fig. 3, is similarly well represented by Equation (7). (The data in this figure implies a slightly greater scatter, due to uncertainties in recovering heat transfer coefficients. Such a process inevitably involves an element of subjective interpretation.) The height of walls in domestic houses are typically about 2.5 m, which is covered by the data displayed in Fig. 3. It is to be expected that heat transfer would be influenced by three-dimensional flow or 'side-wall' effects if the height/width ratio of the walls were relatively large. However, this aspect ratio varies between about 0.5 and 1 in houses, so that the flow and heat transfer are likely to be essentially two-dimensional.

The CIBS Guide[4] does not provide data for surface-averaged, buoyancy-induced heat transfer in the practical case of transition to turbulent flow on vertical surfaces. Its laminar flow data also falls 10 per cent below the asymptotic Nusselt number given by the present correlating Equation (5). This new expression for convection on vertical surfaces, in the form of Equation (7), therefore provides building thermal modellers with a more accurate and convenient basis for their computations.

When the authors developed the present correlating Equations (5) and (7) they were unaware that later work by Churchill and his co-workers[11] had yielded analogous data correlations for buoyancy-driven motion on vertical surfaces.* Churchill & Chu[11] developed their formulation for the two asymptotic conditions $Ra \rightarrow 0$ and $Ra \rightarrow \infty$, with an additional requirement that it should reproduce the analytical results of laminar boundary layer theory in the range $10^5 < Ra < 10^8$. This led to the rather complicated expression:

$$Nu^{1/2} = 0.825 + \frac{0.387\,Ra^{1/6}}{\{1 + (0.437/Pr)^{9/16}\}^{8/27}} \tag{8}$$

Very low Rayleigh numbers ($Ra \rightarrow 0$) are only encountered in practice for rather exotic low Prandtl number fluids, such as liquid metals, which are obviously inappropriate to the built environment. Nevertheless, ESDU[12] has simplified Equation (8) for air flow and, over the limited range of temperatures applicable to naturally-ventilated buildings, it becomes:

$$h_c = 0.134\,L^{-1/2} + 1.11\,\triangle T^{1/6} \tag{9}$$

This expression, together with its counterpart, Equation (8), are found to yield heat transfer coefficients which are about 5 per cent higher than the present correlating Equations, (5) and (7), for high temperature differences and long surfaces (high Rayleigh numbers, $10^9 < Ra < 10^{12}$). Conversely, for relatively low temperature differences and short surface lengths ($10^4 < Ra < 10^7$), the Churchill & Chu/ESDU equations imply coefficients that are about 11 per cent lower than those given by the present correlations. Interestingly, Churchill & Chu[11] themselves argued that heat transfer coefficients similar to those yielded by the present correlating Equation (5) are 'more accurate', in the range $10^4 < Ra < 10^7$, than those of Equation (8). Nevertheless, in view of the inherent uncertainty in the experimental data on buoyancy-driven convection noted above, the Churchill & Chu/ESDU correlating equations may be regarded as alternatives to the present ones for vertical surface flows. The present data correlations, however, have the additional merit of being rather simpler in form, and have been extended to horizontal surfaces in the section that follows.

2.2 Buoyancy-driven convection near horizontal surfaces

The motion induced by a heated horizontal surface is illustrated in Fig. 1b. The relatively hot, lighter fluid on the upper surface has a tendency to be convected upwards in the form of 'plumes' or 'thermals', being replaced by colder, more dense fluid from above. This configuration is therefore gravitationally unstable and, for temperature differences above a minimum or 'critical' value, the buoyancy forces will drive the convective motion. The flow pattern shown for the upper surface in Fig. 1b is an idealisation based on the work of the Soviet scientist M. Mikheyev as given by Al-Arabi & El-Riedy[13]. 'Orderly', laminar flow is observed at low Rayleigh numbers ($Ra < 10^6$), but the convective motion undergoes a transition to turbulence at high values ($Ra > 10^9$). A cooled horizontal surface, where $T_w < T_a$, will give rise to a similar flow pattern, although this develops on the underside of the element. Both arrangements involve a downward heat flow. There is a considerable divergence of opinion in the literature regarding the most appropriate length scale to adopt for these horizontal surface flows. In the present study the characteristic length has been defined as:

$$L = 4A_s/P_s \tag{10}$$

where A_s is the surface area and P_s its perimeter. This is analogous to the 'hydraulic diameter' used in fluid mechanics, and for a square surface the length scale simply becomes the length of one side. Equation (10) has the conceptual merit of yielding appropriate length scales for irregular, non-rectangular surfaces, although elements of this type are not usually found in buildings.

Experimental Nusselt number data obtained in air[13-15] and corrected for the present length scale are shown in Fig. 4, where they are compared with the new correlating Equation (5). The data of Fujii & Imura[16] for water (considered by Yousef *et al*[15] to be amongst the most accurate) has also been included in Fig. 4 in order to augment the rather sparse air data at high Rayleigh numbers. Data for buoyancy-driven convection on horizontal surfaces is seen to display a greater scatter than that for vertical surfaces. This is in part due to the influence of 'edge' effects which tend to increase the heat transfer rate on relatively small surfaces[13,15]. Nevertheless, the present correlating Equation (5) is seen to be representative of the bulk of this data throughout the laminar, transitional and turbulent regimes. A similar

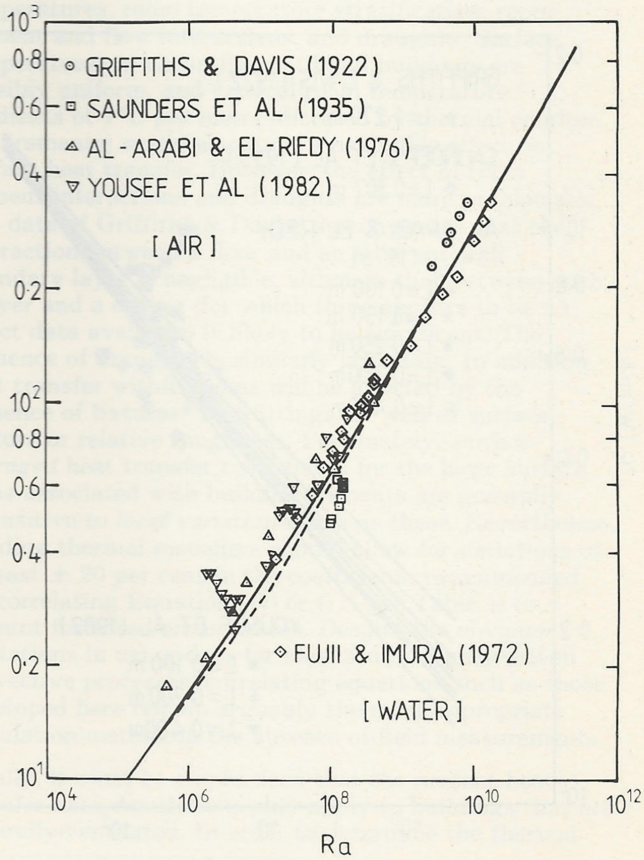

Fig. 4. Nusselt versus Rayleigh number relation for buoyancy-driven convection adjacent to horizontal surfaces. (Solid line: present correlation. Broken line: asymptotic laminar and turbulent behaviour).

*The authors only discovered a reference to the paper of Churchill & Chu[11] when they consulted the Engineering Sciences Data Unit[12] in search of additional data to validate the horizontal surface cases.

conclusion may be drawn in regard to the recovered data for h_c in air plotted on Fig. 5, where they are compared with correlating Equation (7). Yousef *et al*[15] postulated that the very high heat transfer coefficients obtained with their smallest test plate (L = 0.1 m) might have been caused by the presence of a single buoyant plume. This explanation for the anomalous data is open to doubt, but this geometry is, in any case, outside the range of interest to the built environment. The data of Saunders *et al*[14] which falls considerably below the present correlation, is generally regarded (see, for example, Yousef *et al*[15] as being among the lowest of the available data sets. It is clear from Fig. 5 that further accurate data for $L \simeq 3$ m would have been useful for validating the present correlating Equation (7) at conditions directly applicable to buildings.

Both the CIBS Guide[4] and the appropriate ESDU Data Item[12] employ two-part models to correlate data for h_c in air near horizontal surfaces. The former adopts a characteristic length scale equal to the mean value of the two sides of a rectangular surface. This choice is identical to that given by Equation (10) for square surfaces, and differs only slightly for rectangular elements. The length scale adopted by ESDU is equal to 0.25 L as used in the present work. In the laminar regime, the CIBS correlation is almost identical to that given by Equation (7), whereas the ESDU relation yields coefficients which are almost 30 per cent higher. The CIBS and ESDU correlations for the turbulent regime, in contrast, imply coefficients which are not more than 5 per cent above those given by correlating Equation (7).

2.3 Stably-stratified convection near horizontal surfaces

The fluid beneath a heated horizontal surface, such as that illustrated in Fig. 1b, must always be at a lower temperature than that of the surface. This fluid is therefore gravitationally stable, and the layer over which the vertical temperature gradient persists is stably-stratified. A cooled horizontal surface would also give rise to the formation of such a stable layer, but on top of the element. Either configuration may be characterised as having an upward heat flow. Instabilities that would otherwise lead to the onset of turbulence at high Rayleigh numbers are damped under the influence of downward, negative buoyancy forces which restore equilibrium. The analytical results of Singh *et al*[17], based on the solution of the integral boundary layer equations, indicate that the Nusselt number for such laminar stably-stratified layers varies according to a 1/5 power of the Rayleigh number. This differs from the ¼ power-law dependence of buoyancy-driven laminar flow on both vertical and horizontal surfaces. Nevertheless, recent experimental data[16,18,19] has confirmed that the 1/5 power relation holds up to at least $Ra \simeq 10^{11}$, and it is adopted in the present work. Thus, for stable-stratification alone, the standard correlation Equation (3) is used, together with the 1/5 exponent and the empirical coefficient, C, having a value of 0.58. Simplifying this expression for the thermal conditions applicable to naturally-ventilated buildings yields a dimensional formula for the convective heat transfer coefficient:

$$h_c = 0.60 \left(\frac{\Delta T}{L^2}\right)^{1/5} \tag{11}$$

The choice of the characteristic length scale governing heat transfer across stably-stratified layers has again been the subject of divergent opinions in the literature. Equation (10) has been employed in the present work for the reasons outlined in Section 2.2 above. The experimental Nusselt number data for air[8,14,18,19] which is displayed in Fig. 6 has been scaled using this characteristic length. The early measurements by

