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THE BRIS COMPUTER PROGRAM FOR SIMULATING BUILDING THERMAL BEHAVIOUR

Physical basis and principles of data handling

GÖSTA BROWN

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Physical basis and principles of data handling

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CONTENTS 1

Preface

Up to now, a comprehensive description of BRIS, the oldest and most complete Swedish computer program for simulating building thermal behaviour, has not been published in English. In Swedish three articles were issued in 1963 and 1964 in the HVAC-journal VVS. They were also brought together in a reprint in 1964 from the National Swedish Institute for Building Research (see the reference list at the end of Introduction). The first article gives a method of calculating heat and light radiation in rooms, the second deals with calculating heating and cooling loads, while the third shows an application example.

These articles were translated to English in 1965, by Mr. R.M.E. Diamant, at that time lecturer in chemical engineering at the Royal College of Advanced Technology, Salford. These translations are presented here as Chapters 1, 2 and 3 with only a few corrections.

Including these early articles without alterations of the text may seem astonishing, after the passing of 25 years. However, the algorithms used in the program are still the same, even though the manner of presentation of input and output data is different, as well as the data handling in the program. The unit for thermal power is W in the program. In the Chapters 2 and 3 kcal/h has been kept, presumably without any disadvantage for the understanding. Further, for a reader of today, some remarks in the section 'Notes on calculation by means of a computer', at the end of Chapter 2, may seem superfluous. They are however representative of the problems at the time of the origin of the program, thus motivating a preservation.

Since then, the program has been used in a large number of applications in the fields of research and design. The basis of input data, such as for climate and heat transfer through windows etc., has expanded and improved. The computer technology has developed tremendously. All this has suggested that the way of dealing with the thermodynamics, as well as the data handling, be given a more detailed account. This has been done in Chapter 4.

An example shows an application of the program at the end of the data handling section. The printout from this example, including the input data, is given in Appendix B. Appendix A contains a set of blank data forms, while in Appendix C shading coefficients and thermal transmission coefficients of fenestrations are tabulated, for recommended use in the program.

A reader who is most interested in the use of the BRIS program and not so much in its physical background may concentrate his attention to Introduction, Chapter 4 and the Appendices.

Finally, I would like to express my great gratitude to Axel Bring, Engelbrekt Isfält and Ed Sowell.

Mr. Axel Bring is the programmer of BRIS. He has also from the beginning up to now accomplished all program improvements and amplifications, including the forms in Appendix A. He is the author of BRIS programbeskrivning (*Bring*, 1982) (User´s Manual for BRIS) on which the contents of Chapter 4, to a great extent, are based. By introducing the code usage of handling the socalled 'quantity sought' in the program, he has created good conditions for a versatile system simulation to achieve energy efficient control strategies.

Dr. Engelbrekt Isfält has been the most industrious user of BRIS. His interest in the research field of later years has been to gain a good thermal indoor climate by taking advantage of the heat capacity of building structures, in that way using a minimum of power input. This is exemplified by the design of ventilated floor slabs (see Section 4.2.6.).

Professor Edward F. Sowell of the Computer Science Department at the California State University, Fullerton, has been visiting professor at the Department of Building Services Engineering during the last six months. Sowell, who has been active on several ASHRAE technical committees concerned with these matters, has read the manuscript carefully, and I feel highly obliged to him for his comments and criticisms and, also, help in improving the language.

Stockholm, March 1989

Gösta Brown

INTRODUCTION

Some characteristics of the program

The BRIS program, developed at the Royal Institute of Technology, Stockholm, calculates the temperature variations that occur in a room subject to variable external and internal heat gains. An iterative, finite difference method is used to solve a system of heat balance equations, taking into account the effect of heat transfer by radiation between the room surfaces, by convection between the room air and room surfaces, and by conduction between the different material layers of the walls and floors. The surface temperatures and the room air temperatures are used to calculate operative temperatures (below also called effective temperatures) and the comfort number, which is a measure of the asymmetry of the radiation field. The temperatures on the outside surfaces of walls and floor slabs that enclose a room are also calculated. This allows conditions in adjacent rooms, having different heat gains, to be taken into account.

The implementation, using the iterative method, is briefly described as follows. All temperatures prevailing at the beginning of the period being studied, say 24 hours, are assumed to be known. The period is divided into time steps of, for instance, one hour or half an hour. The time step must be short enough so that the variation in temperatures, radiation intensities etc. during the step may be assumed to be linear (and so that oscillations in the calculations do not occur). A calculation, using the system of the heat balance equations, gives the temperatures at the end of one step from the corresponding figures at the beginning of the step. Once the temperatures at the end of the first step have been determined, they are used to calculate the temperatures at the end of the next step, with modified values of coefficients in the equations if necessary. Thus we proceed step by step until we have calculated the temperatures and other sought variables during the entire calculation period.

In the heat balance equation for a material layer, the temperature change of the layer during a time step is assumed to be proportional to the average value of the difference between the temperature gradient at the boundary surfaces of the layer at the beginning and at the end of the time step. In other words, implicit midpoint equations are used.

The iterative solution method is very advantageous in connection with quantities which are nonlinearly dependent on the calculated temperatures. This is the case for the exchange of longwave radiation between the room surfaces, which in BRIS is determined in accordance with the Stefan-Boltzmann law, and for the convective heat transfer at the room surfaces. The stratification at the floor is not like that at the ceiling. Therefore different expressions are used in BRIS for the temperature dependence of the convective heat transfer coefficients valid at wall, ceiling and floor.

Iterative calculations promotes a high degree of accuracy. Versatility is another advantageous property of the program. Versatility is achieved by use of a variable called 'quantity sought' which can be given different code numbers. The number says which of four quantities is to be determined by the program: room air temperature, supply air temperature, supply air flow rate, or heater output. Another value can be added to the code which implies that limitation values are applied to the sought quantity.

Climatic data (solar radiation, outdoor temperature) can be taken from a weather file. For design day calculations, they can also be calculated by the program. In the calculation of solar radiation (radiation from sun and sky, and reflected from the ground), a clear sky is assumed. This is well adapted for cases when an accurate estimation of solar radiation is most urgently needed, i.e. for calculation of cooling load. Reduction in solar radiation due to absorption in the atmosphere and a uniformly obstructed horizon can be taken into account. In design day calculations of heating demand, the reduction factor can be used to simulate the effect of average cloud cover.

Heat loss from external surfaces due to long wave radiation is taken into account for horizontal surfaces by a constant whose value depends on the cloud cover. For vertical surfaces, this radiation is assumed to be included in the surface heat transfer coefficient.

For the calculation of solar heat gain through a window the radiation is considered to be divided in two components. One is transmitted directly through the window without transformation to heat in panes or shading devices. This shortwave radiation is assumed to be emitted diffusely from the inside of the window. The proportions ultimately absorbed at the different room surfaces, after an infinite number of reflections, are calculated from the reflectivities of the surfaces and the room geometry. The other radiation component is absorbed in the window. Part of this absorbed energy is transmitted back to the outside of the window and transferred to its environment, while the remainder is emitted to the room from the inside of the window. This internal transfer occurs in two ways, namely by convection to the room air, and by long wave radiation to the room surfaces.

The program calculates the solar heat gain hour by hour in W/m² for a window with two panes of ordinary window glass. In order to get the heat gain through the fenestration in question, shading coefficients like those shown in Appendix C have to be given as input data, possibly as time dependent data when the shading devices are used during a portion of the calculation period.

The heat balance equation for the room air states that the heat, which is introduced convectively from heat sources in the room and by ventilating air and possibly leakage air (infiltration), is removed by the exhaust air and by convective heat transfer at the room surfaces.

In BRIS, a 'heater' is a versatile room component. It is normally assumed to be a radiator, placed on an exterior wall below a window, and emitting heat to the room air by convection and to the room surfaces by radiation. It is also assumed to have negligible thermal inertia when its heat output is the quantity sought. This standard model of the heater can be modified. It can, for example, be regarded as a cooler. It is then assumed to have no radiative thermal emission and its location in the room is unimportant. Also, heater without thermal output but with thermal inertia can simulate heavy furniture or other equipment in the room.

The flow rate and temperature of ventilating air can be prescribed or calculated in such a way that room climate restrictions are enforced. However, it is also possible in BRIS to include the HVAC system in the simulation. In this case the supply air flow and temperature as well as heating or cooling loads applied in the HVAC system will be a result of the calculations. The variables describing the HVAC system performance are included in the iterative solution method. The capability to handle varying recirculation of room air or varying heat recovery is of special interest.

Restrictions to system heating and cooling capacities can also be introduced. When they come into conflict with room climate restrictions, the system limitations will have precedence, and the resulting room climate will be calculated. The effects of capacity limitations can thus be studied directly in a realistic manner. It may be added that the versatility of the air handling model built in BRIS has made it possible to simulate a number of innovative systems.

The standard output from a BRIS run includes a heat balance report, produced for any day or for specified subperiods during the day. It can also be produced for longer periods.

It is important to observe that the heat balance is reported for <u>the room</u>, not for <u>the room air</u>. The room boundary for this balance report is the wall, ceiling and floor surfaces in the room, and the inside surface of the windows.

During each calculation step, the heat transferred to or from the room through the structure surfaces is determined by calculating the conducted heat through the material layer next to the surface of each structure element. The values are added and specified as 'walls' in the heat balance.

The thermal energy brought to the room through a window is regarded as the difference between the solar radiation energy reaching the inside surface of the window and the energy transmitted from this surface back to the outdoor air, which is caused by the temperature difference between the surface and the air. In the heat balance, the solar radiation and the heat transfer to the outdoor air are specified as separate components.

The heat gain and loss contributions shown are therefore from: solar radiation, window heat transmission, people, lighting, space heater, 'walls', energy brought to the room by supply air, and energy brought to the room by infiltration.

Of these items, 'walls' is of special interest. This quantity of heat gives a clear picture of the heat storage effects of the envelope of the room. During the peak hours, this could be the dominant item in the heat balance, and is often seen to reduce cooling requirements to half or less of the heat gains.

History and usage of the program

Use of computers in the study of temperature variation in buildings started early in Sweden. The first example of such use dates from 1957 when a method was described for computer calculation of the temperatures in an external wall exposed to solar radiation, *Brown* (1957). In the same year another investigation was presented dealing with design outdoor temperatures for heating load calculations, *Adamson, Brown & Hovmöller* (1957). The temperatures proposed in this work were determined on the basis of computer calculated temperature variations in different types of buildings caused by temperature variation outdoors. In both these cases, the calculations were made on the Swedish computer BESK which was installed in 1953.

In a following paper, *Brown* (1962), is pointed out that there are three different ways of calculating temperature variations in buildings: by analytical methods, by electrical analogues and by numerical methods. The analogues were used, for instance, in Denmark and in the Netherlands. The Division of Heating and Ventilating at the Royal Institute of Technology in Stockholm always used numerical methods only, solving difference equation systems with digital computers.

At the same time as the BRIS program began to be utilized, there developed a demand for better solar radiation values. Tables and charts giving irradiation from sun and sky in Sweden on clear days were compiled, *Brown & Isfält* (1969). The algorithms compiled in connection with this publication are used in BRIS.

Brown & Isfält (1974) gives a comprehensive description (in Swedish) concerning solar irradiation and shading devices. A question of vital importance in the context of solar shading devices is the transmission of daylight through the window. The cited report briefly describes methods for daylight calculations. The BRIS program has been used to study the distribution of diffuse daylight in a room with one (or more) window(s) (see Chapter 1). These methods for determination of daylighting levels make it possible, for example, to establish to what extent artificial lighting must be used to meet different illumination requirements, with attendant increase in the cooling load of the room. These questions are further described in *Isfält* (1971) (in English).

At the beginning of the seventies some papers were published abroad, treating the BRIS program and its use. The emphasis of these works was on the interaction which takes place between a building and its climatic installations: *Brown* (1971), *Brown* (1972), *Mandorff* (1971), *Brown & Isfält* (1973).

In *Brown* (1972), a comparison is shown between the measured air temperature in a laboratory room exposed to strong solar radiation during some days through a large window and the air temperature in that room as simulated by BRIS. The agreement was very good.

In *Brown & Isfält* (1973) it is demonstrated that low cooling loads can be achieved by proper use of the heat capacity of buildings. The cooling requirement in rooms with windows of the same size and emitting the same solar energy to the room depends on how the energy is transferred from the inside of the window. This can be mainly by convection to the room air, or mainly by radiation to the room surfaces. The cooling load will be lower in the latter case due to the absorption of energy at peak hours. It is shown by an example that the heat storage effect of the external wall in a room does not have an appreciable influence on room temperature, since its surface (excluding the window area) is small in relation to the sum of the surfaces which enclose the room. The heat capacity of the floors is always large in modern buildings, but they are often provided with floor coverings or suspended ceilings, which obstruct the surface transmission of heat. Direct contact between ventilation air and floor slab can however be achieved if the air is conveyed through ducts in the floor (see Section 4.2.6). Further, it is shown that if the maximum output of a cooling installation is too low to maintain the temperature which is set on the room thermostat, the rise in temperature may nevertheless be relatively moderate owing to the heat storage which occurs in the building structure.

At the National Swedish Institute for Building Research, *S.Mandorff* has developed a method in which computer and manual calculations are combined for determination of high room temperatures as a result of hot outdoor climate (*Mandorff*, 1971). It was shown that on the basis of a few computer calculations (using BRIS), supplemented by simple manual calculations, it is possible to obtain detailed information on the temperature during 24-hour periods for a range of different outdoor temperatures. A more condensed form can be achieved by calculating the cumulative frequency distribution of the indoor temperature. Allowance must be made for the frequency distribution of the outdoor temperature and the related amount of daily solar radiation, if a value which reflects actual conditions is to be obtained. The estimated total of time intervals for which a given temperature will be exceeded can be used as a 'thermal performance index' when assessing the estimated thermal indoor climate. It can be used for comparing different climates resulting from alternative designs of building and mechanical services. - The method is used by The National Board of Public Buildings, and also by consultant firms.

During the last decade, the BRIS program has been extended and improved several times. At the beginning, the programming language was ALGOL. It was run on TRASK, a Swedish computer with a small core memory, necessitating complicated programming structure. Now, the language is ANSI Fortran 77. The program is installed on two computers in Stockholm, namely a PRIME 750 and a Cyber 170/730, available via Tymnet and Euronet.

As regards the capacity of the program, it can be mentioned that the thermal climate in ten coupled rooms can be treated simultaneously. Another point of interest is the accuracy which is inherent in the finite difference method, superior to the accuracy of other methods which may be faster as to heat transfer determinations. We have not changed to any new method, because most of the computer time is spent on the selection of a 'desired' solution from many possible solutions. A long chain of relaxations on temperature restrictions and limitations in installed conditioning capacity is then made at each step.

The BRIS program was employed in several contributions to 'the Fourth International Symposium on the Use of Computers for Environmental Engineering Related to Buildings' in Tokyo. Topics covered included the utilization of the thermal mass of buildings for reducing the size of cooling and heating systems (*Brown & Isfält*, 1983), and the use of versatile system simulations to find optimum energy saving strategies and to study tentative new technological developments (*Bring* , 1983). The design of hollow core concrete slabs (*Andersson et al.*, 1979) may be said to consistute such a new technological development.

The first part of the first mentioned contribution to the symposium gives results from an analytical study on the thermal response of slabs exposed to temperature changes. This issue is more thoroughly treated in *Brown* (1984).

In the design of a building and its HVAC equipment, a manual calculation method can be a help at a preliminary stage. In *Brown* (1987) a method based on BRIS simulations is presented, intended to show in a swift and clear way how the indoor temperature and the energy requirements depend on outdoor climate, air change, heat storage capacity of the space, and some other parameters.

A trend in architecture is the increasing use of glass. In *Lundquist et al.* (1980), glass house projects are described where the BRIS program has been used.

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1. CALCUALTION OF HEAT AND LIGHT RADIATION IN ROOMS

Definitions

The *absorption factor* Ψ_{ij} defines the fraction of radiation from a surface A_i which is absorbed by a surface A_j. In this connection not only the fraction of radiation from A_i, which radiates directly upon A_j and is absorbed there, but also the energy of radiation, which reaches A_j and is absorbed by it after being reflected by all the reflecting surfaces in the room, is included. Radiation from A_i can be in the form of long wave length radiation due to the emissive power of the surface:

$$E_i = \varepsilon_i C_s \left(\frac{T_i}{100}\right)^4$$

where

ε_i = emissivity

C_s = $10^8 \cdot \sigma$, where σ = Stefan Boltzmann's constant, i.e. $5.67 \cdot 10^{-8}$ W/m^{2}K^4

T_i = the absolute temperature of the surface in K

It can also be in the form of short and long wave radiation reflected from the surface A_i.

The absorption factor Ψ_{ij} depends upon the absorptivities, the sizes and forms of all the surfaces in the room, and the positions of all surfaces in relation to each other. If all room surfaces are black ($\varepsilon = 1$) the fraction of the radiation from a surface A_i which is absorbed by a surface A_j is determined only by the size of the two surfaces, by their form and their positions in relation to each other. In such a case one substitutes *angle factor* φ_{ij} for the absorption factor Ψ_{ij}.

Conditions

The following conditions apply to the use of the method:

a) The surfaces in the room are plane and rectangular, do not overlap and are either parallel or perpendicular to each other.

b) The temperature and the absorptivity of a given surface are the same over the whole surface. (A wall, for example can, however, be divided up into parts with various temperatures and absorptivities.)

c) All radiation is emitted and reflected diffuse.

d) The room air does not absorb any part of the radiation.

Equations for calculating the angle factors

A plane *black* surface A_i has an emissive power

$$E_{si} = C_s \left(\frac{T_i}{100}\right)^4$$

As the radiation from A_i against the entire surroundings is equal to $E_{si}A_i$ W, the fraction of radiation which is absorbed by a surface A_j (fig. 1.1.) is equal to:

$$\varphi_{Ai \rightarrow Aj} E_{si} A_i$$

according to the definition of the angle factor. In the same way the fraction of radiation from the surface element dA_i which reaches surface element dA_j is equal to:

$$\varphi_{dAi \rightarrow dAj} E_{si} dA_i$$

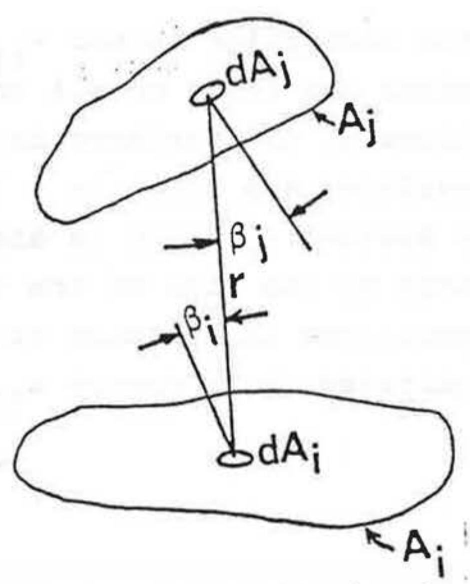

Fig. 1.1. Determination of angle factors for radiation between two surfaces.

According to Lambert´s cosine law (see fig. 1.1, where β_i and β_j are the angles between the normals and the tieline between the surface elements), it is possible to express this fraction of radiation from dA_i as:

$$\frac{E_{si}\,dA_i}{\pi r^2}\cos\beta_i\cos\beta_j\,dA_j$$

and therefore one obtains:

$$\varphi_{dAi \to dAj} = \frac{\cos\beta_i\cos\beta_j}{\pi r^2}\,dA_j \tag{1a}$$

In the same way:

$$\varphi_{dAj \to dAi} = \frac{\cos\beta_i\cos\beta_j}{\pi r^2}\,dA_i \tag{1b}$$

The angle factor for radiation from dA_i against A_j i.e. $\varphi_{dAi \to Aj}$ can be obtained by summating all values of $\varphi_{dAi \to dAj}$ for radiation from dA_i against all surface elements of the surface A_j:

$$\varphi_{dAi \to Aj} = \int_{A_j}\varphi_{dAi \to dAj} = \int_{A_j}\frac{\cos\beta_i\cos\beta_j}{\pi r^2}\,dA_j \tag{2}$$

It is then possible to obtain $\varphi_{Ai \to Aj}$ by forming the average value of $\varphi_{dAi \to Aj}$ by integrating over the surface A_i:

$$\varphi_{Ai \to Aj} = \frac{1}{A_i}\int_{A_i}\varphi_{dAi \to Aj}\,dA_i \tag{3}$$

$$\varphi_{Ai \to Aj} = \frac{1}{A_i}\int_{A_i}\int_{A_j}\frac{\cos\beta_i\cos\beta_j}{\pi r^2}\,dA_j\,dA_i \tag{4a}$$

Similarly:

$$\varphi_{Aj \to Ai} = \frac{1}{A_j} \int_{A_j} \int_{A_i} \frac{\cos \beta_i \cos \beta_j}{\pi r^2} dA_i \, dA_j \qquad (4b)$$

As it is immaterial in which order the integration is carried out in the two equations, the following equation applies:

$$A_i \varphi_{Ai \to Aj} = A_j \varphi_{Aj \to Ai}$$

or, if the following writing simplification is employed $\varphi_{Ai \to Aj} = \varphi_{ij}$, $\varphi_{Aj \to Ai} = \varphi_{ji}$:

$$A_i \varphi_{ij} = A_j \varphi_{ji} \qquad (5)$$

With the help of equation (5) it is easy to calculate φ_{ji} after previously determining φ_{ij}.

An other fundamental rule with the calculation of angle factors for surfaces in a room is, that the sum of the angle factors for radiation from a surface against the remaining surfaces must be equal to 1. If the surfaces are designated 1, 2, 3, ..., n, the sum of the angle factors for radiation from, for example, surface 2 is equal to:

$$\varphi_{21} + \varphi_{23} + \varphi_{24} + \varphi_{25} + \ldots + \varphi_{2n} = 1 \qquad (6)$$

As it is assumed that surface 2 is plane and therefore does not receive any radiation from itself, $\varphi_{22} = 0$.

With the method which is described here for determining angle factors by means of a computer, equations are used in the worked out programme, which only contain terms which can be calculated by means of equations (7) and (8):

$$\varphi_{ij} = \frac{2}{a\,b\,\pi}\left[a\sqrt{b^2+h^2}\,\mathrm{arctg}\frac{a}{\sqrt{b^2+h^2}} - ah\,\mathrm{arctg}\frac{a}{h}\right.$$
$$+ b\sqrt{a^2+h^2}\,\mathrm{arctg}\frac{b}{\sqrt{a^2+h^2}} - bh\,\mathrm{arctg}\frac{b}{h}$$
$$\left. - \frac{h^2}{2}\ln\frac{(a^2+b^2+h^2)\,h^2}{(a^2+h^2)(b^2+h^2)}\right] \quad (7)$$

$$\varphi_{ij} = \frac{1}{\pi}\left[\mathrm{arctg}\frac{b}{h} + \frac{a}{h}\,\mathrm{arctg}\frac{b}{a}\right.$$
$$- \frac{\sqrt{a^2+h^2}}{h}\,\mathrm{arctg}\frac{b}{\sqrt{a^2+h^2}} + \frac{a^2}{4\,bh}\ln\frac{(a^2+b^2+h^2)\,a^2}{(a^2+b^2)(a^2+h^2)}$$
$$\left. - \frac{b}{4\,h}\ln\frac{(a^2+b^2+h^2)\,b^2}{(a^2+b^2)(b^2+h^2)} + \frac{h}{4\,b}\ln\frac{(a^2+b^2+h^2)\,h^2}{(a^2+h^2)(b^2+h^2)}\right] \quad (8)$$

These equations apply (according to *Kollmar-Liese* [1]) for the following two basic cases (see fig. 1.2.):

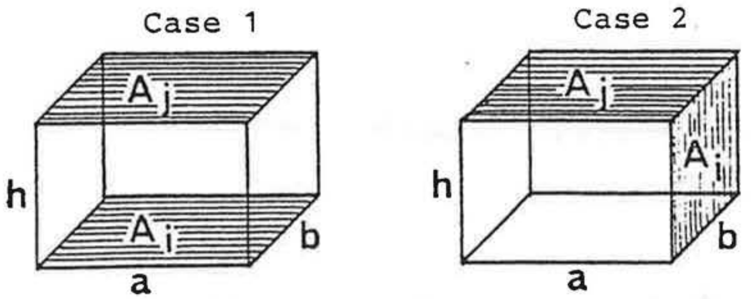

Fig. 1.2.

1) Two parallel, rectangular surfaces which are similar in size, where one lies directly above the other (equ. (7)).

2) Two surfaces facing each other at right angles, which share one edge (equ. (8)).

In order to apply various actual cases to such basic conditions, *angle functions* can be calculated as follows (see *Squassi* [2]).

During radiation from a surface A_a against a surface A_b the angle function is defined as:

$$\Phi_{a \to b} = \varphi_{a \to b} A_a \tag{9}$$

As, according to equation (5), $\varphi_{a \to b} A_a = \varphi_{b \to a} A_b$, one obtains

$$\Phi_{a \to b} = \Phi_{b \to a} \tag{10}$$

With regard to the addition of angle functions, the following is to be observed. If a surface A_a radiates against another surface A_a', which consists of several part surfaces, A_b, A_c ..... A_n,

$$\Phi_{a \to a'} = \Phi_{a \to b} + \Phi_{a \to c} + \ldots + \Phi_{a \to n} \tag{11}$$

and

$$\Phi_{a' \to a} = \Phi_{b \to a} + \Phi_{c \to a} + \ldots + \Phi_{n \to a} \tag{12}$$

apply.

For parallel and perpendicular surfaces respectively, the arrangement according to fig. 1.3, is

$$\Phi_{a \to d} = \Phi_{b \to c} \tag{13}$$

(see *Raber - Hutchinson* [3].

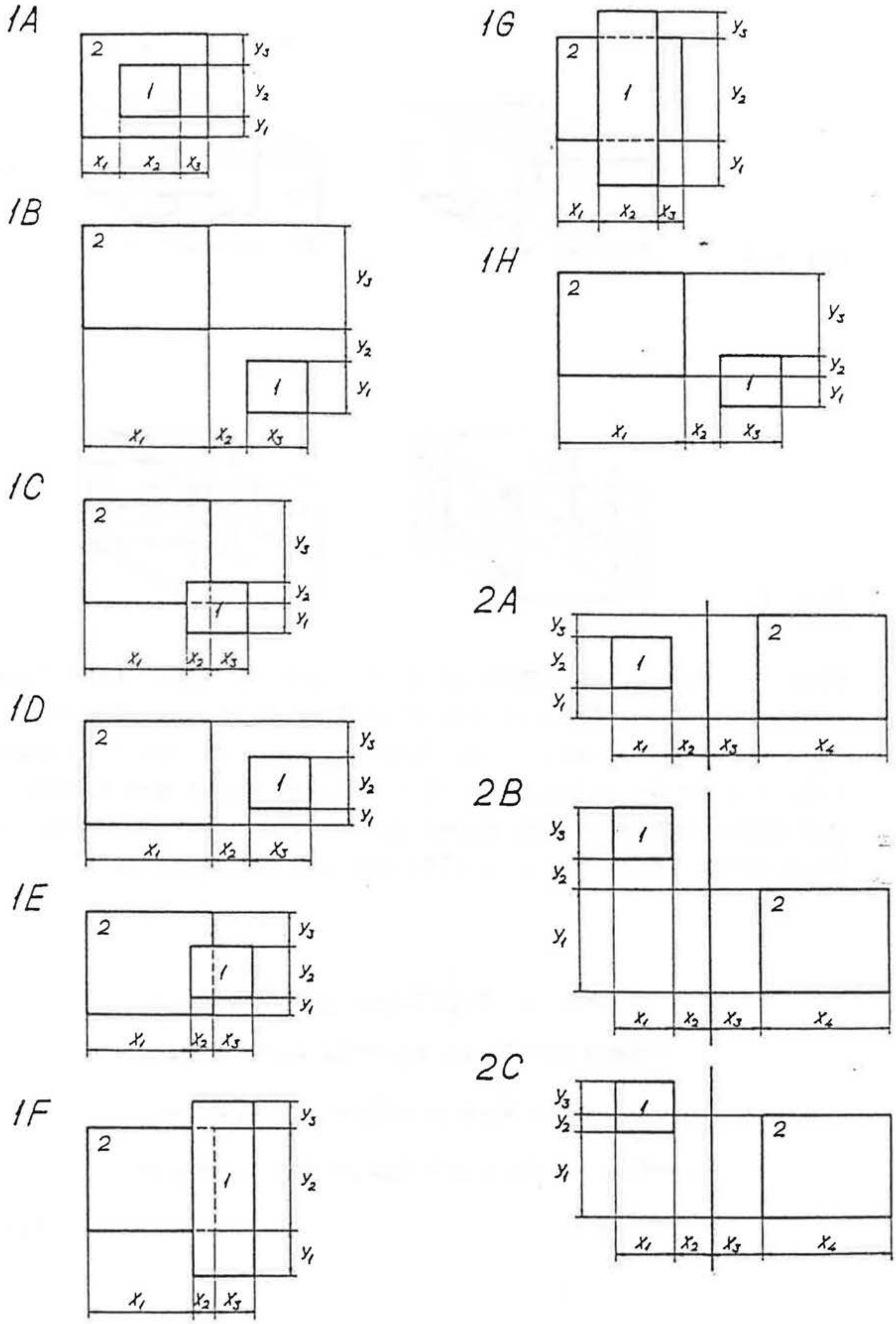

Fig. 1.5. Parameters required for automatic computation of angle factors.

