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A human calorimeter for the direct and indirect measurement of 24 h energy expenditure

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1. A calorimeter for the continuous measurement of heat production and heat loss in the human subject, for at least 24 h, is described. The calorimeter operated on the heat-sink principle for direct calorimetry and an open-circuit system for indirect calorimetry.

2. Sensible heat loss was measured using a water-cooled heat exchanger, and the temperature of water entering the heat exchanger was controlled to maintain a mean temperature gradient of zero across the chamber walls.

3. Evaporative heat loss was determined from ingoing and outgoing wet-and-dry bulb temperatures and air flow-rates.

4. Problems associated with the calculation of evaporative heat loss and the estimation of the volume of incoming air in open-circuit systems are considered.

5. The calibration, limits of accuracy, sources of error and experiments with subjects are discussed.

Whole-body calorimeters are used at present to measure the heat production and heat loss of farm animals over several days or weeks (Wainman & Blaxter, 1958; McLean, 1971; Verstegen, Close, Start & Mount, 1973). However, very little work of a similar type is being undertaken with man. Atwater & Benedict (1905) described a calorimeter which was used for measuring the energy expenditure of human volunteers. Each subject occupied the calorimeter for between 1 and 13 d. The trend during the last 30 years in direct calorimetry work on man has been towards short-term studies, which are often concerned with temperature regulation and the partition of heat loss (Winslow, Herrington & Gagge, 1936; Hardy & DuBois, 1938; Benzinger & Kitzinger, 1949; Benzinger, Huebscher, Minard & Kitzinger, 1958; Short, 1976). The emphasis has been on designing equipment with a rapid response time: the gradient-layer calorimeter described by Spinnler, Jéquier, Favre, Dolivo & Vannotti (1973) claims an over-all response time of less than 3 min.

In indirect calorimetry with human subjects little work has been carried out using respiration chambers. The ventilated hood (Ashworth & Wolff, 1969) is often used for resting subjects, while the Douglas bag (Douglas, 1911) and Kofranyi–Michaelis portable respiration calorimeter are used with active subjects (Passmore & Durnin, 1955).

To study problems of energy regulation and obesity it was necessary to construct a calorimeter which could be occupied for at least 24 h by a subject who was either at rest or active. Since Hardy & DuBois (1940), Bittel & Henane (1975) and Close, Dauncey & Ingram (1976) found differences in heat loss between men and women at different environmental temperatures, it was essential that the temperature of the calorimeter could be altered, and that heat loss could be partitioned into its sensible and evaporative components. Short-term components of energy expenditure such as the thermic effect of a meal are best measured by recording heat production, because of heat storage by the body (Pittet, Chappuis, Acheson, de Techtermann & Jéquier, 1976). Facilities for the measurement of oxygen consumption and carbon dioxide production therefore had to be incorporated. This paper describes the design, operation, calibration and use of a calorimeter operating on the heat-sink principle for direct calorimetry and an open-circuit system for indirect calorimetry.