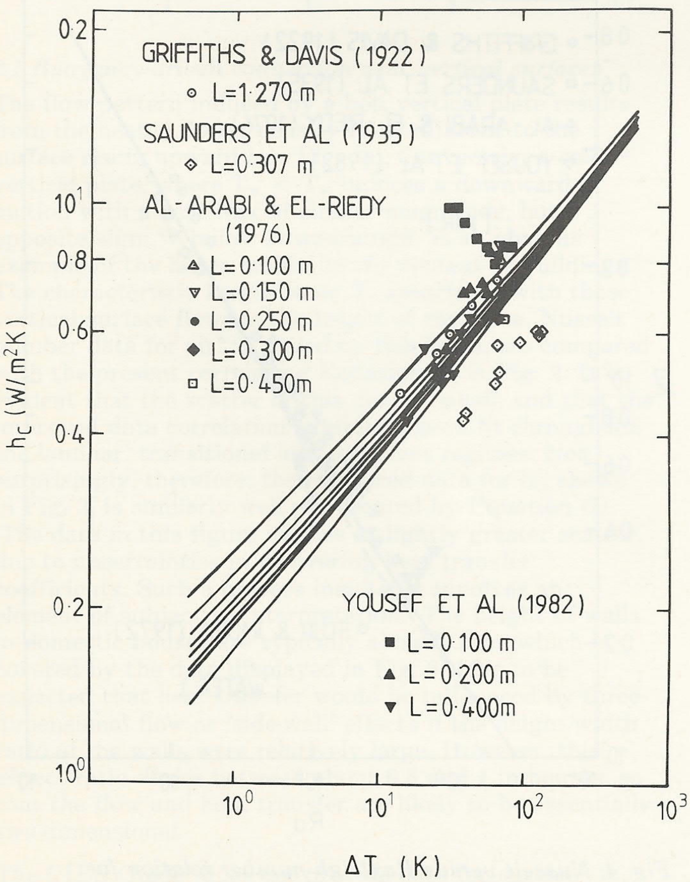

Fig. 5. Heat transfer coefficient for buoyancy-induced air movement adjacent to horizontal surfaces.

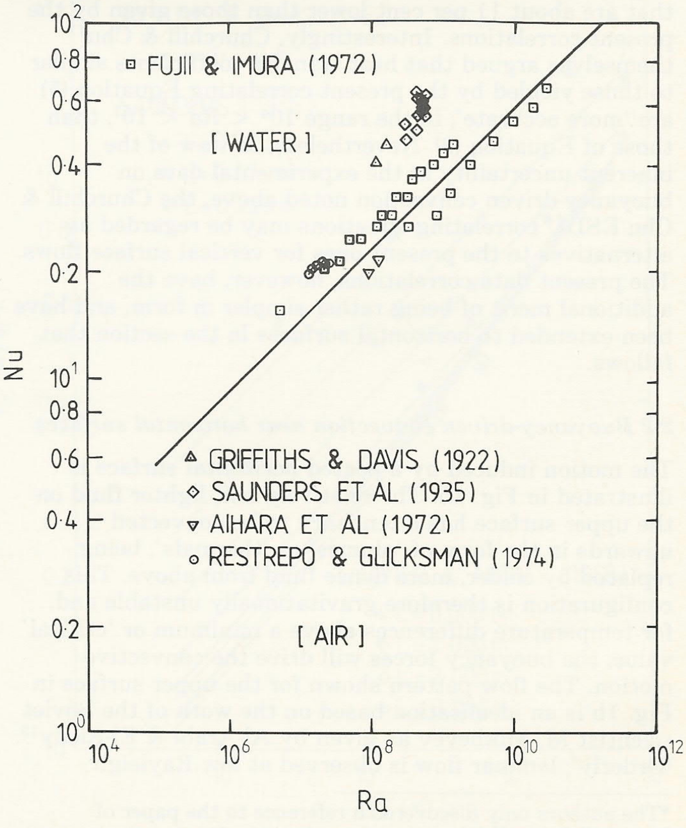

Fig. 6. Nusselt versus Rayleigh number relation for stably-stratified layers near horizontal surfaces.

Saunders *et al*[14], on which the 'standard' data correlations are based[4,12], have been criticised by subsequent investigators[18,19] and are regarded as too high. They used a test plate which simultaneously heated the fluid above and below the surface, so that the flow over the upper surface influenced that near the edges of the lower one[19]. Restrepo & Glicksman[19] have also drawn attention to the influence of the edge thermal boundary conditions (whether they are heated, cooled or insulated) which can alter the heat transfer coefficient by more than ± 20 per cent. In order to give fuller coverage of high Rayleigh number data, the measurements of Fujii & Imura[16] for water have been included in Fig. 6. It can be seen that the present correlating Equation (3) is representative of the more recent measurements, and that the early data of Griffiths & Davis[8] and Saunders[14] are appreciably higher. This is the reason that correlating Equation (11) does not appear to fit the recovered data for h_c in Fig. 7 very closely. In fact the more recent, but rather sparse, data is fairly evenly scattered about the present data correlation.

A data correlation based on the measurements of Saunders *et al*[14] and employing a ¼ power-law dependence has been adopted for both the CIBS Guide[4] and the corresponding ESDU Data Item[12]. These 'standard' correlations, when corrected for the length scale given by Equation (10), are almost identical, but yield heat transfer coefficients which are up to 40 per cent higher than those given by correlating Equations (3) and (11) under conditions typical of the built environment. Indeed the older correlations are over 120 per cent in error at the highest Rayleigh number of their declared range of application. The use of the ¼ power-law is at variance with theoretical results[17] and with recent experimental data[16,18,19], which all suggest a

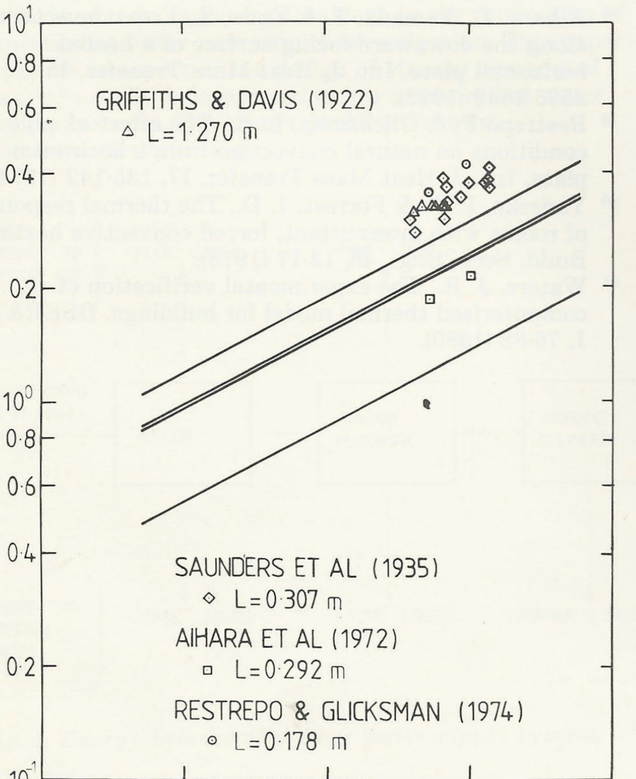

Fig. 7. Heat transfer coefficient for stably-stratified air layers near horizontal surfaces.

1/5 power-law dependence for heat transfer in stably-stratified layers. The new correlating Equations (3) and (11) therefore constitute a significant improvement on these older data correlations.

3 Concluding remarks

Improved data correlations for buoyancy-driven convection from the internal surfaces of naturally-ventilated buildings have been derived on the basis of the model of Churchill & Usagi[6]. These correlating equations provide a smooth fit to data across the full range of laminar, transitional and turbulent airflows, in contrast to the 'two-part' model employed in 'standard' correlations. The new correlating equations thereby avoid the possibility of inducing numerical instability when used in conjunction with the new generation of dynamic building thermal models. The new data correlations are presented in a simplified and more convenient form, Equation (7), from which the convective heat transfer coefficient may be obtained directly in dimensional terms, rather than via the 'exact' Nusselt number relation. They display generally good agreement with the rather limited experimental data available for conditions typical of the built environment. However, it has been shown that the standard, two-part correlation equations used by the CIBS and the ESDU are, in some cases, significantly in error when compared with recent measurements.

The data on which the improved correlating equations are based was obtained from experiments using isolated surfaces. It is therefore appropriate to consider the extent to which this idealisation is valid in the conditions prevailing in 'real' buildings. The ESDU[12] has identified a number of factors which alter the apparent heat transfer rates implied by correlating equations, and those relevant to buildings include non-uniform surface temperatures, room temperature stratification, room element and flow interactions, and draughts. Surface temperatures in naturally-ventilated buildings are sensibly uniform, and vertical room temperature gradients of 1 °C per metre (dictated by thermal comfort requirements) would also have a negligible effect on surface heat transfer. However, the effect of room element interactions and draughts are more problematic. The data of Griffiths & Davis[8] demonstrates that the interaction between a floor and an adjacent wall boundary layer is negligible, although that between such a layer and a ceiling (for which there appears to be no direct data available) is likely to be significant. The influence of draughts is similarly uncertain. In addition, heat transfer within rooms will be affected by the presence of fixtures* and fittings, as well as surface texture or relative roughness. Fortunately, *surface-averaged* heat transfer coefficients for the large surface areas associated with building elements are generally insensitive to *local* variations such as these. Nevertheless, building thermal modellers should allow for variations of at least ± 20 per cent in the coefficients recommended for correlating Equations (7) or (11) (see Table 1) to account for these disturbances. Despite the obvious limitations in using data for idealised buoyancy-driven convective processes, correlating equations such as those developed here remain arguably the most appropriate calculation method in the absence of field measurements.

Finally, it must be emphasised that the present heat transfer data correlations only apply to buildings that are naturally-ventilated. In order to determine the thermal

*It should be noted, incidentally, that the thermal boundary conditions for the plume above a 'radiator' are different from those associated with the present or standard data correlations. Thus, such correlating equations do not apply in this case, and the error involved in their use cannot be readily evaluated.

performance of buildings using forced convective heating and/or mechanical ventilation, the convection coefficient must be calculated via a more elaborate procedure, such as that recently developed by the present authors[5]. Yaneske & Forrest[20] concluded for example, from a field study of over thirty rooms of various shapes and sizes heated by a fan 'convector', that surface coefficients were much higher than those for buoyancy-driven convection. They demonstrated that the use of the latter when selecting heater capacity for a mechanically-ventilated dwelling could result in substantial increases in preheat times. Waters[21] has likewise shown that the accuracy of his implicit finite-difference building thermal model was strongly dependent on the correct choice of internal heat transfer coefficient when simulating mechanically-ventilated structures.

Acknowledgements

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Summary Domestic hot water is commonly supplied from a storage cylinder heated by water circulated from the central heating boiler. The efficiency of hot water production in such systems is dependent on boiler losses, circulation losses, storage losses and losses from the secondary system of distribution pipework to the draw off points. The present work examines the effect of the size of the cylinder heat exchanger on system performance and considers other aspects of system design which affect overall efficiency. A range of cylinder heat exchangers is tested using three boiler designs with 'compressed' draw-off schedules. An optimum heat exchanger surface area has been found for the sizes of boiler employed. An energy balance shows the distribution of energy losses from the system.

