A Rational Manual Method for Determination of Space Temperature Swing Due to Solar Gains

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ABSTRACT

Empirical approximation formulae for maximum storage during the solar input cycle and for phaseshift corrections required for determination of storage at the time of the maximum air temperature excursion, are developed. They parallel the analytical solution for isothermal storage in a convective environment with harmonic air temperature variation. The analytical results for heat sink performance are utilized in making corrections for finite conductivity effects of typical construction materials. As an example, graphs and tables are developed for the manual determination of the maximum space temperature excursion in a direct passive solar heating system. Account is taken of storage in the surrounding surfaces with varying properties as well as furniture and adjacent spaces with known capacitance and coupling to room air. Storage due to solar input and due to air swing cycle is additive but treated separately because of the differences in phase shift and time history. Sample manual calculations are included.

INTRODUCTION

Heating energy and lighting energy can be saved through admission of solar energy through windows to a greater extent than has been customary in the past. Furthermore, such energy savings can be maximized by allowing the space temperature to rise during such solar input cycle. The natural heat storage capacity of the space is thus utilized for the purpose of retaining the excess energy of the early part of the solar input cycle for release later to provide beneficial heating. Designer of the exterior envelopes of buildings that fully utilize such passive solar heating must be able to predict with good accuracy the heat storage effect (or the maximum space temperature excursion) in spaces receiving solar energy.

The methods generally used for prediction of the space temperature swing are computer simulation of mathematical models and "rule-of-thumb" estimates of the mass that may participate in the process. Computer simulation is costly, requires special knowledge and equipment. Simplistic estimates of the active thermal mass can be in error very considerably and are the reason for most of the reported poor performance of passively solar heated structures. There is a need therefore for a more accurate rational approach that could be used by designers manually with the help of charts and tables or simple approximation formulae.

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METHODOLOGY

The proposed method was developed as part of a larger study on passive solar heating for the Ministry of Energy of Ontario[1]. The study included extensive hour-by-hour computer simulations of a complete house model with three zones, utilizing weather tapes and dynamic modelling techniques[2]. Study of the simulation results allows certain insights into the effects of the major parameters and justification for simplifying assumptions. Modelling results are also used to check the proposed manual method and to develop constants for the method. The available analytical solutions provide the guide to the form of the approximation formulae used in the method.

The proposed method is based primarily on the theory of thin walls in a convective environment (isothermal storage in an infinitely conducting medium) adjusted for the finite conductivity effects. The method parallels to some extent the available analytical solutions for harmonic air temperature variation[3]. The typical driving inputs (variation with time of solar gains and of space temperature) are not harmonic. Such empirical input patterns are derived from the simulation results of the complete house model and are used in numerical integration of the basic storage models to trace the storage temperatures. Correction formulae for finite conductivity effects and for phase shift effects are obtained from such numerical modelling with empirical inputs.

The energy storage due to the solar radiation input and the air temperature swing input are calculated separately and are added together after the appropriate phase shift corrections are applied, to obtain the total energy stored at the time of the air swing maximum.

DISCUSSION OF PARAMETERS AND DEVELOPMENT OF APPROXIMATION FORMULAE

An energy input cycle or pulse can be described approximately by its maximum intensity, its duration and the total amount of energy involved in the cycle. If the total amount of energy is divided by the maximum intensity or rate during the cycle, a quantity in units of time is obtained that will be termed the "equivalent duration" of the cycle.

If

Q - Total amount of energy during cycle (W.h)

q - Max. energy rate during cycle (W)

D_t - Equivalent duration of energy cycle (h)

Then

$$D_t = \frac{Q}{q} \qquad (1)$$

As shown in Fig. 1 the equivalent duration equals the base of the rectangle with the same maximum energy rate and the same area (total energy) as those of the actual energy pulse plotted against time.

Thus the maximum rate of the energy cycle and its equivalent duration fully describe a cycle in a family of cycles with similar shape or time history.

Let us next consider an air temperature swing cycle next to a wall consisting of highly conductive material that distributes heat well and is therefore at the same temperature throughout at any given time. Heat will be convected through the air film into the wall as long as the air temperature is higher than the temperature of the wall. When the air temperature drops below the wall temperature later during the cycle, heat flows through the air film in the reverse direction. There is no heat flow at the moment when both temperatures are equal, which coincides with the time of the wall temperature maximum. In Fig. 2 the temperature histories of such an air swing cycle as well as storage temperatures of a "heavy" and "light" wall are shown. Note the phase

shifts between the maxima of the air temperature and the storage temperatures. The air temperature swing cycle can be described in terms of the maximum temperature rise in air t_a and an equivalent duration in hours. Equivalent duration of an air swing cycle is defined as the area under the air temperature -time curve divided by the maximum temperature excursion.

The isothermal storage coupled to air through a convective film can be defined by its thermal capacitance and the coupling coefficient. The ratio of capacitance to coupling is called the time constant (T) and is expressed in units of time.

If

C - capacitance of storage system ($W.h/K$ or $W.h/(K.m^2)$)

h - coupling of storage to room air (W/K or $W/(K.m^2)$)

T - time constant (hours)

Then

$$T = \frac{C}{h} \qquad (2)$$

Note that in the case of surfaces coupled to room air, only the convective coefficient should be included in coupling h. Air film coefficient is given frequently in the combined form that includes the radiative component.

Next the two limiting cases of isothermal storage in a convective environment are considered.