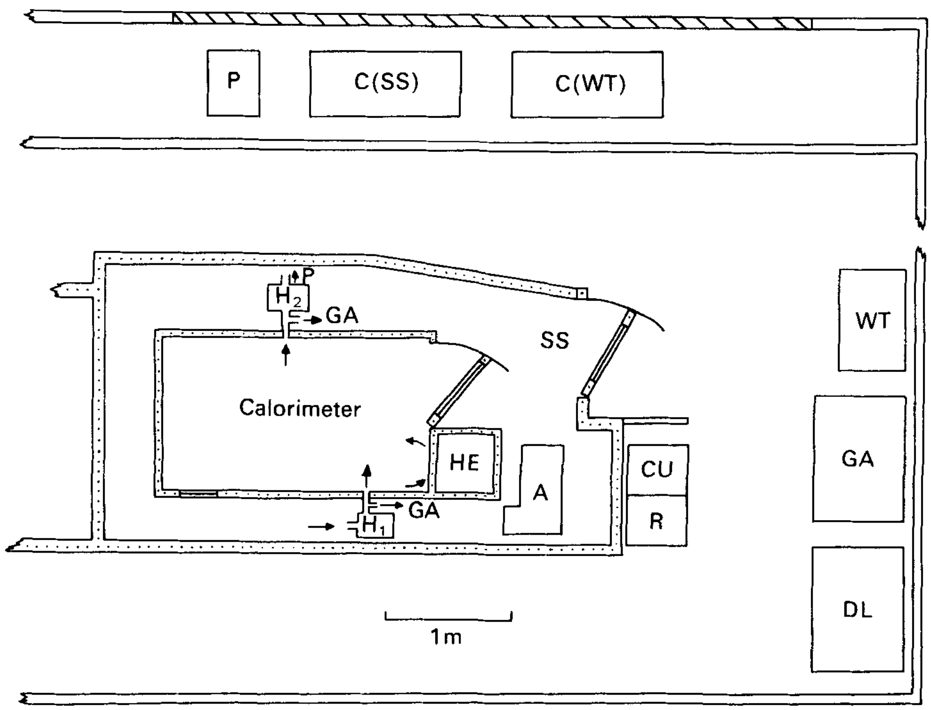

Fig. 1. Design of human calorimeter. HE, heat exchanger; SS, shell space; A, air-conditioning unit; H_1, H_2, hygrometers; P, pump; C(SS), compressor for shell space; C(WT), compressor for water tank; WT, water-cooling tank; GA, gas analysers; DL, data logger; CU, control unit; R, recorder.

DESIGN OF THE CALORIMETER

The design of the calorimeter (Fig. 1) was developed partly from the direct heat-sink calorimeter described by Mount, Holmes, Start & Legge (1967) for work with pigs. The calorimeter stands within a shell, 4·04 m long, 2·21 m wide and 2·05 m high. Air in the shell space was conditioned by recirculating it with two fans over a refrigeration evaporator and an electrical heater bank at a rate of approximately 35000 l/min. Feedback control of the heat input enabled the temperature of the shell to be maintained within ± 0·25° of any set point in the range 10–40°. The outer wall of the shell space was constructed of 100 mm expanded polystyrene, covered on the inside with Al sheeting and on the outside with plywood. The floor was of 25 mm blockboard positioned over 100 mm thick expanded polystyrene.

The calorimeter, 2·08 m long, 1·22 m wide and 1·95 m high, was similar in dimensions to that of Atwater & Benedict (1905). The structure was of 50 mm thick expanded polystyrene, supported on a timber frame and faced with 22 swg Al sheeting. The chamber was entered by a door fitted with a double-glazed window and two small air-tight hatches. It contained a chair, table, folding bed, bicycle ergometer, shelves, telephone, intercom and facilities for the collection of excreta. Illumination was provided by two 60 W lights in the shell space. Air entered the chamber through a duct from the shell, circulated inside the calorimeter and was exhausted through an extract duct. The ventilation rate of 200 l/min was monitored by a Rotameter (GEC-Marconi Process Control Ltd, Croydon, Surrey).