Comparative performance tests of a domestic storage hot water system with a range of cylinder heat exchangers

A. T. HOWARTH, BSc, MPhil, CEng, MIMechE, MCIBS, and M. M. J. AL-AZAWI, BSc, MSc

1 Introduction

As increasingly high standards of insulation are incorporated in the fabric of buildings and as draughts are progressively reduced, so the proportion of energy used for the provision of domestic hot water increases. At the same time space heating systems are becoming more efficient as the heat distribution efficiency is improved by the increasing sophistication of controls.

Methods of reducing energy wastage in the production of domestic hot water now require investigation. This paper examines an aspect of the design of storage systems of the type usually incorporated in low pressure hot water heating systems. The energy paths are shown diagrammatically in Fig. 1.

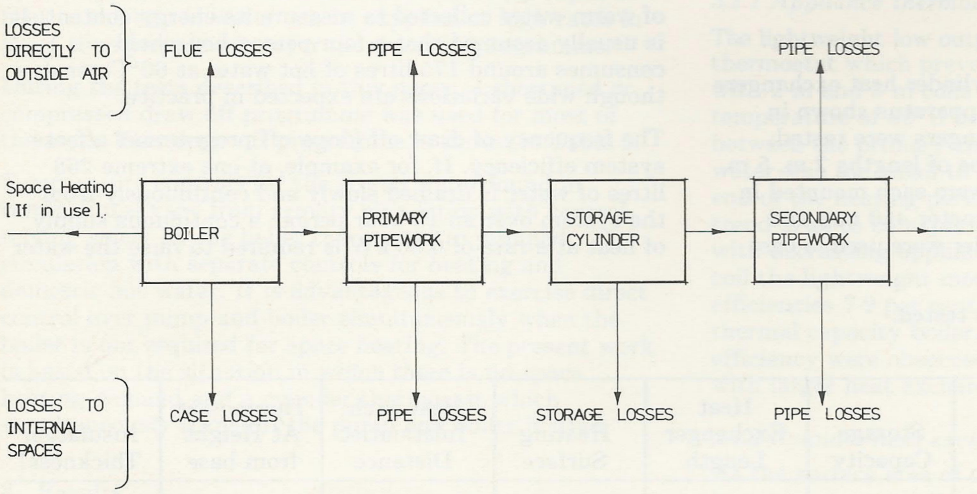

Fig. 1. Energy balance of the hot water supply system.

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The losses from the boiler and primary distribution pipework depend partly on the running times of the boiler and of the circulation system and it is important therefore that heat is transferred as rapidly as possible into the cylinder. This suggests that the heat exchanger within the cylinder should be able to transfer a high proportion of the output of the boiler, so that in summer, and in the heating season when space heating controls are satisfied, the boiler is not idling for long periods. This paper reports a series of tests in which the major variable was the heat transfer rate of the cylinder heat exchangers; varied by employing a range of heat exchanger surface areas.

Previous tests by Whittle and Warren[1] have revealed overall system efficiencies* in the range 18 to 40 per cent

*See Section 3.1 for definition.

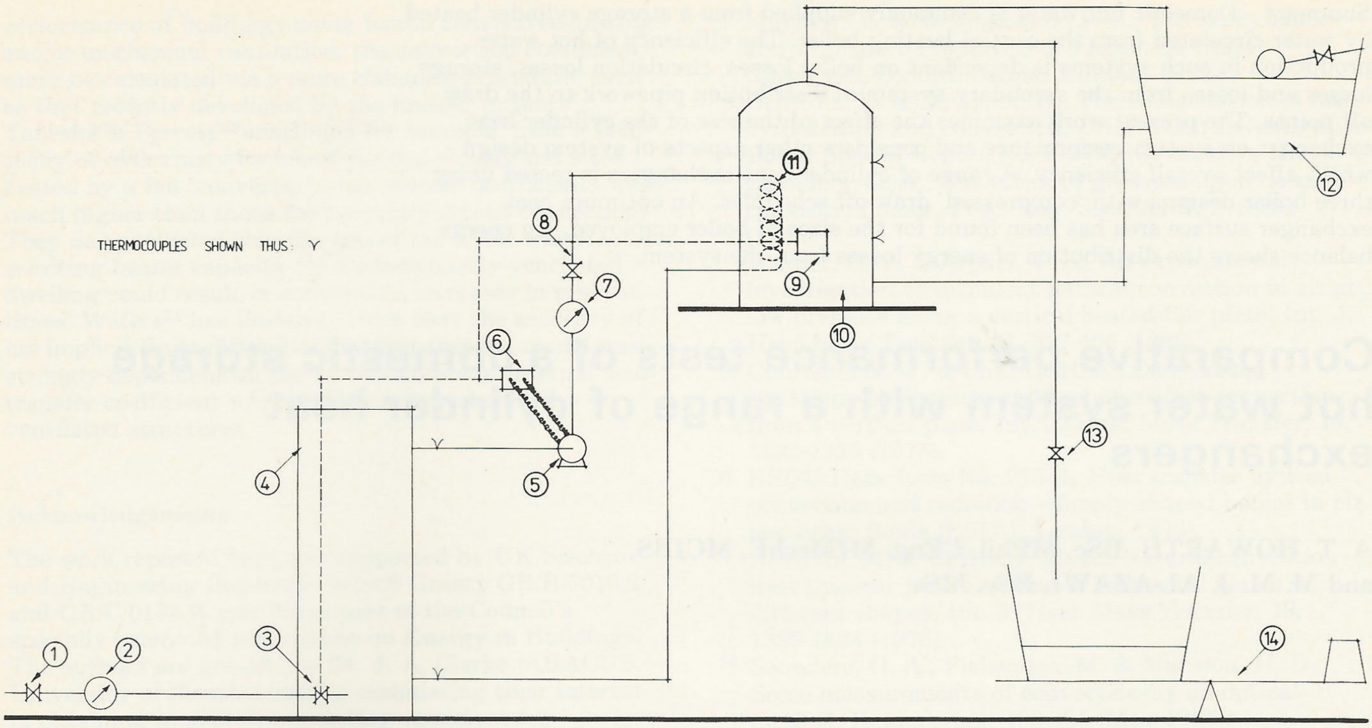

Fig. 2. Test rig layout: (1) Main gas valve, (2) Gas meter, (3) Boiler switch, (4) Boilers (see Table 2), (5) Pump, (6) Pump switch, (7) Flow meter, (8) Flow control valve, (9) Cylinder thermostat, (10) Cylinders (see Table 1), (11) Coil heat exchangers, (12) Feed and expansion tank, (13) Draw off valve, (14) Weigh scales.

with a range of gas fired boilers supplying a BS1566[5] 5 m coil. They employed a sequence of draw offs over 16 hour periods as recommended by BS1250 part 3[4]. Howarth[2] has previously conducted a short series of tests using a compressed draw off schedule lasting approximately three hours with a range of heat exchanger designs. Efficiencies in the range of 36 to 57 per cent were observed, the higher efficiencies applying to the larger heat exchanger surface area. Al Azawi[3] conducted more rigorous tests of different sized heat exchangers under a wider range of conditions; some results of these tests form the subject of this paper.

2 Test procedure

The comparative performance of cylinder heat exchangers was evaluated by tests using the apparatus shown in Fig. 2. Four coiled tube heat exchangers were tested; each of 28 mm diameter copper tube of lengths 2 m, 5 m, 7 m and 10 m (Table 1). The coils were each mounted in BS1566[5] insulated cylinders of diameter 450 mm and height 900 mm. Three types of boiler were used; a cast iron high water content boiler and two low thermal capacity boilers as set out in Table 2.

The performance of hot water systems is affected by its utilisation. Whittle and Warren[1] employed total daily draw off volumes of both 90 litres and 180 litres, finding greater system efficiencies associated with the larger draw off volumes. They employed a BS1250 Part 3[4] draw off profile extended to a realistic sixteen hour working day. In the present tests the draw off volume is a total of 268 litres, the tests being run so that the contents of the cylinder are cold at both the beginning and the end of the tests. Thus part of the volume is, in effect, a drain-down of warm water collected to measure its energy content. It is usually assumed that a four person household consumes around 175 litres of hot water at 60°C per day though wide variations are expected in practice.

The frequency of draw off (draw off programme) affects system efficiency. If, for example, at one extreme 268 litres of water is drained slowly and continuously from the system over an 18 hour period, a continuous supply of heat at a rate of 0.86 kW is required to raise the water

Table 1. Specifications of cylinders tested.

Cylinder no.	Cylinder Diameter	Cylinder Height	Storage Capacity	Heat Exchanger Length	Heating Surface	Heat exch. Inlet/outlet Distance	Thermostat At Height from base	Insulation Thickness†
	mm	mm	l	m	m²	mm	mm	mm
(1)	450	900	120	2	0.176	300	300	20
(2)	450	900	120	5*	0.44	300	300	20
(3)	450	900	120	7	0.615	300	300	20
(4)	450	900	120	10	0.88	450	300	20

* BS Heat Exchanger
† Urethane Foam Insulation

Table 2. Specifications of boilers.

Boiler Type	Boiler Designation	Rated Input	Rated Output	Bench Efficiency	Water Content	Gas Consumption at full load	Pilot Gas Consumption
		kW	kW	per cent	kg	m^3/min.	m^3/min.
HTC ON-OFF	A	20.5	14.8	72	7	0.0337	0.45×10^{-3}
LTC Modulating	B	19.0	14.7	77	2	0.0301	0.26×10^{-3}
LTC ON-OFF	C	10.88	8.2	75	1	0.0200	0.19×10^{-3}

HTC—high thermal capacity
LTC—low thermal capacity

Table 3. Procedure for tests using compressed laboratory schedule.

Sequence	Boiler/Pumps	Draw off volume (Litres)
1	On	—
2	Off(T)	43
3	On	—
4	Off (T)	61
5	On	—
6	Off (T)	34
7	On	—
8	Off (M)	130

T —thermostatically switched off
M—manually switched off

temperature by 50°C. If, at the other extreme, 120 litres is drawn off instantaneously, a much larger rate of input of heat can be absorbed by the stored water in a much shorter time, e.g. 13.93 kW for reheat in ½ hour. In the second case the shorter period of operation considerably reduces the circulation losses and the boiler idling losses, but is only possible when the heat exchanger is capable of transferring the heat. In practice, by the application of time controls, most of the input of heat to the stored water can be supplied at high rates in short time periods whatever the draw off programme.

During the tests described in this paper, a shortened or compressed draw off programme was used for most of the tests. This draw off programme is shown in Table 3.