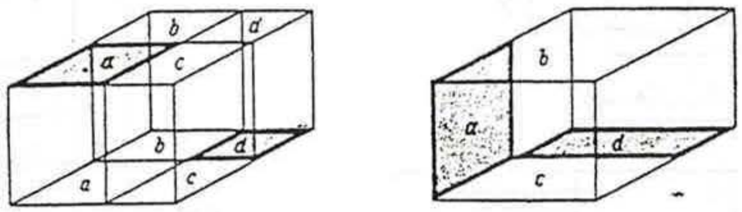

Fig 1.3.

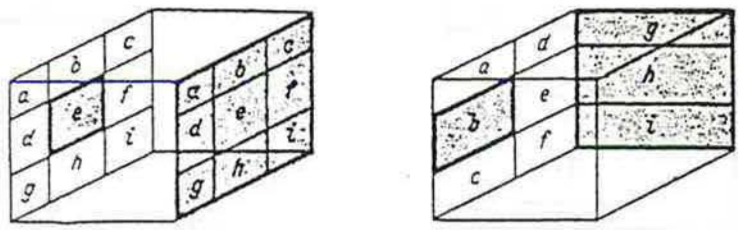

Fig. 1.4.

Fig. 1.4 gives two examples of the use of angle functions, where the angle factors for radiation from a window *e* against a facing wall, and from a window *b* against a side wall are to be calculated. The wall surfaces are subdivided into rectangles as shown in the illustration. With the help of equations (10) - (13) the angle functions:

$$
\begin{aligned}
\Phi_{e \to abcdefghi} = \frac{1}{4}(&\Phi_{abdc \to abde} + \Phi_{bcef \to bcef} + \\
&+ \Phi_{cfhi \to cfhi} + \Phi_{degh \to degh} - \Phi_{ab \to ab} - \Phi_{ad \to ad} - \\
&- \Phi_{cf \to cf} - \Phi_{bc \to bc} - \Phi_{ih \to ih} - \Phi_{fi \to fi} - \\
&- \Phi_{dg \to dg} - \Phi_{gh \to gh} + \Phi_{a \to a} + \Phi_{c \to c} + \Phi_{i \to i} + \\
&+ \Phi_{g \to g})
\end{aligned} \tag{14}
$$

and

$$\Phi_{b \to ghi} = \frac{1}{2}(\Phi_{abdc \to gh} + \Phi_{bccf \to hi} - \Phi_{de \to gh} -$$
$$- \Phi_{cf \to hi} - \Phi_{ad \to g} - \Phi_{cf \to i} + \Phi_{d \to g} + \Phi_{f \to i}) \quad (15)$$

are obtained.

It can be seen that every term in equations (14) and (15) can be calculated with the help of equations (7) and (8), after the angle functions have been exchanged for angle factors according to equ. (9).

Also other cases which can occur in a room are shown in fig. 1.5, where the two cases in fig. 1.4 correspond to 1A and 2A. 1A to 1H inclusive, show parallel surfaces, the surfaces 1 and 2 being projected in the same plane. In the cases 2A - 2C, the two surfaces are inclined towards each other at 90°. In the figure, however, they are turned 90° so that they there lie in the same plane. The corner, which is formed by the two planes of surface 1 and surface 2, is given as a straight line.

The programme includes also the calculation of the angle factor for radiation from a small sphere with surface A_i against a rectangle with surface A_j, with one corner lying over the sphere at a distance h from it. For such a fundamental case that one corner of the rectangle lies directly over the sphere, the following equation has been derived:

$$\varphi_{ij} = \frac{1}{4\pi} \arcsin \frac{ab}{\sqrt{h^4 + h^2(a^2 + b^2) + a^2 b^2}}$$

where a and b are the sides of the rectangle. Here also the reciprocal theorem equ. (5) and the summation rule equ. (6) apply.

Example 1

Let us assume that the angle factors for radiation between all surfaces in a room according to fig. 1.6 are to be determined. Let surfaces 1 and 2 be windows (actually against the inside of the external wall), and 6 and 7 are the glass part and the timber part of a French window, while 3 is a radiator.

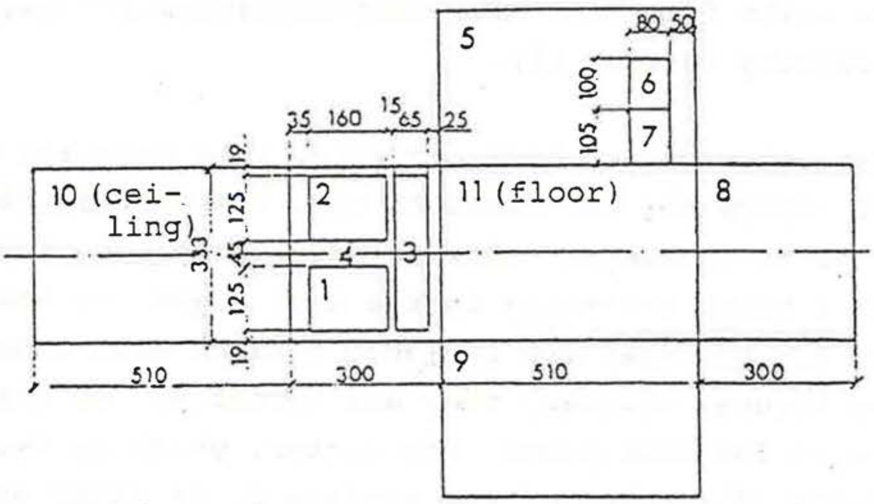

Fig. 1.6.

Table 1.1 summarises how the various angle factors should be calculated. In the top columns of the table, the designation of the surface, which is considered as facing the radiation is given, i.e. the second index number in the designation of the angle factor.

As regards angle factors for surface 1, it is only $\varphi_{1 \to 8}$, $\varphi_{1 \to 9}$, $\varphi_{1 \to 10}$ and $\varphi_{1 \to 11}$ which require to be calculated with the computer. The corresponding columns in the table give the cases in fig. 1.5 which are applicable. $\varphi_{1 \to 1}$ to $\varphi_{1 \to 4}$ inclusive are 0, as surfaces 1-4 do not receive any radiation from surface 1.

According to the table $\varphi_{1 \to 5} = \varphi_{2 \to 9} - \varphi_{1 \to 6} - \varphi_{1 \to 7}$.

The equation can be written $\varphi_{2 \to 9} = \varphi_{1 \to 5} + \varphi_{1 \to 6} + \varphi_{1 \to 7}$.

Table 1.1. Scheme for calculating angle factors for radiation between surfaces in a room according to fig. 1.6.

	1	2	3	4	5	6	7	8	9	10	11
φ_1	0	0	0	0	$\varphi_{2-9}-\varphi_{1-6}-\varphi_{1-7}$	$\frac{A_6}{A_1}\varphi_{6-1}$	$\frac{A_7}{A_1}\varphi_{7-1}$	1A	2A	2A	2A
φ_2	0	0	0	0	$\varphi_{1-9}-\varphi_{2-6}-\varphi_{2-7}$	$\frac{A_6}{A_2}\varphi_{6-2}$	$\frac{A_7}{A_2}\varphi_{7-2}$	φ_{1-8}	2A	φ_{1-10}	φ_{1-11}
φ_3	0	0	0	0	$\varphi_{3-9}-\varphi_{3-6}-\varphi_{3-7}$	2B	2A	1A	2A	2A	2A
φ_4	0	0	0	0	$1-\varphi_{4-6}-\varphi_{4-7}-\varphi_{4-8}-\varphi_{4-9}-\varphi_{4-10}-\varphi_{4-11}$	$\frac{A_6}{A_4}\varphi_{6-4}$	$\frac{A_7}{A_4}\varphi_{7-4}$	$\frac{A_8}{A_4}\varphi_{8-4}$	$\frac{A_9}{A_4}\varphi_{9-4}$	$\frac{A_{10}}{A_4}\varphi_{10-4}$	$\frac{A_{11}}{A_4}\varphi_{11-4}$
φ_5	$\frac{A_1}{A_5}\varphi_{1-5}$	$\frac{A_2}{A_5}\varphi_{2-5}$	$\frac{A_3}{A_5}\varphi_{3-5}$	$\frac{A_4}{A_5}\varphi_{4-5}$	0	0	0	$\frac{A_8}{A_5}\varphi_{8-5}$	$\frac{A_9}{A_5}\varphi_{9-5}$	$\frac{A_{10}}{A_5}\varphi_{10-5}$	$\frac{A_{11}}{A_5}\varphi_{11-5}$
φ_6	2A	2A	$\frac{A_3}{A_6}\varphi_{3-6}$	$\varphi_{6-1+2+3+4}-\varphi_{6-1}-\varphi_{6-2}-\varphi_{6-3}$	0	0	0	2A	1A	2A	2A
φ_7	2B	2B	$\frac{A_3}{A_7}\varphi_{3-7}$	$\varphi_{7-1+2+3+4}-\varphi_{7-1}-\varphi_{7-2}-\varphi_{7-3}$	0	0	0	2A	1A	2A	2A
φ_8	$\frac{A_1}{A_8}\varphi_{1-8}$	φ_{8-1}	$\frac{A_3}{A_8}\varphi_{3-8}$	$\varphi_{8-1+2+3+4}-\varphi_{8-1}-\varphi_{8-2}-\varphi_{8-3}$	$\varphi_{8-9}-\varphi_{8-6}-\varphi_{8-7}$	$\frac{A_6}{A_8}\varphi_{6-8}$	$\frac{A_7}{A_8}\varphi_{7-8}$	0	2A	2A	φ_{8-10}
φ_9	$\frac{A_1}{A_9}\varphi_{1-9}$	$\frac{A_2}{A_9}\varphi_{2-9}$	$\frac{A_3}{A_9}\varphi_{3-9}$	$\varphi_{9-8}-\varphi_{9-1}-\varphi_{9-2}-\varphi_{9-3}$	$\varphi_{9-5+6+7}-\varphi_{9-6}-\varphi_{9-7}$	$\frac{A_6}{A_9}\varphi_{6-9}$	$\frac{A_7}{A_9}\varphi_{7-9}$	$\frac{A_8}{A_9}\varphi_{8-9}$	0	2A	φ_{9-10}
φ_{10}	$\frac{A_1}{A_{10}}\varphi_{1-10}$	φ_{10-1}	$\frac{A_3}{A_{10}}\varphi_{3-10}$	$\varphi_{10-8}-\varphi_{10-1}-\varphi_{10-2}-\varphi_{10-3}$	$\varphi_{10-9}-\varphi_{10-6}-\varphi_{10-7}$	$\frac{A_6}{A_{10}}\varphi_{6-10}$	$\frac{A_7}{A_{10}}\varphi_{7-10}$	$\frac{A_8}{A_{10}}\varphi_{8-10}$	$\frac{A_9}{A_{10}}\varphi_{9-10}$	0	1A
φ_{11}	$\frac{A_1}{A_{11}}\varphi_{1-11}$	φ_{11-1}	$\frac{A_3}{A_{11}}\varphi_{3-11}$	$\varphi_{11-8}-\varphi_{11-1}-\varphi_{11-2}-\varphi_{11-3}$	$\varphi_{11-9}-\varphi_{11-6}-\varphi_{11-7}$	$\frac{A_6}{A_{11}}\varphi_{6-11}$	$\frac{A_7}{A_{11}}\varphi_{7-11}$	φ_{10-8}	φ_{10-9}	φ_{10-11}	0

This equation applies, because surface 5 + surface 6 + surface 7 together are equal in size to surface 9 and have the same position in relation to surface 1 as surface 9 has in relation to surface 2. In addition, there is no need to calculate $\varphi_{1 \to 6}$ and $\varphi_{1 \to 7}$ by means of the computer as they can be obtained from $\varphi_{6 \to 1}$ and $\varphi_{7 \to 1}$ with the help of equation (5). To calculate $\varphi_{4 \to 5}$ the rule, that the sum of the angle factors for a surface is equal to 1, has been used.

According to table 1.1 it is necessary to calculate 31 angle factors out of the 92 by means of the computer. In a table of data, which is provided for the computer operator, the cases as in fig. 1.5 are quoted and the dimensions of x and y, which determine the size and position of the surfaces and their positions in relation to each other, are given. With the parallel surfaces the distances between the surfaces, h, are also to be given. Only cases 1A, 2A and 2B are dealt with in this example.

All calculated angle factors are given in table 1.2. The values can be checked by using the rule, that the sum of all the angle factors of a surface is equal to 1 (equ. (6)). This check cannot be used for surface 4, as the rule has already been used for the calculation of angle factor $\varphi_{4 \to 5}$.

Table 1.2. Angle factors for radiation between surfaces in a room according to fig. 1.6.

	1	2	3	4	5	6	7	8	9	10	11
φ_1	0	0	0	0	0,124	0,004	0,004	0,101	0,318	0,263	0,186
φ_2	0	0	0	0	0,314	0,002	0,002	0,101	0,132	0.263	0,186
φ_3	0	0	0	0	0,195	0,004	0,004	0,099	0,203	0,142	0,353
φ_4	0	0	0	0	0,205	0,003	0,003	0,096	0,210	0,254	0,229
φ_5	0,018	0,046	0,027	0,061	0	0	0	0,124	0,237	0,251	0,236
φ_6	0,011	0,006	0,010	0.014	0	0	0	0,293	0,243	0.217	0,206
φ_7	0,010	0,005	0.009	0,015	0	0	0	0,244	0.220	0,126	0.371
φ_8	0,020	0,020	0,019	0,039	0,170	0,023	0,021	0	0,214	0.237	0,237
φ_9	0,042	0,017	0,025	0,056	0,211	0,013	0,012	0,140	0	0.242	0.242
φ_{10}	0.031	0,031	0,016	0,061	0,202	0,010	0,006	0,139	0,218	0	0.286
φ_{11}	0.022	0,022	0,040	0,055	0.190	0.010	0.018	0,139	0.218	0.286	0

Equations for the calculation of the absorption factors

Let us assume that the radiation from a surface with an area of A_i m^2 is equal $P_i A_i$ W to. Radiation consists of both long wave radiation emitted from the surface due to its emissive power, and of short and long wave radiation, reflected from the surface.

The fraction of the radiation $P_{i\lambda} A_i$ which has a wavelength of λ within a range of wavelengths $\Delta\lambda$, is considered. A surface A_j absorbs $\Psi_{ij\lambda} P_{i\lambda} A_i$ of this, if $\Psi_{ij\lambda}$ is the absorption factor for this monochromatic radiation from surface A_i to surface A_j. Itself, surface A_j gives off radiation $P_{j\lambda} A_j$ with wavelength λ. The net radiation with this wavelength from A_j is therefore

$$q_{j\lambda} = P_{j\lambda} A_j - \sum_{i=1}^{n} \psi_{ij\lambda} P_{i\lambda} A_i \qquad (16)$$

for a room with n surfaces.

An expression for $\Psi_{ij\lambda}$ can be derived in the following way. A fraction $\varphi_{ip} P_{i\lambda} A_i$ of the radiation from A_i with wavelength λ impinges upon (note: not absorbs) a third surface A_p. If $r_{p\lambda}$ is the reflectivity of A_p for radiation with wavelength λ, then $\varphi_{ip} r_{p\lambda} P_{i\lambda} A_i$ is reflected. The percentage of this reflected radiation, which is absorbed by A_j, is of the same magnitude as the fraction of $P_{p\lambda} A_p$ which is absorbed at A_j, i.e. $\Psi_{pj\lambda}$. For this reason the radiation, which comes from A_i, and which is absorbed by A_j after reflection at all surfaces in the room, can be expressed by:

$$\sum_{p=1}^{n} \psi_{pj\lambda} \varphi_{ip} r_{p\lambda} P_{i\lambda} A_i$$

The fraction $\varphi_{ij} a_{j\lambda} P_{i\lambda} A_i$ of $P_{i\lambda} A_i$ is directly absorbed by A_j, if $a_{j\lambda}$ is the absorption factor of surface A_j for radiation with wavelength λ. The total quantity absorbed is $\Psi_{ij\lambda} P_{i\lambda} A_i$, and therefore

$$\psi_{ij\lambda} = \sum_{p=1}^{n} \varphi_{ip} r_{p\lambda} \psi_{pj\lambda} + \varphi_{ij} a_{j\lambda} \tag{17}$$

is obtained.

If it is desired to calculate the fraction of radiation absorbed at surface A_j, originating from A_i, which has a wavelength of λ, in the case of a room with n surfaces, it is possible to use the equation system (17), which consists of n equations with n unknown absorption factors

$$\psi_{1j\lambda},\ \psi_{2j\lambda}, \ldots,\ \psi_{ij\lambda},\ \psi_{jj\lambda}, \ldots,\ \psi_{nj\lambda}$$

Angle factors φ_{i1}, φ_{i2}, ..., φ_{ij}, ..., φ_{in} (note that $\varphi_{jj} = 0$ while $\Psi_{jj} \neq 0$) are included in the equations. Also included are the reflection factors $r_{1\lambda}$, $r_{2\lambda}$, ..., $r_{i\lambda}$, $r_{j\lambda}$, ..., $r_{n\lambda}$ as well as the absorption factor $a_{j\lambda}$ (if A_j is an opaque surface, $a_j = 1 - r_j$).

$\Psi_{ij\lambda}$ does not depend upon the transfer of energy which takes place between surfaces due to radiation of wavelengths other than λ, nor is it affected by either convection or conduction heat transfer. It is therefore possible to study shortwave and longwave radiation in a room separately.

The reflectivities vary with the wavelength of the radiation. The composition of the radiation in a room varies with the source of radiation. The reflectivities of the surfaces are therefore not the same for low temperature radiation, i.e. the radiation which takes place due to the emissive power of the surfaces, as for radiation from illuminating lamps and from the sun.

With low temperature radiation the wavelength distribution varies so slightly, that one can reckon with a certain constant reflectivity for a surface which receives such radiation. The radiation of the sun has a composition, which varies with the height of the sun, and is not the same as the radiation from the sky. In general, the reflectivities of room surfaces are not known with any greater degree of accuracy at any particular wavelength. It is therefore in most cases justified to assume a constant reflectivity for a surface, even with shortwave radiation.

The calculation of the absorption factors for room surfaces is simplified in that the following equations apply for them (according to *Gebhart* [4]):

$$a_i A_i \psi_{ij} = a_j A_j \psi_{ji} \tag{18}$$

$$\sum_{j=1}^{n} \psi_{ij} = 1 \tag{19}$$

These equations correspond to equations (5) and (6) which apply to the angle factors.

Example 2. Absorption factors when surface reflectivities are equal.

Calculate the absorption factors for radiation between surfaces in a room according to fig. 1.7. It is assumed that all surfaces have the same reflectivity and that this varies from 0-1. Table 1.3 gives the angle factors for room surfaces (calculated by means of computer).

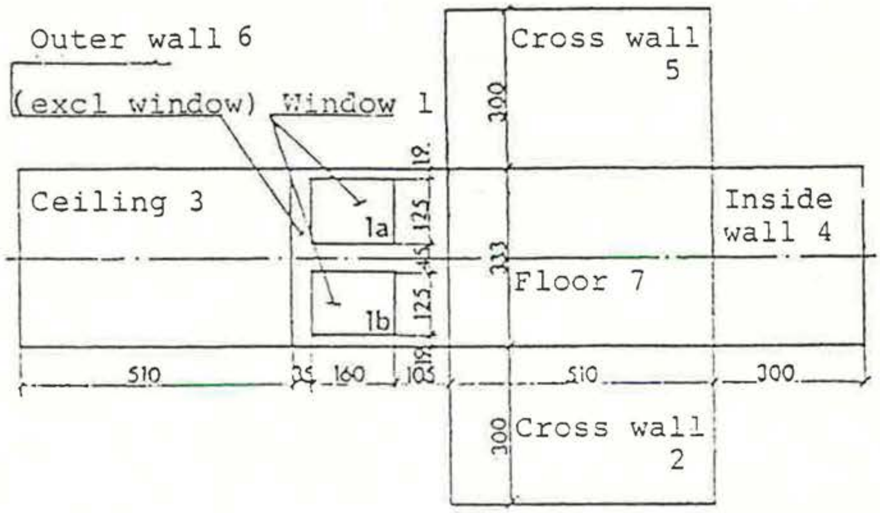

Fig. 1.7.

Table 1.3. Angle factors for radiation between surfaces in a room according to fig. 1.7.

	1	2	3	4	5	6	7
φ_1	0	0,225	0,263	0,100	0,225	0	0,187
φ_2	0,059	0	0,242	0,140	0,236	0,081	0,242
φ_3	0,062	0,218	0	0,139	0,218	0,077	0,286
φ_4	0,040	0,214	0,237	0	0,214	0,058	0,237
φ_5	0,059	0,236	0,242	0,140	0	0,081	0,242
φ_6	0	0,207	0,219	0,097	0,207	0	0,270
φ_7	0,044	0,218	0,286	0,139	0,218	0,095	0

For each of the seven room surfaces an equation system was established, which consists of seven equations which correspond with equation (17). For example, for radiation against the floor it is possible to obtain the absorption factors from

$$\psi_{17} = \varphi_{12} r \psi_{27} + \varphi_{13} r \psi_{37} + \varphi_{14} r \psi_{47} + \varphi_{15} r \psi_{57} + \varphi_{16} r \psi_{67} + \\ + \varphi_{17} r \psi_{77} + \varphi_{17} (1-r)$$

$$\psi_{27} = \varphi_{21} r \psi_{17} + \varphi_{23} r \psi_{37} + \varphi_{24} r \psi_{47} + \varphi_{25} r \psi_{57} + \varphi_{26} r \psi_{67} + \\ + \varphi_{27} r \psi_{77} + \varphi_{27} (1-r)$$

$$\psi_{37} = \varphi_{31} r \psi_{17} + \varphi_{32} r \psi_{27} + \varphi_{34} r \psi_{47} + \varphi_{35} r \psi_{57} + \varphi_{36} r \psi_{67} + \\ + \varphi_{37} r \psi_{77} + \varphi_{37} (1-r)$$

$$\psi_{47} = \varphi_{41} r \psi_{17} + \varphi_{42} r \psi_{27} + \varphi_{43} r \psi_{37} + \varphi_{45} r \psi_{57} + \varphi_{46} r \psi_{67} + \\ + \varphi_{47} r \psi_{77} + \varphi_{47} (1-r)$$

$$\psi_{57} = \varphi_{51} r \psi_{17} + \varphi_{52} r \psi_{27} + \varphi_{53} r \psi_{37} + \varphi_{54} r \psi_{47} + \varphi_{56} r \psi_{67} + \\ + \varphi_{57} r \psi_{77} + \varphi_{57} (1-r)$$

$$\psi_{67} = \varphi_{61} r \psi_{17} + \varphi_{62} r \psi_{27} + \varphi_{63} r \psi_{37} + \varphi_{64} r \psi_{47} + \varphi_{65} r \psi_{57} + \\ + \varphi_{67} r \psi_{77} + \varphi_{67} (1-r)$$

$$\psi_{77} = \varphi_{71} r \psi_{17} + \varphi_{72} r \psi_{27} + \varphi_{73} r \psi_{37} + \varphi_{74} r \psi_{47} + \varphi_{75} r \psi_{57} + \\ + \varphi_{76} r \psi_{67}$$

The equation systems can be solved easily with a computer using a standard programme.

Of special interest is the case where there is only long wave radiation, and where the room surfaces are made from non-metallic materials. Reflection with such surfaces is slight. In general one can estimate r = 0,07. The absorption factors calculated by the computer with this reflectivity are given in table 1.4.

Table 1.4. Absorption factors for radiation between surfaces in a room according to fig. 1.7, when r = 0,07.

	1	2	3	4	5	6	7
φ_1	0.004	0,221	0,258	0,102	0,221	0,006	0,188
φ_2	0,058	0,015	0,239	0,138	0,231	0,080	0,239
φ_3	0.061	0.215	0,018	0,137	0.215	0,076	0,278
φ_4	0.041	0,211	0,234	0,010	0,211	0,059	0,234
φ_5	0.058	0,231	0,239	0,138	0,015	0,080	0,239
φ_6	0.004	0,205	0,217	0,099	0.205	0,006	0,264
φ_7	0.044	0,215	0,278	0,137	0,215	0,093	0,018

The values in table 1.4 do not differ appreciably from the values in table 1.3. The biggest difference is found for ceiling and floor. In both cases these surfaces absorb 1,8 % of the radiation, which they themselves send out, due to reflection at all room surfaces. In addition, it can be seen that while the angle factor for radiation from the ceiling to the floor is 28,6 %, the corresponding absorption factor is 27,8 %, i.e. 100 · (28,6 - 27,8)/28,6 = 2,8 % less.

As this is the largest difference which occurs, it is obvious that *in engineering calculations it is possible to calculate with angle factors instead of absorption factors in a room, when the reflection at the surfaces is slight.* This is the normal case when only longwave radiation occurs.

The absorption factors for all room surfaces are also calculated for reflectivities of 0,25, 0,50, 0,70, 0,85 and 0,95. Some of the results of the calculations are given in figures 1.8 and 1.9.

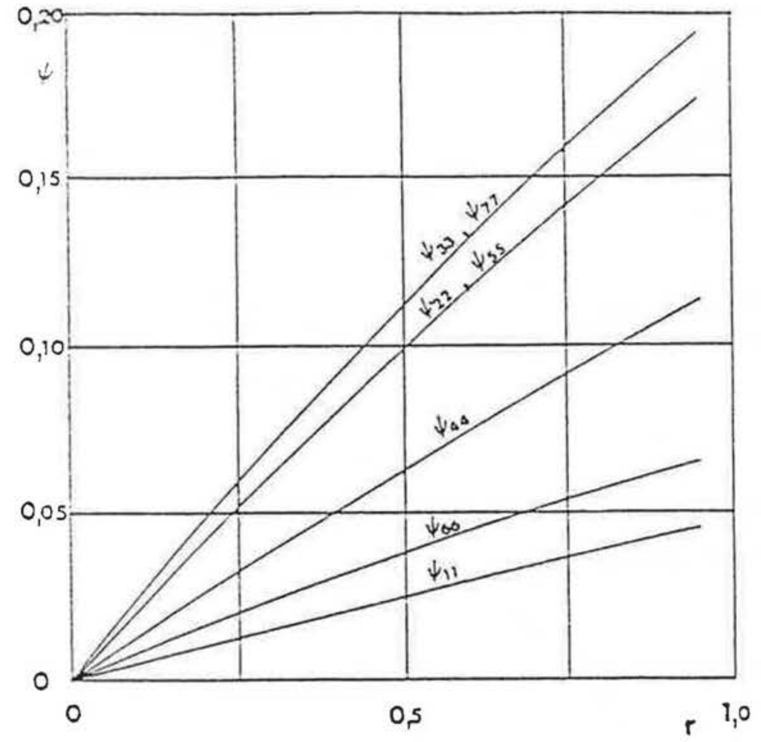

Fig. 1.8. Absorption factors for radiation from a surface in a room according to fig. 1.7 back to the same surface with varying reflectivity of the room surfaces.

Fig. 1.8 shows what fraction of the radiation from a room surface returns to it after reflecting from all room surfaces, with different values of *r*. Naturally, the return reflections from the largest surfaces, the ceiling and the floor, are the biggest items. The return reflections from windows are the least, and it is possible to draw the conclusion from the values of Ψ_{11}, that *only a very small percentage of the solar and sky radiation through the windows is reflected out again.*

Fig. 1.9 shows what quantity of the radiation from the ceiling surface is absorbed by the different room surfaces, with different values of *r*. The values are only slightly related to *r*, if one disregards the cases of radiation against the floor and the radiation back against the ceiling (Ψ_{37} and Ψ_{33}).

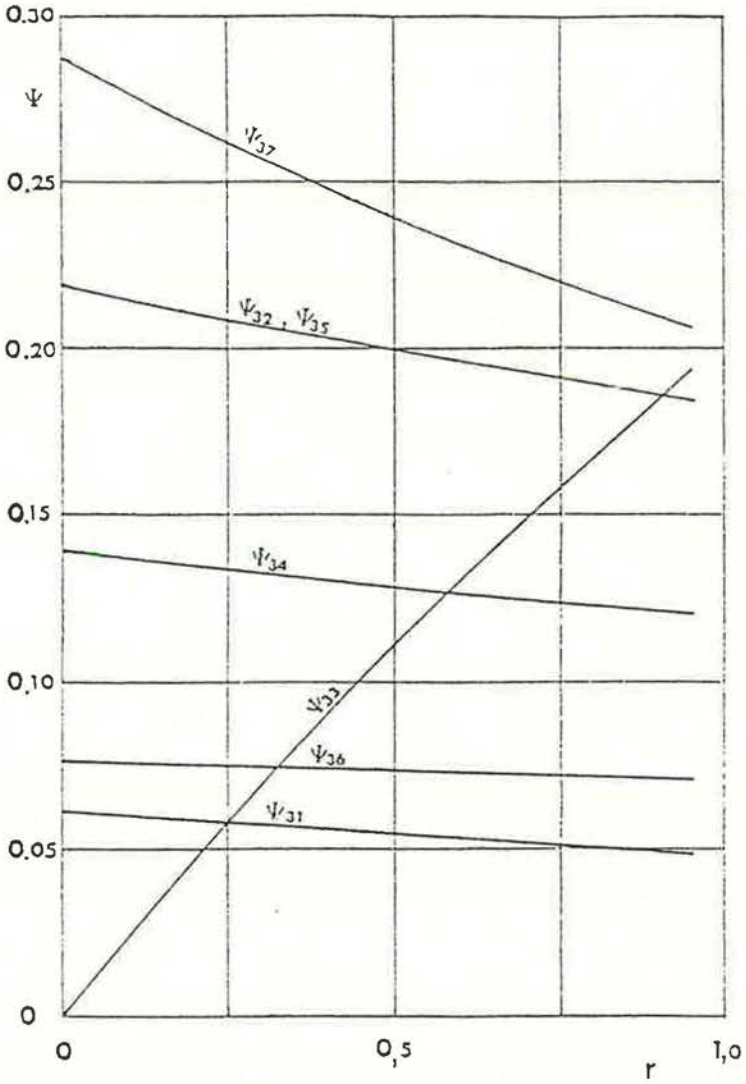

Fig. 1.9. Absorption factors for radiation from ceiling in a room according to fig. 1.7 against various room surfaces, with varying values of the reflectivity of the room surfaces.