The first limiting case is that of a storage component with a very short time constant, e.g. a light wall. It is obvious that the temperature of such a component will follow any variations of the air temperature very closely. The time of maximum heat storage will coincide with the maximum air temperature and the amount of heat stored will be equal to the product of the maximum air temperature excursion and the thermal capacitance. If the maximum heat stored during the cycle is designated by Q_w, then for a very light storage component

$$Q_w = t_a \times C = t_a \times h \times T \qquad (3)$$

where

t_a - max. temperature rise in air

The second limiting case is that of a storage component with a very long time constant, i.e. a heavy component, such as may be represented by a thick water wall. The temperature of the storage medium will barely rise and the amount of heat stored will be the product of the coupling coefficient and the time integrated air temperature swing. The maximum storage will be attained at the very end of the air temperature excursion, since heat will be stored as long as the air temperature is above the storage temperature. As defined previously, the time integrated air temperature swing can be represented by the product of the equivalent duration and the maximum temperature excursion, so that

$$Q_w = t_a \times h \times D_t \qquad (4)$$

for the limiting case of isothermal storage with a very long time constant.

Inspection of the two equations for the two limiting cases of isothermal storage (Eq. 3 and Eq. 4) indicates that in both cases the amount of heat stored is equal to the product of the maximum rate of heat input ($t_a \times h$) and

either the time constant or the equivalent duration of the heat input cycle. In the case of a storage system with a finite time constant the answer must lie somewhere between the two extremes. A suitable simple formula that allows determination of the intermediate values is

$$D_w = (T^{-a} + D_t^{-a})^{-1/a} \qquad (5)$$

where

D_w - equivalent duration of heat input for a storage medium with a finite time constant (hours)

a - empirical power coefficient

The above formula parallels the analytical solution for the harmonic case of isothermal storage[3]. The power coefficient "a" is chosen to provide the best fit to the numerical modelling results.

For a storage system with a finite time constant therefore the maximum stored heat during an air swing cycle is given by

$$Q_w = t_a \times h \times D_w \qquad (6)$$

In the case of the solar input cycle the same formula can be used. Solar input can be converted to a temperature difference cycle by utilizing the sol-air concept[4]. Solar intensity is simply divided by the air coupling coefficient to obtain the equivalent extra air temperature rise.

Maximum stored heat for a solar input cycle becomes

$$Q_w = q_s/h \times h \times D_w = q_s \times D_w \qquad (7)$$

where

q_s - max. rate of solar input during the solar input cycle (W/m^2).

Equations (6) and (7) can be added together after application of phase shift corrections discussed further. For the solar input cycle D_t is simply the total heat of the solar input cycle divided by the max. rate of input during the cycle. Computer simulation results indicate that D_t of about 6 hrs is representative for both cycles (air swing and solar input). Power "a" equal to 1.25 gives a good approximation to numerical modelling results. It should be noted that this value of the coefficient applies only with specific shapes of the air swing and solar input cycles plotted against time. This constant and other constants developed further apply to the case of solar gain through south facing glazing during a sunny winter day.

It is instructive to look again at the limiting cases separately for solar input and air temperature swing input:

i) Solar Input

$Q_w = q_s \times D_t$ for very heavy components

$Q_w = q_s \times C/h$ for very light components

ii) Air Temperature Swing Input

$Q_w = t_a \times h \times D_t$ for very heavy components

$Q_w = t_a \times C$ for very light components

Note that good coupling to room air (large h) can be detrimental for storage of solar energy and beneficial for storage resulting from the air temperature swing. It can also be said that for a combined solar and air swing cycle, storage is mainly determined by capacitance for the light components and by coupling for the heavy components. The above also indicates that for very heavy components the actual capacitance is not important.

The limiting cases discussed above are not approached closely in reality, since even light construction has a time constant that cannot be neglected. Very long time constants are only encountered in the coupling of exceptionally massive components, such as the basement.

The preceding approximation formulae allow the determination of the maximum storage attained during the heat input cycle. Since the main purpose of storage calculations is to determine the air temperature swing it is necessary to also quantify the amount of heat stored at the time of the maximum excursion of air temperature. A phase shift correction factor has to be determined for this purpose. Such correction will be quite severe for the heavy components and slight for light components as it will depend on the length of the time interval between the peak of the air swing cycle and the peak temperature in the storage medium.

Phase shifts during an air temperature swing cycle are indicated in Fig. 2. A good approximation of the phase shift correction factor F_a to be applied to D_w to obtain the air temperature swing input duration is:

$$F_a = .5 + .5\ (D_w/T)^2 \qquad (8)$$

Fig. 3 was obtained from computer simulation results of a complete dynamic model[1]. The combined effects of the air swing input and the solar input are indicated. Note that the solar input cycle maximum precedes the air swing maximum by about two hours, so that the phase shift correction factor in this case is less than unity for very light components, rises to unity and then becomes less than unity again as the time constant of the storage component increases.

The following expression was found to agree well with the numerical integration results (see Appendix B) for the phase shift correction factor F_s to be applied to D_w to obtain the solar input duration:

$$F_s = 1 - .3\ ((.85 - D_w/T)/.85)^2 \qquad (9)$$

The preceding calculation is correct for a storage medium with infinite conductivity, e.g. water walls. An additional adjustment has to be made in order to account for the finite conductivity of the normal construction materials. Since we are dealing with heat input cycles of typical length and pattern, a relatively simple correction can be made to the capacitance of surfaces of concrete, gypsum board, wood, etc.

The following correction factor is based on analytical formulae for heat sink performance of heavy walls subjected to a circular pulse at the surface[3].

$$C_e = C(1 + CR/10)^{-\frac{1}{2}} \qquad (10)$$

where

C_e - equivalent or corrected capacitance $W.h/(K.m^2)$

C - actual capacitance per unit of surface area $W.h/(K.m^2)$

R - internal resistance across the heavy slab (face to face) $K.m^2/W$

The constant of 10 provides the best fit to numerical integration results (see Appendix B).