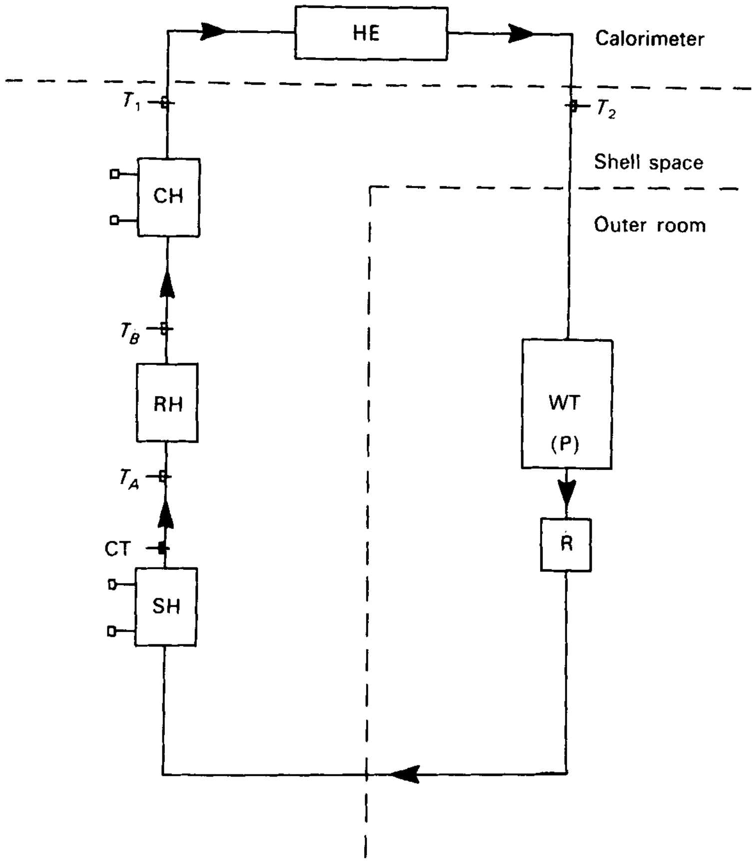

Fig. 2. Water circulation system for a human calorimeter. WT, water-cooling tank containing P, submersible pumps; R, Rotameter; SH, smoothing heater (controllable to 300 W); CT, control thermistor for smoothing heater; RH, reference heater (constant 100 W); $T_B - T_A$, reference-heater thermopile; CH, control heater (controllable to 625 W); HE, heat exchanger; $T_2 - T_1$, heat-exchanger thermopile.

MEASUREMENT OF HEAT LOSS

Sensible heat loss

The calorimeter was maintained at the same temperature as the shell space which could operate at any set point between 10 and 40°. The air within the calorimeter was recirculated at 12000 l/min over a water-cooled heat exchanger. The mean gradient across the walls and ceiling of the calorimeter was measured by a distributed thermopile having one thermocouple to each 0·4 m^2 of wall area. The thermopile provided the input to a three-term (proportional, derivative and integral) controller whose load was the 625 W control heater in the water-line before the heat exchanger (Fig. 2). The temperature of the water entering the heat exchanger and thus the rate of sensible heat extraction from the calorimeter were thereby controlled to maintain a gradient of zero across the walls. The rate of heat extraction was a function of the coolant mass flow-rate and the increase in water temperature across the heat exchanger.

Water leaving the heat exchanger returned to a tank where it was cooled to a temperature below that which was required for maximum heat removal from the calorimeter. Temperature fluctuations in the cooled water were smoothed by a 300 W heater regulated by a three-term controller, to give a constant temperature as sensed by a thermistor in the water leaving the heater. A reference heater then introduced a constant 100 W into the water-line. The temperature difference across the reference heater indicated the constancy of the water flow-rate, which was maintained at a nominal 1 l/min. In addition, it could be used to determine the heat extracted (W) by the heat exchanger from:

$$(T_2 - T_1)/(T_B - T_A) \times 100,$$

where $T_2 - T_1$ is the temperature difference across the heat exchanger and $T_B - T_A$ is the temperature difference across the reference heater (see Fig. 2).

The temperature differences across the heat exchanger and reference heater were measured by four-couple thermopiles in the water-line.

Evaporative heat loss

The difference in water vapour pressure between the air entering and leaving the calorimeter was proportional to the evaporative heat lost from the subject. Water uptake by the air circulating through the chamber was determined from the continuous monitoring of the wet-and-dry bulb temperatures of the ingoing and exhaust air. The temperatures were measured by thermocouples, paired for similar accuracy, in two identical Perspex hygrometers through which samples of air were drawn continuously at a nominal rate of 6 l/min, as indicated by Rotameters.

MEASUREMENT OF HEAT PRODUCTION

An open-circuit system was used to estimate O_2 consumption and CO_2 production for the computation of heat production (Fig. 3). Samples of air entering and leaving the calorimeter were carried in butyl-rubber tubing for gas analysis. The samples of air were passed at a rate of 2 l/min through drying tubes containing silica gel, and solenoid valves directed samples of ingoing and outgoing air through the analysers. A paramagnetic O_2 analyser (Taylor Servomex Ltd, Crowborough, Sussex) and infra-red CO_2 analyser (Analytical Development Co. Ltd, Hoddesdon, Herts.) were used for gas analysis.

RECORDING OF RESULTS

Twenty-four thermocouples were used to measure temperatures in the calorimeter, shell space and water-line. The accuracy of the thermocouples was $\pm 0.1°$ and the thermocouple reference was $0.0 \pm 0.1°$ (Churchill Instrument Co. Ltd, Greenford, Middx.). Temperatures could be read on a meter or observed on a chart recorder.