The curves in fig. 1.9 deviate only slightly from straight lines. This was also shown to be the case when corresponding curves were drawn from the other room surfaces. By extrapolating the curves it is found that Ψ_{ij} approaches value $A_j/\Sigma A$ when r approaches 1 (ΣA is the total area of the room surfaces). In a room where all the surfaces have the same reflectivity one obtains therefore the following approximate equation for the calculation of the absorption factor for radiation from surface A_i against surface A_j:

$$\psi_{ij} \approx (1 - r)\,\varphi_{ij} + \frac{A_j}{\Sigma A}\, r \qquad (20)$$

The largest deviation from a straight line was shown by the graph for Ψ_{33} (and Ψ_{77}), see fig. 1.8. When r = 0,5, the error is about 10 % when one applies equation (20).

Example 3. Variation of daylight intensity in a room.

Give the variation of daylight with the distance from a window-wall in a room according to fig. 1.7. The light distribution is diffuse and the reflectivities for visible radiation are the following for room surfaces 1 to 7 inclusive: r_1 = 0,20, $r_2 = r_4 = r_5 = r_6$ = 0,60, r_3 = 0,75, r_7 = 0,25. The variation is studied in such a way, that the absorption factors are calculated for radiation from window surfaces against horizontal, upwards facing, black surfaces. These are 1 cm^2 in area and placed 80 cms above the floor, (the height of a writing desk) in points a - l, distributed round the room according to fig. 1.10.

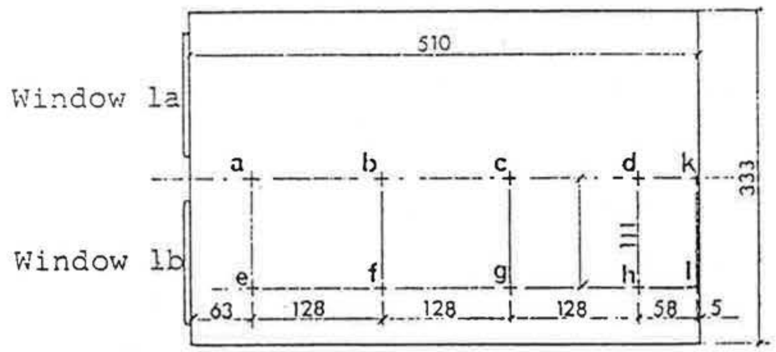

Fig. 1.10. Location of surface elements for study of variation of daylight intensity in a room as shown in fig. 1.7.

The angle factors for radiation from the surface elements against room surfaces 1a, 1b, 2, 3, 4, 5, 6 and 7 are first calculated by means of the computer. The angle factors for radiation from the room surfaces against the surface elements are then calculated with the help of equation (5). The angle factors at the windows were proviously calculated by equation:

$$\varphi_{y \to 1} = \varphi_{y \to 1a} + \varphi_{y \to 1b}$$

so that it is possible subsequently to treat both windows as one surface (index $_y$ indicates one of the surface elements a - l). The calculation gives the values in table 1.5.

Table 1.5 Angle factors for radiation between surfaces in a room according to fig. 1.7 and the surface elements a - l which are positioned in the room according to fig 1.10.

	a	b	c	d	e	f	g	h	k	l
φ_1	$4.698 \cdot 10^{-6}$	$1.827 \cdot 10^{-6}$	$0,670 \cdot 10^{-6}$	$0,296 \cdot 10^{-6}$	$4,810 \cdot 10^{-6}$	$1,535 \cdot 10^{-6}$	$0.589 \cdot 10^{-6}$	$0,271 \cdot 10^{-6}$	$0.225 \cdot 10^{-6}$	$0,200 \cdot 10^{-6}$
φ_2	0.860	1,103	1,103	0,860	2,039	2.374	2,374	2.039	0,650	1,316
φ_3	2,272	2,919	2,919	2,272	1,915	2,453	2,453	1,915	1,764	1,495
φ_4	0,214	0,468	1,200	3,301	0.917	0,413	1,001	2,807	4,862	4,792
φ_5	0.860	1,103	1,103	0,860	0,411	0,515	0,515	0,411	0,650	0,337
φ_6	2,368	0,782	0,334	0,160	1,469	0,645	0,296	0,147	0,112	0,109
φ_7	0	0	0	0	0	0	0	0	0	0

To determine the absorption factor Ψ_{1y} for radiation from the windows against a surface element y, the following equation system (according to (17)) is then solved:

$$\psi_{1y} = \varphi_{12} r_2 \psi_{2y} + \varphi_{13} r_3 \psi_{3y} + \varphi_{14} r_4 \psi_{4y} + \ldots + \varphi_{17} r_7 \psi_{7y} + $$
$$+ \varphi_{1y} (1 - r_y)$$
$$\psi_{2y} = \varphi_{21} r_1 \psi_{1y} + \varphi_{23} r_3 \psi_{3y} + \varphi_{24} r_4 \psi_{4y} + \ldots + \varphi_{27} r_7 \psi_{7y} +$$
$$+ \varphi_{2y} (1 - r_y)$$
$$\psi_{3y} = \varphi_{31} r_1 \psi_{1y} + \varphi_{32} r_2 \psi_{2y} + \varphi_{34} r_4 \psi_{4y} + \ldots + \varphi_{37} r_7 \psi_{7y} +$$
$$+ \varphi_{3y} (1 - r_y)$$

. . . . .

$$\psi_{7y} = \varphi_{71} r_1 \psi_{1y} + \varphi_{72} r_2 \psi_{2y} + \varphi_{73} r_3 \psi_{3y} + \ldots + \varphi_{7y} (1 - r_y)$$

In this case r_y constitutes the reflectivity of the surface element. When $r_y = 0$, the absorption factors given in table 1.6 are obtained. To check the values of the absorption factors, all absorption factors for radiation from the surface element against the room surfaces are calculated for each surface element with the help of equ. (18) and summated. The sum was 1, which corresponded with equation (19).

Table 1.6. Absorption factors for radiation between surfaces in a room according to fig. 1.7 and surface elements a - l which are positioned in the room according to fig. 1.10.

	a	b	c	d	e	f	g	h	k	l
ψ_1	$6,171 \cdot 10^{-6}$	$3,572 \cdot 10^{-6}$	$2,484 \cdot 10^{-6}$	$2,047 \cdot 10^{-6}$	$6,298 \cdot 10^{-6}$	$3,274 \cdot 10^{-6}$	$2,384 \cdot 10^{-6}$	$2,014 \cdot 10^{-6}$	$1,914 \cdot 10^{-6}$	$1,874 \cdot 10^{-6}$
ψ_2	2,302	2,674	2,723	2,487	3,318	3,781	3,821	3,508	2,276	2,849
ψ_3	3,388	4,086	4,132	3,566	3,066	3,680	3,717	3,247	3,112	2.872
ψ_4	1,702	2,115	2,347	4,773	1,671	2,054	2,642	4,298	6,192	6,114
ψ_5	2,302	2,674	2,723	2,487	1,892	2,153	2,192	2,082	2,276	1,991
ψ_6	3,713	2,370	1,988	1,771	2,831	2,233	1,938	1,754	1,677	1,664
ψ_7	1,737	1,919	1,971	1,933	1,695	1,893	1,936	1,904	1,892	1.870

Initially only the absorption factors for radiation against surface elements *a* to *h* inclusive were calculated, but the calculation was then completed, so that the absorption factors for radiation against surface elements *k* and *l* were also obtained. In this way it was possible to observe, whether reflection against inner wall 4 should cause an increase of the values adjacent to that wall. The values in table 1.6 show, that this is not the case up to a distance of 5 cm from the wall.

Absorption factors and angle factors for radiation from the windows against surface elements *a* - *l* are introduced in diagram fig. 1.11. With the help of the plotted points in the diagram, curves were drawn afterwards, which show the variation of the factors with the distance from the window wall.

If the intensity of radiation which is emitted at the inside surfaces of the windows is I, then it is $10^4 \cdot I_{\Psi_{1y}}$ at the surface element, as Ψ_{1y} is the fraction of radiation from the window surface, which is absorbed by the black surface element y of size 1 cm^2 and intensity is reckoned per m^2. The strength of illumination on a surface is the flow of light (radiation flow) which impinges upon 1 m^2. The curves relating to the absorption factors in fig. 1.11 give therefore the percentage ratio between the intensity of illumination upon a horizontal surface 80 cm above the floor, and the intensity of illumination upon a vertical surface, just inside the window, after multiplying by 10^6. The curves of the angle factors give the corresponding value in a black room.

This quotient falls from 6 % about 0,7 m from the window, to just under 2 % at the internal wall, when the walls have the reflectivities mentioned previously. In the case of black room surfaces, it falls from 4,5 % to 0,2 % or slightly more. The difference between the values in the light and black room, respectively, at the same distance from the window is nearly constant at about 1,75 %, when the distance is more than 1,5 m.

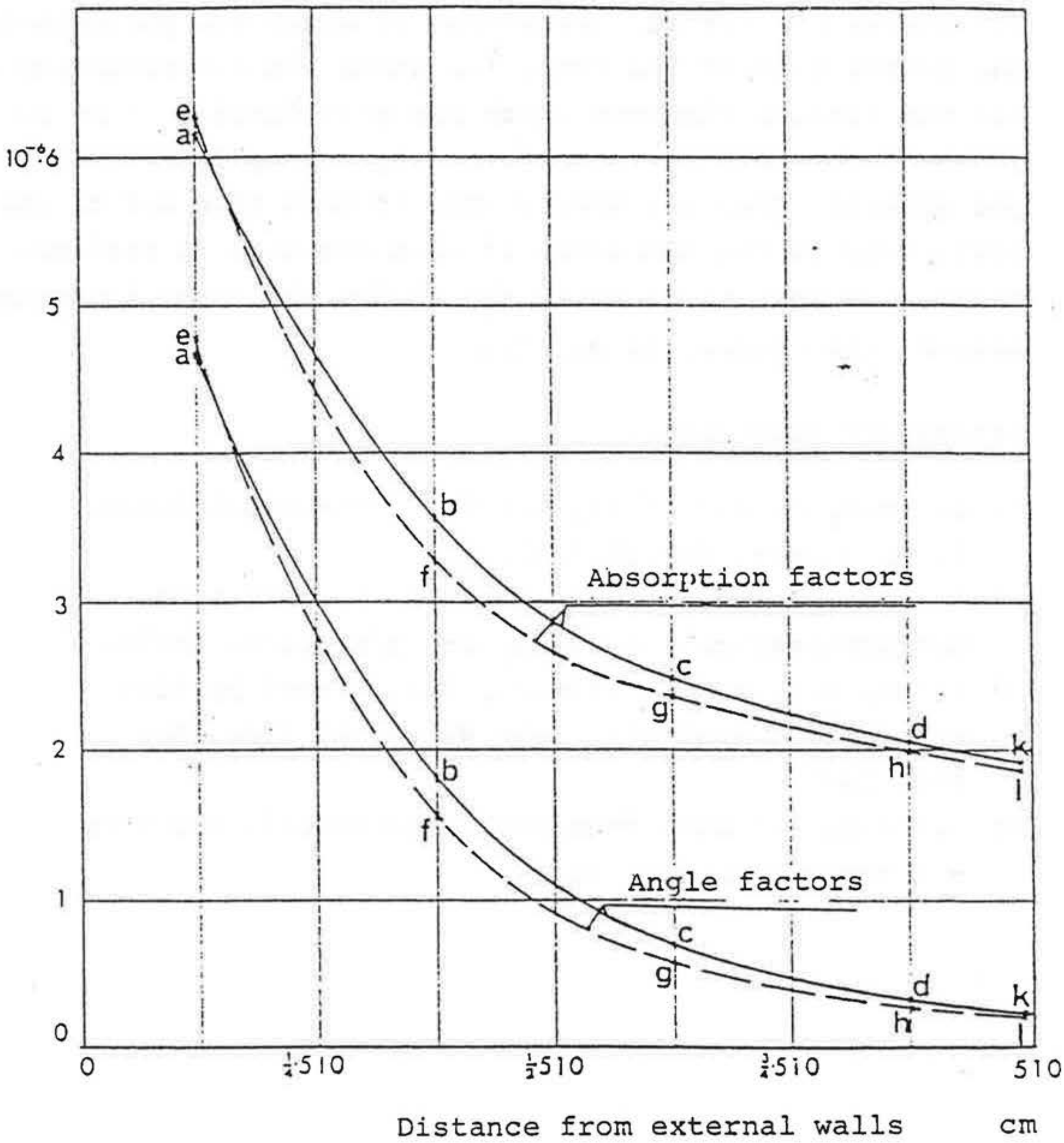

Fig. 1.11. Absorption and angle factors for radiation from windows in a room as shown in fig. 1.7 to black surface elements of 1 sq.cm area placed horizontally face up at a height of 80 cm above floor level along the centre-line of the room (continuous lines) and at a distance of 111 cm from the centre-line (broken lines).
The upper curves also show the intensity of the light falling on the elements. This, expressed as a percentage of the intensity of illumination of a surface element placed vertically immediately inside the window, amounts in fact to 10^6 times the absorption factor. The lower curves indicate similarly the intensity of illumination in a black room.

The curves for the surface elements, which are positioned at the centre line of the room, lie above the corresponding curves for surface elements which are positioned a third of the width of the room to the side of the centre line, as long as the distance from the window wall is more than 0,7 m. The relationship is the opposite, if this distance is smaller. The reason for this is probably the shadow thrown by the frame between the windows 1a and 1b.

Literature references:

1) *Kollmar, A.* and *Liese, W.:* Die Strahlungsheizung. R. Oldenburg, Munich 1957.
2) *Squassi, F.:* Die Einstrahlzahlen in Wohnräumen. Gesundheitsingenieur 1957, Nr. 5/6, pages 69-72.
3) *Raber, B.F.* and *Hutchinson, F.W.:* Panel Heating and Cooling Analysis. John Wiley and Sons. New York 1947.
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2 CALCULATION OF HEATING AND COOLING LOADS

INTRODUCTION

To find the cooling requirements, when the dimensions of an air conditioning plant are to be determined, it is necessary to allow for the considerable variation in loading which takes place during a 24-hourly period. As regards an office building , for example, not only do the external temperature vary. Solar radiation may cause large quantities of heat to enter into the room during working hours, especially through the windows. The heating effect from lights and from people inside the offices at any given time, must also be introduced into the calculations. The building shell (walls and flooring) is heated, when the air temperature inside the room rises. As it falls afterwards, the shell gives off heat. The result is a delay and reduction of the oscillation of the room temperature.

The necessary data for the calculations are set out in tables 2.1 and 2.2. The quantities given in table 2.1 are considered to vary with the time of the day and should therefore be given as a function of time. Table 2.2 contains quantities which can, in general, be considered unaffected by time. The calculations are made, however, slightly more complicated if one assumes that, for example α_y, a_y or m_f in table 2.2 have different given values at different times of the day.

This chapter is intended to give a calculation technique, which makes it possible to investigate theorectically, how the various factors in table 2.1 affect the cooling requirements with different shell and window constructions.

When solving the equations, the values in this table are unaffected variables, when θ_r, the temperature of the air in the room, is to be determined. There is no reason why one cannot, however, calculate the cooling effect H instead, when θ_r is given for different times during the day. It is also possible to calculate θ_i or G_i.

The method of calculation can be used for determining both heating and cooling requirements. Although it is most important to consider temperature variations in a building when calculating the dimensions of a cooling system, such variations are also quite valuable to know, when determining the dimensions of a heating system, for instance if one wishes to study the use one can make of solar radiation and of artificial illumination as a source of heat.

Table 2.1. Quantities dependent upon time

Quantity	Symbol	Units
Outside air temperature	θ_l	^oC
Equivalent outside air temperature	θ_e	^oC
Solar and sky radiation against a facade	I	W/m^2
Solar and sky radiation (short wave radiation) transmitted directly through a window	I_T	W/m^2
Solar and sky radiation energy absorbed at the inner surface of a window	I_V	W/m^2
Solar and sky radiation, transmitted directly through a window other than the one considered	I'_T	W/m^2
Radiation from a light source spread over a surface (for example a ceiling surface)	I_{be}	W/m^2
Radiation from a point source light supply	B	W
Temperature of ventilation air	θ_i	^oC
Ventilation air flow rate	G_i	kg/h
Flow rate of air supplied to the room at outside air temperature	G_l	kg/h
Heat transferred directly to the room air from heater, lights, human beings etc.	H_c	W
Thermal output of heaters or coolers in the room	H	W

Table 2.2. Quantities independent of time

Quantity	Symbol	Units
Time interval	Δt	h
Thickness of layer	Δx	m
Area	A	m^2
Volume	V	m^3
Thermal diffusivity	a	m^2/s
Thermal conductivity	λ	W/m^oC
Density, weight per unit volume	ρ	kg/m^3
Thermal resistance of air layer	m_l	$m^2\ ^oC/W$
Thermal resistance of window including the external but excluding the internal surface resistance	m_f	$m^2\ ^oC/W$
Absorptivity of facade surface for solar and sky radiation	a_y	dimensionless
Surface heat transfer coefficient at the outside of an external wall, including the effect of long wave radiation	α_y	$W/m^2\ ^oC$
Angle factor for radiation between two surfaces in a room	φ	dimensionless
Absorption factor for radiation between two surfaces in a room	Ψ	dimensionless
Emissivity	ε	dimensionless

HEAT TRANSFER UNDER STEADY-STATE TEMPERATURE CONDITIONS

Thermal conduction in the interior of a wall or flooring

Let us assume that the heat flow is one-dimensional and takes place through a homogeneous material in a direction which is perpendicular to the wall surface. The material can be considered as being divided into layers parallel with the wall surface according to fig. 2.1. The temperatures θ_{n-1}, θ_n and θ_{n+1} are the average temperatures which exist in each specific layer.

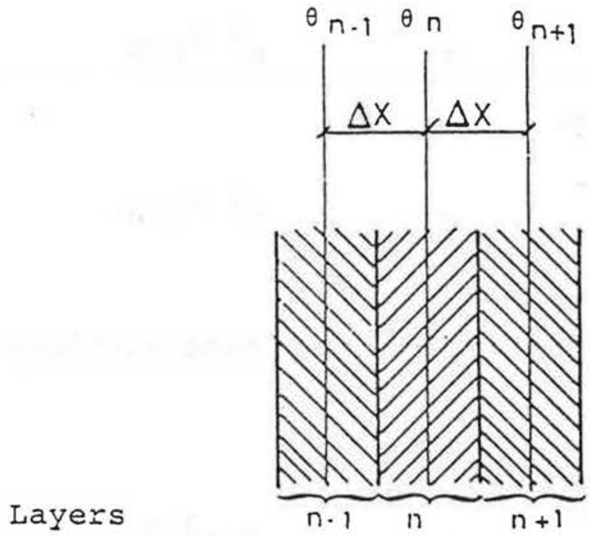

Fig. 2.1.

If $\theta_{n-1} > \theta_n$, then the following amount of heat is transferred to layer n between layers n - 1 and n, per m^2 of wall surface and s:

$$\frac{\lambda}{\Delta x}\left(\theta_{n-1} - \theta_n\right)$$

where λ is the thermal conductivity of the material. Between layers n and n + 1 the following quantity of heat is led from layer n:

$$\frac{\lambda}{\Delta x}\left(\theta_n - \theta_{n+1}\right)$$

As the temperatures are constant, layer n is neither heated nor cooled, and the heat transferred to the layer is always equal to that which is transferred from it.

Heat transfer with respect to facade and room surfaces

Facade surfaces

When the sun shines upon an external wall, it is useful to calculate with a fictional or "equivalent" outside air temperature θ_e, which includes a term representing the effect of solar and sky radiation.x)

To calculate θ_e, one equates the two expressions of how much heat is being supplied to the wall surface per m^2 and s:

$$\alpha_y (\theta_e - \theta_y) = \alpha_y (\theta_l - \theta_y) + a_y I \qquad (1)$$

In this case α_y stands for the surface heat transfer coefficient at the wall surface (including the effect of long wave radiation), θ_y is the temperature of the wall surface, θ_l is the outside air temperature, I is the intensity of solar and sky radiation in W/m^2 and a_y is the absorptivity of the surface for this radiation. From this the following is obtained:

$$\theta_e = \theta_l + \frac{a_y I}{\alpha_y} \qquad (2)$$

When the values of α_y and a_y are known, it is possible to calculate θ_e as a function of θ_l and I by means of the computer together with the remaining computer operations.

Room surfaces (excluding windows)

When a section of a wall surface in a room is considered, heat is transferred (fig. 2.2) from the surface by conduction into the wall, to the surface by convection from the room air, and to and from the surface by means of radiation. The illustration shows a case where the air is warmer than the surface and the surface is warmer than the material in the body of the wall.

x) θ_e is sometimes called sol-air temperature.

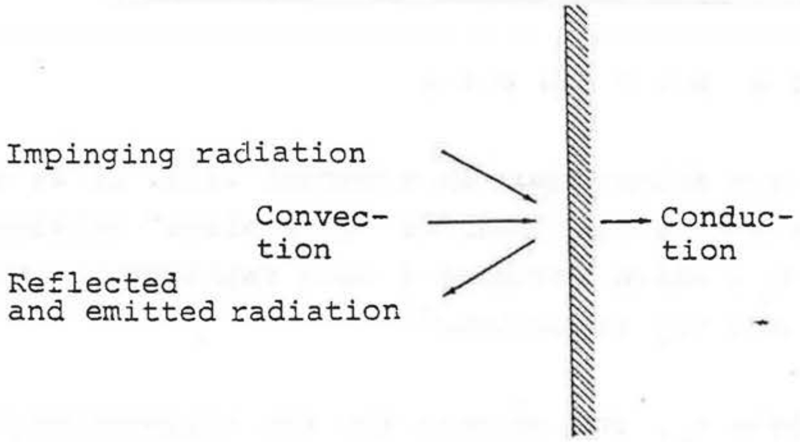

Fig. 2.2.

Heat radiation at room surfaces

In order to solve the complicated mathematical problems which arise during the calculation of radiation between surfaces in a closed room, the term "absorption factor" has been introduced. The absorption factor Ψ_{ab} defines the fraction of radiation from a surface A_a which is absorbed by a surface A_b. Included is not only the fraction of radiation from A_a, which radiates directly upon A_b and is absorbed there, but also the energy of radiation, which reaches A_b and is absorbed by it after being reflected by all the reflecting surfaces in the room.

Ψ_{ab} is dependent upon the absorptivities of all the surfaces within the room as well as their sizes, shapes and positions in relation to each other (geometry of room).

If all surfaces are black ($\varepsilon = 1$), so that they absorb all incident radiation, the fraction of radiation from a surface A_a which is absorbed by a surface A_b is determined only by the size, shape and position of the two surfaces. This must be the case, as no part of the radiation is reflected by any other surface in the room, prior to impinging on the receiving surface. In such a case one refers to the angle factor φ_{ab} instead of the absorption factor Ψ_{ab}.

In chapter 1 methods are related for the calculation of absorption factors and angle factors for the cases of radiation between plane rectangular surfaces in a room, where the surfaces do not shield each other, and are either parallel or perpendicular to each other. Diffuse emission and reflection of all radiation was assumed in these cases. These same assumptions and methods are incorporated in the program for calculating heating and cooling loads.

Radiation from a surface can consist of both long wave radiation, emitted due to the emissive power of the surface, and of short and long wave radiation reflected at the surface. Long wave radiation is emitted from all the surfaces in the room, walls, floor and ceiling, as well as the inside surfaces of windows, radiators, lamps, furniture and human beings according to equation:

$$E = \varepsilon C_s \left(\frac{T}{100}\right)^4 \tag{3}$$

where

E = the emissive power of the surface in W/m^2
ε = the emissivity of the surface, dimensionless
$C_s = \sigma \cdot 10^8$, where σ = Stefan Boltzmann's constant, i.e. $5.67 \cdot 10^{-8}\ W/m^2\ K^4$
T = the absolute temperature of the surface in K

The inside of a window, (window pane or window drape) or the surface of a lamp also emits short wave radiation, when it is light outside or when the lamp is lit, as the case may be.

Long wave radiation:

In an example given in chapter 1 it is shown, that when long wave radiation between room surfaces is considered, sufficient accuracy is obtained if angle factors are used instead of absorption factors, i.e. if the reflection of the surfaces is ignored. The largest error in the example was found to be between the ceiling and the floor. In this case it was found that the angle factor was 2.8 % larger than the absorption factor, if the reflectivity of all surfaces is $r = 0.07$. Therefore the program uses angle factors instead of absorption factors for long wave radiation.

The net quantity of long wave radiation which a surface A_n receives from a surface A_u can then be calculated in the following way.

Let us assume that the surface A_u sends out a total of P_u W as long wave radiation. Of this, surface A_n absorbs $\varphi_{un} P_u$ W. Surface A_n gives off P_n W and A_u absorbs $\varphi_{nu} P_n$ W of this. The following relationship applies for angle factors;

$$\varphi_{un} A_u = \varphi_{nu} A_n$$

The net radiation between A_u and A_n is therefore;

$$P_{un} = \varphi_{un} P_u - \varphi_{nu} P_n = \varphi_{nu} A_n \left(\frac{P_u}{A_u} - \frac{P_n}{A_n}\right) \qquad (4)$$

As it is assumed that the reflection between all room surfaces is equal to zero,

$$\frac{P_u}{A_u} = E_u, \quad \frac{P_n}{A_n} = E_n,$$

where E_u and E_n, respectively, are calculated according to equation (3). If the emissivities of both surfaces = ε, then equation (4) gives;

$$P_{un} = 5.67 \ \varepsilon \varphi_{nu} A_n \left[\left(\frac{T_u}{100}\right)^4 - \left(\frac{T_n}{100}\right)^4\right]$$

If the quantity

$$f_{un} = \frac{\left(\frac{T_u}{100}\right)^4 - \left(\frac{T_n}{100}\right)^4}{\theta_u - \theta_n} \qquad (5)$$

is introduced, and it is observed that $f_{un} = f_{nu}$, one obtains as expression for the radiation energy, measured in W, which is transferred between surfaces A_u and A_n the following:

$$P_{un} = 5.67\,\varepsilon\,\varphi_{nu} f_{nu}\, A_n\,(\theta_u - \theta_n) \tag{6}$$

When surfaces are considered, the temperatures of which differ only slightly from the ordinary room temperatures, for example, ordinary wall and flooring surfaces, one can approximate by giving f_{nu} and ε constant values. The value of f_{nu} varies only slightly with the temperature:

$$\theta_u = 9°,\ \theta_n = 20° \quad : \quad f_{nu} = 0.95$$
$$\theta_u = 19°,\ \theta_n = 20° \quad : \quad f_{nu} = 1.00$$
$$\theta_u = 28°,\ \theta_n = 20° \quad : \quad f_{nu} = 1.05$$

It is possible to write $\varepsilon = 0.93$ for ordinary room surfaces (also glass). If one accepts that $f_{nu} = 1$ (not used in the program) one obtains

$$P_{un} = 5.3\,\varphi_{nu}\,A_n\,(\theta_u - \theta_n) \tag{6'}$$

Short wave radiation:

Room surfaces reflect short wave radiation to a much greater extent than long wave radiation, provided the surfaces are not very dark. The approximation would therefore be less accurate, if one should also substitute angle factors for absorption factors for short wave calculations. It is therefore assumed that one knows the reflectivities of the room surfaces for the short wave radiation which is transmitted directly through the windows and for the short wave radiation which is emitted from the lamps within the room. It is then possible to calculate the absorption factors for radiation from these sources against the room surfaces.

A surface of area A_n m^2 absorbs P_{fn} W of the solar and sky radiation I_T W/m^2, which is transmitted directly through a *window* with a surface area of A_f m^2, according to equation

$$P_{fn} = \psi_{fn}\,A_f\,I_T \tag{7}$$

One obtains different expressions for the fraction of short wave radiation from *illumination sources*, which is absorbed by surface A_n, whether one considers that the lamps are evenly distributed over a larger surface, for example over the whole ceiling surface, or if they are concentrated in certain positions.

In the first case one obtains

$$P_{be,n} = \psi_{be,n} A_{be} I_{be} \tag{8}$$

In this case I_{be} constitutes the radiation energy in W per square meter of the surface A_{be} on which the lamps are positioned, and $\Psi_{be,n}$ the absorption factor for radiation from this surface against A_n.

If, in the letter case, one has a source of illumination, which contributes B W of radiation energy in a point for which the absorption factor for radiation against surface A_n is equal to $\Psi_{B,n}$, then the radiation absorbed by surface A_n can be expressed by

$$P_{B,n} = \psi'_{B,n} B \tag{9}$$

The total net radiation, measured in W, which is absorbed by surface A_n at a certain instant is then, according to equations (6) - (9)

$$P_n = A_n \left[5.67\ \varepsilon \Sigma \varphi_{nu}\ f_{nu}\ (\theta_u - \theta_n) + \Phi_n\right] \tag{10}$$

where

$$\Phi_n = \frac{1}{A_n} (\Sigma\, \psi_{fn} A_f I_T + \Sigma\, \psi_{be,n} A_{be} I_{be} + \Sigma\, \psi_{B,n} B) \tag{11}$$

θ_u is the temperature of a surface (also window surface) in the room from which long wave radiation falls against the surface in question A_n. Φ_n is short wave radiation from windows and illumination, measured in W/m^2, which is absorbed by surface A_n. The sign Σ indicates that one must add the absorbed long wave net radiation from all surfaces A_u in the room, which are in radiation exchange with A_n, and the absorbed short wave radiation from all windows and sources of illumination, respectively.