The actual sequence of calculation is not the sequence chosen for clarity of the preceding presentation. First the capacitance correction must be applied to the unit area capacitance of each type of wall, floor and ceiling. Next the time constant is determined using the corrected capacitance for each surface. Then the duration factors and phase shift corrections for both the air swing input and, where applicable, the solar input are calculated so that the final corrected duration factors are obtained. Tables Nos. 1 and 2 are examples of such calculations for various materials. Figs. 4 and 5 give the final corrected durations in graphical form. Note that duration shown for concrete block is that derived from face to face resistance and average density given for 200 mm block. Durations shown for other thicknesses of concrete block are therefore approximate and should be used as a guide only, since rib spacing, dimensions and material of specific blocks may affect the calculated duration value. It should be noted that in these tables only the first column is exact. The remaining columns are calculated and rounded.

As discussed further, some of the components participate in the air swing cycle only and are represented by lumped capacitance and lumped coupling. This applies to the representation of basement, of adjacent rooms not receiving sunlight and of furniture. Fig. 6 may be used for determination of the duration factor for these components which are fully described by their respective time constants.

Thirty two manual calculations were carried out to check the accuracy of the method. The results were compared to the results of the dynamic computer simulation of the house model[1]. A specific sunny January day was selected for this purpose. Calculations included a range of south window areas from 8 to 16 m^2 (in a house with 100 m^2 floor area), double glazing and triple glazing, conventional exterior frame walls and 200 mm concrete block exterior walls with exterior insulation, internal gains of 250 W and 125 W, infiltration rates of .25 and .375 air changes per hour. Room temperature rise ranged up to 10 K. The maximum error in the manually calculated temperature rise was .89 K and the probable error was .32 K (standard deviation of .475 K).

APPLICATION TO DIRECT PASSIVE SOLAR HEATING

The mechanism of heat storage and heat exchange in a direct passive solar system is quite complex. Exact calculation by computer simulation is not practical because of the intricate ever changing pattern of the insolated area, the uncertainties introduced by the habits of the occupants and details of decoration and furnishing. The maximum error introduced by simplified representations of the variables that are most difficult to simulate can, however, be estimated either by simulation of extreme (limiting) assumptions or from general considerations. Simplifying assumptions that follow allow development of models that have a more general applicability.

The colour or solar absorptivity of the surfaces receiving radiation (e.g. walls, ceiling, floor) and position of the thermal mass within the enclosure are relatively unimportant. This was tested with the dynamic computer model[1] and can be also deducted from the following considerations.

Due to the cavity effect, the effective absorptivity of the window will be in the range of .9 to .95 if the room finishes have an absorptivity of about .5 (typical of most normal light coloured finishes). Therefore finishing the room in dark colours cannot improve the overall absorptivity by more than a few percent since the effective window absorptivity cannot exceed unity. This effect can be observed by looking at a window from the outside and noting the very dark appearance of window without drapes. The albedo (or reflectance) of a window will be found to be 5 to 10% for a room of reasonable depth and with normal finishes. The method for calculation of the effective absorptance of cavity openings is given in Ref. 4.

The cavity effect together with the radiative coupling between the surfaces of the room in both the shortwave (solar) and the longwave regions is important in another respect. The emissivity and absorptivity of normal finishes is high in the low temperature radiation region, making the longwave radiative coupling between surfaces quite effective. Such radiative coupling tends to

even out temperature differences between surfaces that can see each other. Solar heat reaching any surface, whether absorbed or reflected, is largely distributed to other surfaces.

Participants in the room heat exchange mechanism can be classified into two broad groups:

1. Surfaces that receive solar radiation, see each other and therefore participate in the radiant heat exchange between themselves as well as in the convective heat exchange with the room air.

2. Surfaces and other massive components that participate in the convective heat exchange with the room air only.

Walls, floors and ceilings fall into the first category. Furniture falls into the second category. Placement of furniture does not significantly alter the overall radiant heat exchange in the room. A table placed on the floor will shade a portion of the floor and it can be said that the upper portion of the table becomes floor while the shaded floor becomes furniture, which exchanges heat by convection only. Thus to assess the effect of furniture it is only necessary to estimate its weight and surface area. Adjacent spaces that do not receive sunlight also fall into the second category and can be similarly represented by their thermal mass and a coefficient representing the degree of coupling to the air of space under consideration.

Simplifying assumptions discussed in the preceding paragraphs indicate that it is satisfactory to assume that the solar gains are distributed uniformly over all the interior surfaces of the room. Such an assumption will simplify the calculations and will introduce little error. The proposed manual method is suitable however for both uniform and nonuniform distribution of solar gains since storage in each component is calculated separately.

The time period that needs to be considered in the temperature swing calculations is that from the start of solar gains to the time of the air excursion maximum.

The quantity of heat that needs to be stored during this time is the sum of solar gains and internal gains from which the heat losses are deducted. Suggested rules for such calculations are given in Reference 1.

Adequacy of design of a passive solar heated structure can be assessed by comparing the heat that needs to be stored with the heat stored in the structure with an assumed maximum air temperature excursion. In Appendix A the predicted air temperature swing for a house is calculated. In this instance it is necessary to separate the heat storage component that is due to the air temperature swing from the component due to the solar input cycle, so that the heat quantity involved in the air swing cycle is isolated, thus allowing the determination of the space temperature excursion.