A data-logging system (Solartron Electronic Group Ltd, Farnborough, Hants) and paper tape punch (Facit-Addo Ltd, Rochester, Kent) were used to record, every 5 min, thermocouple and thermopile potentials, gas concentrations and barometric pressure.

ANALYSIS OF RESULTS

The information stored on paper tape was analysed using an IBM 1130 computer. The main functions of the programs were as follows:

Sensible heat loss

The increase in water temperature across the heat exchanger was proportional to the sensible heat lost from either an electrical calibration heater (see p. 563) or a subject. During a calibration test, the voltage, V, from the thermopile across the heat exchanger was obtained with the calorimeter empty and with a calibration heater of known power input, P_I. A

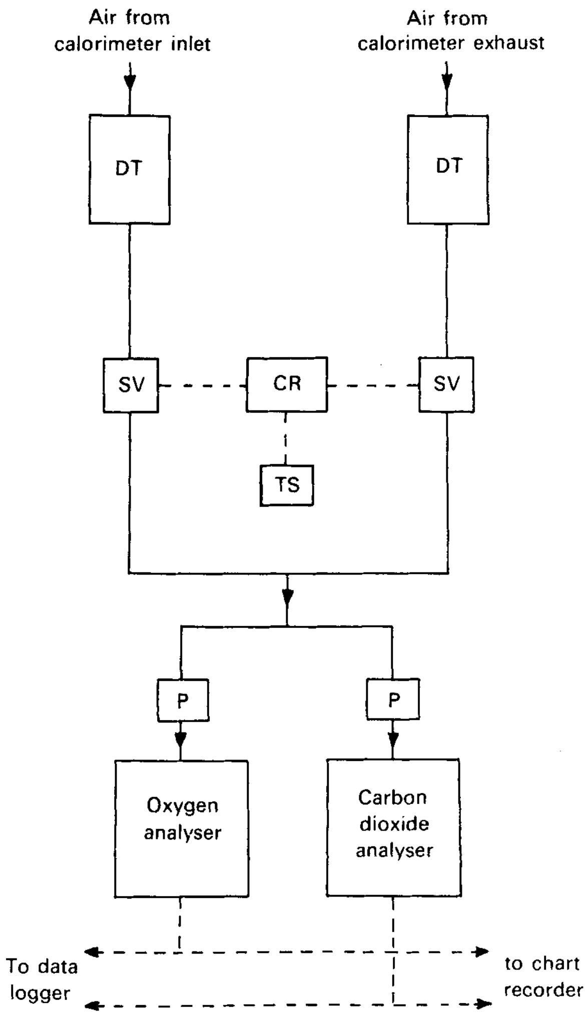

Fig. 3. Gas analysis system for a human calorimeter. DT, drying tube; SV, solenoid valve; CR, change-over relay; TS, time switch; P, pump.

'least-squares' line for P_I *v.* V was calculated, stored in the computer, and used to convert thermopile voltages to the corresponding amounts of heat while a subject was in the calorimeter. During transient conditions a correction was made for any heat exchange through the walls. A correction for heat storage in the wall material during transient conditions could also be made.

Evaporative heat loss

For the calculation of evaporative heat loss, the wet-and-dry bulb temperatures were derived from the corresponding thermocouple voltages, V, by a quadratic equation in V. The increase in water vapour pressure, VP, was estimated from the wet-and-dry bulb temperatures of the ingoing and exhaust air. If t' and t are the wet-and-dry bulb temperatures respectively and P is the air pressure in the hygrometer, then:

$$VP = SVP - kP(t - t'),$$

where SVP is the saturated vapour pressure at t'.

The *SVP* was obtained from a cubic equation in t', derived by multiple regression analysis from tables of *SVP* and accurate to 0·001 mmHg in the temperature range 15–25°. The precise value of k, the psychrometer 'constant', under the experimental conditions used, was unknown, and the problems involved in its determination have been discussed by Monteith (1954). Using the results of Awberry & Griffiths (1932), Whipple (1933) found a mean value of $6{\cdot}6 \times 10^{-4}$/deg. for k. However, this applies where the relative air velocity is at least 3 m/s, whereas that over the hygrometers used with the calorimeter was only 0·8 m/s. This velocity was more nearly equivalent to that of air over a Stevenson screen, in which k is taken as 8×10^{-4}/deg. This value for k was therefore used in the calculation of water vapour pressure, and a number of calibrations have indicated that its use was justified.