Temperature controls for the stored water found in common use vary from gravity circulation with control relying solely on the boiler thermostat to pumped circulation with separate controls for heating and domestic hot water. It is advantageous to exercise direct control over pump and boiler simultaneously when the boiler is not required for space heating. The present work is based on the situation in which there is no space heating demand and a cylinder thermostat which simultaneously turns off the pump and boiler is used.

3 Results

3.1 System Efficiency

The overall efficiency of the system is defined as:

$$\frac{\text{the energy of drawn off water above mains temperature}}{\text{energy in gas consumed}}$$

throughout the test. There is no allowance for losses in the dead legs between the cylinder and the draw off point. The range of measured system efficiency is 32 to 65 per cent (Fig. 3), the following factors affecting the performance:

(1) appliance thermal capacity,
(2) cylinder heat exchanger surface area,
(3) primary water flow rate, and
(4) draw off water schedule and quality.

3.1.1 Appliance thermal capacity

The lightweight low output boiler (C in Fig. 3) has a pre-set thermostat which prevented the contents of the cylinders with 2 m and 5 m coils from being heated to a thermostat temperature of 60°C because the temperature difference between the primary circulating water and the stored water was too small to allow heat transfer towards the end of the heating period. With this exception the general trend is as to be expected—increasing system efficiency with decreasing appliance thermal capacity. With the 2 m coil the lightweight modulating boiler (B) gave system efficiencies 7-9 per cent points higher than the high thermal capacity boiler. Slightly smaller improvements of efficiency were observed when the same boilers were used with larger heat exchangers.

3.1.2 Cylinder heat exchanger surface area

As the surface area of the heat exchanger is increased its heat transfer capacity is increased and the rate of heat transfer in the cylinder is increased. Thus boiler cycling is decreased and heat-up times are reduced. The effect on system performance is illustrated by Fig. 4 which confirms the validity of the simplified tests and estimations reported previously[2]. Here, the overall system efficiency is plotted against the heat transfer rates of the range of heat exchangers tested. There is clear indication that the advantages of increasing heat exchanger surface area beyond 10 m are diminishing.

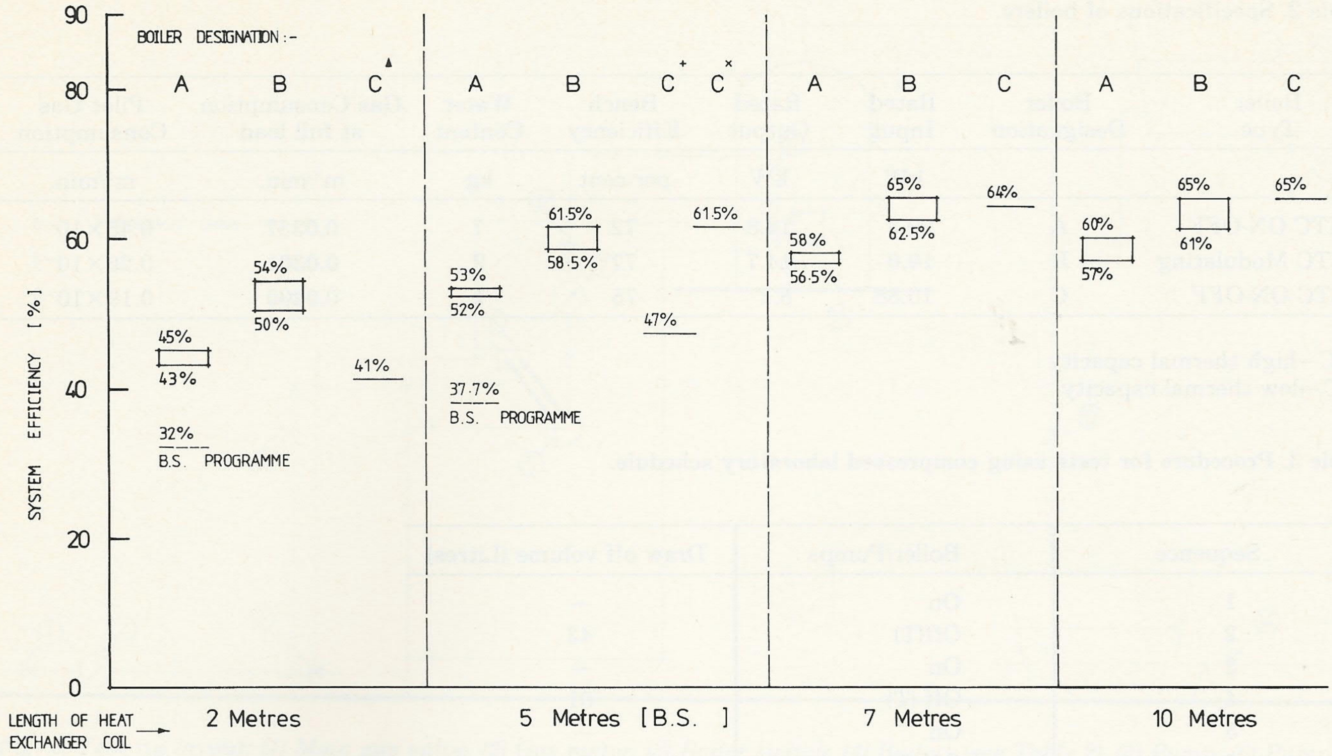

Fig. 3. Graphical presentation of the system efficiency results. (Cylinder thermostat set at 60°C except where shown.)

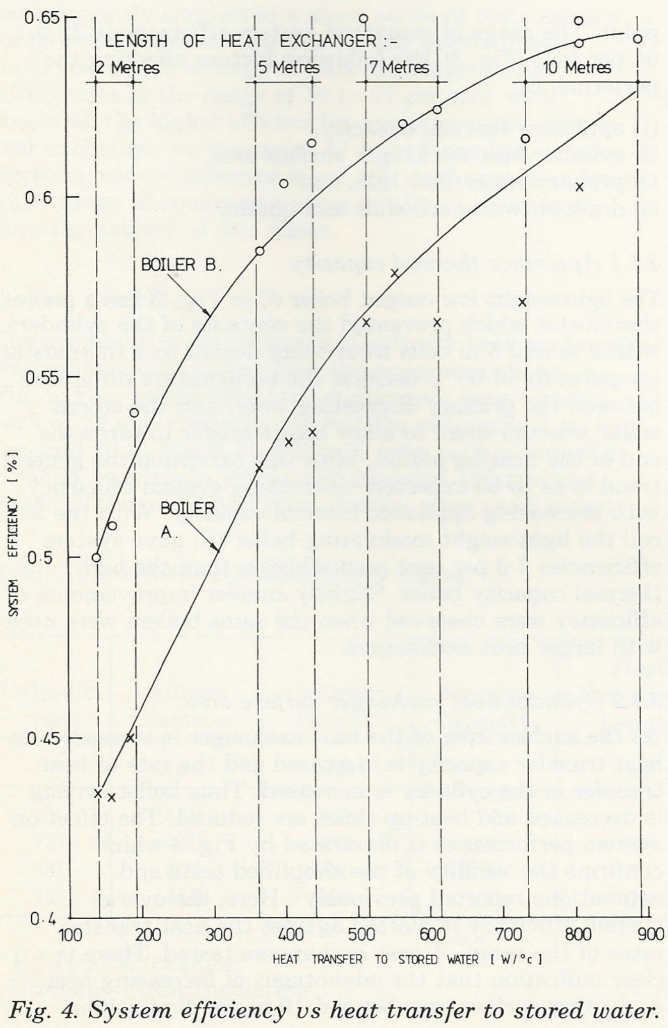

Fig. 4. System efficiency vs heat transfer to stored water.

An alternative method of presenting the results is shown in Fig. 5 where the losses are shown in more detail. It can be seen that the distribution losses from the primary pipework play a major role in the reduction of losses achieved by shortening the heat up times of the cylinders.

3.1.3 Primary water flow rate

As the pitch of the coils decreases, so the overall heat transfer coefficient of the heat exchanger varies. The overall heat transfer coefficients have been estimated (3) at between 910 and 1000 W/m^2 °C for the 2m coil and between 840 and 880 W/m^2 °C for the 10m coil at a primary flow rate of 15 litres/minute. Improvements of up to 20 per cent in heat transfer coefficient occurred when the flow rate was increased to 20 litres/minute, the improvements being most noticeable for the shorter coils. The effect of these variations is illustrated in Fig. 6 which shows the increasing efficiencies observed along with the distribution of losses in six tests with the 5 m coil. It should be noted that a tendency for pipe losses to increase with flow rate is largely offset by the shorter warm up times and hence early pump switch off arising from the improved coil heat transfer.

3.1.4 Draw off water schedule

This variable produces the greatest variations as shown by Fig. 7. The reduction of system efficiency is clearly seen at lower draw-off volumes both when a 2 m coil and a 5 m coil were used. The improvement achieved by the use of the longer coils is still evident. These results bear comparison with the previous results of Whittle and Warren[1], whose even lower efficiencies occurred because of (a) losses in their secondary pipework and (b) longer operating times.

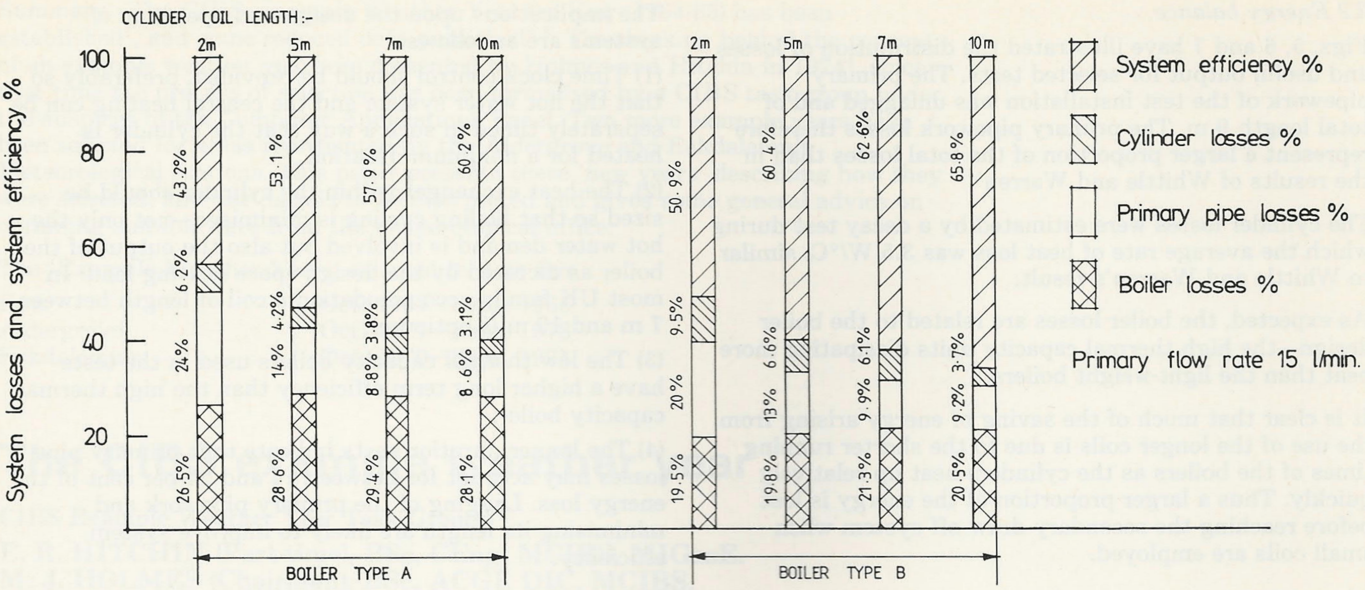

Fig. 5. Effect of cylinder heat exchanger length on system losses.