Convection heat transfer at room surfaces

Heat transfer coefficients between a surface and air, due to natural convection can be calculated by means of equation

$$Nu = c(Gr\ Pr)^n$$

where Nu, Gr and Pr are *Nusselt's*, *Grashof's* and *Prandtl's* numbers and c and n are constants. The temperature changes in a room are slight enough to make it possible to consider the properties of the air to be constant. Than one obtains:

$$\alpha_k = c\,(\Delta\theta)^n\,L^{-m} \tag{12}$$

where

α_k = coefficient of heat transfer by convection, W/m^2 ^{o}C

$\Delta\theta$ = temperature difference between air and surface, ^{o}C

L = a characteristic length in m

c, m, n are constants.

Measurements from which the constants can be determined were carried out by *Min et al.* 1956 in USA /1/. These were carried out in a room with floor or ceiling heating. From the results one obtains the following specific correlations for rooms:

walls: $\alpha_k = 1.88\ \Delta\theta^{0.32}\ L^{-0.05}$ (13)

warm ceiling, cold floor: $\alpha_k = 0.20\ \Delta\theta^{0.25}\ L^{-0.25}$ (14)

cold ceiling, warm floor: $\alpha_k = 2.42\ \Delta\theta^{0.31}\ L^{-0.08}$ (15)

In equation (13) L = the height of the room. In equation (14) and (15) L = 4 times the area of the ceiling divided by its perimeter.

The measurements were carried out in a room without air exchange or other disturbances. In a usual room where these occur, the α_k values may be somewhat higher than those given by equations (13) - (15), especially when temperatur differences are small and particularly when there is warm ceiling or a cold floor. In such a case equation (14) gives very small values, which is to be expected, because the difference between the temperature of the surface and the air cannot cause any convection air currents.

The computer carries out a linear interpolation between values of α_k given in the following table. The values are calculated with the help of equations (13) - (15), where

$$\Delta\theta = |\theta_r - \theta| \tag{16}$$

with respect to windows and wall surfaces,

$$\Delta\theta = \theta_r - \theta \tag{17a}$$

with respect to ceiling surfaces, and

$$\Delta\theta = \theta - \theta_r \tag{17b}$$

for floor surfaces. θ_r is the temperature of the air within the room, and θ is the temperature of the surface.

$\Delta\theta$ ^oC	α_k kcal/m^{2}h ^oC x)	
	Windows, walls	Floor, ceiling
< 0	-	0.5
0	0.5	0.5
0.5	1.4	1.6
2	2.1	2.4
7	3.1	3.6
30	4.9	5.7

x) 1 kcal/m^{2}h ^oC = 1.163 W/m^2 ^oC

T h e i n t e r n a l s u r f a c e o f a w i n d o w

Heat gain through a window due to solar and sky radiation per m^2 of window surface and hour can be subdivided into two parts, I_T and I_V. Here, I_T is the radiation which is transmitted directly to the room. This fraction does not heat the glass panes (or shading devices). I_V is the fraction of radiant solar energy, which is first converted into heat within innermost glass pane (or possibly internal shading device). One part of I_V passes to the room by means of long wave radiation to the room surfaces and by means of convection to the room air, another transmits back through the window to its outside.

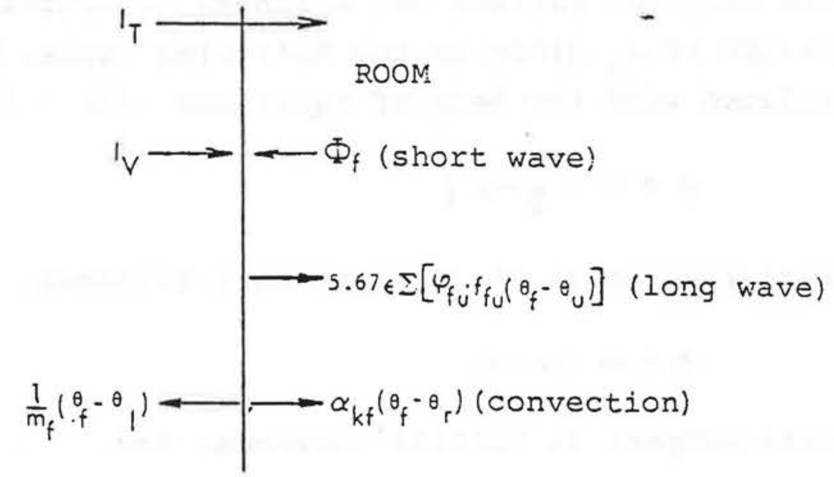

Fig. 2.3.

Fig. 2.3 shows the thermal exchange at the internal surface of the window. From the inside of the room, the window pane receives short wave radiation from the window pane itself, due to reflection at all room surfaces, from any other windows present, and from artificial lighting. Corresponding with equation (11), the window in question A_f absorbs the following quantity of this per m^2 apart from, generally, a small, transmitted, fraction:

$$\Phi_f = \frac{1}{A_f}(\psi_{ff} A_f I_T + \Sigma\psi'_{ff} A_f' I_T' + \Sigma\psi_{be,f} A_{be} I_{be} + \\ + \Sigma\psi_{B,f} B) \qquad (18)$$

ψ'_{ff} designates the absorption factor for radiation I_T back towards A_f. In addition, A_f' is the surface of an other window in the room, I_T' is the directly transmitted solar and sky radiation through this window, and ψ'_{ff} is the absorption factor of this radiation against A_f.

The long wave radiation from the window pane against a room surface A_u is equal to

$$5.67\ \varepsilon\ \varphi_{fu} f_{fu} (\theta_f - \theta_u)$$

(Watt per m^2 window surface)
while the convection heat transfer between the pane and the room air is equal to

$$\alpha_{kf}(\theta_f - \theta_r)$$

The heat transfer through the window from the innermost glass pane to the external air is equal to

$$\frac{1}{m_f}(\theta_f - \theta_l)$$

where m_f is the thermal resistance of the window including the boundary layer resistance on its outside, but excluding the boundary layer resistance on its inside surface. θ_l is the temperature of the external air.

The heat balance for the internal surface of the window is then:

$$I_V + \Phi_f = 5.67\ \varepsilon\ \Sigma\ [\varphi_{fu} f_{fu} (\theta_f - \theta_u)] + \\ + \alpha_{kf}(\theta_f - \theta_r) + \frac{1}{m_f}(\theta_f - \theta_l) \qquad (19)$$

Heat balance for room air

Heat is supplied to the air within the room by means of injected ventilation air (flow rate G_i kg/h), which is heated or cooled to a temperature θ_i $^{\circ}$C, and by air leaking in from the outside (G_l kg/h) at a temperature of θ_l $^{\circ}$C.

The room air also receives heat from lighting, human beings, etc. and also fram heaters in the room, if present. If one can neglect the heat storage of these heat sources, and the heat supply due to radiation from their surfaces, it is possible to introduce their power directly into the heat balance for the room air. If there are coolers in the room, their power is designated by a minus sign.

Heat is removed with the exhaust air and also by convective heat transfer at all room surfaces (including windows) if these are colder than the room air.

The heat balance is then:

$$0.24\,(G_i\,\theta_i + G_l\,\theta_l) + H_c = 0.24\,(G_i + G_l)\,\theta_r + \\ + \Sigma\alpha_k A\,(\theta_r - \theta) \qquad (20)$$

H_c is the heat transferred directly from the lights, heaters etc. to the air. A is the area of a room surface in m^2 and θ is its temperature in ^oC. Room surfaces also include the surfaces of heaters or coolers, if present. Furthermore, in equ. (20) 0.24 is the specific heat of air in kcal/kg ^oC (1 kcal = 4.19 kJ).

HEAT BALANCE EQUATIONS UNDER NONSTEADY-STATE TEMPERATURE CONDITIONS

Assembly of equations

To calculate *the temperature in the interior of a wall or a floor*, the following equation is employed:

$$K_1\,\theta_{n-1,t} + K_2\,\theta_{n,t} + K_3\,\theta_{n+1,t} = K_4\,\theta_{n-1} + \\ + K_5\theta_n + K_6\,\theta_{n+1} \qquad (A)$$

The values of the coefficients are given in table 2.3.
The temperatures at facade and room surfaces are calculated by means of:

$$-F_1\,\theta_{n\pm1,t} + F_2\,\theta_{n,t} - F_3\,\theta_{r,t} - \Sigma F_4\,\theta_{u,t} = \\ = F_1\,\theta_{n\pm1} + F_5\,\theta_n + F_3\,\theta_r + \Sigma F_4\,\theta_u + F_6 \qquad (B)$$

The coefficients are given in table 2.4.
The temperature of the room air can be obtained, in accordance with equation (20), from:

$$L_1\,\theta_{r,t} + \Sigma\,L_2\,\theta_t = L_3 \qquad (C)$$

where

$$L_1 = 0.24\,(G_{i,t} + G_{l,t}) + \Sigma\alpha_k A$$
$$L_2 = -\alpha_k A$$
$$L_3 = 0.24\,(G_{i,t}\,\theta_{i,t} + G_{l,t}\,\theta_{l,t}) + H_{c,t}$$

(0.24 = the specific heat of air in kcal/kg ^oC)

Table 2.3 Coefficients in equ. (A)

Cofficients	K_1	K_2	K_3	K_4	K_5	K_6
In homogeneous material	-1	$2\left(\frac{\Delta x^2}{a\Delta t}+1\right)$	-1	1	$2\left(\frac{\Delta x^2}{a\Delta t}-1\right)$	1
At the boundary surface between two materials	$-\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}$	$\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}\cdot\frac{\Delta x^2_{n-1}}{a_{n-1}\Delta t}+\frac{\Delta x^2_{n+1}}{a_{n+1}\Delta t}+\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}+1$	-1	$\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}$	$\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}\cdot\frac{\Delta x^2_{n-1}}{a_{n-1}\Delta t}+\frac{\Delta x^2_{n+1}}{a_{n+1}\Delta t}-\frac{\lambda_{n-1}}{\lambda_{n+1}}\cdot\frac{\Delta x_{n+1}}{\Delta x_{n-1}}-1$	1
At the surface of an air layer of which the other surface temperature is θ_{n+1} *)	-1	$\frac{\Delta x^2_n}{a_n\Delta t}+\frac{1}{m_l}\cdot\frac{\Delta x_n}{\lambda_n}+1$	$-\frac{1}{m_l}\cdot\frac{\Delta x_n}{\lambda_n}$	1	$\frac{\Delta x^2_n}{a_n\Delta t}-\frac{1}{m_l}\cdot\frac{\Delta x_n}{\lambda_n}-1$	$\frac{1}{m_l}\cdot\frac{\Delta x_n}{\lambda_n}$

*) If the surfaces of the air layer have temperatures θ_n and θ_{n-1} instead of θ_n and θ_{n+1}, then the coefficients K_1 and K_3 change places with eath other in equation (A), as do K_4 and K_6.

Table 2.4. Coefficients in equ. (B)

Coefficients	F_1	F_2	F_3	F_4	F_5	F_6
Facade surface	1	$\frac{\Delta x^2_n}{a_n \Delta t} + \frac{a_y}{\lambda_n} \Delta x_n + 1$	0	0	$\frac{\Delta x^2_n}{a_n \Delta t} - \frac{a_y}{\lambda_n} \Delta x_n - 1$	$\frac{a_y}{\lambda_n} \Delta x_n (\theta_e + \theta_{e,t})$
Facade surface with thin layer outside an air gap	1	$m_l \cdot \frac{2 s_n c_n \rho_n}{\Delta t} +$ $+ m_l a_y + 1$	0	0	$m_l \cdot \frac{2 s_n c_n \rho_n}{\Delta t} -$ $- m_l \alpha_y - 1$	$m_l \alpha_y (\theta_e + \theta_{e,t})$
Wall, floor or ceiling surface in the room	1	$\frac{\Delta x^2_n}{a_n \Delta t} + \frac{a_{kn}}{\lambda_n} \Delta x_n + 1 +$ $+ 5.67\, \varepsilon \frac{\Delta x_n}{\lambda_n} \Sigma \varphi_{nu} f_{nu}$	$\frac{a_{kn}}{\lambda_n} \Delta x_n$	$5.67\, \varepsilon \frac{\Delta x_n}{\lambda_n} \varphi_{nu} f_{nu}$	$\frac{\Delta x^2_n}{a_n \Delta t} - \frac{a_{kn}}{\lambda_n} \Delta x_n - 1 -$ $- 5.67\, \varepsilon \frac{\Delta x_n}{\lambda_n} \Sigma \varphi_{nu} f_{nu}$	$\frac{1}{A_n} \frac{\Delta x_n}{\lambda_n} [\Sigma \psi_{fn} A_f (I_T + I_{T,t}) +$ $+ \Sigma \psi_{be,n} A_{be} (I_{be} + I_{be,t}) +$ $+ \Sigma \psi_{B,n} (B + B_t)]$
Heater or cooler in room	0	$2 \frac{c_h \rho_h V_h}{\Delta t A_h} + a_{kh} +$ $+ 5.67\, \varepsilon \Sigma \varphi_{hu} f_{hu}$	a_{kh}	$5.67\, \varepsilon \varphi_{hu} f_{hu}$	$2 \frac{c_h \rho_h V_h}{\Delta t A_h} - a_{kh} -$ $- 5.67\, \varepsilon \Sigma \varphi_{hu} f_{hu}$	$\frac{1}{A_h} [H + H_t + \Sigma \psi_{fh} A_f (I_T + I_{T,t}) +$ $+ \Sigma \psi_{be,h} A_{be} (I_{be} + I_{be,t}) + \Sigma \psi_{B,h} (B + B_t)]$
The internal surface of a window *)	0	$a_{kf} +$ $+ 5.67\, \varepsilon \Sigma \varphi_{fu} f_{fu} +$ $+ \frac{1}{m_f}$	a_{kf}	$5.67\, \varepsilon \varphi_{fu} f_{fu}$	—	$\frac{1}{A_f} (\psi_{ff} A_f I_{T,t} + \Sigma \psi_{f'f} A_f' I'_{T,t} +$ $+ \Sigma \psi_{be,f} A_{be} I_{be,t} + \Sigma \psi_{B,f} B_t) + I_{V,t} + \frac{1}{m_f} \theta_{l,t}$

*) Note that in the case of a window one writes $\theta_{n+1} = \theta_n = \theta_r = \theta_u = 0$ in equ. (B).

Derivation and explanation of equations

In the equations, which apply to nonsteady-state temperature conditions, one must consider the heat which is stored or removed from bodies which have heat capacity. Apart from the walls and flooring, one must also consider the heat storage of furniture, and of the heating and cooling system, in certain cases.

The temperature in the interior of a wall or flooring.

In homogeneous material

One considers the material as being subdivided into layers which are parallel to the wall surface (se fig. 2.1.). Let us assume, that layer n in fig. 2.1 has temperature θ_n $^\circ$C at a time 0, and that the adjoining layers $n-1$ and $n+1$ have temperatures θ_{n-1} and θ_{n+1} $^\circ$C.

Under the influence of temperature gradients at the boundary surfaces of the layer n, it is assumed that the temperature of the layer after a time increment Δt h has been altered to $\theta_{n,t}$. At the same time the temperatures of the adjoining layers have become $\theta_{n-1,t}$ and $\theta_{n+1,t}$.

If the layers are sufficiently thin and the period of time is short enough, then the temperature gradients at the two boundary surfaces of the layer n have average values during this period of time of approximately

$$\frac{\theta_{n-1} - \theta_n + \theta_{n-1,t} - \theta_{n,t}}{2 \Delta x}$$

and

$$\frac{\theta_n - \theta_{n+1} + \theta_{n,t} - \theta_{n+1,t}}{2 \Delta x}$$

respectively.

The heat balance calculated per m^2 of wall surface is then the following for layer n

$$(\theta_{n,t} - \theta_n)\,\Delta x c\rho = \Delta t\,\frac{\lambda}{\Delta x}\cdot\frac{1}{2}\,[\theta_{n-1} - \theta_n + \\ + \theta_{n-1,t} - \theta_{n,t} - (\theta_n - \theta_{n+1} + \theta_{n,t} - \theta_{n+1,t})] \tag{21}$$

where c is the specific heat in J/kg^{o}C
and ρ is the density in kg/m^3.

By multiplying the equation by $2\Delta x/\lambda\Delta t$ and shifting the temperatures which are to be calculated, i.e. those that apply at a time Δt, to the left hand side of the equation, the following is obtained:

$$-\theta_{n-1,t} + 2\left(\frac{\Delta x^2}{a\Delta t} + 1\right)\theta_{n,t} - \theta_{n+1,t} = \\ = \theta_{n-1} + 2\left(\frac{\Delta x^2}{a\Delta t} - 1\right)\theta_n + \theta_{n+1} \tag{21a}$$

where

$$a = \frac{\lambda}{c\rho} \tag{22}$$

(a is the thermal diffusivity).

Equ. (21a) is the same equation as equ. (A) with coefficients according to table 2.3.

At the boundary layer between two different materials

The two different materials are designated n-1 and n+1. The thickness, thermal conductivity, specific heat and density of the layer are

Δx_{n-1}, λ_{n-1}, c_{n-1} and ρ_{n-1}

for the first material, and

Δx_{n+1}, λ_{n+1}, c_{n+1} and ρ_{n+1}.

for the other material.

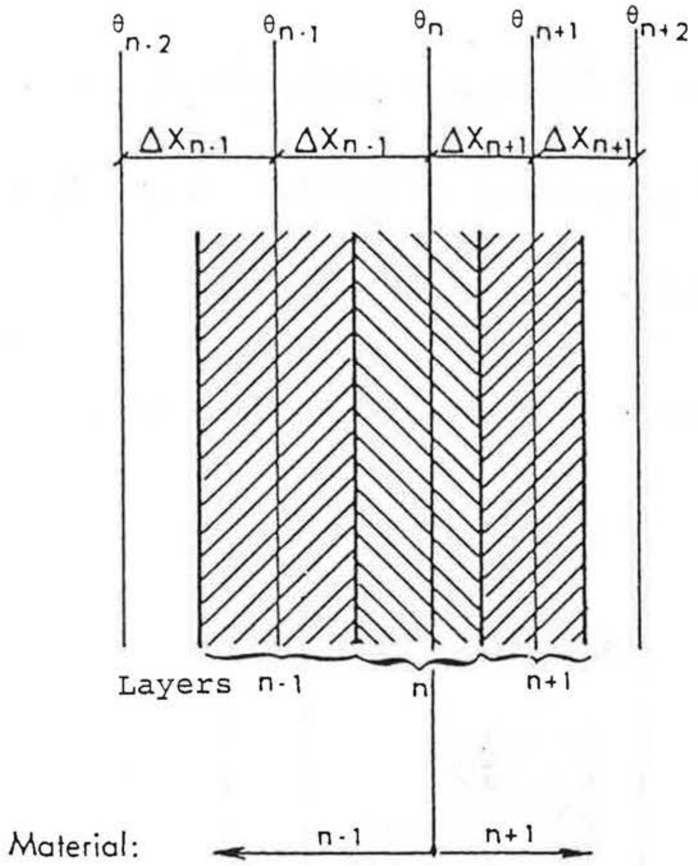

Fig. 2.4.

The temperature of the boundary surface is θ_n. The layer n (see fig. 2.4) with thickness $\frac{\Delta x_{n-1} + \Delta x_{n+1}}{2}$, is assumed to have this temperature.

The heat balance for layer n is

$$(\theta_{n,t} - \theta_n)\left(\frac{\Delta x_{n-1}}{2} c_{n-1}\, \rho_{n-1} + \frac{\Delta x_{n+1}}{2} c_{n+1}\, \rho_{n+1}\right) =$$

$$= \Delta t\, \frac{1}{2}\left[\frac{\lambda_{n-1}}{\Delta x_{n-1}}\left(\theta_{n-1} - \theta_n + \theta_{n-1,t} - \theta_{n,t}\right) - \right.$$

$$\left. - \frac{\lambda_{n+1}}{\Delta x_{n+1}}\left(\theta_n - \theta_{n+1} + \theta_{n,t} - \theta_{n-1,t}\right)\right] \qquad (23)$$

After rearrangement one obtains equation (A).

<u>At the surface of an air space</u>

Let us assume that the temperatures of the two boundary surfaces of the air space are θ_n and θ_{n+1}, see fig. 2.5. Approximately, θ_n and θ_{n+1} are also the temperature of half the material layers closest to the air space. The heat balance for layer n with thickness $\Delta x/2$ is then:

$$(\theta_{n,t} - \theta_n)\frac{\Delta x_n}{2} c_n \rho_n = \Delta t \frac{1}{2}\left[\frac{\lambda_n}{\Delta x_n}(\theta_{n-1} - \theta_n + \theta_{n-1,t} - \theta_{n,t}) - \frac{1}{m_l}(\theta_n - \theta_{n+1} + \theta_{n,t} - \theta_{n+1,t})\right] \tag{24}$$

After rearranging, one obtains equation (A).

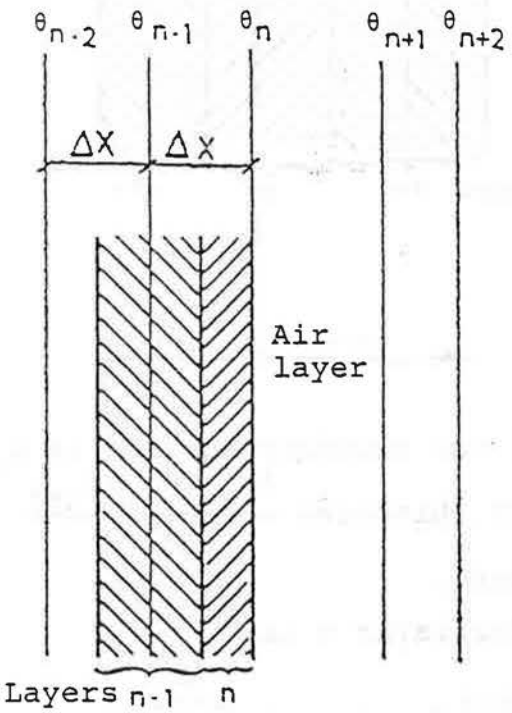

Fig. 2.5

If the temperatures of the two boundary surfaces of the air gap are θ_n and θ_{n-1}, instead of θ_n and θ_{n+1}, the coefficients will be unchanged. The coefficients K_1 and K_3, however, exchange positions with each other in equation (A), as do K_4 and K_6.

Layers of dense, highly conductive material can be treated in a special way. Let us assume that at the surface with temperature θ_n there is a sheet of such a material, asbestos cement or something similar. One can then consider, that the temperature in the material does not vary with the distance from the surface. Under such conditions the left hand side of the heat balance equation above is changed to:

$$\left(\theta_{n,t} - \theta_n\right) \left(\frac{\Delta x_n}{2} c_n \rho_n + s_x c_x \rho_x\right)$$

while the right hand side of the equation remains unchanged. In this case s_x designates the thickness of the sheet, while c_x and ρ_x designate the specific heat and density of the material from which the sheet is constructed. The sheet stores heat, so that the thermal diffusivity is reduced from

$$a_n = \frac{\lambda_n}{c_n \rho_n}$$

to

$$a_n = \frac{\lambda_n}{c_n \rho_n + \frac{2 s_x}{\Delta x_n} c_x \rho_x} \qquad (25)$$

T h e t e m p e r a t u r e s o f f a c a d e a n d r o o m s u r f a c e s

Facade surfaces

At the outside of an external wall (see fig. 2.6) a heat balance is established for a material layer closest to the surface, which has a thickness $\Delta x/2$. One obtains:

$$\left(\theta_{n,t} - \theta_n\right) \frac{\Delta x_n}{2} c_n \rho_n =$$
$$= \Delta t \cdot \frac{1}{2}\left[a_y \left(\theta_e - \theta_n + \theta_{e,t} - \theta_{n,t}\right) - \right.$$
$$\left. - \frac{\lambda_n}{\Delta x_n} \left(\theta_n - \theta_{n\pm 1} + \theta_{n,t} - \theta_{n\pm 1,t}\right)\right] \qquad (26)$$

α_y and θ_e have the same meaning as given previously. Equation (B) is obtained from equation (26) with coefficients according to table 2.4.

If a facade surface according to fig. 2.6 is covered by a sheet with a thermal resistance that can be neglected, the coefficient of thermal diffusivity a_n can be calculated according to equation (25).

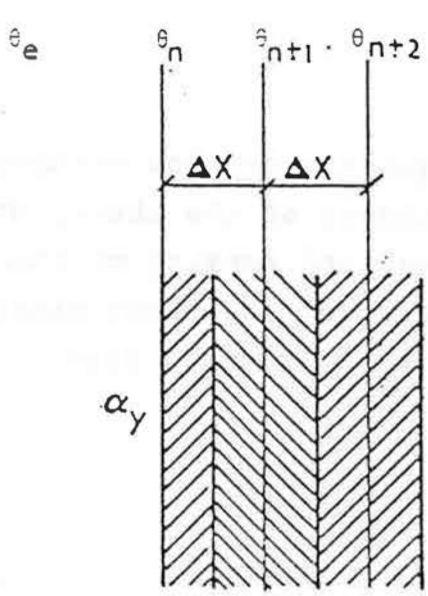

Fig. 2.6.

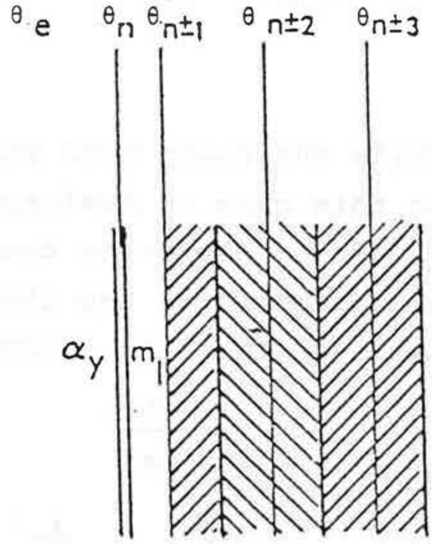

Fig. 2.7.

If, as is shown in fig. 2.7, there is a facade where a sheet with a thermal resistance low enough to be neglected is separated from the rest of the wall by an air gap with thermal resistance m_l, one obtains the following heat balance equation:

$$(\theta_{n,t} - \theta_n)\, s_n\, c_n\, \rho_n =$$
$$= \Delta t \cdot \frac{1}{2}\Big[\alpha_y (\theta_e - \theta_n + \theta_{e,t} - \theta_{n,t}) -$$
$$- \frac{1}{m_l}(\theta_n - \theta_{n\pm1} + \theta_{n,t} - \theta_{n\pm1,t})\Big] \qquad (27)$$

<u>Wall, floor and ceiling surfaces in the room</u>

The heat balance for a surface layer with thickness $\Delta x/2$ is the following (see fig. 2.2 and equations (10) and (11)):

$$(\theta_{n,t} - \theta_n)\frac{\Delta x_n}{2} c_n \rho_n = \Delta t \cdot \frac{1}{2}\Big\{\alpha_{kn}(\theta_r - \theta_n + \theta_{r,t} -$$
$$-\theta_{n,t}) + 5.67\varepsilon \Sigma [\varphi_{nu} f_{nu}(\theta_u - \theta_n + \theta_{u,t} - \theta_{n,t})] +$$
$$+ \frac{1}{A_n} [\Sigma\psi_{fn} A_f (I_T + I_{T,t}) +$$
$$+ \Sigma\psi_{be,n} A_{be} (I_{be} + I_{be,t})$$
$$+ \Sigma\psi_{B,n}(B + B_t)] -$$
$$- \frac{\lambda_n}{\Delta x_n}(\theta_n - \theta_{n\pm 1} + \theta_{n,t} - \theta_{n\pm 1,t})\Big\} \qquad (28)$$

As before, it is possible to use equ. (25) for the calculation of a_n in the coefficients F_2 and F_5, if the surface is covered by a sheet which has a thermal resistance small enough to be neglected.

Heaters and coolers in the room

A heater or cooler always has a heat storage capacity. If this capacity can be neglected, as well as the radiation heat transfer from the heating or cooling element surfaces, it is possible to introduce the thermal output of the heater or cooler into the heat balance for the room air as the quantity $H_{c,t}$ in equ. (C).

If it is impossible to make this approximation, but if one can assume that the temperature in the heater or cooler is the same in all its components, θ_h at a time 0 and $\theta_{h,t}$ after a time interval Δt, then one obtains a heat balance equation for the whole apparatus which is as follows (the volume of the apparatus is assumed as V_h and its surface as A_h):

$$(\theta_{h,t} - \theta_h) V_h c_h \rho_h = \Delta t \cdot \frac{1}{2}\Big\{(H + H_t) -$$
$$- \alpha_{kh} A_h (\theta_h - \theta_r + \theta_{h,t} - \theta_{r,t}) -$$
$$- 5.67\,\varepsilon A_h \Sigma [\varphi_{hu} f_{hu}(\theta_h - \theta_u + \theta_{h,t} - \theta_{u,t})] +$$
$$+ \Sigma\psi_{f,h} A_f (I_T + I_{T,t}) +$$
$$+ \Sigma\psi_{be,h} A_{be} (I_{be} + I_{be,t}) +$$
$$+ \Sigma\psi_{B,h}(B + B_t)\Big\} \qquad (29)$$

H and H_t are the thermal output of the heating or cooling element in W.