SUMMARY AND CONCLUSIONS

For cases of energy input cycles of known or typical pattern with time it is possible to develop simple approximation formulae that take account of the major parameters involved in storage of heat in building components. The maximum air temperature swing can be predicted with good accuracy by manual methods utilizing such approximation formulae or charts and tables derived from the formulae. Determination of the constants for the approximation formulae for a given energy input pattern involves numerical integration and curve fitting analysis. The worked out example is that of solar energy input through south facing glazing in winter, i.e. direct passive solar heating. Results of the manual calculations compare well with the results of detailed dynamic simulation models.

APPENDIX A

Calculated Example

Calculate the temperature swing in the south living area of a two storey house with the following characteristics:

1. Given Data

   a. Southern living space (includes the living-dining area on first floor and southern bedrooms on second floor)

Windows, triple glazed	16 m^2
Heavy sidewalls, 200 mm block	44 m^2
Heavy south wall, 200 mm block	19 m^2
Light partitions, floor, ceilings equivalent to 23 mm of gypsum board	160 m^2
Furniture of 290 W.h/K capacitance and 196 W/K coupling to room air	

Heat Loss Coefficient:

Glass	29 W/K
Walls	16 W/K
Roof	6 W/K
Infiltration at .25 AC	20 W/K
Total	71 W/K

Daytime internal gains	250 W
Maximum solar gain rate	700 W/m^2 of glass

   b. Northern third of house (includes kitchen, hall, closets, stairs and northern part of second floor)

Capacitance	4470 W.h/K
Coupling	300 W/K

   c. Basement

Capacitance	8480 W.h/K
Coupling	200 W/K

2. Calculation Procedure

   a. Calculate Heat to be Stored

Solar gains during 6 hrs from start of solar gains to air temperature maximum
= max. rate x 5 hrs. x glass area
= 700 x 5 x 16 = 56,000 W.h.

The product of max. rate and 5 hrs approximates the accumulated energy received at varying intensity during the first 6 hrs of the solar cycle[1].

Internal gains during 6 hrs
=250 x 6 = 1,500 W.h.

Total gains 57,500 W.h.

Heat losses during 6 hrs
=71 (23° - (6.3^o) x 6 = 12,482 W.h.

Heat to be stored (Q_{total}) 45,018 W.h.

23^oC is the assumed average room temperature during the 6 hours.
-6.3^oC is the mean January outdoor temperature[1].

b. Calculate Solar Input Storage

Max. Solar Intensity on surfaces
$=700 \times 16/(44 + 19 + 160) = 50.22\ W/m^2$

Heavy sidewalls solar input duration	4.15 hrs(Fig. 4)
Light partitions etc. solar input duration	1.54 hrs(Fig. 4)

Heavy solar input storage = 4.15 x 50.22 x 63 =	13,130 W.h.
Light solar input storage = 1.54 x 50.22 x 160 =	12,374 W.h.
Total solar input storage (Q_{solar})	25,504 W.h.

c. Calculate Sum of Air Swing Storage Factors (Sum of products of duration and coupling of each storage component)

Convective coefficient for surfaces 3 $W/(K.m^2)$

	Time Constant hrs.	Duration hrs.	Storage Factor W.h/K
Heavy Sidewalls		2.61 (Fig. 5)	2.61x63x3 = 493.3
Light Parts. etc.		1.32 (Fig. 5)	1.32x160x3= 633.6
Furniture	290/196 = 1.48	1.15 (Fig. 6)	1.15x196 = 225.4
N. Part of House	4470/300 = 14.9	2.65 (Fig. 6)	2.65x300 = 795.0
Basement	8480/200 = 42.4	2.85 (Fig. 6)	2.85x200 = 570.0

Sum of Air Swing Storage Factors = 2717.3

d. Calculate Temperature Swing

Total stored heat equals the sum of solar input storage and air swing input storage

$$Q_{total} = Q_{solar} + Q_{air}$$

Also, air swing input storage is the product of the sum of air swing storage factors and the air temperature swing. Therefore

$$t_a = \frac{Q_{total} - Q_{solar}}{\text{Sum of Air Swing Storage Factors}} = \frac{45{,}018 - 25{,}504}{2717.3} = 7.18^{o}C$$

e. Summary of Heat Storage

Heavy walls 493.3 x 7.18 + 13,130 =	16,672 W.h
Light partitions, etc. 633.6 x 7.18 + 12,374 =	16,923 W.h
Furniture 225.4 x 7.18 =	1,618 W.h
N. part of house 795 x 7.18 =	5,708 W.h
Basement 570 x 7.18 =	4,093 W.h
Total =	45,018 W.h
	or 162,065 kJ

A more detailed discussion of such calculations is given in Reference 1.

APPENDIX B

Response of an Elementary Storage Component to a Known Input Cycle

Variation with time of the temperature of a storage component can be calculated for any given empirical cycle of solar input or air temperature swing input. Elementary finite differences models can be used with fine time steps (e.g. 1 min.) to obtain high accuracy.

Response to solar input of a storage component with isothermal capacitance C and convective coupling h can be calculated as follows:

- Read hourly intensity of given solar input
- Begin loop repeated P times (say 60 times) during the hour
- Interpolate input intensity q to that applicable at the particular time (fraction of the hourly interval)
- If rise of storage temperature above ambient temperature equals t (and t' for the preceding time interval), then

$$t = t' + (q - h \times t')/P/C$$

- End of loop
- Read intensity of next hour and begin loop again.