The volume fraction of water vapour in the air is given by VP/P (Dalton's law of partial pressures), where P is the atmospheric pressure. Thus the volume (l/min) of water vapour is given by $F \times VP/P$, where F is the volume air flow-rate (l/min). The increase in the volume of water vapour, W (l/min), is given by:

$$W = (F_o \times VP_o/P_o) - (F_i \times VP_i/P_i),$$

where the incoming and outgoing conditions are distinguished by the suffices i and o respectively, and F_i was corrected to outgoing dry-bulb temperature, T_o, and pressure, P_o.

The outgoing flow rate, F_o, was monitored using a Rotameter, and corected for changes in air density. F_i was usually different from F_o, and the method for deriving F_i is developed later.

The evaporative heat loss, E (W), was then calculated from:

$$E = W \times \left(\frac{273{\cdot}18 P_o}{760 T_o}\right) \times \frac{18}{22{\cdot}414} \times (2493 - 2{\cdot}26 T_o) \times \frac{1}{60},$$

where the four multipliers have the effect of (1) correcting the water vapour volume from outgoing conditions to standard temperature and pressure (STP), (2) converting the water volume to a mass (g/min), (3) multiplying by the latent heat of vaporization (J), which is assumed to be a linear function of T_o, and (4) converting J/min to W.

O_2 consumption and CO_2 production

In open-circuit systems a systematic error occurs when the respiratory quotient is less than unity (Misson, 1974), and when water vapour is produced. Under these conditions the volume flow-rate of air entering was greater than that leaving by a volume equal to the difference between O_2 consumption and CO_2 production. Similarly, the ingoing volume was decreased by the production of water vapour. Thus, although the volume of air leaving the calorimeter, F_o, was controlled, the volume entering, F_i, was not. F_o and F_i can only be related by assuming the conservation of nitrogen, and other inert gases in the ventilating, air. Let f_O and f_C be the concentrations of O_2 and CO_2 respectively in the dry air. By definition, the N_2 concentration, f_N, is given by:

$$f_N = 1 - f_O - f_C.$$

Also, let the dry-air flow-rates in and out be F_{di} and F_{do}, related to F_i and F_o by:

$$F_d = F(1 - VP/P), \tag{1}$$

since a proportion, VP/P, of the wet air is water vapour. Note that, like F_i, F_{di} was also corrected to outgoing temperature and pressure. Now, if the N_2 flow-rate is F_N, at outgoing temperature and pressure, then:

$$F_N = F_o(1 - VP_o/P_o) \times f_{No}, \tag{2}$$

$$= F_i(1 - VP_i/P_i) \times f_N. \tag{3}$$

It follows that:

$$F_i = \frac{F_o(1 - VP_o/P_o) f_{N_o}}{(1 - VP_i/P_i) f_{N_i}}.$$

The decrease in O_2 passing through the calorimeter is given by:

$$F_{di} f_{O_i} - F_{do} f_{O_o},$$

while for CO_2 the increase is:

$$F_{do} f_{O_o} - F_{di} f_{C_i}.$$

Substituting for F_{do} and F_{di}, from equations nos. 1, 2 and 3, these functions simplify to:

$$O_2 \text{ consumption} = F_N(f_{O_i}/f_{N_i} - f_{O_o}/f_{N_o}),$$

$$CO_2 \text{ production} = F_N(f_{C_o}/f_{N_o} - f_{C_i}/f_{N_i}).$$

Values were then corrected to STP.

The incoming shell space air was sampled on one in four occasions, and for each outgoing calorimeter sample the incoming conditions were obtained by linearly interpolating between adjacent shell samples.

The energy per unit volume of the consumed O_2 (kJ/l) is given by:

$$16{\cdot}494 + 4{\cdot}628 \times \text{RQ},$$

where RQ, the respiratory quotient, is rate of CO_2 production:rate of O_2 consumption (Bell, Davidson, & Scarborough, 1968).