Equation (29) also applies for the heat balance of an object within the room, e.g., furniture or similar. Under these conditions $H = H_t = 0$.

The internal surface of a window

It is assumed that the window has no thermal capacity. For this reason the temperatures can be calculated from the values which apply after the time interval Δt. Equation (19) can therefore be used directly.

In order to convert this equation into equation (B) one must equate the temperatures in equation (B) with 0 at time 0. In this way one obtains the coefficients given for the inside of a window in table 2.4.

NOTES ON CALCULATION BY MEANS OF COMPUTER[x)]

When solving the preceding heat balance equations, one begins by considering that the temperature is known at every position of the building where the temperature changes are to be calculated, at one given initial moment. By solving the equations of forms (A), (B) and (C) which are contained in the equation system, the temperatures after a chosen interval of time Δt is obtained. These temperatures are then used for determining the temperatures after a further time interval by employing the same equation system, but with changed values for the coefficients, if necessary, and so on. The calculations are carried out step by step, until the variations of the temperatures during the whole period of analysis have been determined.

In general, the temperature conditions are not known at any given time before the calculations have been carried out.

x) It should be noted that this was written more than 25 years ago (in Swedish).

This, however, is not of any real importance, because after a few days the initial values of the temperatures have little influence upon the calculated values, and therefore the initial values can be chosen quite freely. If the temperature conditions in a building are to be studied where the external temperature, solar radiation etc vary periodically with a 24-hourly period (the method of calculation is, however, not limited to such a case), the calculations can be stopped after the values calculated for the last 24-hourly period correspond with those of the previous 24-hourly period within the accuracy required. It is obvious that the calculations must continue longer if a greater accuracy is required, if the initial values have been chosen badly, or if the building structure is heavy.

Even if one is only interested in the variation of the room air temperature, it is still necessary to determine the temperatures of the room surfaces as well as the temperatures of the interior of walls and flooring. For the calculations of temperatures in a single room a large number of equations of form (A) - (C) is needed, several tens. Moreover, if temperatures in many rooms are to be determined simultaneously, there may be several hundred equations to be solved. If a suitable method, say the relaxation method (see for example *Shaw* /4/) is used for the solution of such large equation systems, the computer costs need not be excessive, provided that the calculations are carried out with a suitable machine, and that the program uses the machine to its full capacity. This metod of calculating room temperatures with regard to unsteady-state temperature conditions in a building is, according to the writer´s opinion, the best to use. There are naturally also other possibilities, see for example *Andersson* /5/ and *Brown* /6/.

The equations (A) through (C) are of the implicit form. In the last named paper a numerical method of calculations is mentioned, which is similar to the one described here. By using that method, however, explicit difference equations are used. For each temperature to be determined, there is an equation in which the temperature applying after the time interval Δt is expressed as a function of previous values. For this reason the new temperatures can be determined without the need to solve the equations for all the temperatures existing at the same time simultaneously. This would appear to be an advantage. The disadvantage of the explicit method is, however, that one must take Δt as a very short period of time in some cases, so that the calculations will not be subject to instability. When Δt is small, the required computer time can be large, and the calculation becomes more expensive. The upper limit of Δt while maintaining stability can be calculated from the surface heat transfer coefficient, the thermal diffusivity and the thickness of the layer Δx, for the various walls and floors, a calculation which can be quite complex. Although the implicit formulation used is more stable, it is obvious that one must not use too large values for Δt, if all variations in presupposed temperatures and other given values are to influence the results in a correct fashion.

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3 CALCULATION OF HEATING AND COOLING LOADS- AN EXAMPLE

A sketch, showing the dimensions of the room concerned is given in fig. 3.1. Table 3.1 gives the various values for walls and flooring. The window is a double glazed (coupled frames) type with Venetian blinds between the panes (inclination of slats = 45°). The Venetian blinds are lowered at 12:30 p.m. and drawn up at 6:30 a.m.

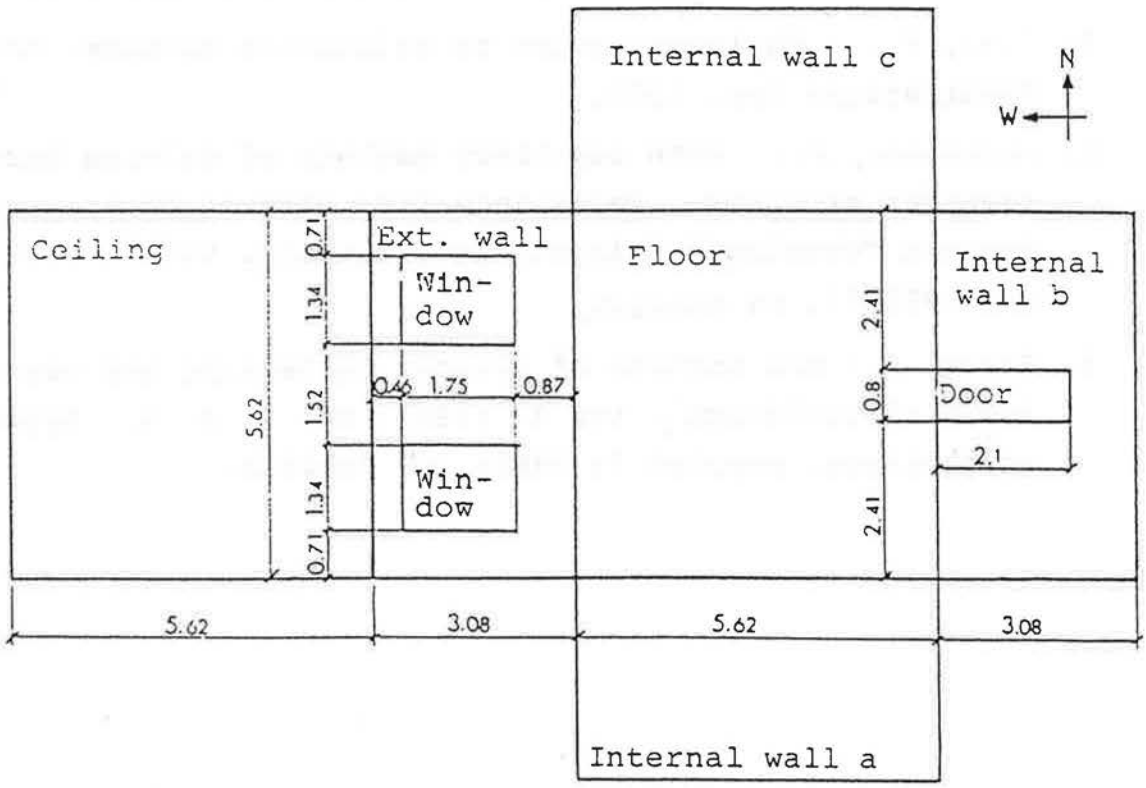


Fig. 3.1. Dimensions of the boundary surfaces of the room. The walls and ceiling are drawn in the plane of the floor.

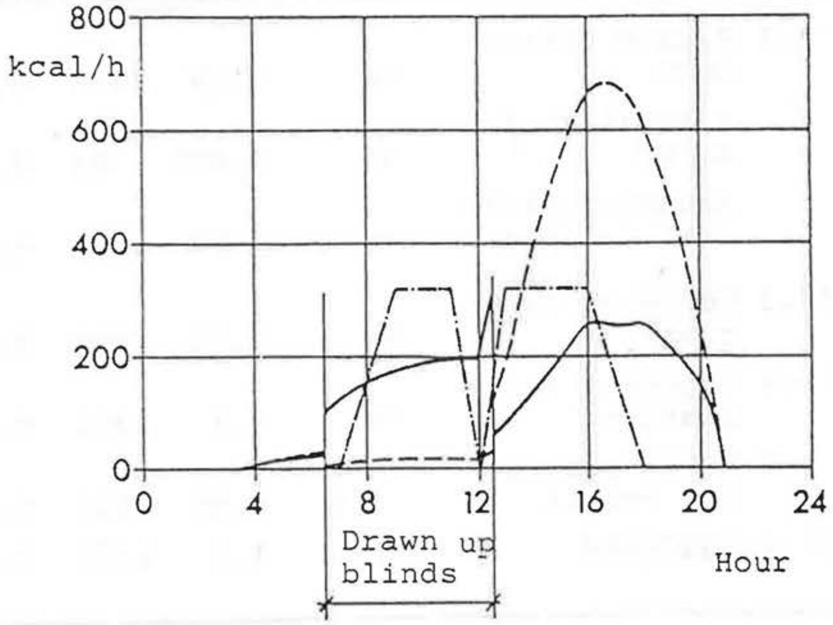
It is assumed that the room lies on the west side of a building, which is situated at a latitude of 60° north, and that the calculations are being carried out on a clear day around the 15th July. It is assumed that the external temperature and the radiation from the sun and sky were the same during the previous 24-hourly periods as during the day in question (24-hourly periodicity). It is also assumed that the temperatures in the rooms lying next to the room which is being studied, and the ones above and below, vary in the same way as the temperatures of the room in question. In the room outside wall *b* (the side facing the corridor) the temperature is, however, constant at 22^{o}C.

Table 3.1. Dimensions and data regarding materials of walls and flooring (1 kcal = 4.19 kJ, 1 kcal/h = 1.163 W)

Room surface	A m^2	Material	Thickness cm	λ $\frac{kcal}{mh\ ^oC}$	ρ $\frac{kg}{m^3}$	c $\frac{kcal}{kg\ ^oC}$
External wall	12.6	Facade brickwork	12	0.60	1600	0.20
		Mineral wool layer	10	0.035	45	0.21
		Internal brick leaf + plaster	14	0.50	1600	0.20
Internal walls a and c	17.3	Cellular concrete	10	0.13	600	0.21
Internal wall b	15.6	Concrete and plaster	18	1.3	2300	0.21
Door	1.7	Fibre plates and Wellit	3.6	0.05	200	0.41
Flooring	31.6	Concrete	20	1.3	2300	0.21

With normal external temperatures, the quantity of ventilation air is 360 kg/h between 7 a.m. and 10 p.m., and the air is taken directly from the outside, so that the air which is blown in, has the temperature of the external air. During the rest of the day the fans are switched off and under those conditions the quantity of air which is flowing in, is equal to 60 kg/h. During periods of hot weather, the fans are left running the whole day long so that the building is cooled as much as possible by the night air.

Heat exchange between the room and its surroundings takes place by means of ventilation and by heat transmission through walls and windows. In addition, there is a heat gain due to solar and sky radiation through the window, and heat given off by human beings. There is a maximum of 4 people in the room and it is assumed that each of them gives off 80 kcal/h (93 W) in the form of sensible heat (latent heat in the form of moisture from the skin and water vapour in the air breathed out do not alter the room temperature). During the day when the room was being studied, no artificial lights were required, and there were no other sources of heat.



—— Solar and sky radiation transmitted directly
---- Solar and sky radiation transmitted indirectly
-·-·- Heat from human beings

Fig. 3.2. Heat which is being supplied to the room from human beings and due to solar and sky radiation.

Figure 3.2 shows how the heat given off by human beings is likely to vary during a 24 hour period. The quantity of solar and sky radiation, which is transmitted directly through the window, and the indirectly transmitted radiation of this type, which is supplied to the room by heating up the inside glass pane in the window, is also shown here. For windows without Venetian blinds (i.e. drawn up blinds), the radiation data were obtained from precalculated tables.

In the case of windows with Venetian blinds, the influence of these upon the thermal inflow has been calculated according to

Ozisik and Shutrum /1/. Their values correspond well with those which were obtained by the writer when carrying out his own measurements. The thermal resistance of the window, including the external surface resistance but excluding the internal resistance, is taken as 0.29 $m^2h^oC/kcal$ (0.25 $m^{2o}C/W$) when the Venetian blinds are drawn up and 0.36 $m^2h^oC/kcal$ (0.31 $m^{2o}C/W$) when they are lowered.

It may be of interest to note that when the Venetian blinds are drawn up, the radiation which is transmitted directly, is greater than the radiation which is transmitted indirectly, while the opposite applies when the Venetian blinds are lowered.

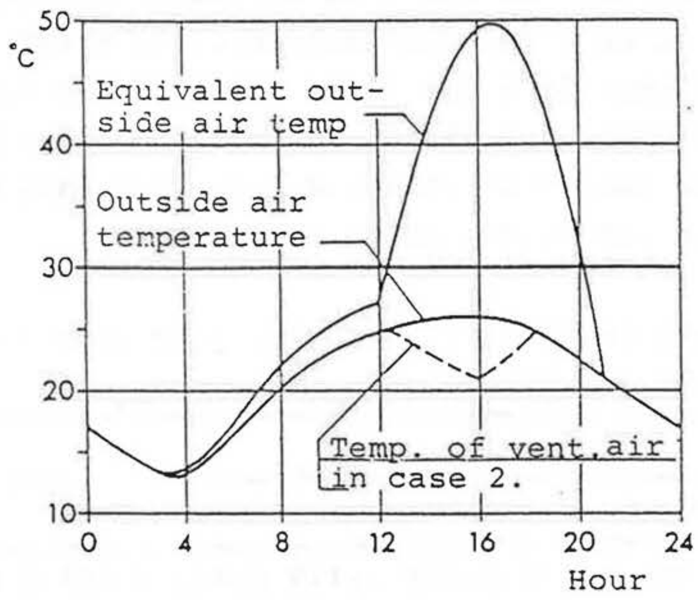


Fig. 3.3. External air temperature and equivalent outside air temperature.

When heat transmission through an external wall exposed to the sun is calculated the equivalent external temperatur is obtained by adding the term $a_y I/\alpha_y$ to the external air temperature (see fig. 3.3). The I value of the solar and sky radiation against the facade is measured in $kcal/m^2h$ (W/m^2), a_y is the absorptivity of the wall surface for this radiation and

α_y is the surface heat transfer coefficient (including also the effect of long wave radiation). During the whole time when the facade is exposed to the sun, it is assumed that α_y = 14 kcal/m^{2}h $^{\circ}$C (16.3 W/m^2 $^{\circ}$C) and a_y = 0.5. On the side facing the corridor, of the wall *b* and the door, it is assumed that the coefficient of surface heat transfer, including radiation is 6 kcal/m^{2}h $^{\circ}$C (7.0 W/m^2 $^{\circ}$C) for the whole 24 hour period.

As far as radiation between the room surfaces is concerned, it is presumed that corresponding to the assumptions made in chapter 2, for long wave radiation emitted due to the emissivity of the surfaces ε = 0.93 at all surfaces, while reflection at the surfaces is zero for all long wave radiation. Solar and sky radiation, which is transmitted directly through the windows is, however, reflected diffuse at all room surfaces. The reflectivity for such radiation is 60 % for wall surfaces, 50 % for door surface, 75 % for ceiling, 25 % for floor and 20 % for the inside of a window. The absorption factors, which can be obtained from these reflectivities are given in table 3.2. They have been calculated according to the method given in chapter 1.

Table 3.2. Absorption factors Ψ_{nu} for radiation from a surface A_n against a surface A_u.

Room surface	n	Ψ_{nu}							
		$u = w$	a	b	c	d	t	g	f
Ext. wall	w	0.045	0.118	0.098	0.118	0.012	0.124	0.451	0.034
Int. wall	a	0.086	0.060	0.107	0.107	0.012	0.124	0.439	0.065
Int. wall	b	0.080	0.119	0.056	0.119	0.007	0.126	0.436	0.057
Int. wall	c	0.086	0.107	0.107	0.060	0.012	0.124	0.439	0.065
Door	d	0.073	0.100	0.051	0.100	0.007	0.116	0.481	0.072
Ceiling	t	0.079	0.109	0.099	0.109	0.013	0.059	0.471	0.061
Floor	g	0.096	0.128	0.115	0.128	0.018	0.157	0.292	0.066
Window	f	0.046	0.119	0.098	0.119	0.017	0.132	0.434	0.035

Alternative conditions for the calculations

Six different calculations of the temperature variations during the 24-hour period of 15 July vere made. The conditions were varied in the following way:

Case 1 No cooling of the ventilation air takes place. Its flow rate is constant during the 24-hour period.

Case 2 Cooling of ventilation air when required so that the temperature of the room air does not rise above 25°C. Constant flow rate of ventilation air.

Case 3 No cooling, ventilation off during the night between 10 p.m. and 7 a.m. The leakage air flow rate is 60 kg/h, while the ventilation air flow rate is 360 kg/h during the day.

Case 1a The same as case 1, but the convective heat transfer coefficients at the ceiling and flooring surfaces are assumed to have constant values during the 24-hour period. The average 24-hourly values which can be calculated in case 1 are used.

Case 1b The same as case 1 but α_k = 2.5 kcal/m^{2}h °C (2.9 W/m^2 °C) at ceiling and flooring surfaces for the whole day.

Case 1c The same as case 1 but the instantaneous value of the equivalent outside air temperature is constant and equal to the average 24-hourly value of the instantaneous values in case 1.

Cases 1a, 1b och 1c were studied to see how certain simplifications of the conditions affect the result of the calculations. A calculation according to the method can, naturally, be carried out quicker and therefore cheaper, if it is possible to use such simplifications without reducing accuracy too much.

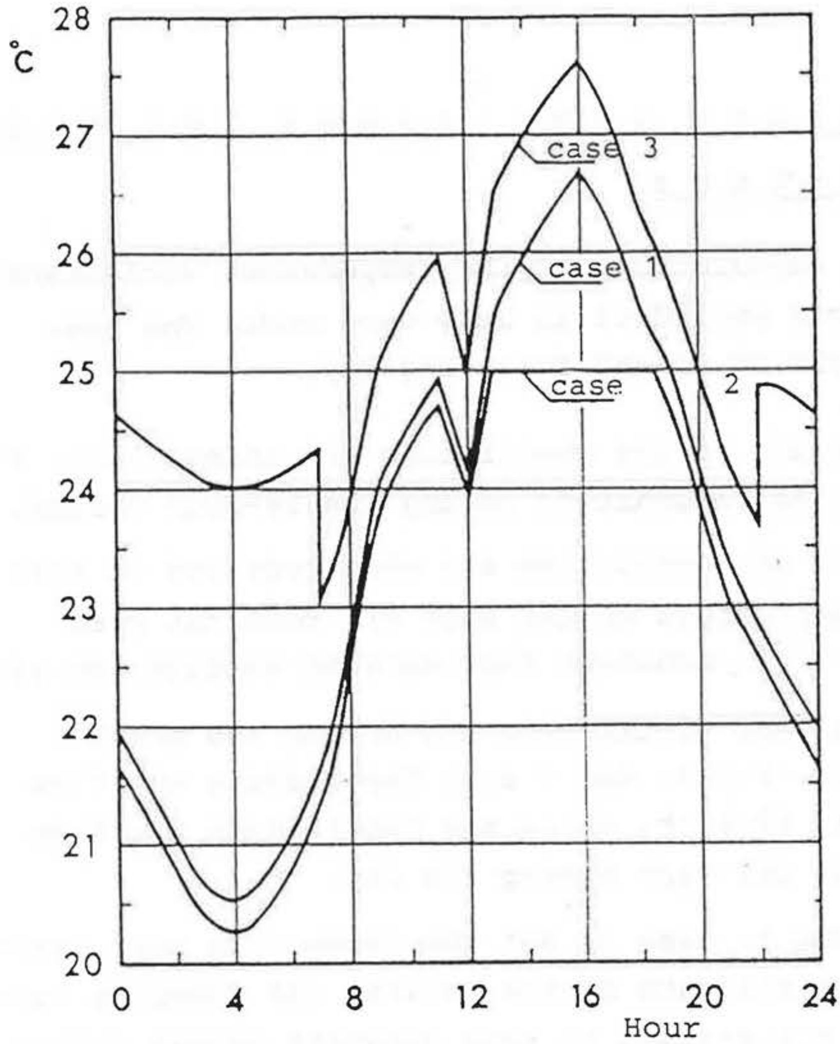


Fig. 3.4. Temperature of air in room in cases 1-3.

R e s u l t s o f c a l c u l a t i o n s

Diagram fig. 3.4 shows how the temperature of the air in cases 1, 2 and 3 varies during a 24 hour period.

In case 1 the temperature approaches the maximum value of 26.7°C at 4 p.m. In spite of the fact that, according to fig. 3.2 radiation inwards through the window continues to rise somewhat, the room temperature falls quickly after that time, evidently because heat given out by the occupants is reduced rapidly.

In case 2, it is seen from the graph in fig. 3.4 that it is necessary to cool the ventilation air between about 12:45 p.m. and 6:30 p.m. so that the temperature of the room air should not rise above 25°C. The temperatures to which the ventilation air must be cooled, are inserted into the diagram in fig 3.3.

The calculations show, that the maximum cooling effect occurs at 4 p.m. and is 440 kcal/h (512 W). The entire cooling requirement is 1470 kcal (6160 kJ).

In case 3 one misses the cooling of the walls and floor of the room, caused by the cool night air. During office hours the room temperature is therefore about 1°C higher than in case 1.

The thermal comfort of human beings depends not only upon the temperature of the air within the room, but also upon the temperature of the surfaces of the room, with which the body is in constant heat exchange due to radiation.

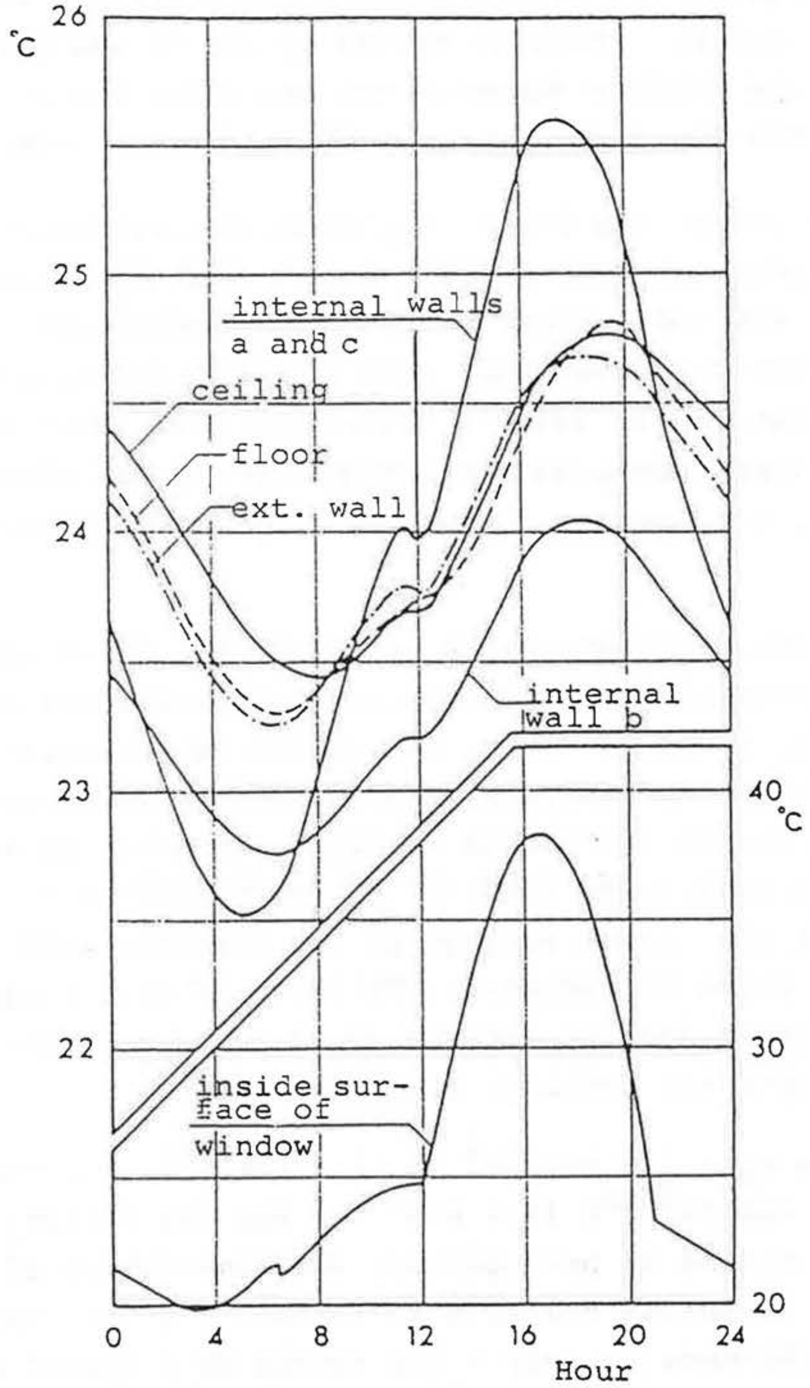


Fig. 3.5. Temperatures of room surfaces in case 1.

Fig. 3.5 shows how the temperatures of the room surfaces vary during a 24-hour period in case 1. The window panes and the blinds are assumed to have no thermal capacity. They absorb solar radiation, and therefore the temperature of the inner window surfaces varies strongly. The thermal capacity of the opaque building materials causes the variations of surface temperatures to be less than of the room air. In spite of the fact that the average 24-hourly temperatures of wall and flooring surfaces are higher than the temperature of the air within the room (with the exception of the temperature at the surface of wall *b*, which is cooled by the cooler air from the corridor), the surface temperatures are lower during working hours than the temperature of the air within the room.

The temperature during the day is higher on the surface of the comparatively light internal walls *a* and *c*, than on the surfaces of the flooring and the heavier walls. The disadvantage with light walls is not only that they have less ability to even out temperature variations of the air within the room. They also have a higher surface temperature during the day and therefore contribute to give a feeling of a higher room temperature.

Fig. 3.6 shows the temperatures at six different times of the day in the cross-section of the flooring and walls. The external wall dampens, as can be seen, the strong temperature variations on the outside exceedingly. The variations which occur, however, on the inside are caused mainly by temperature variations of the air within the room and of other room surfaces with which the inside surface of the external wall exchanges heat by means of radiation. This can be seen from fig. 3.5, where the graph for the external wall has nearly the same shape as the graphs for ceiling, floor and wall *b*.

In case 1a, when α_k = 1.5 kcal/m^2 ^{o}C (1.7 W/m^2 ^{o}C) for the floor surface and 1.2 kcal/m^{2}h ^{o}C (1.4 W/m^2 ^{o}C) for the ceiling surface during the entire 24 hour period, the temperature of the air in the room is nearly the same as in case 1 at all times of the day. The difference is only a few tenths of a degree at night, while during the day the difference is even smaller. In the same way the deviations with regard to ceiling and floor temperatures were quite insignificant.

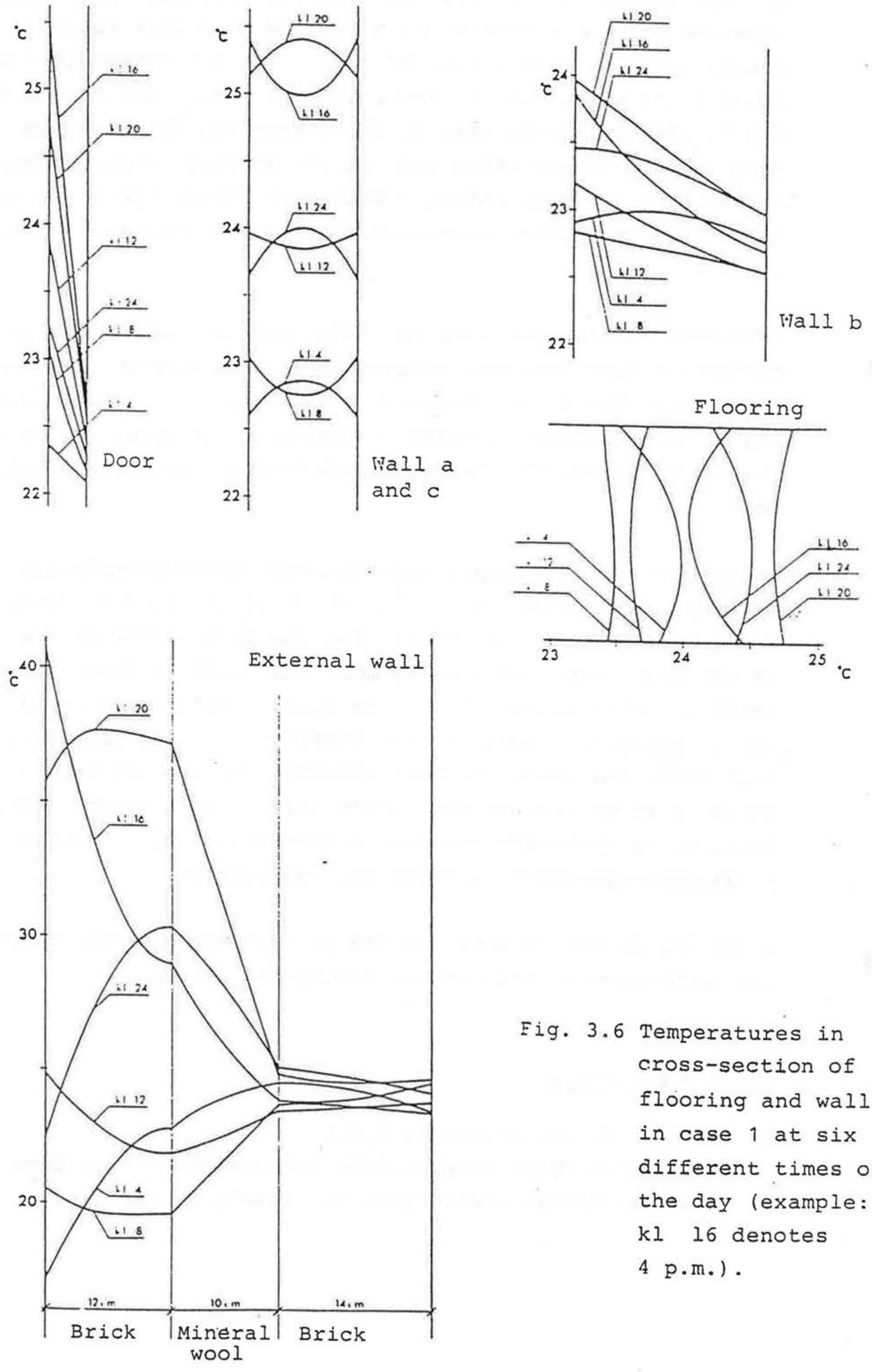


Fig. 3.6 Temperatures in cross-section of flooring and walls in case 1 at six different times of the day (example: kl 16 denotes 4 p.m.).

