Proposed Streamlined Residential Heating

Energy Budget Analysis by a

Variable Temperature Design Method

Gunard E. Hans
Member ASHRAE

INTRODUCTION

Growing emphasis on energy conservation in buildings calls for better house designs that supplement fossil fuel energy with solar heat gains. Traditional design procedures and building code provisions make it difficult to recognize the sun as a heat source. Advantages of solar-oriented design can be conclusively demonstrated only by procedures based on heating energy budget analysis.

Detailed solar-oriented house design requires lengthy longhand calculations or computerized analysis; simplified passive solar design methods[1] often depart from established engineering design practices. For rapid evaluation of the thermal performance of solar-oriented houses by a procedure within accepted practice, a simplified design process, the "variable temperature design" method, attempts to quantify all solar heat gain and storage effects in terms of an equivalent design temperature credit. Because the magnitude of such credit for different buildings varies with their construction features, the resulting heating load calculations are based on variable rather than fixed indoor "design" temperatures. The fixed temperatures have been standardized by code references, for all houses regardless of their solar energy collection potential.

ENERGY BUDGET DESIGN METHODS

Available Options for Residential Design

Growing interest in passive solar energy utilization has led to the introduction of a variety of new concepts for the analysis of solar house designs. The most commonly cited of such procedures is the Solar Load Ratio Method[1], developed for analysis of thermal storage wall passive solar designs. Its relative simplicity, however, also interferes with its application to other building types. Typical passive solar design analyses are focused on room temperature control considerations and do not lend themselves to seasonal energy budget calculations.

The most commonly used conventional energy budget analysis procedure is the Degree-Day Method, which is a single-measure procedure that has gained popular acceptance because of its simplicity rather than its accuracy. The multiple-measure Bin Method offers greater accuracy but is more time consuming and has remained relatively unpopular in the residential design field. The Modified Degree-Day Procedure may be viewed as a compromise between the two; it also requires only a single calculation, but permits adjustments for variations in actual heating load distributions by simple correction factors. These three procedures are part of recognized design methodology[2].

Gunard E. Hans is an architect at the Forest Products Laboratory, Forest Service, U.S. Department of Agriculture. The Laboratory is maintained at Madison, Wis., in cooperation with the University of Wisconsin.

Conventional thermal design is based on assumed steady-state conditions that allow heat loss calculations as a product of the nominal heat loss rate and the applicable temperature differential:

$$H = (h_t + h_s)(t_i - t_o) \tag{1}$$

where

H = heat loss, Btu/h

h_t = building envelope transmission losses, Btu/h·°F

h_s = sensible heat loss due to air infiltration, Btu/h·°F

t_i = indoor air temperature, °F

t_o = outdoor air temperature, °F

Estimates of the heating budget (or net heating load) based on the Degree-Day Method are calculated by a similar procedure as:

$$Q = (h_t + h_s) \times 24DD \tag{2}$$

where

Q = net heating load, Btu

24 = number of hours per degree-day

DD = number of degree-days for the period under consideration

Degree-day numbers are commonly calculated to a base of 65° F that provides for a uniform 7° F solar and internal heat gain credit at an indoor design temperature of 72° F. Use of a standardized heat gain credit for all buildings, however, does not allow more detailed recognition of solar gain and storage effects in different structures. Past proposals for modeling such effects within recognized design methodology have included modifications to thermal loading (in the form of adjustments to basic degree-day numbers) as well as to the response of a building to load variations (in the form of "effective U values"). Nevertheless, such proposals still have not resulted in accepted new design methods, and changes to basic U values would require complex reference tables[3].

The Heat Balance Concept[2] provides the necessary mechanism for a flexible thermal performance analysis procedure with no modifications to established design coefficients. This concept allows characterization of the thermal response of a house in terms of its "break-even" temperature. The "break-even" temperature is that outdoor temperature at which heat losses from the conditioned space (at any indoor temperature rather than a fixed reference as in conventional degree-day calculations) are equal to the corresponding internal and solar heat gains. This concept provides the basis for further development of the "variable temperature design" method.

Proposed Variable Temperature Design Method

The variable temperature design concept is based on the premise that it is more appropriate and practical to model the effects of solar heat gains and storage in buildings by adjustments to design temperatures rather than by variations in design coefficients that have become identified with conventional design procedures (such as the physical properties quantified in terms of an R value or its reciprocal).

To allow simplified consideration of solar heating effects, this proposal also introduces a number of other design concepts not directly related to the variable temperature design concept. Thus, solar heat gains through opaque wall surfaces may be estimated on the basis of any desired method, as long as they are quantified in units of heat (such as Btu). To permit such calculations both glazed and opaque portions are quantified in terms of a "design glass area" ($\overline{DGA}$), which is the aggregate of actual and equivalent glass areas. This method offers the advantage of estimating such gains for both glazed and opaque portions of the building envelope in a single step and on the basis of the same solar radiation data.

The solar gain estimated to be received through such glass area is identified as the "nominal solar gain" (Q_{SG}) before adjustments for losses attributable to reflection from interior surfaces. The portion estimated to be retained is identified as "useful solar gain" (Q_{SU}). The ratio between useful and nominal solar heat gains is identified as the "solar acceptance factor" ($\overline{SAF}$), and, conversely, an estimated $\overline{SAF}$ value is used in predicting likely useful solar heat gains from calculated nominal solar gains. This factor is intended to be applied to modeling solar heat gain and storage effects in an essentially unoccupied house maintained at typical occupancy temperatures. Further adjustments for occupancy conditions (such as shading windows by day and protecting them by insulated covers at night) can be modeled more effectively as an independent consideration.