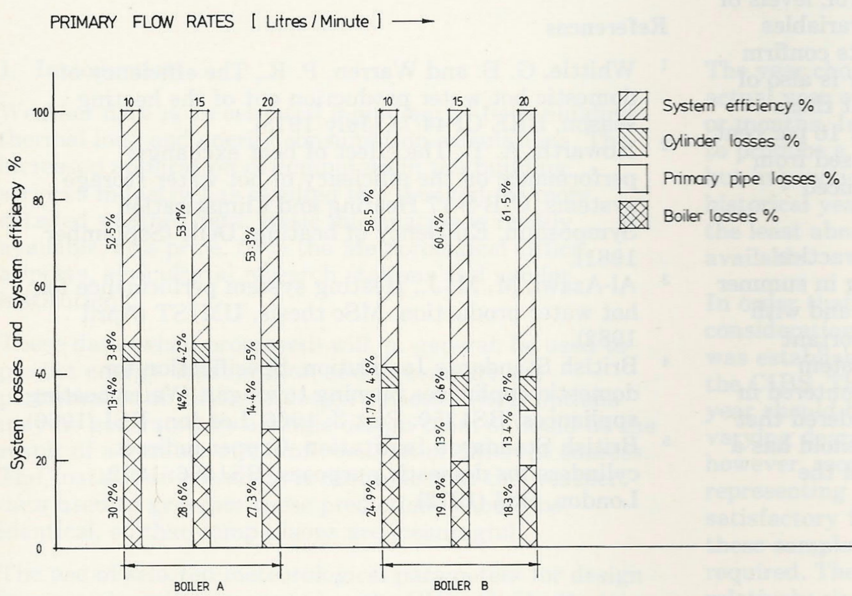

Fig. 6. Effect of primary flow rate on system losses (5 m heat exchanger).

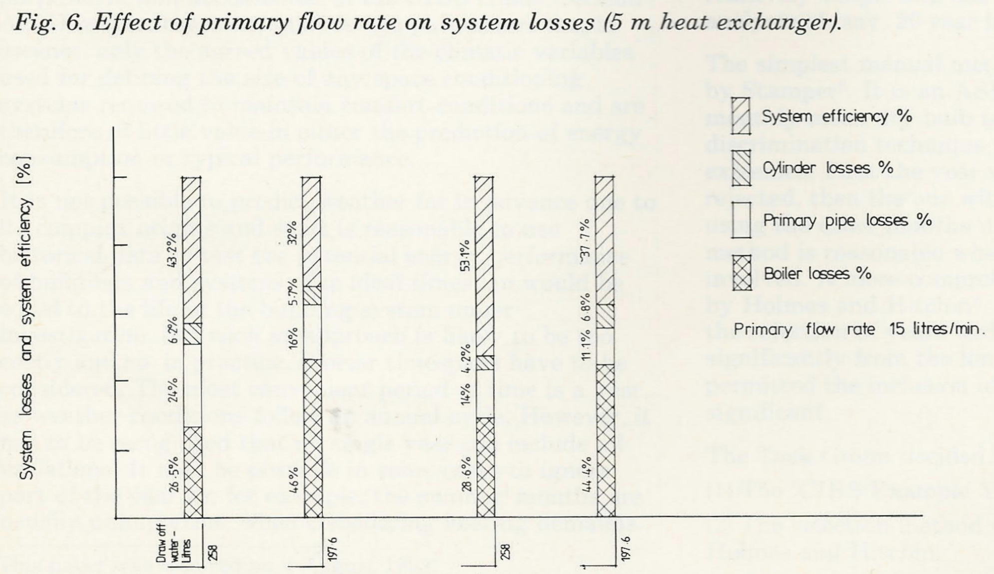

Fig. 7. Effect of draw off water programme on system losses.

3.2 Energy balance

Figs. 5, 6 and 7 have illustrated the distribution of losses and useful output for selected tests. The primary pipework of the test installation was unlagged and of total length 9 m. The primary pipework losses therefore represent a larger proportion of the total losses than in the results of Whittle and Warren[1].

The cylinder losses were estimated by a decay test during which the average rate of heat loss was 3.5 W/°C, similar to Whittle and Warren's result.

As expected, the boiler losses are related to the boiler design—the high thermal capacity units dissipating more heat than the light-weight boilers.

It is clear that much of the saving of energy arising from the use of the longer coils is due to the shorter running times of the boilers as the cylinders heat up relatively quickly. Thus a larger proportion of the energy is lost before reaching the secondary draw off system when small coils are employed.

4 Conclusions

The results of the tests with the BS draw off programme show similarities with the results of Whittle and Warren[1] who suggest that changes in methods of control, levels of insulation and draw off quantities are major variables affecting system efficiency. The present results confirm this and indicate that heat exchanger capacity is also of importance. There is clear evidence throughout these results that energy savings of between 13 and 16 per cent were achieved when the coil length was increased from 2 m to 5 m and a further increase to 7 m produced further savings of between 6 and 8 per cent.

Such results are most closely represented in practice when the pumped primary system is operating in summer with simultaneous control of pump and boiler and with time clock control, the latter being a very important feature. In winter when the central heating system operates, savings may be less than those encountered in the present tests. It should, however, be considered that a modern newly constructed four person household has a *design* heat loss of around 7 kW, less than half the instantaneous heat transfer to a cold cylinder.

The implications upon the design and operation of systems are as follows:

(1) Time clock control should be provided, preferably so that the hot water system and the central heating can be separately timed in such a way that the cylinder is heated for a minimum duration.

(2) The heat exchanger within the cylinder should be sized so that boiling cycling is minimised—not only the hot water demand is involved but also the output of the boiler as dictated by the design space heating load. In most UK family accommodation a coil of length between 7 m and 10 m is optimum.

(3) The low thermal capacity boilers used in the tests have a higher long term efficiency than the high thermal capacity boiler.

(4) The longer duration tests indicate that primary pipe losses may account for between 14 and 24 per cent of the energy loss. Lagging of the primary pipework and minimising its length are likely to improve system efficiency.

Acknowledgement

The authors acknowledge the Department of Building, UMIST where the experimental work was carried out.

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Summary The CIBS Example Weather Year for Kew (1964/65) has been established[1], and some reduced data published[2,3]. The thinking behind the concept of an example weather year was presented by Holmes and Hitchin in 1974[4], since that time the process of selection has been formalised by a CIBS task group under the auspices of the Computer Applications Panel. Two more example years have been selected for areas represented by the Aldergrove and Eskdalemuir Meteorological stations. This paper presents these 'new years' describing how they were selected, how the data is to be interpreted and gives some general advice on obtaining suitable data from the meteorological office.

The 'Example Weather Years' selected to date are:

Kew	Oct. 1964–Sept. 1965
Aldergrove	Oct. 1977–Sept. 1978
Eskdalemuir	Oct. 1970–Sept. 1971

The CIBS example weather year

CIBS Example Weather Year Task Group:

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1 Introduction

Weather data is an essential ingredient of both building thermal load and energy consumption calculations. The increased application of computers in the building services industry has resulted in a demand for more detailed weather data. Basic information is readily available, at a price, from the Meteorological Office, airports, agricultural research stations and similar institutions.

These data (when processed) will, in general, be used to predict energy consumption and to compare the performance of different design options. These options may be generated by a single design team or could be the result of a number of submissions from different sources. The Institution considers it desirable that the weather data used to generate these predictions should be identical, so that comparisons are meaningful.

The use of selected meteorological parameters for design purposes is well documented in the CIBS Guide, Section A2. However, these external design parameters are, in essence, only the agreed values of the climatic variables used for defining the size of any space conditioning systems required to maintain comfort conditions and are therefore of little value in either the prediction of energy consumption or typical performance.

It is not possible to predict weather far in advance due to its complex origins and so, it is reasonable to use historical data to test the potential energy performance of buildings and systems. The ideal timespan would be equal to the life of the building/system under investigation, but such an approach is likely to be too costly and so, in practice, shorter timespans have to be considered. The most convenient period of time is a year, as weather conditions follow an annual cycle. However, it has to be recognised that no single year can include all variations. It may be possible in some cases to ignore part of the year as, for example, the summer months are usually unimportant when considering heating demands.

This paper was received on 4 August 1983.

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The year chosen to represent the weather could be an actual year or one synthesised from selected days, weeks or months. In theory, a synthesised year should be able to produce a better statistical representation of the past, but in practice, this is very difficult to realise[5]. For an historical year, selection becomes a matter of choosing the least abnormal year from those for which data are available.

In order that this subject might be given serious consideration, the Example Weather Year Task Group was established by the Computer Applications Panel of the CIBS. The Task Group was unanimous that a real year should be used and many methods of selection with varying degrees of complexity can be envisaged. It was, however, recognised that a year chosen as suitable for representing weather in S.E. England may not be satisfactory for other sites in the UK. Thus, a number of these samples or 'Examples' of the UK climate would be required. The selection method should therefore be relatively simple and not require the purchase and analysis of, say, 20-year hourly weather tapes.

The simplest manual method is probably that described by Stamper[6]. It is an ASHRAE method based on monthly mean dry bulb temperature and the discrimination technique used is one of rejecting extremes. First the year with the warmest January is rejected, then the one with the coldest August and so on, using the other months until only one year remains. This method is reasonable when only a single parameter is involved. A more comprehensive method is that described by Holmes and Hitchin[4]. This method basically involved the rejection of years with months that differed significantly from the long term mean. The method permitted the inclusion of any parameters thought to be significant.

The Task Group decided that:—

(1) The 'CIBS Example Year' should be a real year.

(2) The selection method should be that proposed by Holmes and Hitchin.

(3) The year should start on 1 October as this was considered to be the beginning of the heating season. Thus:

(a) Continuous data are available for both air-conditioning and heating during periods of high load.
(b) Simultations can commence in a period of low load, thus minimising 'start-up' errors.

These proposals were fully discussed at a CIBS discussion meeting held in February 1979.