Response to an empirical air temperature swing input is obtained again by reading the given air temperature rise for each time interval (t_{air}). The following algorithm can be used in a similar loop to that of the preceding example:

$$t = t' + (t_{air} - t')\ h/P/C$$

To allow for conductivity effects (non-isothermal storage), a multilayered representation of the storage component is used. For a two layer scheme with resistance R between nodal points and capacitance C of each node, the appropriate algorithm for solar input is:

$$t_1 = t'_1 + (q + (t'_2 - t'_1)\ /R - t'_1 \times h)\ /P/C$$

$$t_2 = t'_2 + (t'_1 - t'_2)\ /R/P/C$$

Algorithms for more layers are similarly derived.[2]

Sum of temperature rises of all layers multiplied by their capacitances gives the total heat stored in the multilayer storage component at any given time. Comparison of the heat stored in a multilayer model to that stored in a single layer (isothermal) model allows the determination of correction due to the effects of finite conductivity of the storage medium.

APPENDIX C

Effects of Infrared Heat Exchange Between Surfaces

It may appear that in the case of the solar input cycle the film coefficient "h" should include the radiative component, in which case the coupling between the storage and the room air would be greater than with the convective component only. In transient storage cycles discussed in the present paper the contribution of the long wave heat exchange between surfaces is quite dissimilar to the effects of such radiation in the more familiar steady state cases. In the case of the sol-air temperature calculation for an exterior surface, the surface faces a highly absorptive medium of infinite capacitance. In the case of the steady state heat loss coefficient of a wall, the interior surface of the wall faces large interior surfaces with zero (steady state) capacitance. Neither of these conditions apply to the case of transient energy storage in surfaces surrounding a room.

Let us first consider the simpler transient storage cycle where all the surfaces have uniform thermal capacitance per unit of surface area. If solar energy is distributed uniformly then all the surfaces will be at the same temperature at any given time and obviously there will be no net radiant heat exchange during the storage cycle. If the surfaces are not at the same temperature (e.g. due to non-uniform distribution of solar energy) then a definable quantity of heat will be transferred by radiation from the hotter surfaces to the cooler surfaces. The drop in the temperature of the hotter surfaces caused by such heat flow will result in an equivalent increase in the temper-

ature of the cooler surfaces, so that the overall convective flow of stored heat to the room air will not change. It can be readily seen that this will be true even if the sizes of the hotter and cooler surfaces are not equal, since then the drop and the increase in the respective temperatures will be inversely proportional to the area ratio while the convective heat flow is directly proportional to the area ratio.

If some of the surfaces are thermally heavier than other surfaces, then the infrared heat flow between heavy and light surfaces will change the net heat flow from the storage surfaces to the room air. Paradoxically though the overall coupling of storage to room air will be reduced in the case where the lighter surfaces are hotter (the more likely case), since radiant transfer will drop the temperature of the light surfaces more than it will raise the temperature of the heavy surfaces. The coupling of overall storage to room air will be increased only in the case of a heavy surface surrounded by cooler light surfaces.

The same infrared radiation effects apply equally to the energy storage in surfaces resulting from room temperature swing, only here it is not possible to have the case of hotter heavy surfaces during the charging part of the cycle.

It follows from the preceding discussion that in a typical room the infrared radiation between surfaces is likely either to have no effect on the overall storage, or to enhance it, depending on the relative surface areas of heavy and light components and the distribution of the solar input intensity on such components. It also follows that more accurate results will be obtained if the radiative component is omitted from the film coefficient between storage surfaces and room air for both the solar storage and the air swing storage calculations.

A different consideration is the infrared coupling of the surfaces to the carbon dioxide and moisture in room air. It is small due to the room dimensions but not negligible even under winter conditions and should be included in the film coefficient[1].

Infrared radiation to the cool window surface can be accounted for in the window U factor when room heat losses are calculated (see Appendix A). It may be noted that even with a large window the angle factor from any typical interior surface to such window is in the order of .05 to .08.

REFERENCES

1. Residential Passive Solar Heating, Review and Development of Design Aids, Ontario Ministry of Energy Publication, 1980.

2. Cuplinskas, E.L., "Thermal Response Calculations by the Finite Differences Method", ASHRAE Transactions, 83,2: 118, 131, 1977.

3. Rohsenow, W.M. and Hartnett, J.P., Handbook of Heat Transfer, McGraw-Hill, New York, 1973, Section 3.

4. Gebhart, B., Heat Transfer, McGraw-Hill, New York, 2nd Ed., 1971, 144 - 145.

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SOLAR INPUT DURATION FACTOR FOR A SURFACE

SOLID CONCRETE
CONDUCTIVITY 1.7 W/(m.K) VOLUMETRIC CAPACITANCE 2060 kJ/(K.m3)