In case 1b the values were, however, so different from those obtained in case 1 that we must examine them more closely: during the time from 1 p.m. to 5 p.m. the air temperature was found to be about 0.4 ^{o}C lower, between 1 a.m. and 5 a.m. about 0.4 ^{o}C higher than in case 1. The reason for this must be, that in case 1b the floor and ceiling surfaces cool the air more during the day, and heat them more during the night since the coefficient of surface heat transfer is greater than in case 1.

From this can be seen that the value of the coefficient of convective heat transfer influences the temperature calculations, but that it can be considered adequate to use an average 24-hourly value in order to obtain a sufficiently accurate result. Of course the average value must be accurately calculated.

In case 1c it was assumed that the equivalent outside air temperature was constant at 27 ^{o}C, which is the average 24-hourly value in case 1. It was found that the temperature of the air in the room never differed by more than 0.03 ^{o}C from the corresponding value in case 1. In the case of an external wall, where the temperature variations are dampened so appreciably as in this case, and where the heat transfer through the wall is so small in comparison to that which takes place through the windows, it is obviously completely unnecessary to calculate using a variable equivalent outside air temperature.

Cases 1a, 1b and 1c show how the program can be used to study the importance of selected modelling assumptions.

Reference

1. *Ozisik, N. and Schutrum, L.F.:*
Solar Heat Gains through Slat-Type Between-Glass Shading Devices. Trans. ASHRAE Vol. 66 (1960), p. 359.

4 PRESENT-DAY STATE OF THE PROGRAM

4.1 Physical basis

4.1.1 Climatic data

Outdoor climte is described by outdoor temperature and solar radiation.

Observed data from a climate file can be used for aperiodic simulations. An optional calendar can be utilized to locate weekends and public holidays so that different profiles of time dependent data can be used for these different days.

For a periodic simulation, for a specified day of the year, the outdoor air temperature can be given by the 24-hour mean temperature, the amplitude of the temperature variation during the day, and the hour when the maximum temperature is reached. The temperature oscillation is then treated as sinusoidal.

The solar radiation, direct from the sun and diffuse from the sky, is calculated based on the desired location of the building, given by latitude and longitude. Latitude is positive North of the equator, longitude is positive West of Greenwich. In addition to that, the time meridian is required to relate the location to the correct time zone. Rather than using the reference longitude, the time lag after Greenwich is entered. As an example, Central European time is normally entered as 23 h. The value should be corrected for daylight saving time, when relevant.

The reduction of incoming solar radiation due to absorption in the atmosphere is taken into account by a reduction factor entered by the user (maximum value 1.0, for a clear and dry day). It can also be used to approximate the effects of average cloud cover in this context.

The radiation from sun and sky is reflected from the ground. Typical values of the ground surface reflectivity are 0.25 for summer conditions and 0.8 for snow cover.

The BRIS program calculates the solar radiation for an entirely unobstructed horizon, or for a horizon hidden to a specified altitude between 0 and 90 degrees. (Detailed shading from shading devices or from nearby buildings can at present only be treated by preparatory runs of separate shading programs.) Incident radiation values in W/m^2 are obtained for horizontal building surfaces (roofs) and for facades with orientations given by their azimuth.

The absorbed fraction of the irradiation at the outer surface of an external wall or a roof is determined by the absorptivity for shortwave radiation, for instance 0.5 for yellow and light red brick, 0.7 for dark red brick and 0.9 for a dark roof cover.

Heat is emitted from building surfaces by longwave radiation to the sky and by convection to the outdoor air. For the radiation losses, different cloud cover constants are assumed which are dependent on cloudiness. They are relevant for horizontal or somewhat tilting surfaces only. For vertical surfaces, the radiative heat transfer is assumed to be included in the surface heat transfer coefficient.

4.1.2 Heat transfer through windows

According to Chapter 2, the shortwave radiation from sun and sky can be subdivided into two parts. One, I_T , is directly transmitted through the window, possibly after reflections between the window panes, but not absorbed by them or by the shading device, if any. The other, I_V, is absorbed, i.e. transformed to heat. The temperature rise of the innermost pane effects the heat balance of the room, and the room climate.

Again, one portion of I_V is transferred by longwave radiation to surfaces in the room and by convection to the room air. Another portion of I_V is transmitted in the opposite direction through the window back to its outside.

In BRIS the treatment of heat transfer through a window exposed to sun will be the following.

Each fenestration combination (number of panes, types of shading, etc.) has to be described by two shading coefficients F_1 and F_2 , plus a thermal transmission coefficient U W/m²°C, see table in Appendix C. Here F_1 denotes total transmitted solar radiation and F_2 direct transmitted solar radiation, both relative to total transmitted radiation for a window with two panes of ordinary glass, I_{ref}.

In the result output from the BRIS computation, I_{ref} W/m² and $I_{tot} = (I_V + I_T)A_f$ W are recorded. In addition, values of the temperature of the internal surface of the window, θ_f °C, and the heat transfer P_{out} W from this surface to the out-door air, are given (in fact - P_{out} out because the aim is to sum up all heat transferred into the room). It should be noticed that θ_f (calculated from equation (19) in Chapter 2) is of great importance in connection with the effective temperature as measured at different points in the room, especially close to the window. The relations between the mentioned quantities are discussed below.

It is assumed that m_f is the thermal resistance of the window including the external but excluding the internal surface resistance, which we call m_i. The total resistance is

$$m_{tot} = 1/U = m_f + m_i \quad (1)$$

Independent of other thermal transmission through the window, the portion of I_V which is transferred to the room is the ratio between the thermal resistance m_f and the total thermal resistance m_{tot} . We can write this ratio $1-U_{mi}$, and we get

$$(1 - U_{mi})\ I_V = (F_1 - F_2)\ I_{ref} \quad (2)$$

Also, we have

$$I_T = F_2\ I_{ref} \quad (3)$$

It is assumed in the program that m_i = 0.11 m²°C/W when I_{tot} and P_{out} are to be determined. (In other cases, the convective and radiant heat transfer at the inside of a window are calculated according to Chapter 2).

Then we get

$$I_{tot} = (I_V + I_T)A_f = \left(\frac{F_1 - F_2}{1-0.11U} + F_2\right)\ A_f\ I_{ref} \quad (4)$$

Further, we have

$$P_{out} = A_f(\theta_f - \theta_l)/m_f = A_f(\theta_f - \theta_l)/(1/U-/0.11) \quad (5)$$

9C

$I_{tot} - P_{out}$ is the contribution from the window heat transfer to the heat balance of the room. Note that in the program the radiant portion is assumed to be emitted diffusely to the room surfaces.

4.1.3 Thermal radiation between surfaces in rooms

Angle and absorption factors (see Chapter 1) for wall, floor and ceiling surfaces, are always determined in BRIS runs and given in the result output if wanted.

Using these factors, examples of determinations of diffusely emitted daylight show good agreement with measured values. Best calculation results have been reached when floor and ceiling surfaces and partition wall surfaces have been divided in subsurfaces (stripes), located at different distances from the window in question *(Brown & Isfält* 1974)[x].

In passing, it can be mentioned that angle and absorption factors can be approximately obtained from simple equations that in addition to the areas and absorptivities of the room surfaces also contain 'location coefficients'. Values of such coefficients have been determined for radiation from a window in rooms of different shapes (*Brown* 1984)[x].

Effective temperature

The aim of the quantity 'effective temperature' is to take into account the effect of the radiative heat exchange betwen the human body and the room surfaces, in order to give a better measure of the comfort in a room than the room air temperature only.

The BRIS program calculates an effective temperature by considering not only the temperatures of the surfaces in the room, but also the angle factors for radiation from the room

x) See References in Introduction.

surfaces to six surfaces of a small hypothetical cube placed at the point in the room for which the effective temperature is required. The sides of the cube are parallel to the surfaces of the room.

Seen from the surfaces of the cube, different parts of the surfaces of the room appear to be screened, with the result that the irradiation varies from one surface of the cube to another.

A comfort rating is also determined, defined as the maximum difference of effective temperature in different directions from the calculation point. The comfort factor can be said to constitute a measure of the asymmetry of the radiation field, because it represents the difference between the temperature of the warmest and the coldest surface on the cube. A high value, in other words, is an indicator of poor comfort.

If the air temperature in an office is not cooled during the working hours of the day, the effective temperature is usually lower than the air temperature (Swedish summer climate conditions). The graph in fig. 4.1 gives an example showing the difference between the air and the effective temperatures in a point near a window facing the north in a room that is first assumed to be in a lightweight building (continuous line), then in a massive building (broken line).

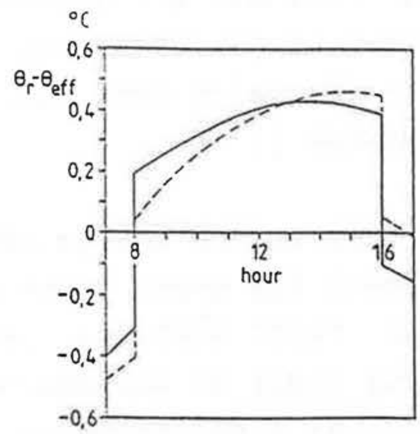


Fig. 4.1. Difference between room air temperature and effective temperature. An example.

When BRIS calculates comfort, the variations are restricted to the six sides of a cube. A sphere would normally give a slightly higher variations.

BRIS also calculates a frequency distribution for the effective temperature. This can serve as a basis for the calculation of cumulative frequencies over typical climate variations.

The effective temperature is calculated and printed for all intervals included in the output listing. The frequency distribution, however, is normally required just for the period when the room is occupied. The desired period is indicated by a reference in form type 3 (see Appendix A).

4.1.4 Convection heat transfer at room surfaces

In BRIS, the convective heat transfer at every room surface, the inner window surface included, is calculated by using the formula

$$\alpha_k = a + b(abs(\theta_r - \theta))^c$$

where θ_r and θ are the temperatures of room air and surface respectively. The values a , b and c may be given as constants or as time dependent parameters. Normally used values of a , b and c are those given in Chapter 2.

Usually, the room air is regarded in BRIS as one homogeneous mass with the same temperature throughout the room. Values of temperature gradients at ceiling and floor surfaces, constant or time dependent, can however be added to the average room air temperature, thus taking vertical stratification into account. These values are added only for the determination of the convective heat transfer at these surfaces.

4.1.5 Heat sources in a room

A nominal heat gain from people is assumed to be valid at 20°C. The heat is thought to decrease linearly with increasing room air temperature, reaching a zero value at 40°C. Temperature dependence can be suppressed by giving a negative input value; the absolute value of the gain will then be used in the calculations. Note that latent gains are not treated by the program.

Heat gain from lights is normally treated as emitted by a typical unventilated, fluorescent ceiling light. BRIS will then distribute the gain evenly over the ceiling, assuming 20 % as visible light, 25 % as long wave radiation and 55 % as convective heat. Other assumptions may be done in the input, for instance a portion of thermal energy from lights may be used to heat return air.

The standard heater used by BRIS is a radiator placed on an exterior wall below a window. It will be given a size to fit the available space, and is assumed to have negligible thermal inertia. The radiator (only one per room) gives heat to the room air by convection and to the room surfaces by radiation.

The standard heater model can be modified in the following ways:

A cooler can be represented as a heater with negative thermal output. It is normally treated as a convective unit, i.e. a unit with negligible radiation exchange with the room sur-faces. All heat is then transmitted directly to the room air, and the location of the heater in the room is insignificant.

A quantity 'surface enlargement factor' is used to govern the convective heat transfer from a heater. Its size is 1 for a convector, 2 for an entirely flat heater. Typical values are in the range 3 - 4.

The thermal capacity of the heater, in J/°C, although normally negligible, may be given a non zero value. However, BRIS will always use zero capacity when heater output is the quantity sought.

The thermal inertia of extra heavy furniture or other equipment in the room can be simulated by the heat capacity of a heater with a zero thermal output.

The output of a heater can be made dependent on the outdoor climate by use of an 'outdoor temperature factor' f, to simulate existing control systems where water supply temperature to the heater is influenced by the outdoor air temperature θ_l. The output for different air temperatures is given by a nominal output power multiplied by f.

How f depends on θ_l will be described by BRIS in the following way.

For two or three values of θ_l the corresponding values of f are given as input data. For temperatures higher than the highest or lower than the lowest of these θ_l -values the f -values do not vary; between the given θ_l -values the variation of f is linear.

4.2 Data handling

4.2.1 General information

Data for the calculations are recorded on printed forms, see Appendix A. The following types of forms are used:

0 General Information

1 Room

2 Wall or Floor/Ceiling

3 Facades plus Room Dependent Data

4 Profiles of Time Dependent Data

5 Air Handling

6 Ventilated Floor Slab (TermoDeck)

7 Auxiliary Output Control (provisional)

The forms may be used in many different combinations. Data from the forms are entered into a data file. Any suitable editor may be used. When old cases have to be revised, new forms may be completed or the old forms updated, but the normal practice is to note the changes in an input data listing and then update the data file directly via the editor.

User's Manual for BRIS (*Bring*, 1985) may be referred to for detailed information about using the forms. However, some survey of the data handling characterizing the program will be given here.

Data must be expressed in SI-units. The unit of time is hours instead of seconds and thus the unit of energy is Wh rather than Joule (1 Wh = 3600 J). However, specific heat capacity is measured in J/kg°C.

Flexibility has been a goal during the development of the BRIS program, to make calculations possible for a wide variety of buildings and installations. The different forms are linked via cross references in the forms themselves, and the combinations of forms making up a particular case can vary within wide limits.

Different box styles in the forms are used to indicate data classes. For values that always remain constant during the calculation, single line boxes are used, and for values that may vary in time double line boxes are used. In the double line boxes there could be given a) a constant value or b) a reference to a table describing the profile of the variation, as defined in a form of type 4.

All rooms, walls, windows etc. are indentified by numbers. Each type of element has a separate numbering, from one and upwards. All individual room surfaces (corresponding to walls, ceilings, floors, doors, windows and possible lights and heaters) are numbered separately for each room; every room has a surface number 1 etc. This numbering is local, in contrast to the others.

The following restrictions apply to element numbers:

Type of element	Highest number permitted
room	10
wall	60
facade	x
window	x
effective temperature	x
profile of time dependent data	50
room surface	18

x The sum of the highest numbers assigned for rooms, walls, facades, windows, effective temperatures, and profiles must not exceed 200. The total number of room surfaces in the data file must not exceed 100.

As seen in Appendix A, the form type 0 for GENERAL INFORMATION contains data for the building as a whole. Examples are calculation accuracy, length of simulation, output control, outdoor climate.

The length of time to be simulated for a periodic calculation is specified by the calculation period. The same period is simulated repeatedly, in general until a recurrent state has been reached according to a convergence criterion described below. The typical period length is 24 hours, but in principle any period length can be used, a few hours or several days. The simulation can also be aperiodic, i.e. performed just once for an arbitrary length of time, for instance a winter or a summer period, or an entire year. A steady state case can also be run, treated as a periodic case, with calculation period put equal to the calculation step (and print step).

Calculation step is the interval between the successive time coordinates where the program makes complete heat balance calculations. The choice of intervals is based on considerations of accuracy and computer time. Typical intervals are one hour or half an hour. The shorter value is suitable when the case contains thin wall layers (around ten millimeters or less).

Printing step is the interval at which output is produced. It can be longer than the calculation step. Output may also be printed for specified times only.

The difference equations used by BRIS to represent the heat balance equation are solved for each time step by iteration, as mentioned in Chapter 2. In each iteration, the variable values are changed systematically to bring them closer to the solution. The difference between two successive sets of variable values are compared with a relaxation tolerance. When all changes are less than the tolerance, the iteration is stopped and the calculation proceeds to the next time step.

Since the different variables in a calculation may be of very different magnitude, it is reasonable to strive for some type of relative accuracy. In BRIS, this has been solved by calculating a dimensionless change for each variable, by the formula

$$\frac{\text{ABS (new value - old value)}}{\text{20 + ABS (new value)}}$$

The value 20 represents a typical magnitude for most calculated temperatures. The dimensionless approach will avoid excessive accuracy demands when loads or air flows take on large values.

In a periodic simulation, the whole calculation period is iterated. For each new period, the calculation is repeated at all the time steps in the period. At the end of a period, each variable value is compared to the value of the same variable at the end of the previous period. A recurrent state is deemed to be reached when all the differences observed fall below the period tolerance. (The value of the period tolerance is irrelevant in an aperiodic simulation.)

The dimensionless expression described above is used, also when checking for period convergence. To make period convergence possible, the period tolerance must obviously be larger than the relaxation tolerance.

When starting a simulation, all temperature variables are often given the same initial value. This value should be chosen close to the mean temperature of the building. The number of periods required to reach a recurrent state will depend on how well this value was originally estimated. A carefully considered start value is especially significant for heavy constructions with large thermal inertia.

Energy totals can be calculated separately for different subperiods of each simulated day. Examples of interesting subperiods are: business hours/after hours, ventilation on/off, etc. Up to six different subperiods can be chosen.

As shown in form type 0, the output of results from a simulation run can be controlled by using control codes. A complete printout is shown in Appendix B and is discussed in 4.2.7.

In the remaining part of form type 0 climatic input data are to be entered. They are treated in 4.1.1.

4.2.2 Room data

Data for rooms are entered in forms type 1 for ROOM. If you want to study several different and interrelated rooms, e.g. a corner room and an adjacent room, one form of type 1 is to be completed for each room.

The wall, ceiling and floor surfaces have fixed numbers: 1 - 6 (as shown in a layout on the back of the form). Subsurfaces may be introduced as parts of these main surfaces to show the locations of windows, wall heaters etc. Width and length of all surfaces and their reflectivities are to be noted in this form.

In form type 1 the flow rate and temperature of the infiltration air are also entered, together with heat gains from people and lights, the code number of the quantity sought, and the four quantities that can be sought: room air temperature, supply air temperature, supply air flow rate and heater output. Guidance in selecting appropriate code numbers for quantity sought and for restrictions, if any, is provided in Section 4.2.5.

All these quantities, including the flow rate of infiltration air, i.e. the spontaneous ventilation in the room, can be regarded constant or time dependent. During fan operation time the infiltration can sometimes be disregarded or treated with the forced ventilation.

Heat emitted from people and lights and from heaters in a room is treated in Section 4.1.5.

The handling of supply air is studied in Section 4.2.4. The supply air temperature is the temperature of the air reaching the room, after all processing. See Section 4.2.5 for discussion of quantity sought.

4.2.3 Wall and floor structure data

Each section of a wall or floor slab (below referred to as wall) is described in a copy of a form type 2 for ENVELOPE SECTION. The section lists a sequence of homogeneous layers making up the wall. One of the sides of the wall will always be a wall surface in the room under consideration; the other side, the outside, may be a wall surface or a facade. In this context, facade denotes a surface where radiation and temperature conditions are known and supplied as input data. One of two alternative lines of data boxes in the form are to be entered, depending on whether the outside is a facade or a wall surface.

The different homogeneous layers (solid materials or air spaces) of the wall are to be listed in a table in the form. Layer thickness, thermal conductivity, density and specific heat capacity of the materials are given here. If standard materials are used (brick, concrete etc.), they can be specified by codes. The material properties that are used in that case are given at the bottom of the form.

Material layers will often have to be divided into thinner <u>sublayers</u> during the calculation. Too thick sublayers will mean loss of accuracy; thin layers will increase the accuracy but will require the time step to be kept short to avoid oscillations.

A reasonable balance between conflicting needs for accuracy and short calculation time can be found using time steps of 0.5 or 1 hour. With a time step of 0.5 h, suitable sublayer thicknesses are 0.04 - 0.05 m for concrete and rock wool, 0.03 - 0.04 for brick, and 0.025 - 0.03 for wood. In thick walls, sublayers can be made thicker without difficulty. Suitable sublayer thickness varies with the square root of the time step. Normal slabs of glass, plasterboard and particle board are not split into sublayers. Sublayer thicknesses of the standard materials are chosen by BRIS.

4.2.4 Air handling control

The form type 5 for AIR HANDLING should be used when it is assumed that the heating and cooling capacity available for air handling is not unrestricted, i.e. when the prescribed or calculated supply air temperatures may not be achieved at all times. Otherwise, the data given in form type 1 are sufficient for the simulation.

In the simple case when the current room gets air from another room, only the numbers of the two rooms have to be introduced in the form 5. Other boxes in the form are meant for the normal case when the room is supplied by air from an air supply system.

Temperature rise of the supply air due to fan operation is specified, and also possible temperature changes in the supply air duct system, resulting from leakage or bad insulation.

System code numbers specify which system type is to be applied: recycling, recovery, recycling and recovery, or evaporative cooling, when either of these features is used. In these cases, temperature rise due to fan action on return air is also entered.

In the alternative of a recycling system, the minimum proportion fresh air is specified. This proportion may be held constant, but will often vary during the calculation period due to occupancy variations. In such a case BRIS will assure that the fresh air rate will vary from the specified limit and upwards. With this way of operation, satisfying comfort can be achieved with a minimum of energy demand.

In the case of a recovery system, the maximum temperature efficiency of the heat exchanger is specified in the form. The program will assume that the efficiency is variable from zero up to the limit specified, and best possible comfort with a minimum of energy demand will be achieved in this way.

An evaporative cooler is assumed to operate at fixed cooling efficiency when the cooling is on. When such a cooler is off, it will act as a heat exchanger in a recovery system. For this case the dewpoint of the outdoor air must also be given.

Available power for heating and cooling of supply air is introduced in form 5. By code numbers, constant or time dependent, the periods are specified when heating and cooling may be used, or when a maximum temperature difference between the room air and the supply air is required for comfort reasons. This temperature difference is also given in form type 5. Comfort restrictions on the supply air can also be introduced as fixed temperature limits, independent of the room air temperature, in form type 1. Both the temperature limits and the code for quantity sought could be used to vary the impact of the restrictions during the day.

If the aim of the simulation is to estimate the maximum capacity demand for heating or cooling, a large value that is certain to be sufficient under worst conditions may be entered in the form type 5. BRIS calculates then the cooling or heating capacity if requested, by use of quantity sought. The power can also be assigned proposed realistic values, in order to study room climates resulting from intentional undersizing.

It is often desirable, for instance in office buildings, to use time limits for fan operation, during both heating and cooling seasons.

During the heating season, the fans can be run at night with 100 % recycling with the motive of redistributing excess heat stored in sunlit zones during the day. The fan operation would be discontinued when temperatures in the warmer zones fall below a certain limit. Another scheme is to stop the fans after working hours and start them again (with heating) if temperatures drop too low. They may be stopped once again if the temperature then rises above a limit.

In hot weather, it is often desirable to use cool night air for ventilation, until the room air temperature has fallen sufficiently. When the fans have been stopped, energy released from the envelope may possibly heat the room air so that the fans would need to be started again.

Only fixed limits of the room air temperature may be used to control the fan operation. In hot weather, the switch-on limit would be higher then the switch-off limit. In the heating season, both alternatives are possible.

4.2.5 Use of the concept 'quantity sought'

With the aid of quantity sought it is possible to make the most of a BRIS simulation. The full flexibility of the program can then be reached. Quantity sought, first mentioned in Section 4.2.2, is given a detailed, more exhaustive description in User's Manual, with many examples referring to realistic cases. The principle of its use will be shown briefly below.

The following four variables in the heat balance equation of the room air have been selected as suitable 'unknown' quantities in BRIS. Each of them is specified by a code digit:

Quantity Sought	Code
T_r = Room air temperature	1
T_i = Supply air temperature	2
G_i = Supply air flow rate	3
H = Heater output	4

In simpler calculation problems, a single variable is chosen as the quantity sought. All the others should then be given predefined values.

If different quantities are sought at different times during the calculation period, the code digit for quantity sought valid during the time interval in question must be specified in a form type 4. For instance, if the heater output is to be sought during hours 7 - 17 and the air temperature during all other hours of the day, the code digit will be 4 during 7 - 17 and 1 during 0 - 7 and 17 - 24.

Values must be given to all quantities, also a value to the quantity that is sought. This latter value is disregarded by BRIS during the calculation of the quantity sought (unless it is used as a limit value).

It is not unusual that a limit value is specified for a quantity whose value varies during the day. During some of the working hours of a winter day, room air temperature may become higher than a lower comfort limit, owing to heat from people, light etc.

In an example, say that this lower limit is 22 °C. BRIS will then determine the heat demand to keep 22°C as long as heat is needed, i.e. when the calculated heater output is positive. When no extra heat is required i.e. when the heater output should become negative to keep 22°C, BRIS will calculate the air temperature instead of heat demand.

The limits for the variables may vary during the calculation period. When describing cases with limiting conditions, quantity sought will be specified on form type 4 by one or more two-digit code numbers. One two-digit number will be given for each variable involved. The first digit specifies the variable, as mentioned before. The second digit specifies the restriction that should be applied to the variable:

Function	Restriction code
Minimum value specified	6
Minimum and maximum value specified	8
Maximum value specified	9
Preferred value specified (values may be higher or lower)	0

When more than one quantity is sought, a code chain is formed, consisting of two or more two-digit code numbers. BRIS will treat the variables in the order they appear in the chain.

In our example the heater output had a minimum value of 0. The two-digit code to describe this is 46. If this condition fails, BRIS will calculate the room air temperature. For this quantity we had a lower limit value (of 22°C), and the code for this condition is 16, and the code chain will be 46 16. In this example, the order of the variables is irrelevant. The solution will be found also if we enter 16 46 for quantity sought.

If the heater has a limited power, it has a maximum as well as a minimum output limit. The code number will be 48 instead of 46, and the code chain 48 16. BRIS will now calculate the room air temperature if the maximum power of the heater is too low to keep 22°C in our example.

The code chain could have been written also 48 10 in this case, i.e.: Calculate the heater output within an interval and with the room air temperature at a preferred value; if no solution is found, set the heater output to its nearest limit and calculate the temperature.

The room air temperature is special among the quantities sought. When we specify limits for the room air temperature they are tentative restrictions, desirable to keep for comfort reasons. Limits for other quantities, on the other hand, normally correspond to physical restrictions which can not be exceeded.

The above mentioned difference in kind, between the room air temperature and the other variables, is reflected in the calculation strategy in BRIS. If the specified set of restrictions can not be upheld, BRIS will find a solution by relaxing the restrictions on the room air temperature.

4.2.6 Ventilated floor slab (TermoDeck)

Ventilated hollow core concrete floor slabs utilized according to the TermoDeck patent, can be simulated by BRIS (see the list of references in Introduction: *Andersson et al. (1979)*).

In the simplest case, there is a single 'deck' supplied with air from a separate air handling system. In other cases, one deck may supply flooring (and possibly ventilation air) to more than one room. In such cases, the deck is described as two or more separate decks. The physical unity of the deck is modelled by letting exhaust air from one deck become supply air to another.

The supply air to the hollow core slab may be supplied by fresh air and return air. Code figures in the form type 6 specifies the proportion of return air (in a recycling case) and whether the supply air is to be heated or cooled.

The return air is assumed to come from a specified room when recycling is used. When heating or cooling occurs the available capacity is supposed to be unlimited.

In case of recycling, heating or cooling of the desired supply air temperature is derived as a function of outdoor temperature, via one of three control curves.

BRIS will always try to keep the desired supply temperature by varying the recycling only, before any heating or cooling is applied. Heating or cooling will be used when required and allowed by the system code. The cooling and heating rates used can be found in the result output.

The <u>control curves</u> are used for different periods of the day. They are labelled <u>day curve, night curve,</u> and <u>morning</u>

cooling curve. The 24 hour calculation period is split into three corresponding subperiods: day, night, and morning.

The day and night curves are always used during the corresponding subperiods, whereas the morning subperiod is intended for optional adjustment, depending on prevailing conditions. Thus, the night curve continues to be used during the morning, until an optional switch is made, or the morning changes to day. The switch can be made, either to the morning cooling curve, or to the day curve.

The decision to switch over from the night curve is based on a reading of the exhaust temperature from the deck. This temperature is compared with a reference temperature, which in turn is defined as a function of outdoor temperature, via a fourth control curve. If the deviation of the exhaust temperature from the reference temperature exceeds a certain tolerance, a switch in control curve is made. The morning cooling curve is selected if the exhaust temperature is too high and the day curve if the exhaust temperature is too low.

The entire morning ritual can be suppressed by making the tolerance high. This is normal practice in the heating season.

Three points are used to describe each curve. The subperiods will be chosen in such a way that they follow each other in the intended order: day, night, morning, day, ...

When more than one TermoDeck is simulated, BRIS will use the same curves for all decks. The curves need thus only be specified once, and the curve definition part can be left blank for the other decks.

The effective heat transfer, taking place between the duct surfaces and the air ventilating the ducts, will be described in the form by two parameters: Film coefficient and Effective surface.