Full details of the selection method are given in Section 3 and it is the intention of the Task Group to select an Example Year for each of the Degree Day Regions in the UK for which suitable data are available. These years will be given the title 'CIBS Example Weather Year'.

The CIBS Example Weather Year should be used unchanged throughout each degree day region. This is to ensure that comparison between different system and building configurations can be made from a common basis for weather. It is intended that the use of real weather data should ensure that no system will be 'favoured' due to unusual combinations of weather.

2 Selection procedure

The object of the procedure is to select a year for which the monthly mean values of a number of pertinent parameters do not differ by more than a specified number of standard deviations from their long term mean. No weighting is given to any single parameter because this would involve some form of constraint on the generality of application of the selected year (and could bias design to a particular building type). One combination is used; that between wind speed and temperature which takes the form of Jackmans Infiltration Number[7] and, whilst directly related to ventilation, it covers all cases where wind speed and temperature appear as multiples. For example, convective heat loss from the outside of the building or solar collectors.

The parameters covered are:

Global radiation on a horizontal surface,
Diffuse radiation on a horizontal surface,
Daily mean wind speed,
Mean maximum dry bulb,
Mean minimum dry bulb,
Mean dry bulb, and
Infiltration number (windspeed × (18.0 — dry bulb)).

Hitchin and Holmes[4] proposed that the number of standard deviations used as the filter should be two. That is, if the weather parameters were normally distributed about the mean, 95 per cent of the values would pass through the filter. Experience has shown that a small increase in this band width (about 0.2) allows a very large number of years to pass the test whilst a similar decrease means that all years are likely to fail. There is always the possibility that more than one year will pass the test, so the final selection, from the years that pass, will be the one with the lowest total deviation of all parameters from the long term mean values.

The original Kew year was selected using a hand-held calculator. However, the amount of data to be handled really necessitates the use of a computer and a suitable program has been prepared, using an improved final rejection technique. This program was used by the Task Group to select both the Aldergrove and Eskdalemuir years. The selection for Kew was not repeated using the program, as the Kew Year will be reconsidered as part of the rotational re-examination of chosen years every ten years. Further data will be included in the reconsideration.

The method used by the Task Group is as follows:—

(1) Obtain monthly mean values for the past 25 years (if available) using the 'Monthly Weather Summary' published by the Meteorological Office. Some of the data may be missing. These can be ignored if the year is rejected on another parameter, but can be obtained at a later date on request from the Meteorological Office if not.

The following algorithm is now followed:

(2) Read in data—set any missing values to 999.

(3) Using data for each January, calculate the following for each weather parameter (i.e. direct solar, dry bulb, etc.), (where the value is 999, move to next January).

Infiltration number—this then is treated as a weather parameter

TVAL : parameter
TOT : sum of values (ΣTVAL)
TOT2 : sum of squares of TVAL
SUMT: total number of values of the parameter for January

At the end of all data for January:

Mean
TBAR = TOT/SUMT

Standard deviation

$$\mathrm{SDEV} = \sqrt{\left(\frac{\mathrm{TOT2} - \mathrm{TBAR}^* \,\mathrm{TOT}}{\mathrm{SUMT} - 1}\right)}$$

(4) Repeat (3) for all months.

(5) Elimination Process: starting at the 1 October. Calculate for this and each of the next 11 months, for each parameter.

(a) Difference from long term mean. (Note that any missing data should assume the long term mean value.)
DELTA = (TBAR — TVAL)/SDEV.
If the absolute value of DELTA is no greater than 2, carry on, if not move to next October and start again as the year can be rejected.

(b) Calculate: SUMDEV : Sum of deviation of all parameters for the year.

(c) If all months of the year pass this test, the year is a 'potential example weather year.'

(6) If more than a single potential example year is selected, the year with the lowest total deviation (minimum SUMDEV) becomes the CIBS Example Weather Year. Should this year contain missing data, these items must be obtained and the procedure repeated before selection can be confirmed.

The above procedure is the minimum necessary to select an Example Weather Year, simple modifications can be introduced to give some form of data vetting and to show why a particular month was rejected. In addition, long term means, average deviation and other statistical information can be obtained. An example of the analysis of rejections is given in Fig. 1 for the Aldergrove and Eskdalemuir data, where the frequency of rejection of a particular parameter is given.

3 Weather data

This can be obtained on an hourly basis from the Meteorological Office at Bracknell. Department Met 01C supplies solar data and Department Met 03 supplies temperature data.

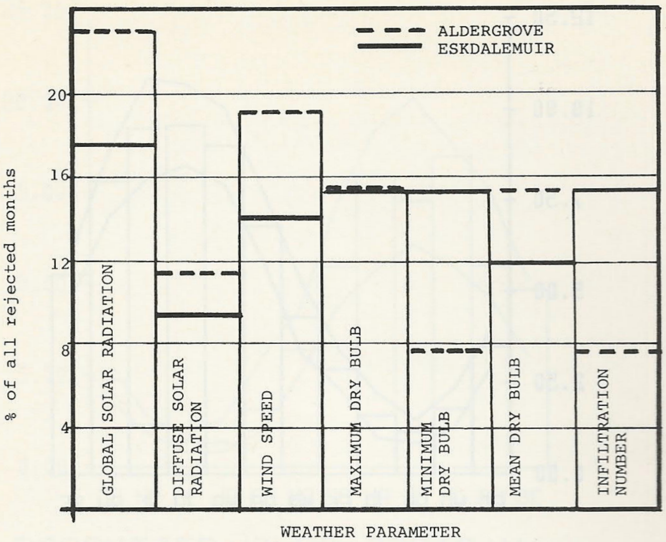

Fig. 1. Distribution of rejections.

The address is: Meteorological Office, Met 01C/Met 03, Eastern Road, Bracknell, Berkshire, RG12 2UR. Telephone: 0344-20242 extn. 2587 (1C), extn. 2278 (3).

The information requested can be supplied as computer print-out or as a magnetic tape formatted to the user's requirements. It is charged at a price per record (single reading) and its use is controlled by a very restrictive copyright agreement, which precludes CIBS (or a nominated organisation) from holding a ready formatted CIBS Example Weather Year tape.

Care must be taken to ensure that the data is read at the hour or averaged over the hour centred on the hour.

4 Concluding comments

In response to the need for agreed weather data for energy consumption calculations and evaluations of the relative performance of various building and system design strategies, the Example Weather Year Task Group of the Computer Application Panel have proposed a method of selection of the 'CIBS Example Weather Year'. The Example Weather Year consists of 24 × 365 hourly values of meteorological variables, starting on the 1 October.

So far, CIBS Example Weather Years have been selected for three Degree Day Regions. Work on the selection of Example Years for further regions is in progress.

In the meantime, for relative performance studies in regions so far not covered, the CIBS Example Weather Year of the nearest available region should be used.

The selection will be reviewed every ten years, wherever possible, based on the previous 25 years' weather data.

It needs to be stressed that, if the methodology proposed in this paper is used for the identification of example years for any other purposes, it cannot be considered as a CIBS Example Weather Year.

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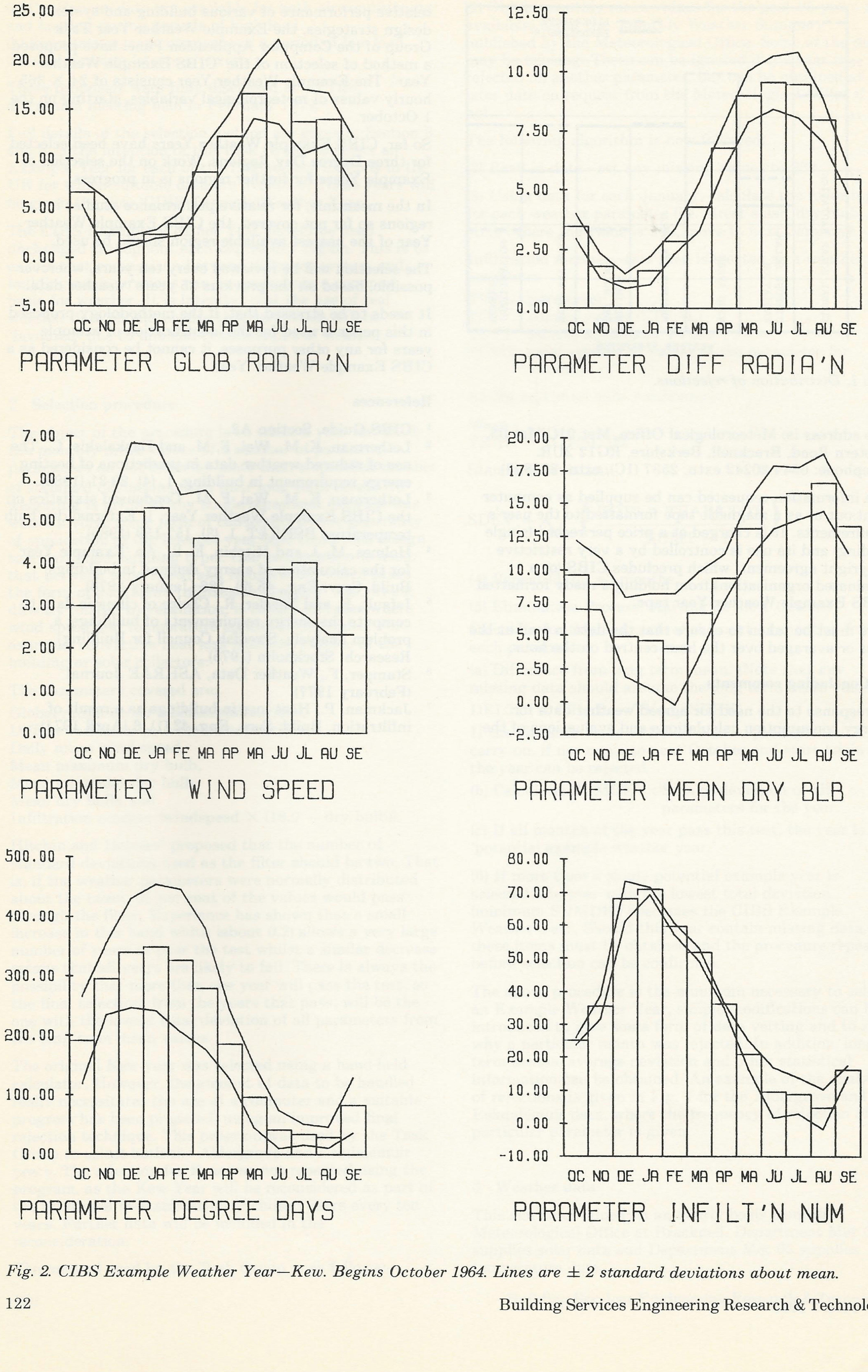

Fig. 2. CIBS Example Weather Year—Kew. Begins October 1964. Lines are ± 2 standard deviations about mean.