CAPACIT'CE /UNIT AREA kJ/(K.m2)	THICK NESS mm	CAPACIT CORR'N	TIME CONST hrs	EQUIV DURATION hrs	PHASE CORREC'N	CORREC'D DURATION hrs
10	5	0.9996	0.9256	0.8597	0.9974	0.8575
12	6	0.9994	1.1105	1.0132	0.9984	1.0116
14	7	0.9992	1.2953	1.1606	0.9991	1.1595
16	8	0.9990	1.4800	1.3019	0.9996	1.3014
18	9	0.9987	1.6645	1.4373	0.9999	1.4372
20	10	0.9984	1.8489	1.5671	1.0000	1.5671
25	12	0.9975	2.3091	1.8683	0.9993	1.8670
30	15	0.9964	2.7679	2.1390	0.9975	2.1337
35	17	0.9952	3.2251	2.3823	0.9949	2.3701
40	19	0.9937	3.6804	2.6016	0.9915	2.5795
45	22	0.9921	4.1336	2.7995	0.9876	2.7648
50	24	0.9902	4.5844	2.9785	0.9833	2.9289
60	29	0.9860	5.4779	3.2885	0.9741	3.2034
70	34	0.9811	6.3591	3.5458	0.9645	3.4200
80	39	0.9755	7.2263	3.7616	0.9549	3.5920
90	44	0.9693	8.0779	3.9440	0.9457	3.7297
100	49	0.9626	8.9125	4.0995	0.9368	3.8406
110	53	0.9552	9.7290	4.2332	0.9285	3.9306
120	58	0.9474	10.5262	4.3488	0.9208	4.0042
130	63	0.9390	11.3033	4.4496	0.9135	4.0648
140	68	0.9303	12.0594	4.5378	0.9068	4.1150
150	73	0.9212	12.7941	4.6155	0.9006	4.1568
160	78	0.9117	13.5068	4.6843	0.8949	4.1918
170	83	0.9020	14.1974	4.7454	0.8895	4.2213
180	87	0.8919	14.8656	4.8000	0.8846	4.2463
190	92	0.8817	15.5114	4.8490	0.8801	4.2675
200	97	0.8713	16.1349	4.8930	0.8759	4.2857
300	146	0.7639	21.2182	5.1643	0.8472	4.3753
400	194	0.6639	24.5871	5.2863	0.8326	4.4012

Based on a convective surface coefficient of 3 W/(K.m2)

TABLE 1

AIR TEMPERATURE SWING DURATION FACTOR FOR A SURFACE

GYPSUM BOARD
CONDUCTIVITY .16 W/(m.K) VOLUMETRIC CAPACITANCE 880 kJ/(K.m3)

CAPACIT'CE /UNIT AREA kJ/(K.m2)	THICK NESS mm	CAPACIT CORR'N	TIME CONST hrs	EQUIV DURATION hrs	PHASE CORREC'N	CORREC'D DURATION hrs
10	11	0.9903	0.9169	0.8524	0.9321	0.7945
12	14	0.9861	1.0957	1.0011	0.9175	0.9185
14	16	0.9812	1.2719	1.1423	0.9033	1.0318
16	18	0.9757	1.4454	1.2759	0.8896	1.1350
18	20	0.9695	1.6158	1.4021	0.8765	1.2290
20	23	0.9627	1.7828	1.5213	0.8640	1.3144
25	28	0.9435	2.1841	1.7896	0.8357	1.4956
30	34	0.9215	2.5598	2.0197	0.8113	1.6385
35	40	0.8974	2.9083	2.2163	0.7904	1.7517
40	45	0.8718	3.2290	2.3843	0.7726	1.8422
45	51	0.8453	3.5221	2.5278	0.7575	1.9149
50	57	0.8183	3.7887	2.6506	0.7447	1.9740
60	68	0.7647	4.2482	2.8466	0.7245	2.0624
70	80	0.7131	4.6217	2.9926	0.7096	2.1237
80	91	0.6648	4.9245	3.1031	0.6985	2.1676
90	102	0.6204	5.1701	3.1879	0.6901	2.1999
100	114	0.5800	5.3702	3.2540	0.6836	2.2243
110	125	0.5434	5.5342	3.3062	0.6785	2.2431
120	136	0.5102	5.6694	3.3481	0.6744	2.2579
130	148	0.4803	5.7819	3.3822	0.6711	2.2697
140	159	0.4533	5.8760	3.4101	0.6684	2.2793
150	170	0.4288	5.9554	3.4332	0.6662	2.2871
160	182	0.4065	6.0228	3.4526	0.6643	2.2936
170	193	0.3863	6.0805	3.4690	0.6627	2.2991
180	205	0.3678	6.1301	3.4830	0.6614	2.3037
190	216	0.3509	6.1730	3.4950	0.6603	2.3077
200	227	0.3354	6.2104	3.5054	0.6593	2.3110
300	341	0.2309	6.4140	3.5605	0.6541	2.3289
400	455	0.1752	6.4902	3.5806	0.6522	2.3353

Based on a convective surface coefficient of 3 W/(K.m2)