CALIBRATION

Sensible heat loss

An electrical heater was used to simulate sensible heat loss. It was supplied from a constant voltage source via a variable-ratio transformer and a wattmeter. Several power inputs between 0 and 200 W were selected, and for each input a corresponding output from the heat exchanger thermopile was obtained. The output plotted as a function of power input produced a straight line calibration. Fig. 4 gives an example of results obtained at 28°.

The full calibration was carried out before each series of experiments. It was checked during individual experiments using two heat input levels, usually 0 and 125 W; each level was maintained for approximately 7 h, allowing approximately 30 min for equilibrium to be reached.

Since the proportions of radiative and convective heat lost from the calibration heater were different from those lost from man, a series of sensible heat tests using different sources of heat were carried out. Using a convective heater of power input 94 W, the recovery was 93·95 W, using a radiative heater of 60 W, the recovery was 59·88 W.

Evaporative heat loss

Water was evaporated by an electrical heater to simulate evaporative heat loss. The water was supplied via a continuous-infusion pump from a measuring cylinder which was used to monitor the volume of water evaporated. This volume was also calculated from the difference in vapour pressures between air entering and leaving the calorimeter. In addition to the heat being used for evaporation, some heat was liberated directly from the heater. Both the power supplied to the heater and the sensible heat loss were measured; the difference between these was the power used to evaporate the water.

A series of evaporative tests were carried out over 24 h, at 28° and 22°, using an evaporation rate of either 0·3 or 0·7 ml/min. This was equivalent to approximately 12 or 28 W,

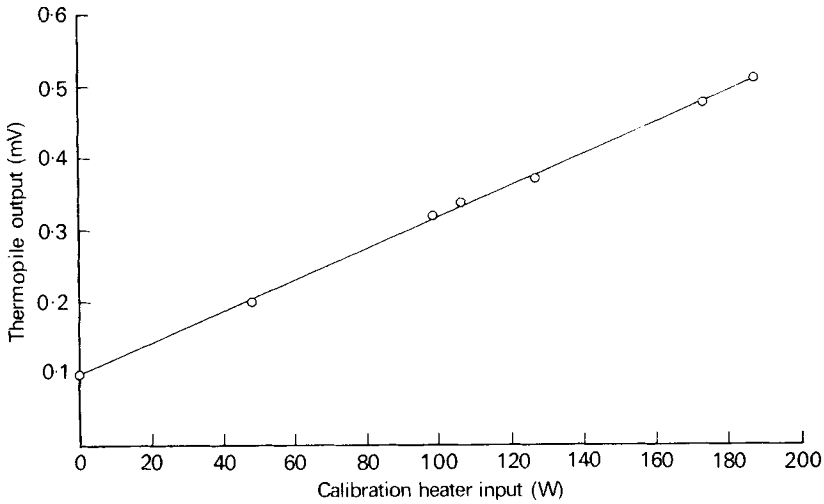

Fig. 4. Example of sensible heat loss calibration at 28°, measured using a human calorimeter, to show the relationship between calibration heater input and output of the heat-exchanger thermopile (see Fig. 2). The output of 0·1 mV at zero calibration represents the heat input from the air-circulation fan in the heat exchanger.

which could be expected from a resting or slightly-active subject. The difference between the measured power used to evaporate the water and that calculated to evaporate the volume of water pumped from the measuring cylinder was ±2 %.

O_2 and CO_2 analysers

The O_2 analyser was calibrated using O_2-free N_2 as the zero reference and atmospheric air as the span reference. The CO_2 analyser was calibrated with atmospheric air and a commercial mixture of CO_2–N_2 (approximately 5:95, v/v). The composition of this reference gas was checked with a Lloyd–Haldane gas analyser.

Calibration of the ventilation rate

The flow-rate of air leaving the calorimeter was calibrated before and after each series of experiments. With the flow-rate adjusted to its nominal value of 200 l/min, indicated by the Rotameter, the air was drawn through a Parkinson Cowan dry gas meter. The time taken for 2000 l to flow through the gas meter was measured and the true flow-rate was calculated. The ambient temperature, barometric pressure and pressure drop across the gas meter were measured at the time of calibration so that the flow-rate could be corrected to STP.

The calibration of the dry gas meter was checked at a flow-rate of approximately 5 l/min against a wet gas meter, to ensure that it remained within its nominal 1 % accuracy. The accuracy of the dry-gas meter was the limiting factor in determining the true ventilation rate through the calorimeter.