Recommended values are based on laboratory measurements and subsequent experience with installed TeromDeck systems. They can only be obtained from Strängbetong AB, Stockholm.

4.2.7 Result output

4.2.7.1 Exemple

In Appendix B, an example of a complete printout from a BRIS run is given. It can be seen that the input data (pages B.1 and B.2) are echoed to the output in the same order as they are filled in on the forms (Appendix A). Headlines are entered in this input account to facilitate the reading.

From the heading on each result output page (pages B.3-B.9) it can be seen that the room simulated is an office facing south with a concrete floor slab. The day considered is a day in July, and cooling was used, at a maximum rate of 500 W.

Input data entered in accordance with Form type 0 show that the steps used were, 0.5 h for the calculation and 1 h for the printing. The simulation was made for a 24-hour period, using tolerances of 0.001 for the relaxations and 0.003 for the period. Climatic data are valid for July 15 and for Stockholm, with daylight saving time used (time meridian 22 instead of 23 and outdoor temperature maximum at 16 hours). The horizon was unobstructed.

The first line of data for the room shows the room dimensions. The 'K' at the beginning of the second line denotes that the temperature of the infiltration air is the same as that of the outdoor air. The quantity sought and gains from people and lights are time dependent, varying during the day. They are specified via profiles in Form type 4. We give them numbers 1, 2 and 3 (T1, T2 and T3), respectively.

On the third data line, for quantities that may be sought, the only specified values are the temperature and the flow rate of the supply air. The temperature, 15°C, is a minimum value, a fact that can be seen from the value of T1 at the top of page B.2 showing a code number 26 of the quantity sought for the whole 24-hour period (2 stands for supply air temperature and 6 for minimum value specified, according to 4.2.5). The flow rate, 180 kg/h, is constant during the period. The other quantities are indicated by 'N' only, stating that they are not given any values during the calculation. The cooling is thought to be furnished to the room by the supply air and not by a cooler in the room. Otherwise the power rate should have been entered under 'Heater output', with a negative sign.

All properties of wall and floor slab material layers are described by codes, explained in Form type 2, Appendix A.

On the data line specifying window parameters in conformity with Form type 3, values on the shading coefficients F1 and F2 and the transmission coefficient U have been taken from Appendix C. They are constant, thus valid for the whole 24-hour period, and in accordance with coefficients of a triple glazed window with Venetian blinds between external panes.

Effective temperatures for two points in the room have been requested, one directly in front of the window and 1 m from it (No. 2), the other in the middle of the room (No. 1). They are requested during that part of the day when people are present, and reference is therefore made to profile T2 on Form type 4.

We will now take a look at the data concerning air handling. They are arranged according to Form type 5 at the bottom of the input list. We see that the source of the return air is room No. 1, the return air fan gives an air temperature rise of 1°C, and recycling is used (system code figure is 1). The code for minimum fresh air is time dependent (T6).

On the next line: The difference between the room air temperature and the supply air temperature must not be larger than 8°C (for comfort reasons), available capacity for cooling of supply air is 500 W, no heating of the air is used, and the code for air handling control is time dependent (T7).

Next line: The control period for the supply air fan operation is time dependent (T4); fan operation is optional between 17 and 8. If the fan is running when the fan operation is optional, it will stop when the room air temperature falls to 21.5°C and start again when the temperature reaches 23.5°C (outside the optional time period the fan will be running continuously).

Next line: The supply air temperature will rise 1°C due to fan operation and 0°C due to leakage or bad insulation in the duct system.

Among the time dependent data, we have already treated T1, the code of quantity sought (which in fact is not time dependent here because the code number 26 is valid during the whole calculation period). T2 shows that heat gain from people is 200 W from 8-11 o'clock, 100 W from 11-13 and 200 W again from 13-17. Here it has been assumed that the gain from one person is 100 W (at 20°C air temperature). The gain from lights is 240 W between 8 and 17, according to T3.

The operation of the supply air fan is optional only from midnight to 8 o'clock and from 17 to midnight, indicated by a non-zero value for T4 during that period. From 8-17 the value is zero, and the fan is always running.

In T5 is shown that an account of energy requirements is wanted for two sub-periods of the day, one for the time between 8 and 17 and another for the rest of the day.

In T6, it can be seen that a value of 30% is chosen for the minimum proportion of outdoor air in the supply air, valid between 8 and 17. No conditions are made for other parts of the day.

Finally, in T7, a code for air handling control is specified for the time interval 8-17. The code number, 120, is a sum of two numbers: 20 and 100. Here 20 means that cooling may be used, 100 that comfort restriction applies. In this case the restriction is that the difference between the room air temperature and the supply air temperature must not be larger than 8°C.

Output data. The first page, B.3, confirms room dimensions and climatic data given on page B.1 and B.2, and continues by showing calculated outdoor air temperatures and values of radiation on the facade and through an unshaded window with two panes of ordinary glass. The page ends listing the room surfaces with their numbers and areas, and numbering corresponding walls and windows.

The next page shows the diurnal variation of room air and supply air temperatures, supply air flow rate and energy input by supply and infiltration air. Further, gains from people and ligths are listed, and in the last column the rate of energy transferred by conduction through the surface layers of all the walls and the floor slab together. These gains and the heat transferred through the room sur-faces are thermal energy quantities brought to the room (positive values) or removed from the room (negative values), some partially by radiation, causing no heating or cooling of the room air.

On page B.5 the following variables are listed: outdoor temperature, recirculated ventilation air (%), thermal energy (W) required for supply air heating or caused by fan operation, thermal energy (W) required for supply air cooling, effective temperatures (two in this example) including comfort rating values (defined in 4.1.3).

The two following pages give surface temperatures. Page B.6 shows the temperatures of the six wall and floor slab surfaces in the room, while B.7 shows the temperature of the innermost window pane. Additionally on page B.7 the radiation I_{tot} and the heat transfer - P_{out} are listed, both defined in Section 4.1.2. Their sum is the contribution of the window heat transfer to the heat balance of the room.

The temperatures of the facade surface and the surfaces between the different material layers in the walls are listed on page B.8.

The last printout page, B.9, begins with an account of the percentage distribution of the effective temperatures during that period of the day when the room is occupied. Mean values for this period are also given (mean values for the entire 24-hour period are given on page B.5).

Page B.9 also shows 'Energy totals' for subperiod 1 (when the room is not occupied), for subperiod 2 (when the room is occupied) and for the entire 24-hour period. It can be observed that values for the entire day can be obtained by multiplying the corresponding power mean values on pages B.4, B.5 and B.7 with the factor 24/1000. The mean room air temperature for the day is also given on page B.4 with two decimals.

Further, it can be noticed that 'Heater' indicates energy required for a space heater. 'Air-cool' specifies the calculated heat removed from supply air, not the electric energy required to achieve this cooling.

In the heat balance of the room, under the headings 'Sun' and 'Window', values of I_{tot} and $-P_{out}$ for the periods in question are given.

Finally, from the column 'Change' one can observe the number of days, in this case five, for which the calculations have been executed before the specified period tolerance was reached.

Let us now study the results of the simulation. Looking at figures on page B.4, one can see that the supply air rate is zero between hours 3 and 6 in the morning. This is caused by the condition for fan operation when the room is not occupied: switch-off when the room air temperature falls below 21.5°C and switch-on when the temperature 23.5° is reached again. The reaction of the air temperature can be observed from the column 'Room air' on page B.4, but more clearly from the graph in fig. 4.2.

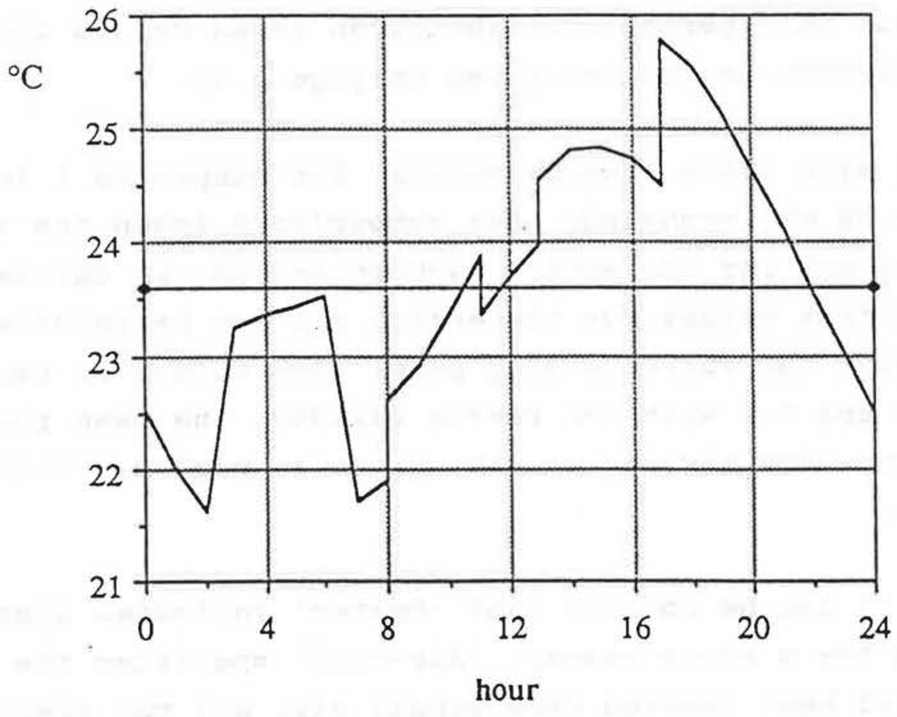


Fig. 4.2. Room air temperature.

The second of the two lines for 8 o'clock in the printed tables concerns the conditions since the room has been occupied. The cooling of the supply air begins at eight, see page B.5. The cooling rate, however, is less than the maximum value 500 W. More cooling would have caused a lower supply air temperature than 15°C. This temperature was given as a minimum in connection with the code number 26 for quantity sought.

This limitation is replaced at 10 o'clock by another comfort restriction that states that the difference between the room air temperature and the supply air temperature must not be larger than 8°C. The maximum cooling rate of 500 W is not used until 15 o'clock, when this temperature difference restriction no longer applies.

From page B.5 can be seen that 70% of the ventilation air is recirculated at 16 and 17 o'clock. At this time of the day the outdoor air is warmer than the indoor air, and a lower, and thus more comfortable, indoor temperature will be achieved by recirculation with the same power input. The stipulated minimum proportion of 30% of fresh air is applied, however.

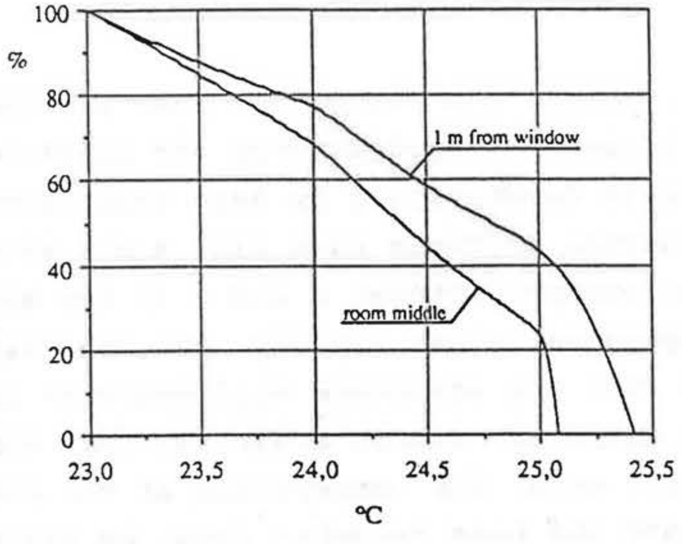


Fig. 4.3. Percentage distribution of effective temperature.

When the room is occupied, the room air temperature reaches its peak value, 24.85°C, at 15 o'clock, and so do the two effective temperatures, with 25.08 and 25.42°C. The highest effective temperature is that closest to the hot window pane (see fig. 4.3). Both are influenced by the temperatures of all the room surfaces, which are a little warmer on average than the air temperature, and therefore the effective temperatures are a little higher than the air temperature.

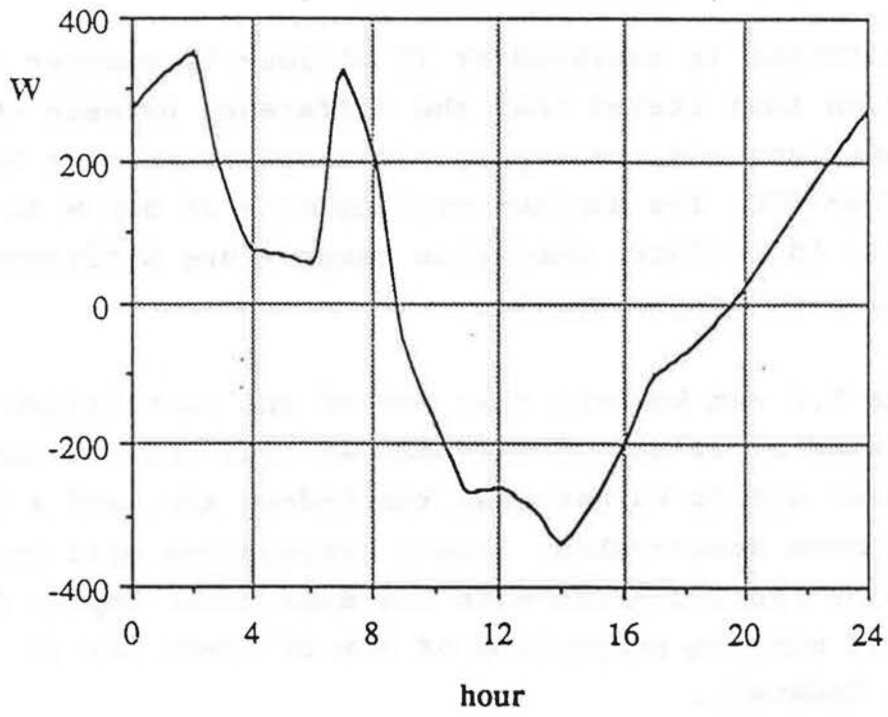


Fig. 4.4. Heat conducted through surface layers of walls and floor slabs.

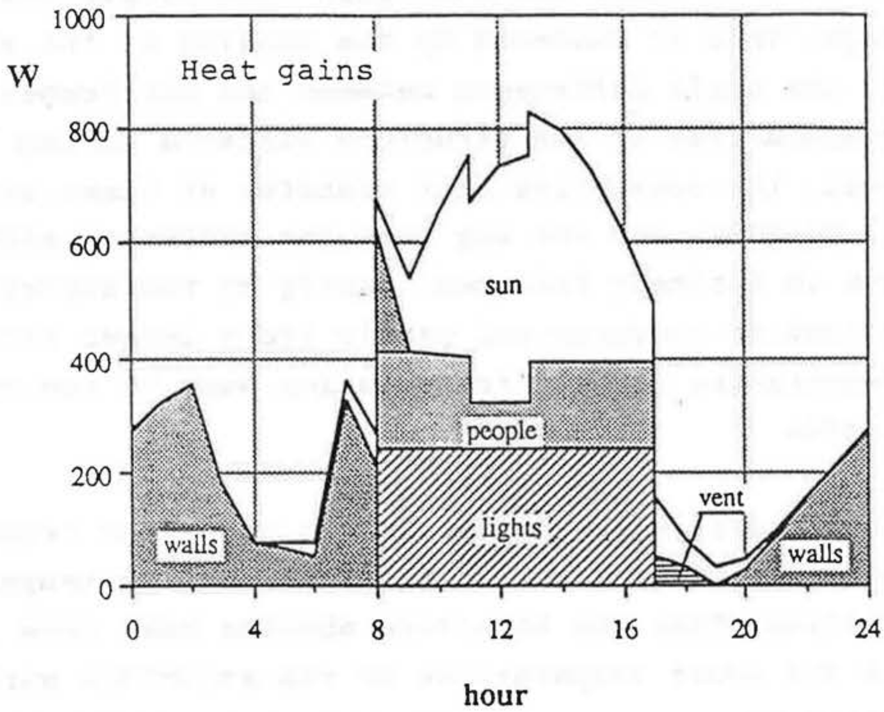
During the night, before 3 o'clock, the room air temperature is falling because of cold outdoor air supplied by the ventilation. This is counteracted by heat transferred to the air from the structure surfaces (see fig. 4.4). After the fan operation has stopped between 2 and 3 in the morning, the room air temperature rises rapidly, and the quantity of heat transmitted from the structure will decrease up to 6 o'clock. The fan operation starts again. At that time the supply air is still cold, the temperature of the room air falls promptly, and the heat transfer from the structure is considerable.

From 8 to 17 o'clock the heat gain from people and lighting is large. This is balanced by the cooling of the supply air. Due to the small difference between the air temperature and the temperatures of the structure surfaces during this time interval, the convective heat transfer at these surfaces is small. Roughly, one can say that the radiation energy from the sun is entirely balanced, partly by the absorption at the structure surfaces and partly (to a lesser extent) by the temperature induced transmission back to the outside of the window.

In other cases, where no cooling is used, the temperature of the room air will be higher than the surface temperatures in the daytime. Then the structure absorbs heat from the air. Due to the lower temperatures of the structure surfaces the effective temperature (measured not too close to the window) will be lower than the air temperature.

The heat balance of the room, for the subperiods 1 and 2 and for the entire 24-hour period, can be studied in the tabular statement on page B.9. One can notice that the sums of the positive items (heat gains) and the negative items (heat losses) balance each other strictly, as would be expected.

Fig. 4.5 shows how heat gains and losses vary during the day, and how they are composed of the different components of heat gain. Gains and losses show some discrepancies, especially at times of strong load changes.



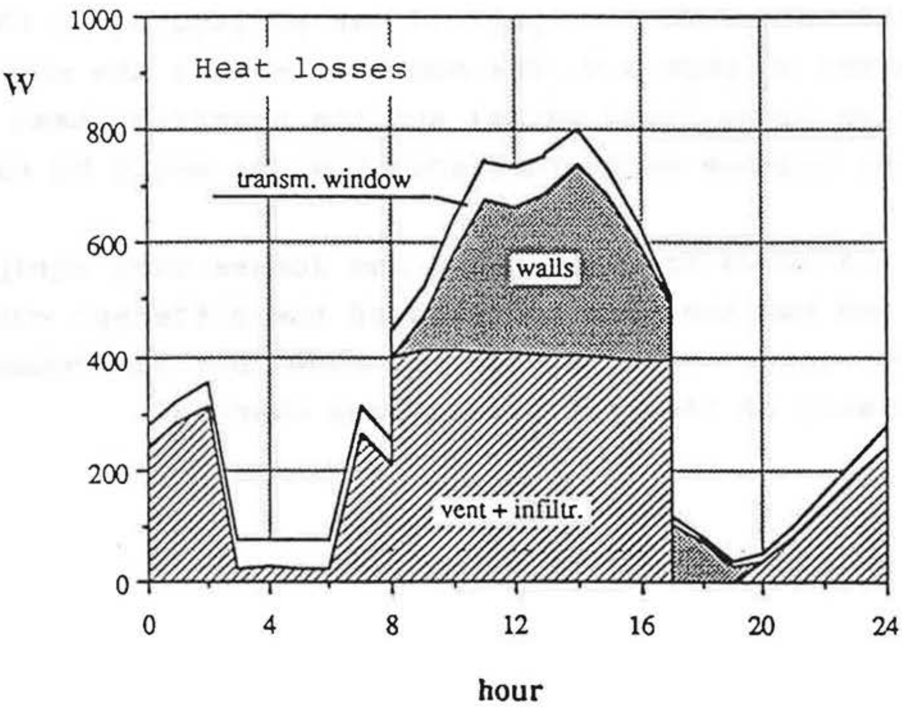


Fig. 4.5. Diurnal variation of the heat balance parameters for the room.

4.2.7.2. Frequencies for heating and cooling powers

A file can be produced by BRIS to yield frequency tables of heating and cooling power values. Load from heaters/coolers in the rooms are included as well as loads from heating/ cooling and fan operation in the air handling. Heat from lighting is not included. Heating and cooling rates are shown separately. The file is obtained if a special output control code is included in the Form type 0 for General information. Values are sampled for each hour during the entire calculation period.

Tables are printed for the entire calculation period and for individual months. The values in the tables specify the number of hours during which different heating or cooling rate values have been reached or exceeded during the relevant period.

The heating/cooling rate intervals to be used are selected by BRIS in such a way that the total number of intervals does not exceed fifty. The interval size is selected as a suitable power of ten multiplied by one, two or five.

Days and nights are presented separately and added together. The separation of days and nights could be useful, e.g. when reduced power prices apply at certain times. A Form type 7: Auxiliary output control has to be used, and the two hours marking the transitions between day and night are to be entered there.

Reason may exist to include weekends in either days or nights depending on type of building simulated (office/ residential), schedules of power prices, etc. Code 1 entered in the form makes weekends group with days, while code 2 makes weekends group with nights.

4.2.7.3 Plot file

This is an optional file that contains data intended for subsequent plotting. It is produced if a special output control code is included in the form for General information.

The format is not adapted to any particular plotter or plotter program. The user must do any editing required to suit particular plotting facilities.

BRIS

Form type 0 for GENERAL INFORMATION

Form code
0

Heading (2 mandatory lines of max 72 chars each)

' '
' '

Calculation step h (1)	Printing step h (1)	Calculation period, h 0=aperiodic (24)	1) Calculation length for periodic or aperiodic calculation	Relaxation tolerance, relative (.001)	Period tolerance, relative (.002)	2) Outdoor air temperature °C	3) Start value °C	Sub-periods for energy

4) Output control codes. Enter -1 to get operator dialogue.

Hours when to print. Enter -1 to get regular printout with 'printing step'.

Climatic data (remaining part of form may be omitted)

Day	Month

Observed climatic data

Name of climate file
' '

5) or Synthetic climatic data

Latitude	Longitude	Time meridian	Reduction factor (0-1)	Outdoor temperature mean °C	amplitude °C	top at, h

Radiation

Facade outside main surface Room no.	Surface no.	6) Azimuth	Horizon (0-90°)	Ground reflectivity (0-1)

1) Calculation length in design day calculation
Max number of days
alt 1. Enter 0= Iterate until periodic
alt 2. Enter n= Discontinue after n days
alt 3. Enter -n= Dialogue after n days.
For aperiodic calculation give total length
n= number of days, -n= number of hours

2) Enter K to refer to climatic data in this form

3) Start value
alt 1. Enter normal value = Approximate mean tempereture
alt 2. Enter -1000 = Dialogue, file name or constant value requested at run time

4) Standard output codes (add 10 for profiles)
0= Energy summary
1= Energy summary + room variables
2= Case 1 + boundaries in walls
3= Case 2 + all wall layers
4= Case 3 +more decimals

5) Example: Stockholm = Latitude 59.3, Longitude -17.9, Time meridian 23

6) Azimuth examples:

South	= 0	or S
Southeast	= -45	or SE
Southsouthwest	= 22.5	or SSW

Calendar (relevant for aperiodic calculations only)

First Sunday day number

Day numbers for holidays

98	End marker

BRIS

Form type 1 for ROO[illegible]

Form code	Room number	Width x m	Depth y m	Height z m	Multiplier (1)	Room name (optional) max 15 chars
1						'

Infiltration 1) Temperature °C	Infiltration Flow kg/h	Quantity sought	Gain from people W	Gain from lights W	2) Room air gradients $dT_{r,floor}$ °C	2) Room air gradients $dT_{r,ceiling}$ °C

Code (xy) for quantity sought

x = Variable code	Variable
1	room temp
2	supply air t[illegible]p
3	supply air flow
4	heater output

y = Restriction code	Function
6	Minimum
8	Interval
9	Maximum
0	Preferred V[illegible]e

Quantities that may be sought

x=1 Room air temperature °C Value 1	x=1 Value 2	x=2 Supply air temperature °C 1) Value 1	x=2 Value 2	x=3 Supply air flow kg/h Value 1	x=3 Value 2	x=4 Heater output W Value 1	x=4 Value 2

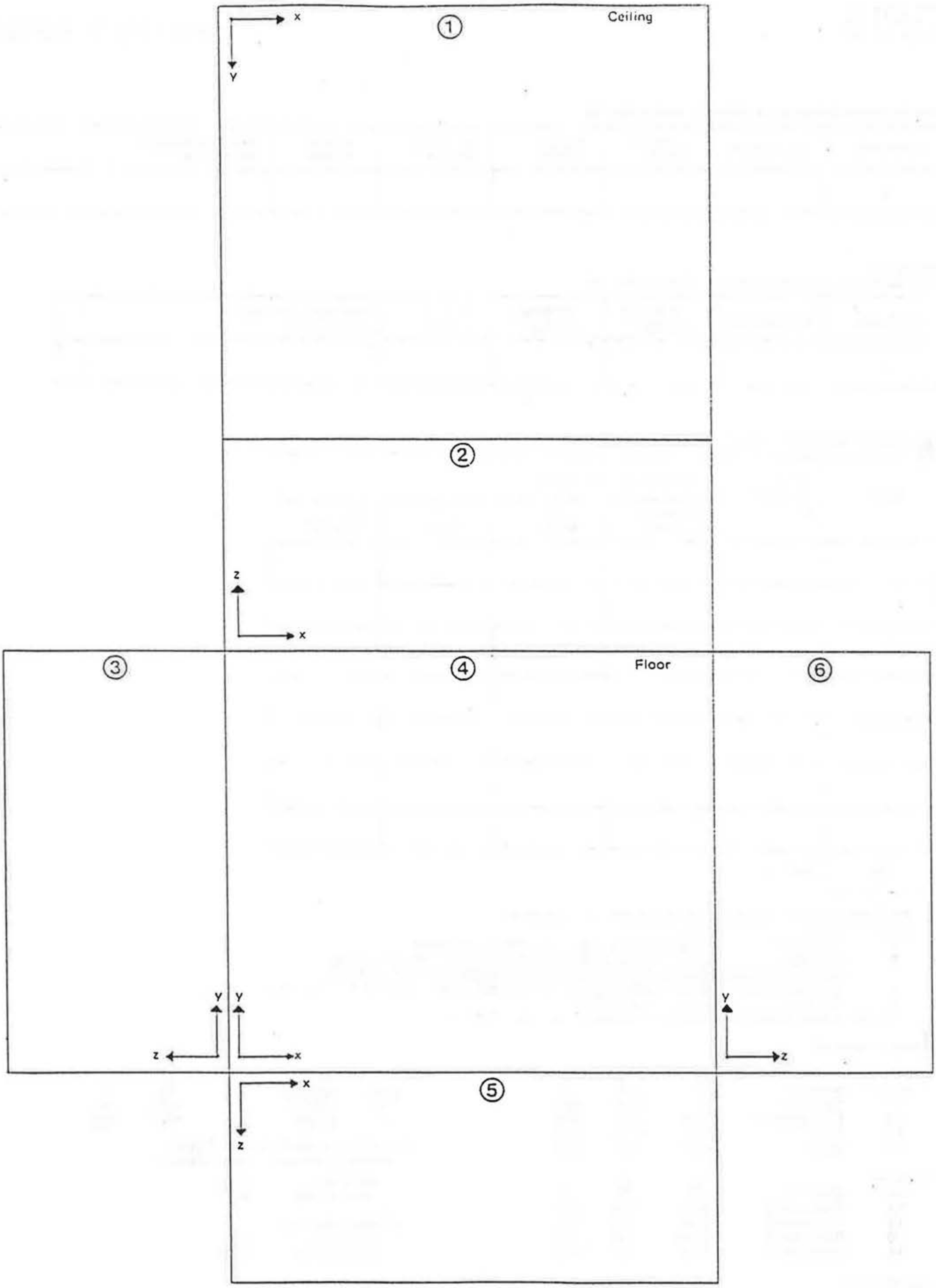
Reflectivities for main surface 1-6

Surface 1 (0.7)	Surface 2 (0.6)	Surface 3 (0.6)	Surface 4 (0.3)	Surface 5 (0.6)	Surface 6 (0.6)

Description of sub-surfaces, parts of main surfaces

Sub-surface no	Part of main surface no	Corner closest to origo x m	y m	z m	Size x-direction m	y-direction m	z-direction m	Reflectivity
98	End marker							

1) Enter K to refer to climatic data in form 0
2) Leave blank when not relevant



Notes: The layout shows the room surfaces with the main surfaces numbered. All subsurfaces must be rectangular, and may not overlap or cover each other. The position of a sub-surface is defined relative to the coordinate system local to the surrounding main surface. The coordinates of the corner closest to origo are specified.