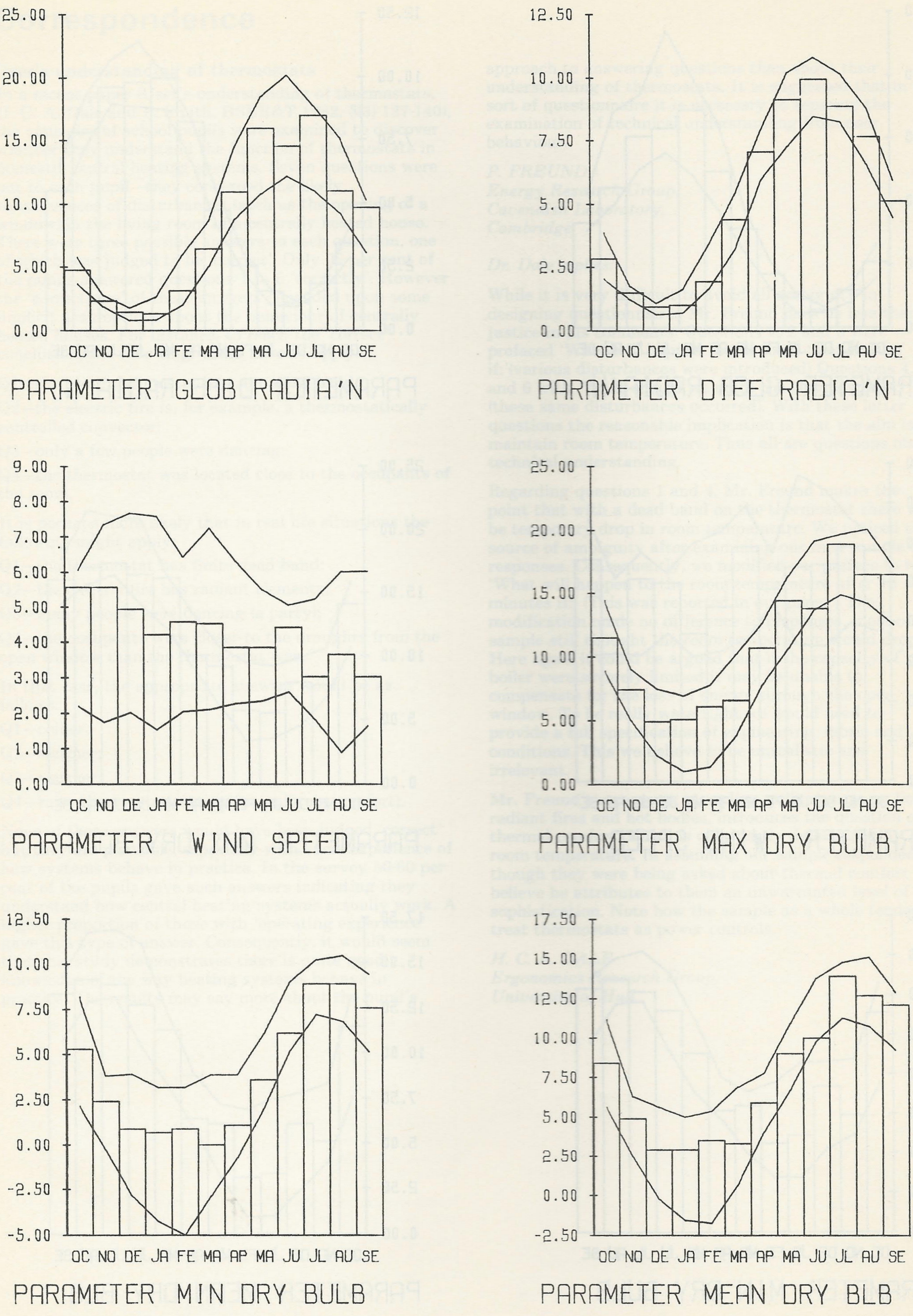

Fig. 3. CIBS Example Weather Year—Eskdalemuir. Begins October 1970.

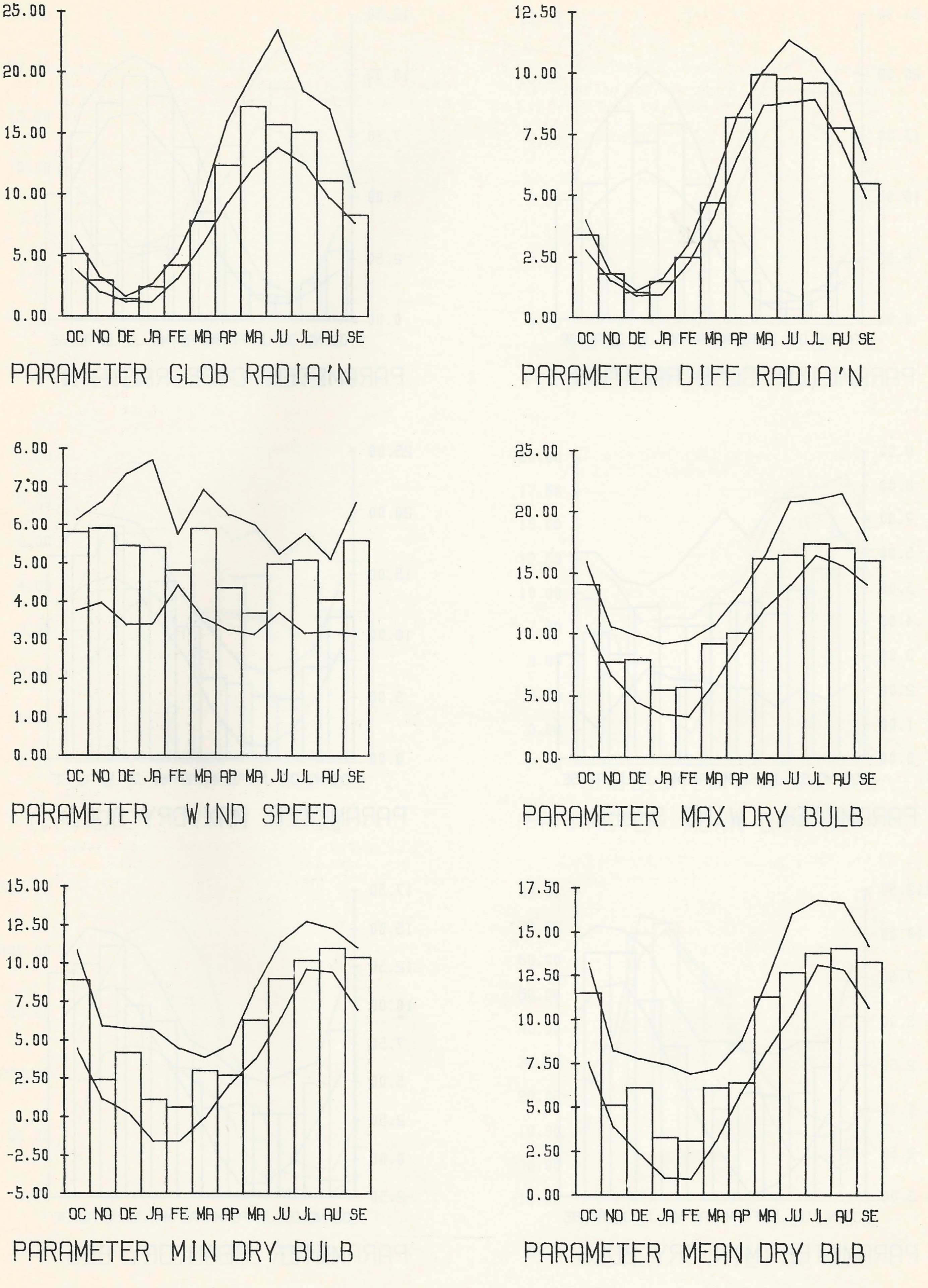

Fig. 4. CIBS Example Weather Year—Aldergrove. Begins October 1977.

Correspondence

User's understanding of thermostats

In a recent paper (User's understanding of thermostats, H. C. A. Dale and P. Smith, BSER&T 1982, 3(3) 137-140), the attitudes of school pupils were examined to discover whether they understand the function of thermostats in domestic central heating systems. Seven questions were put to each pupil—they concerned the likely consequences of disturbances (such as the opening of a window) in the living room of a centrally heated house. There were three possible answers to each question, one of which was judged to be 'correct'. Only 22 per cent of the pupils answered questions 1 to 6 'correctly'. However the 'correctness' of these answers depended upon some implicit assumptions about the behaviour of centrally heated houses. For instance, to reach the 'correct' conclusion, the following would have to apply:

Q1—the room thermostat has zero dead band;

Q2—the electric fire is, for example, a thermostatically controlled convector;

Q3—only a few people were dancing;

Q4—the thermostat was located close to the occupants of the room.

It is perhaps more likely that in real life situations the following might apply:

Q1—the thermostat has finite dead band;

Q2—the electric fire has radiant elements;

Q3—many people were dancing (a party);

Q4—the occupants were closer to the draughts from the open window than the thermostat was.

In that case, the appropriate answers would be as follows:

Q1—colder;

Q2—warmer;

Q3—warmer;

Q4—raise thermostat setting (to maintain comfort).

None of these answers would have been judged 'correct' but they are plausible, especially with some experience of how systems behave in practice. In the survey 50-60 per cent of the pupils gave such answers indicating they understand how central heating systems actually work. A higher proportion of those with 'operating experience' gave this type of answer. Consequently, it would seem that this study demonstrates there is quite good knowledge of the way heating systems behave in practice. The results may say more about the pupil's approach to answering questions than about their understanding of thermostats. It is suggested that in this sort of questionnaire it is necessary to separate the examination of technical understanding from user behaviour.

P. FREUND
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Cavendish Laboratory,
Cambridge.

Dr. Dale replies:

While it is very difficult to avoid all ambiguity in designing questionnaires Mr. Freund does us less than justice in his comments. Questions 1, 2 and 3 were prefaced 'What will happen to the room temperature if:'(various disturbances were introduced) Questions 4, 5 and 6 with 'What should we do to the thermostat if:' (these same disturbances occurred). With these latter questions the reasonable implication is that the aim is to maintain room temperature. Thus all are questions about technical understanding.

Regarding questions 1 and 4, Mr. Freund makes the point that with a dead band on the thermostat there will be temporary drop in room temperature. We noticed this source of ambiguity after examining our first sample of responses. Consequently, we modified our preface to read 'What will happen to the room temperature after 15 minutes if:' (This was reported in our paper.) The modification made no difference to responses, most of our sample still thought the room temperature would drop. Here again it could be argued that if the capacity of the boiler were severely limited it may be unable to compensate for the loss of energy through the open window. To be really watertight we would need to provide a full specification of engineering, fabric and conditions. This we believe to be unrealistic and irrelevant.