USE OF CALORIMETER WITH SUBJECTS

An example of results obtained while a subject was in the calorimeter is given in Fig. 5. The calorimeter has proved acceptable to eighteen male and female volunteers aged from 23 to 62 years, weighing between 47 and 107 kg, who have taken part in more than sixty experiments, each lasting 27–40 h, during the last 18 months. The substitution of gas

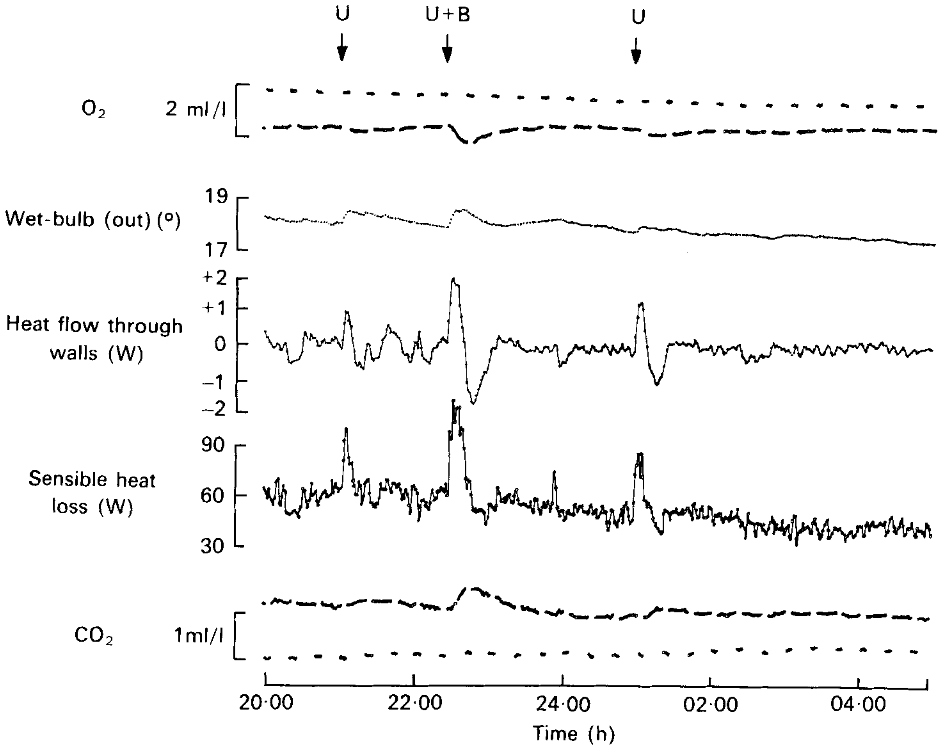

Fig. 5. Part of a chart record obtained at 28° while a subject was in the human calorimeter from 09.30 hours on day 1 to 15.30 hours on day 2 of the study period. B, subject going to bed; U, collection of urine sample. The activities of the subject were: 20.00–22.25 hours, sitting watching television; 22.25 hours–22·35 hours, preparation for bed; 22.35 hours onwards, asleep, apart from urine sample. Oxygen and carbon dioxide analysers sampled ingoing air for 5 min and outgoing air for 15 min. Both readings were proportional to barometric pressure, the CO_2 analyser was also affected by temperature. Evaporative heat loss was obtained from the ventilation rate and wet-and-dry bulb temperatures of ingoing and outgoing air (H_1 and H_2 of Fig. 1). For clarity, only the outgoing wet-bulb temperature is shown. The trace of heat flow through the walls illustrates the operation of the zero-gradient controller. A thermopile across the walls was used to control the calorimeter temperature. When heat flowed out (i.e. positive gradient) at the start of activity, the calorimeter temperature was reduced for a short period to return the heat flow to zero. Sensible heat loss was derived from the temperature difference across the heat exchanger (T_2-T_1 of Fig. 2). It was calibrated with a series of known power inputs.

analysers for gravimetric techniques and the use of data logging and computing systems made many more experiments possible than were originally done by Atwater & Benedict (1905). They carried out only twenty-two experiments covering 60 d during the 12 years in which their calorimeter was developed.

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