TABLE 2

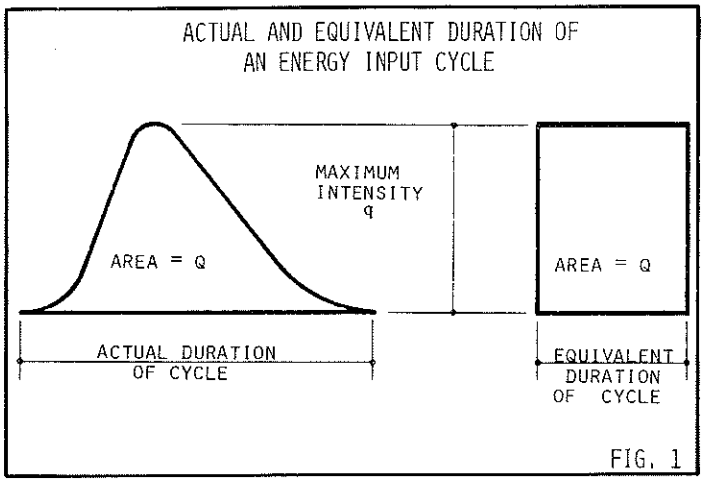


FIG. 1

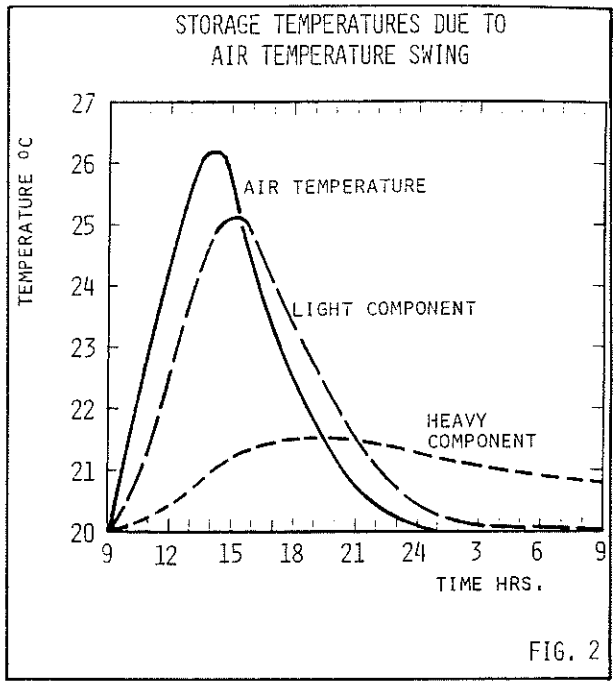


FIG. 2

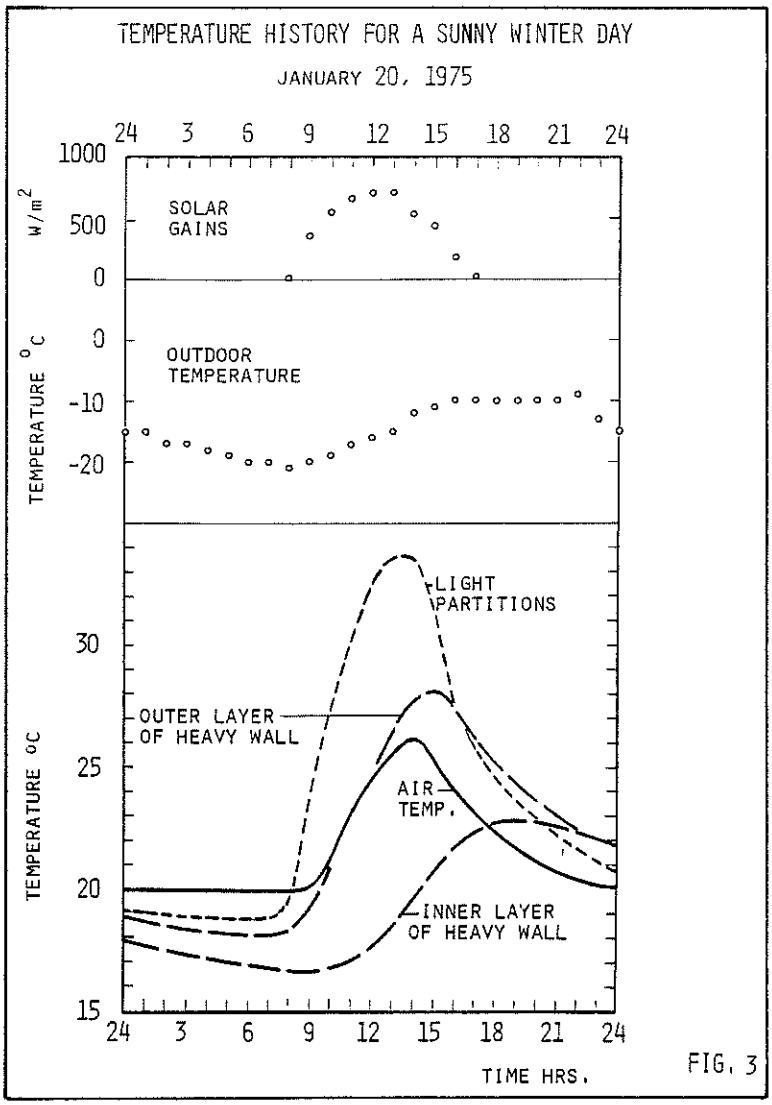
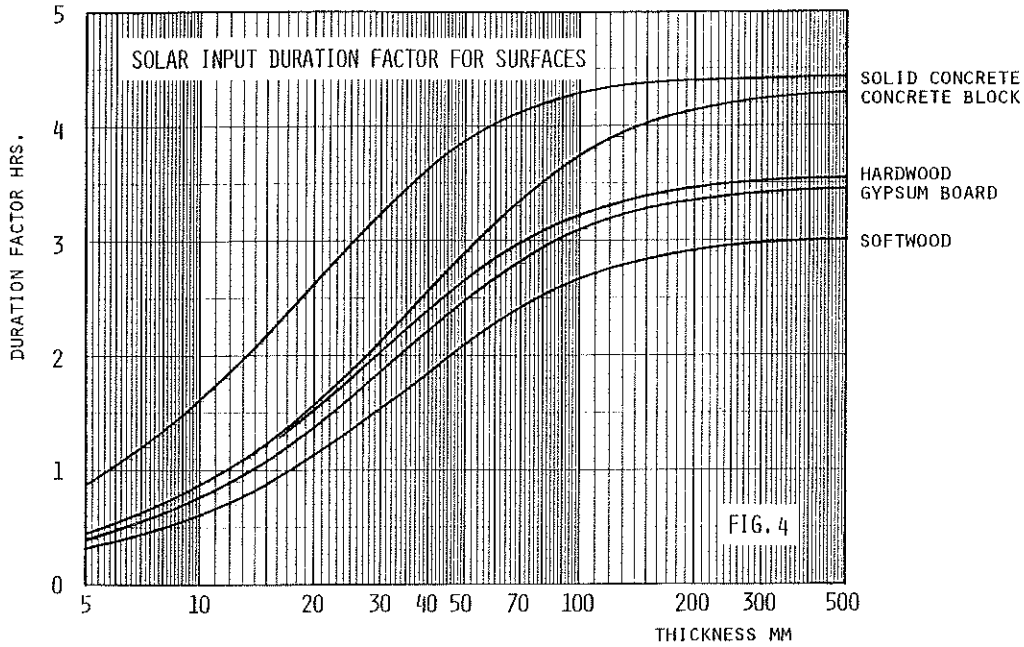
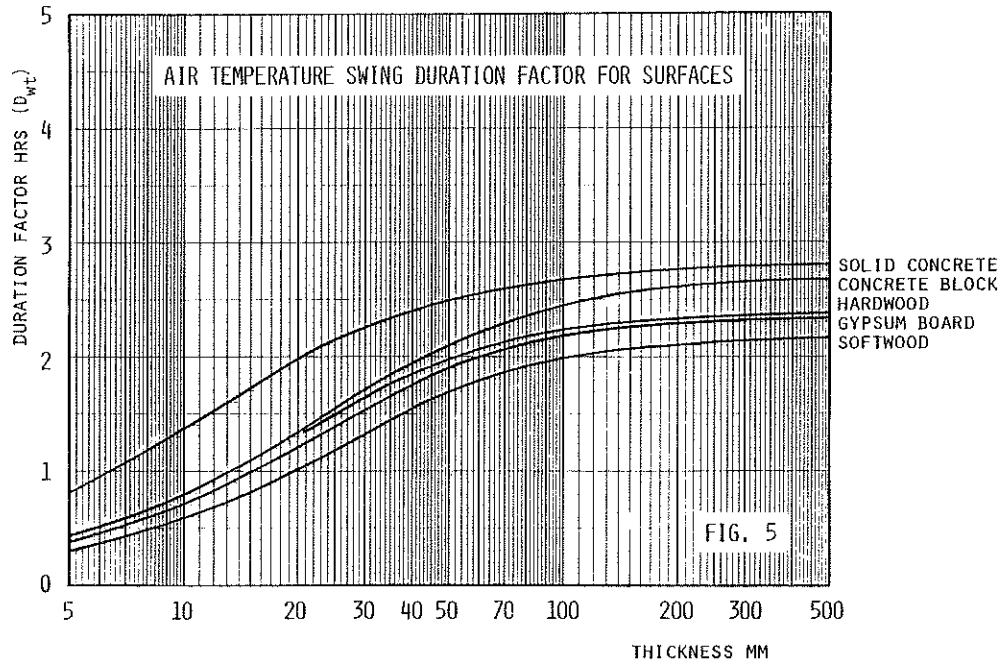
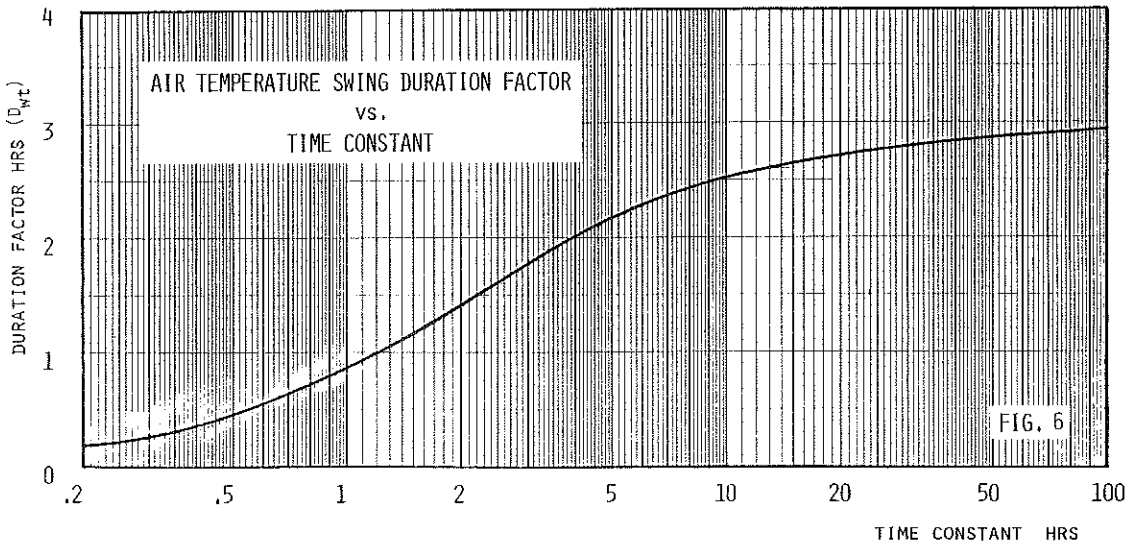


FIG. 3


SOLAR INPUT DURATION FACTOR FOR SURFACES
DURATION FACTOR HRS.
THICKNESS MM
SOLID CONCRETE
CONCRETE BLOCK
HARDWOOD
GYPSUM BOARD
SOFTWOOD
FIG. 4


AIR TEMPERATURE SWING DURATION FACTOR FOR SURFACES
DURATION FACTOR HRS (D_{wt})
THICKNESS MM
SOLID CONCRETE
CONCRETE BLOCK
HARDWOOD
GYPSUM BOARD
SOFTWOOD
FIG. 5


AIR TEMPERATURE SWING DURATION FACTOR
VS.
TIME CONSTANT
DURATION FACTOR HRS (D_{wt})
TIME CONSTANT HRS
FIG. 6