Mr. Freund's comments regarding radiant sources, both radiant fires and hot bodies, introduces the question of thermal comfort, which is only partly determined by room temperature. In assuming our sample responded as though they were being asked about thermal comfort we believe he attributes to them an unwarranted level of sophistication. Note how the sample as a whole tended to treat thermostats as power controls.

H. C. A. DALE
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University of Hull.

New Publications

BRITISH STANDARDS
Published by the British Standards Institution, 101 Pentonville Road, London, N1 9ND.

BS 864: Capillary and compression tube fittings of copper and copper alloy
864: Part 2: 1983 Specification for capillary and compression fittings for copper tubes, 16 pp, A4 size, £14.00.

Specifies requirements for capillary fittings and compression fittings for use with copper tubes of specified dimensions. Supersedes BS 864: Part 2: 1971. (ISBN 0 580 12881 4)

BS 5991: 1983 Specification for indirect gas fired forced convection air heaters for space heating (60 kW up to 2 MW input): safety and performance requirements (excluding electrical requirements), (2nd family gases), 24 pp, A4 size, £14.00.

Safety and performance requirements and methods of test for permanently installed open flued space heating appliances intended for industrial and commercial applications, operating on 2nd family gas. No current standard is superseded. (ISBN 580 11638 7)

BS 5588: Fire precautions in the design and construction of buildings
5588: Part 3: 1983 Code of practice for office buildings (formerly CP 3: Chapter IV: Part 3), 52 pp, A4 size, £19.50.

Planning, construction and protection of escape routes for occupants to use in the event of fire. Recommendations for fire precautions in engineering services and auxiliary accommodation, and for fire alarms and procedures, extinguishing and fire fighting equipment, water supplies, access for fire fighting and smoke control. Advice on the management of completed buildings (Supersedes CP 3: Chapter IV: Part 3: 1968 (ISBN 0 580 11991 2)

BS 6022: 1983 Classification of automatic steam traps (ISO 6704), 4 pp, A4 size, £7.00.

Classification of the main types of steam traps according to the mode of operation of their obturation device. No current standard is superseded. (ISBN 0 580 13316 8)

BS 6340: Shower units
6340: Part 1: 1983 Guide on choice of shower units and their components for use in private dwellings, 8 pp, A4 size, £6.00.

Provides details of minimum requirements to ensure that the facility provided is adequate and safe. Does not cover shower units in boats and caravans nor for the use of disabled persons. No current standard is superseded. (ISBN 0 580 11949 1)

6340: Part 2: 1983 Specification for the installation of shower units, 4 pp, A4 size, £4.50.

Minimum performance requirements, in addition to the performance details given in Part 1, to ensure the facility is adequate and safe. No current standard is superseded. (ISBN 0 580 11956 4)

BS 6380: 1983 Guide to low temperature properties and cold weather use of diesel fuels and gas oils (classes A1, A2 and D of BS 2869), 12 pp, A4 size, £10.50.

Provides basic guidance to help users overcome problems which may be encountered in very severe winters. No current standard is superseded. (ISBN 0 580 13322 2)

BS 5991: 1983 Specification for indirect gas fired forced convection air heaters for space heating (60 kW up to 2 MW input): safety and performance requirements (excluding electrical requirements), (2nd family gases), 24 pp, A4 size, £14.00.

Safety and performance requirements and methods of test for permanently installed open flued space heating appliances intended for industrial and commercial applications, operating on 2nd family gas. No current standard is superseded. (ISBN 580 11638 7)

BS 6350: 1983 Specification for gas heated fish and chip frying ranges, 16 pp, A4 size, £14.00.

Requirements and tests for gas heated fish and chip frying ranges of the type used in establishments whose main purpose is the preparation of fried fish and chips. No current standard is superseded. (ISBN 0 580 11905 X)

Amendments

BS 5839: Fire detection and alarm systems in buildings

5839: Part 1: 1980 Code of practice for installation and servicing Amendment No 4 £7.00. AMD 4242

New work started

Safety of domestic gas appliances. Part 5: Gas fires
Will amend BS 5258: Part 5. GSE/17

Safety of domestic gas appliances. Part 10: Flueless space heaters (excluding catalytic combustion heaters). Part 11: Flueless catalytic combustion heaters
Will amend BS 5258: Parts 10 and 11. GSE/17

Methods of measuring the performance of household electrical appliances
Thermal storage electric water heaters
Will replace BS 3999: Part 2: 1967 by one technically equivalent to the recent international publication. LEL/94

Petroleum fuels for land oil engines and boilers
Will revise BS 2869 to take account of the publication of BS 2000: Part 364 and the amendment of Part 218. PTC/2

Specification for safety of commercial electrical appliances using microwave energy for heating foodstuffs
Will update BS 5175 to accord with Amendments 1 to 5 of BS 3456: Part 1. LEL/161

Unfired fusion welded pressure vessels
Will prepare Amendment No. 3 which constitutes the third technical updating of this edition of BS 5500 containing, as well as a few corrections, revisions to the Appendix concerning fatigue and dished ends. PVE/1

Specification for servicing valves for water services
Will prepare a specification for types (a) spherical ball, (b) plug, and (c) screwdown, copper alloy servicing valves, intended to facilitate maintenance or servicing of a water fitting or appliance relating to BS 1010: Part 2 and BS 5412: Part 3. SEB/2

BS 1212: Float-operated valves. Part 4: Floats for use in small cisterns (NWC listed)
Will cover NWC approved float-operated valves not already covered by other Parts of BS 1212 for use, for example, in small cisterns. SEB/2

BS 1968: Floats for ballvalves (copper)
Will revise BS 1968: 1953 to update references. SEB/2

Shower heads and related equipment
Will specify requirements for the materials, dimensions and functional testing of domestic shower heads and related equipment to be used in conjunction with shower controls complying with BS 1415: Part 1, BS 1010: Part 2, and BS 5412: Part 3. SEB/37

BUILDING SERVICES RESEARCH AND INFORMATION ASSOCIATION
Available from BSRIA, Old Bracknell Lane West, Bracknell, Berks. RG12 4AH.

Heat recovery and heat pumps in buildings, by Gavin Hamilton and John Kew. BSRIA Technical Note TN4/83, £9.00.

This new Technical Note from BSRIA provides a guide to methods of heat recovery which may be used in buildings. It describes air-to-air heat recovery, heat pumps and heat recovery, refrigeration plant heat recovery, destratifying fans, hot waste gas heat recovery and liquid to liquid heat recovery. Each type of equipment is discussed, with illustrations, including its relative advantages and disadvantages and its suitability for various applications. The publication notes factors which affect the viability of heat recovery and technical and economic obstacles to its progress. Areas where further progress will promote the wider acceptance and use of heat recovery techniques are identified.

Market information is presented based on questionnaire response received from 125 companies including market leaders. With building services buyers and specifiers in mind a major section is devoted to a directory of suppliers which tabulates trade names, the original manufacturer and the year the equipment was introduced to the U.K.

BUILDING RESEARCH ESTABLISHMENT
Available from HMSO.

Explosibility assessment of industrial powders and dusts by Peter Field, a new reference book from the Fire Research Station, £2.50 plus 27p postage.

The potential hazards of combustible dusts are becoming increasingly acknowledged by industry—indeed both manufacturers and users of combustible powders have certain legal responsibilities under the Factories Act 1961 and the Health and Safety at Work etc. Act 1974. This book, an update of Fire Research Technical Paper No. 21 on Explosibility Tests for Industrial Dusts, discusses the dust explosion hazard in general terms.

The book indicates the basic requirements of explosion prevention and protection techniques. The experimental methods used by the Fire Research Station to determine explosibility and explosion parameters are described and a wide range of explosibility data is listed. The list includes metals, foodstuffs, agricultural products, chemicals, plastics, rubber, confectionery, dyestuffs, soaps and detergents, cosmetics, wood products, leather and pharmaceuticals. Although the risks associated with some of the powders are well-known in some areas of industry in others they are not well understood.

CHARTERED INSTITUTE OF BUILDING
Available from Sales Office, CIoB, Englemere, King's Ride, Ascot, Berks, SL5 8BJ.

Catalogue of publications, free of charge.
The Chartered Institute of Building has recently published its new catalogue of publications. This details the growing range of material on technical and managerial aspects of building now being published by the Institute. Over 150 titles are included ranging from monographs to individual papers published through the Technical Information Service.

Publications are listed under subject headings, including Building Law, Education, Estimating, Management, Maintenance, Site Management and Surveying. There is also a general section dealing with the Institute's journal 'Building Technology and Management' and 'Construction Computing', as well as the various information services currently available.

DISTRICT HEATING ASSOCIATION
Available from DHA, Bedford House, Stafford Road, Caterham, Surrey, CR3 6JA.

Planning for CHP heat, Conference Proceedings, £30.00.
Following the successful DHA 5th National Conference held at the Palace Hotel, Torquay, on 21-23 June, DHA Publications Ltd. announce that bound copies of the Conference Proceedings are available. The Proceedings comprise fifteen papers and their discussion, bound in a presentation binder. Following after the Atkins Report and the Select Committee on Energy Report on CHP, the Conference technical papers constitute the most recent examination of the factors involved in planning for CHP/DH systems.

The papers include: sizing and analysing DH networks, regional heat grids and heat storage, heat mains reliability in the U.K., and a national refuse incineration/recovery policy. There are three papers on development in Denmark, Sweden, and Finland; and three papers on who should market CHP heat.

AMERICAN SOCIETY OF HEATING, REFRIGERATION AND AIR-CONDITIONING ENGINEERS
Available from ASHRAE, 1791 Tullie Circle, N.E., Atlanta, GA 30329, USA.

ASHRAE Handbook, 1983 Equipment Volume, 80.00 dollars including postage.
ASHRAE has published the 1983 Equipment volume of ASHRAE Handbook. It joins 1982 Applications, 1981 Fundamentals and 1980 Systems as the books of the current ASHRAE Handbook series.

The Equipment volume describes types of equipment available for heating, ventilating, air conditioning and refrigerating (HVAC&R) applications; discusses principles of equipment operation, construction, performance testing and rating; and reviews equipment selection considerations.

The 1983 volume replaces the 1979 edition. It includes a new chapter on space heaters and expanded chapters on duct construction and compressors. Chapters on furnaces, unitary and packaged terminal air conditioners, and unitary and packaged terminal heat pumps have been rewritten. In all, there are 45 chapters in the hard cover book.

First published in 1922, the ASHRAE Handbook incorporates findings from ASHRAE research and draws upon the practice of more than 1,200 Society members selected to contribute on the basis of their professional qualifications.

Building Services Engineering Research & Technology

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[2] Smith, A. B. & Jones, C., 'Studies on Air Jets', JIHVE, **15**, 123 (1960).

[3] Brown, E. F., *Principles of Heating and Ventilating,* (ABC Press, London, 1970).

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