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Form type 2 for ENVELOPE SECTIOI

Internal section bordered by surface (a) and surface (b)

Form code	Wall number	(a) Room number	Surface number	(b) Room number	Surface number	Wall name (optional) max 15 chars
2						' '

Alternative:
External section bordered by facade (a) and surface (b)

Form code	Wall number	(a) Facade number	(b) Room number	Surface number	Wall name (optional) max 15 chars
2					' '

The envelope section described from (a) to (b)

Code 1)	Total Thickness m	For layer code =1, enter:			
		Number of sub-layers	λ W/(m°C)	ρ kg/m³	c_p J/(kg °C)
98	End marker				

1) The different layers in an envelope section can be described by:

Code	Type	Note
1	Solid layer	Enter thickness, sub-layers, thermal properties
2	Air space	Enter heat resistance ($m^2 K/W$) in column 'Total Thickness'
3	Symmetry surface	Appears alone on last line, enter b-side as room 0, surface 0.
4	Film coefficient	Enter (profiles for) parameters a,b,c, to get film coeff. $= a + b \cdot (\text{abs}(T_r - T_{surf}))^c$
5	TD layer	Enter slab air number
11–26	Standard material	Material described below, enter thickness

Standard materials

Structuring

Code	Material	λ	ρ	c_p
11	Concrete	1.5	2300	880
12	L/W concrete	0.15	500	1050
16	Brick	0.58	1500	840
17	Wood	0.14	500	2300

Insulating

Code	Material	λ	ρ	c_p
19	Rockwool 1	0.04	200	750
13	Rockwool 2	0.04	50	750
20	Rockwool 3	0.045	16	750
14	Polystyrene	0.04	20	1400

Wall covering

Code	Material	λ	ρ	c_p
15	Plasterboard	0.22	900	840
21	Chip board	0.14	650	1350
22	Hard fibre board	0.13	1000	1350
23	Medium fibre board	0.08	600	1350
24	Soft fibre board	0.052	300	1350

Facade cladding

Code	Material	λ	ρ	c_p
18	Render	0.80	1800	790
26	Stone	3.0	2700	880

Heat resistance for air spaces ($m^2 K/W$)

Vertical or horizontal, non reflective *

Ventilated	0.07
Unventilated	0.17

Vertical, reflective*

Ventilated	0.28
Unventilated	0.62

Horizontal, reflective*

Heat flow up (winter)	0.5
Heat flow down(summer)	1.4

*A reflective air space is bounded by at least one low emittance surface (~0.05)

Facades

Form code	Facade number	Absorptivity a	Cloud cover (0–8 for roof, 9 for walls)	1) Radiation W/m2	1) Outdoor temperature °C	Film transmission coefficient W/(m2 °C)
31						
31						
31						
31						

1) Enter K to refer to climatic data in form [C]

Form code	Room number
32	

(Mandatory if data are entered below.)

Windows

Form code	Window number	Surface number	1,2) Total radiation	3) F1 %	4) F2 %	1) Outdoor temperature	U–value W/m2	5) Parameters for customized film transmission coefficient a	b	c
33										
33										
33										

Notes 1) - 5): see overlea[f]

Lights

Form code	Surface number	Gain from light, W	Percentage distribution: 1) Long wave radiation	Visible light	To return air
34					
34					

Note 1): see overleaf

Heater

Form code	1) Surface number	2) Thermal capacitance J/K	1) Surface enlargement factor	3) Outdoor temperature factor (nominal output in form Type 1) T_1	f_1	T_2	f_2	T_3	f_3
35									

Notes 1) — 3): See overleaf

Effective temperature

Form code	Effective temperature number	x m	y m	z m	Profile for time restriction
36					
36					
36					
36					
98	End marker				

Notes on form code 33

1) Enter K to refer to climatic data in form type 0
2) Total transmitted radiation through two panes of ordinary glass, W/m^2
3) Total transmitted radiation through the actual window combination, as fraction of double pane radiation above, %
4) Direct transmitted radiation through the actual window combination, as fraction of double pane radiation above, %
5) Customized film coefficient = $a + b * (abs(T_{room} - T_{window}))^c$
   Leave blank if not applicable.

Notes on form code 34

1) Typical values for normal fluorescent light

	unventilated	ventilated
Long wave radiation	25	10
Visible light	20	20
To return air	0	70
Convectively to room air	55	0

Notes on form code 35

1) An entirely flat radiator has surface enlargement factor =2

   For a convective heater, enter:
   surface number = 0
   surface enlargement factor = 1

2) Enter capacitance = 0, when heater output is sought.

3) When outdoor temperature factor is defined
   H = nominal output * factor
   otherwise
   H = nominal output

   Define a curve by two or three points ($T_{outdoor}$, factor)

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Form type 4 for PROFILES of TIME DEPENDENT DATA

Form code	Profile number						
4							
From hour	To hour	1) Time step	2) Value (s)				
98	End marker						

1) Enter time step for progression of values,
or 0 for interval with constant profile,
or S for chain of codes for quantity sought

2) Enter list of values,
or constant value,
or chain of codes for quantity sought

Form code	Profile number						
4							
From hour	To hour	1) Time step	2) Value (s)				
98	End marker						

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Form type 5 for AIR HANDLING

Form code	Room number
5	

Outdoor temperature is specified in form type 0

Recycling, recovery, evaporative cooling

Source of return air (room no.)	Temperature rise of return air from fan operation °C	System code 1=Recycling 2=Recovery 3=Recycling +recovery 4=Evaporative cooling	Min fresh air / not applicable / Min fresh air / Recovery when off	not applicable / Max efficiency / Max efficiency / Cooler efficiency	not applicable / not applicable / not applicable / Dew point

Air handling

Maximum cooling sub temperature °C	Available effects (W)		Code for air handling control
	Cooling	Heating	

Fan control

Fan control period	Temperature limits(°C) in room air for	
	Switch-on	Switch-off

Temperature change

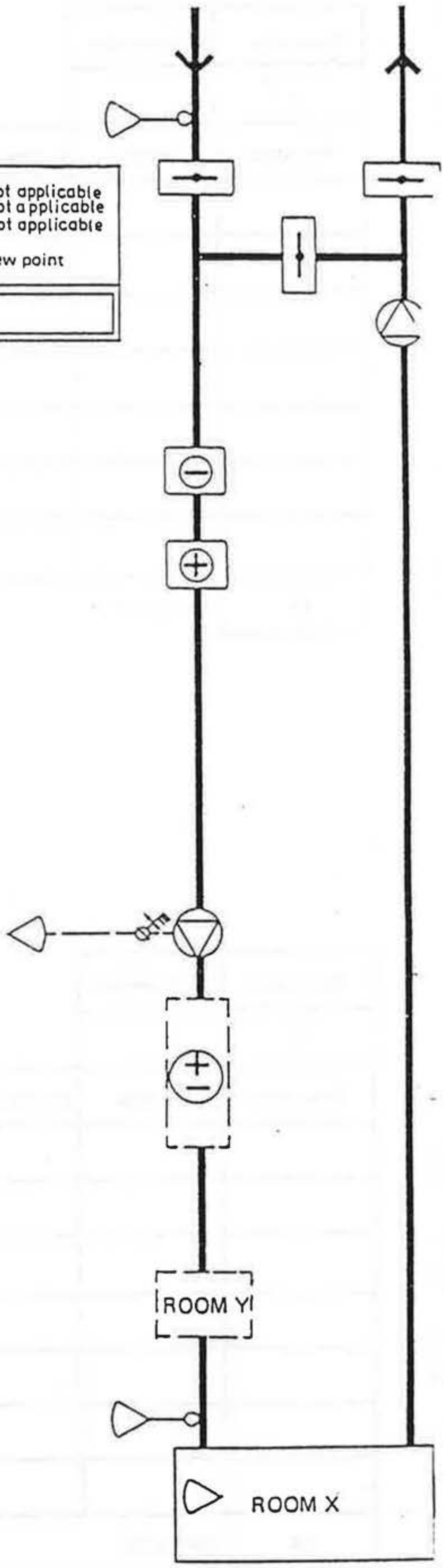
from fan operation	other change

Air from other room

Room number	
98	End marker

Supply air temperature is specified in form type 1

Room air temperature is specified in form type 1



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Form type 6 for VENTILATED FLOOR SLAB

Form code	Slab air number
6	

Notes 1) – 9) see table and figure

1) Code	9) System control
1	Fixed recycling
2	Variable recycling
3	Variable recycling + heating
4	Variable recycling + cooling

Air flow kg/h	Recycle portion %	1) Code for system control	Temperature rises °C 2) return air	3) supply air

4) Air source (slab air number or 0)	5) Source of return air (or 0)	4) Temperature of fresh air (or N)	6) Heat transfer in the ducts. Film coefficient W/(m²K)	Effective surface m²

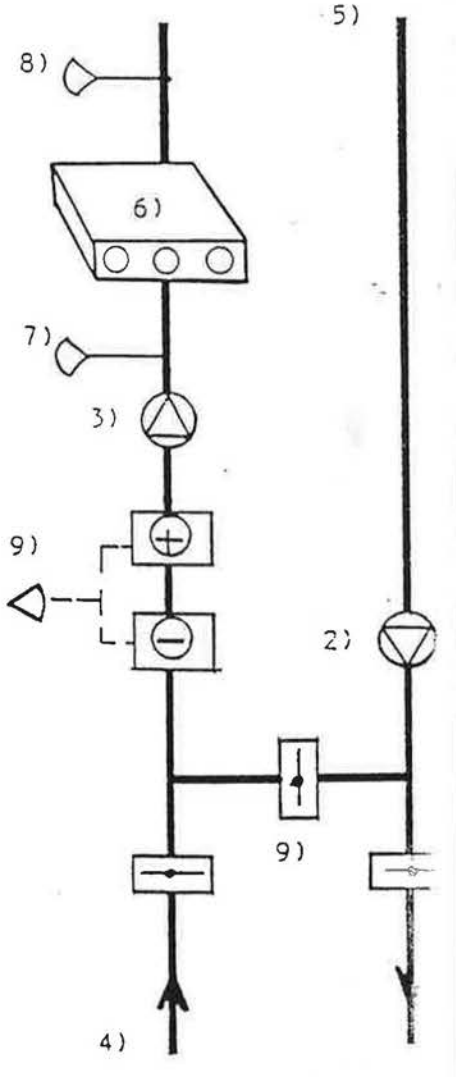
The reminder of the form may be left blank.

Control curves for supply air temperature 7)

Hours of sub-period change night–morning	morning–day	day–night	Tolerance for morning switch °C

Curve	Point 1 $T_{outdoor}$	T_{supply}	Point 2 $T_{outdoor}$	T_{supply}	Point 3 $T_{outdoor}$	T_{supply}
day						
night						
morning cooling						
reference for switch		$T_{exhaust}$ 8)		$T_{exhaust}$		$T_{exhaust}$

98	End marker



BRIS

Form type 7 for AUXILIARY OUTPUT CONTROL

Duration of heating and cooling loads

Form code
7

Hours of sub-period change		Code for w/end. 1= day 2= night
day starts	day ends	

98	End marker

```
0
'OFFICE SOUTH, CONCRETE FLOOR SLAB'
'JULY, COOLING POWER MAX 500 W'
*
*CALC.ST. PRI.ST. CALC.PER. LENGTH R-TOL. P-TOL. OUTDOOR TEMP. START SUB
 .5       1        24          0      .001    .003          K          23     T5
*
*OUTPUT CONTROL CODE
2
*HOURS TO PRINT
-1
*CLIMATIC DATA (DAYLIGHT SAVING TIME)
15 07
59.5 -18 22 0.8  20 6 16
*RADIATION
*ROOM SURF. AZI.  HOR. GRND REFL.
  1    2     S     0      0.2
98
* ROOM
1 1  3.6 4.5 2.7 1 '3 MODULES'
*
*INFILT. SOUGT PEOPLE LIGHTS
  K 10    T1     T2     T3
*
*ROOM AIR TEMP.   SUPPLY AIR TEMP.   FLOW    HEATER
 N     N              15        N       180 N     N  N
*
* REFLECTIVITY
0.7 0.6 0.6 0.3 0.6 0.6
*
* SUB-SURFACES
7 2  0.6 X 0.8  2.4 X 1.2  0.25
98
*
* WALLS
2 1  1 1 2 'EXT.WALL'
16 0.12
13 0.2
15 0.026
98
*
2 2  1 3 1 6 'PARTITIONS'
15 0.026
2 0.16
15 0.026
98
*
2 3  1 5 0 0 'INT.WALL'
15 0.026
13 0.025
3
98
*
2 4  1 4 1 1 'FLOOR'
11 0.2
98
*
* FACADE
31 1 0.8 9 K K 16
*ROOM NUMBER
32 1
* WINDOW
33 1 7 K 30 9 K 1.9
*
*EFF.TEMP
36 1 1.8 2.25 1.4 T2
36 2 1.8 3.5 1.2 T2
98
```

```
* TIME DEPENDENT DATA
* QUANTITY SOUGHT CODE
4 1
0 24 5  26
98
* HEAT FROM PEOPLE
4 2
0 8 0 0
8 11 0 200
11 13 0 100
13 17 0 200
17 24 0 0
98
* HEAT FROM LIGHTS
4 3
0 8 0 0
8 17 0 240
17 24 0 0
98
*
*FAN CONTROL PERIOD
4 4
0 8 0 1
8 17 0 0
17 24 0 1
98
*
*SUB-PERIODS FOR ENERGY
4 5
0 8 0 1
8 17 0 2
17 24 0 1
98
*
*MIN % OF OUTDOOR AIR
4 6
0 8 0 0
8 17 0 30
17 24 0 0
98
*
*CODE FOR AIR HANDLING CONTROL
4 7
0 8 0 0
8 17 0 120
17 24 0 0
98
*
*AIR HANDLING
5 1
1 1 1 T6  N
8 500 0 T7
T4 23.5 21.5
1 0
98
```

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY, COOLING POWER MAX 500 W

ROOM DIMENSIONS

WIDTH	3.60 M	VOLUME	43.7 M3
LENGTH	4.50 M		
HEIGHT	2.70 M		

WEATHER DATA

LATITUDE	59.50
LONGITUDE	-18.00
TIME MERIDIAN	22.00
DATE	15. 7
REDUCTION	.80

	FACADE 1	WINDOW 1
AZIMUTH (EAST OF S)	.0	.0
GROUND REFL. (0-1)	.20	.20
HORIZON (DEG)	0.	0.

HOUR	OUTDOOR	RADIATION ON FACADE OR THRU 2 PANES (W/M2)	
0	17.0	.0	.0
1	15.8	.0	.0
2	14.8	.0	.0
3	14.2	.0	.0
4	14.0	.0	.0
5	14.2	13.9	9.3
6	14.8	31.6	21.2
7	15.8	50.8	34.1
8	17.0	99.7	51.1
9	18.4	250.8	124.4
10	20.0	388.3	242.7
11	21.6	496.1	343.2
12	23.0	563.1	405.2
13	24.2	582.6	422.9
14	25.2	552.5	395.5
15	25.8	476.0	324.4
16	26.0	360.7	217.6
17	25.8	218.7	101.6
18	25.2	69.2	46.3
19	24.2	45.9	30.8
20	23.0	27.7	18.6
21	21.6	10.3	6.9
22	20.0	.0	.0
23	18.4	.0	.0
24	17.0	.0	.0
MEAN	20.0	177.00	116.57

SRF NR	AREA M2	CORRESPONDS TO		
1	16.20	WALL	4	FLOOR
2	6.84	WALL	1	EXT.WALL
3	12.15	WALL	2	PARTITIONS
4	16.20	WALL	4	FLOOR
5	9.72	WALL	3	INT.WALL
6	12.15	WALL	2	PARTITIONS
7	2.88	WINDOW	1	

OFFICE SOUTH, CONCRETE FLOOR SLAB
HAV, COOLING POWER MAX 300 W

ROOM NR 1 3 MODULES DATE 15. 7

HOUR	ROOM AIR DEG C	--SUPPLY AIR-- KG/H	DEG C	PEOPLE WATT	Q-VENT WATT	INFILTR WATT	LIGHTS WATT	WALLS WATT
	✓	✓	✓	✓	✓	✓	✓	✓
	22.51	180.00	18.00	.00	-227.54	-15.44	.00	274.19
1.	22.03	180.00	16.76	.00	-265.64	-17.57	.00	320.55
2.	21.61	180.00	15.80	.00	-292.59	-19.06	.00	353.78
3.	22.25	.00	.00	.00	.00	-25.31	.00	179.90
4	23.37	.00	.00	.00	.00	-26.24	.00	75.98
5.	23.45	.00	.00	.00	.00	-25.89	.00	68.05
6.	23.50	.00	.00	.00	.00	-24.42	.00	54.29
7.	21.72	180.00	16.75	.00	-250.33	-16.71	.00	327.42
8	21.90	180.00	18.00	.00	-196.57	-13.73	.00	210.89
	22.62	180.00	15.00	173.84	-383.89	-15.73	240.00	210.89
9	22.43	180.00	15.00	170.80	-399.51	-12.52	240.00	-61.81
10	23.38	180.00	15.38	166.17	-403.28	-9.47	240.00	-171.01
11.	23.90	180.00	15.90	160.97	-403.29	-6.58	240.00	-257.89
	23.38	180.00	15.38	83.11	-403.19	-5.11	240.00	-267.89
12	22.71	180.00	15.71	81.44	-403.28	-2.00	240.00	-259.85
13	23.99	180.00	15.99	80.04	-403.26	.70	240.00	-284.57
	24.55	180.00	16.55	154.52	-403.20	-.86	240.00	-284.57
14	24.81	180.00	16.81	151.95	-403.25	1.09	240.00	-337.64
15	24.85	180.00	15.87	151.47	-402.17	2.64	240.00	-284.07
16	24.75	180.00	16.90	152.50	-395.42	3.50	240.00	-201.09
17	24.50	180.00	16.67	154.97	-394.64	3.62	240.00	-102.31
	25.72	180.00	26.80	.00	51.31	.05	.00	-102.31
18.	25.54	180.00	26.20	.00	33.11	-.96	.00	-70.74
19.	25.15	180.00	25.24	.00	4.57	-2.55	.00	-27.10
20	24.64	180.00	24.00	.00	-32.21	-4.59	.00	26.29
21.	24.12	180.00	22.55	.00	-79.00	-7.19	.00	90.46
22	23.56	180.00	21.00	.00	-129.25	-9.99	.00	156.82
23.	23.02	180.00	19.45	.00	-180.05	-12.81	.00	217.29
24.	22.50	180.00	18.00	.00	-226.90	-15.42	.00	273.48
MEAN	23.60	153.75	15.95	53.23	-218.71	-10.98	90.00	7.29

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY, COOLING POWER MAX 500 W

ROOM NR 1 3 MODULES DATE 15. 7

HOUR	OUTDOOR DEG C	RECIRC% (0/1)	AIR HANDLING HEAT+FAN	COOL	EFF-TEMP-NR 1 DEG C	COMFORT	EFF-TEMP-NR 2 DEG C	COMFORT
	✓	✓	✓	✓	✓	✓	✓	✓
	17.00	.00	100.80	.00	23.37	.23	23.28	.48
1	15.75	.00	100.80	.00	23.01	.29	22.90	.56
2	14.80	.00	100.80	.00	22.67	.34	22.55	.62
3.	14.20	100.00	.00	.00	23.45	.31	23.34	.56
4	14.00	100.00	.00	.00	23.58	.26	23.47	.55
5	14.20	100.00	.00	.00	23.64	.21	23.55	.48
6	14.50	100.00	.00	.00	23.70	.17	23.62	.38
7	15.75	.00	100.80	.00	22.66	.21	22.59	.35
8	17.00	.00	100.80	.00	22.69	.16	22.66	.20
	17.00	.00	100.80	151.20	23.06	.16	23.04	.14
9	18.45	.00	100.80	224.10	23.37	.15	23.45	.39
10.	20.00	.00	100.80	283.10	23.76	.35	23.98	1.03
11	21.55	.00	100.80	335.17	24.19	.54	24.52	1.54
	21.55	.00	100.80	361.57	23.93	.52	24.24	1.50
12	23.00	.00	100.80	417.67	24.21	.65	24.59	1.84
13	24.24	.00	100.80	466.24	24.46	.71	24.87	1.96
	24.24	.00	100.80	438.19	24.74	.72	25.16	2.01
14	25.20	.00	100.80	473.31	25.00	.73	25.40	1.92
15	25.80	.00	100.80	500.00	25.08	.64	25.42	1.63
16	26.00	70.00	100.80	500.00	25.02	.49	25.27	1.17
17	25.80	70.00	100.80	500.00	24.94	.29	24.97	.62
	25.80	.00	100.80	.00	25.48	.30	25.62	.66
18	25.20	.00	100.80	.00	25.34	.18	25.41	.36
19	24.24	.00	100.80	.00	25.11	.12	25.15	.21
20.	23.00	.00	100.80	.00	24.81	.05	24.82	.06
21	21.55	.00	100.80	.00	24.48	.03	24.45	.12
22	20.00	.00	100.80	.00	24.11	.10	24.05	.27
23	18.45	.00	100.80	.00	23.73	.17	23.66	.38
24.	17.00	.00	100.80	.00	23.36	.23	23.26	.47
MEAN	20.00	19.69	86.10	146.98	24.02	.31	24.09	.74

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY, COOLING POWER MAX 500 W

ROOM NR 1 3 MODULES DATE 15. 7

HOUR	SURF 2 WALL	SURF 3 WALL	SURF 6 WALL	SURF 5 WALL	SURF 4 FLOOR	SURF 1 CEILNG
	24.25	23.98	23.98	23.99	24.50	24.65
1	23.89	23.65	23.65	23.66	24.34	24.53
2	23.54	23.33	23.33	23.34	24.16	24.38
3.	23.46	23.26	23.27	23.28	24.05	24.25
4.	23.70	23.54	23.55	23.55	24.10	24.20
5.	23.78	23.68	23.68	23.68	24.08	24.15
6	23.91	23.76	23.76	23.76	24.07	24.11
7	23.32	23.31	23.31	23.32	23.87	24.01
8	23.13	23.16	23.15	23.16	23.76	23.92
	23.13	23.16	23.15	23.16	23.76	23.92
9.	23.49	23.59	23.59	23.58	23.89	23.92
10.	23.75	23.98	23.98	23.96	24.00	23.97
11	24.09	24.41	24.41	24.38	24.15	24.07
	24.09	24.41	24.41	24.38	24.15	24.07
12	24.29	24.66	24.66	24.62	24.27	24.17
13	24.54	24.93	24.93	24.88	24.42	24.30
	24.54	24.93	24.93	24.88	24.42	24.30
14	24.94	25.31	25.31	25.26	24.59	24.46
15.	25.21	25.49	25.49	25.44	24.73	24.59
16.	25.37	25.51	25.50	25.47	24.82	24.69
17.	25.41	25.39	25.38	25.36	24.87	24.75
	25.41	25.39	25.38	25.36	24.87	24.75
18	25.46	25.32	25.32	25.30	24.85	24.85
19	25.47	25.21	25.21	25.20	24.86	24.87
20	25.35	25.06	25.06	25.06	24.86	24.86
21.	25.16	24.85	24.85	24.85	24.82	24.84
22	24.89	24.58	24.58	24.59	24.74	24.80
23.	24.58	24.29	24.28	24.29	24.63	24.73
24.	24.23	23.97	23.97	23.97	24.49	24.63
MEAN	24.37	24.35	24.35	24.34	24.39	24.42

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY COOLING POWER MAX 500 W

ROOM NR 1 3 MODULES DATE 15. 7

	---WINDOW (SRF 7)--		
HOUR	DEG C	NET RAD.	TRANSM
	22.11	.00	-35.36
1	21.57	.00	-40.24
2	21.12	.00	-43.69
3	21.33	.00	-49.46
4	21.41	.00	-51.25
5	[illegible]	9.54	-52.07
6	22.29	21.69	-51.14
7	22.26	34.87	-45.02
8	22.96	52.34	-41.44
	23.37	52.34	-44.11
9	25.72	127.37	-50.25
10	29.95	249.52	-61.92
11	31.64	351.40	-69.79
	31.16	351.40	-68.54
12	32.02	414.36	-70.68
13	33.99	432.97	-67.44
	34.19	432.97	-66.35
14	34.01	404.92	-60.98
15	32.73	332.12	-47.93
16	30.53	222.82	-31.46
17	27.85	104.06	-14.24
	28.02	104.06	-15.42
18	26.47	47.45	-8.30
19	[illegible]	[illegible]	-10.[illegible]
20	24.95	19.01	-13.48
21	24.06	7.00	-17.46
22	23.29	.00	-22.73
23	22.68	.00	-29.31
24	22.10	.00	-35.28
MEAN	25.95	119.34	-41.15

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY, COOLING POWER MAX 500 W

TEMPERATURES IN ENVELOPE

HOUR	WALL 1 FACADE	WALL 1 BNDRY	WALL 1 BNDRY	WALL 2 BNDRY	WALL 2 BNDRY	WALL 3 BNDRY	WALL 3 BNDRY
	21.62	29.31	24.58	24.09	24.09	24.10	24.13
1.	20.11	27.51	24.20	23.76	23.76	23.78	23.80
2.	18.91	25.89	23.81	23.44	23.44	23.45	23.48
3.	17.75	24.19	23.54	23.23	23.23	23.24	23.24
4.	16.99	22.73	23.70	23.48	23.48	23.49	23.46
5.	16.91	21.44	23.77	23.64	23.64	23.64	23.63
6.	17.25	20.42	23.78	23.73	23.73	23.73	23.73
7.	16.14	19.73	23.47	23.47	23.47	23.46	23.52
8.	19.79	19.40	23.15	23.20	23.20	23.21	23.22
9.	24.49	19.62	23.33	23.49	23.49	23.48	23.45
10.	30.50	20.77	23.57	23.85	23.85	23.83	23.79
11.	36.28	22.95	23.87	24.27	24.27	24.23	24.19
12.	41.14	25.91	24.12	24.56	24.56	24.51	24.49
13.	44.58	29.28	24.41	24.84	24.84	24.79	24.77
14.	45.24	32.65	24.83	25.21	25.21	25.15	25.13
15.	45.96	35.62	25.21	25.44	25.44	25.39	25.37
16.	43.77	37.04	25.47	25.51	25.51	25.47	25.47
17.	39.90	39.06	25.62	25.43	25.43	25.41	25.42
18.	34.80	39.14	25.73	25.35	25.35	25.34	25.34
19.	31.45	38.16	25.77	25.24	25.24	25.24	25.25
20.	29.30	35.60	25.70	25.11	25.11	25.11	25.12
21.	27.16	34.56	25.52	24.92	24.92	24.92	24.94
22.	25.85	33.03	25.26	24.67	24.67	24.68	24.70
23.	23.26	31.15	24.93	24.39	24.38	24.40	24.42
24.	21.62	29.30	24.57	24.07	24.07	24.09	24.11
MEAN	28.88	28.63	24.47	24.35	24.35	24.34	24.34

OFFICE SOUTH, CONCRETE FLOOR SLAB
JULY, COOLING POWER MAX 500 W

DATE 15. 7

PERCENTAGE DISTRIBUTION OF EFFECTIVE TEMPERATURE

EFF.TEMP 1		EFF.TEMP 2	
DEG C	%	DEG C	%
23.0-	100.0	23.0-	100.0
23.5-	94.9	23.5-	87.8
24.0-	68.5	24.0-	77.3
24.5-	44.4	24.5-	58.7
25.0-	23.4	25.0-	43.4
25.5-	.0	25.5-	.0

MEAN VALUES
EFF.TEMP 24.34 24.52
COMFORT .50 1.32

ENERGY TOTALS (KWH) PERIOD 15. 7 - 15. 7 MEAN OUTDOOR TEMP 20.0

ENERGY BOUGHT

ROOM NR 1

PER	HRS	ROOM-AIR	(TOTAL)	LIGHTING	HEATER	AIR-HEAT	AIR-COOL
1	15	23.4	1.16	.00	.00	1.16	.00
2	9	24.0	6.59	2.16	.00	.91	3.53
T	24	23.6	7.75	2.16	.00	2.07	3.53

HEAT BALANCE

ROOM NR 1

PER	HRS	ROOM	PLUS	MINUS	SUN	PEOPLE	LIGHT	HEAT	WINDOW	WALLS	VENT	INFILT
1	15	23.4	2.33	-2.36	.24	.00	.00	.00	-.50	2.09	-1.55	-.21
2	9	24.0	6.06	-6.04	2.62	1.28	2.16	.00	-.49	-1.92	-3.60	-.03
T	24	23.6	6.48	-6.48	2.86	1.28	2.16	.00	-.99	.17	-5.25	-.24

INPUT DATA APPB

RESULT OUTPUT UAPPB

PROGRAM FEE 73.20 SEK

STARTING VALUE = 23.0

NR OF VARIABLES = 49

CHANGE ITERATIONS

1.150	199	442
.075	206	464
.040	211	478
.014	214	486
.002	214	490

Appendix C: Shading Coefficients and U-values for Windows

Shading coefficients F1 and F2 plus heat transmission coefficients (U-values) for different types of fenestration.

F1 specifies total transmitted solar radiation and F2 direct transmitted solar radiation, both relative to total transmitted radiation for a window with two panes of ordinary glass.

Fenestration type		F1 %	F2 %	U-value W/(m^2 °C)	1)
Single glazed window without shading		112	109	5.9	
Double glazed window without shading		100	93	3.1	
Double glazed window with outer pane heat absorbing	2)	60	40	2.1-2.7	3)
Double glazed window with metal coating on outer pane	2)	40	30	2.1	
Triple glazed window without shading		91	80	2.1	
Double glazed window with external Venetian blinds	4)	14	8	3.1	
Double glazed window with Venetian blinds between panes	4)	39	11	2.6	
Double glazed window with internal Venetian blinds	4)	65	14	3.1	
Triple glazed window with external Venetian blinds	4)	11	6	2.1	
Triple glazed window with Venetian blinds between external panes	4)	30	9	1.9	
Triple glazed window with Venetian blinds between internal panes	4)	48	11	1.9	

Fenestration type		F1 %	F2 %	U-value 1) W/(m²°C)
Double glazed window with drapes between panes, drape quality:				
light	compact - loose	31-62	17-54	2.3
medium	compact - loose	40-62	10-43	2.3
dark	compact - loose	46-63	4-33	2.3
Double glazed window with internal drapes, drape quality:				
light	compact - loose	44-71	19-20	2.6
medium	compact - loose	59-75	11-48	2.6
dark	compact - loose	70-82	5-37	2.6

1) Valid for glass part, excluding frame, and for sunlit window

2) Variations depending on type and make

3) Lower value applies for gold or copper coating

4) Slat inclination angle 45°