Those variations in the thermal performance of an occupied house attributable to active intervention by the occupants but not quantifiable in terms of room temperatures (e.g., thermostat setting) are modeled through the introduction of an "energy management factor" ($\overline{EMF}$).

Estimates of these factors ($\overline{DGA}$, $\overline{SAF}$, and $\overline{EMF}$) do not need to be based on the suggested methods of derivation; they may be calculated by any procedure offering the desired level of accuracy or sophistication. Their use in further calculations permits partial recognition of load-and-response relationships: The $\overline{DGA}$ value is based on estimates of "equivalent" glass areas that also include consideration of the heat storage capacity of opaque wall construction, and $\overline{SAF}$ value is influenced by the heat storage capacity of both building envelope and core heat storage capacity and by thermal coupling to room air. The variable temperature design procedure, therefore, represents a modified steady-state design procedure that is fully compatible with conventional steady-state design. To warrant a variable temperature approach the period of investigation must be sufficiently long for elimination of transitory effects and for achieving an averaged balance at which heat no longer is gained in or recovered from storage. Analysis of available field data on light-frame wood structures suggests that the minimum length of such design period may be 5 days.

VARIABLE TEMPERATURE DESIGN

Figure 1 illustrates the basic conventional thermal design process. Heating loads are calculated as a function of indoor-outdoor temperature differential defined by a straight line whose slope is determined by the nominal heat loss rate (Btu/h·°F) and zero-load intercept by the break-even temperature. While heating systems are based on full load calculations under indoor and outdoor "design" temperatures, heating energy budget calculations are based on 65° F (degree-day base line) or break-even temperature (also known as "balance" temperature) zero-load intercepts. Variable temperature design follows the same process but redefines some reference points for increased flexibility.

Basic Relationships

These relationships have been derived from established thermal design procedures and represent adaptations of existing design concepts.

Heat Balance. Steady-state design conditions assume equalization of internal heat exchange with storage during the period under consideration (such as 5 days). Under such conditions, the quantity of heat lost from the building equals the quantity supplied by the heating system and gained from all other sources:

$$Q_{HL} = Q_{HH} + Q_{HG} \tag{3}$$

where

Q_{HL} = heat lost from the building, Btu

Q_{HH} = heat supplied by the heating system, Btu

Q_{HG} = heat gained from all other sources, Btu

Heat Loss. The quantity $(h_t + h_s)$ of equations (1) and (2) may be referred to as the "nominal heat loss" rate estimated under presumably unchanging air infiltration conditions (which actually vary with indoor-outdoor temperature differentials and changing wind pressures) and unchanging U values (which also change with the temperature of the building envelope). Heat loss calculations based on such assumptions are represented by:

$$Q_{HL} = q_{hl}(t_{\bar{i}} - t_{\bar{o}})\ T \qquad (4)$$

where

q_{hl} = $(h_t + h_s)$, nominal heat loss rate, Btu/h·°F

$t_{\bar{i}}$ = average indoor temperature for period, °F

$t_{\bar{o}}$ = average outdoor temperature for period, °F

T = length of period under consideration, h

Heat Gains. Accurate quantification of the net heating load (Q_{HH}) also requires accurate quantification of heat gains from all sources other than a thermostatically controlled heating system. Such gains include the body heat of occupants, waste heat from lights and appliances, uncontrolled heat from wood fire and other auxiliary heat sources, and the useful portion of solar heat gains. The aggregate of other than solar heat gains is commonly identified as "internal heat gain". Total recognizable heat gains, therefore, can be expressed as:

$$Q_{HG} = Q_{IG} + Q_{SU} \qquad (5)$$

where

Q_{IG} = internal heat gain, Btu

Q_{SU} = useful solar heat gain, Btu

Heat Gain Temperature Credit. Consistent with the break-even temperature concept, heat gains in variable temperature design are converted into an equivalent temperature credit to be applied against a design indoor-outdoor temperature differential as follows:

$$t_c = \frac{Q_{HG}}{q_{hl} \times T} \qquad (6)$$

where

t_c = heat gain temperature credit, °F

Floating Room Temperature. The assumed straight-line relationships between heating load and indoor-outdoor temperature differential permit application of the calculated heat gain temperature credit at any point along that line. When applied to the outdoor temperature level, this credit allows modeling floating room temperatures that can be sustained without further heat input from the heating system, and represents the actual heating threshold. This floating room temperature, the basic reference in variable temperature design, is assumed to rise above the average outdoor temperature, for any period under consideration, by the magnitude of the heat gain temperature credit computed for that period:

$$t_{\bar{f}} = t_{\bar{o}} + t_c \qquad (7)$$

where

$t_{\bar{f}}$ = average floating indoor temperature, °F

Heating Load. The resulting net heating load is proportional to the differential between the floating room temperature and the actual room temperature sustained with further heat input. If this heat is supplied by the heating system, the actual room temperature may be equated with the average thermostat setting for the period. If heat is supplied by sources that are manually controlled, such as a portable electric heater, it may be referred to as a "targeted" room temperature. This final calculation for the period under investigation (e.g., 5 days or the entire heating season) is represented by:

$$Q_{HH} = q_{h1}(t_{\bar{t}} - t_{\bar{f}})\ T \tag{8}$$

where

$t_{\bar{t}}$ = average thermostat setting or targeted indoor temperature, °F.

Supporting Relationships

The variable temperature design process relies on certain supporting relationships recognized in terms of design factors based on established procedures but representing new concepts. As noted, the derivation of such factors or calculation of their numerical values is not necessarily limited to the methods suggested below.

Design Glass Area ($\overline{DGA}$). Solar heat gains through opaque wall and roof surfaces traditionally have been disregarded as a relatively minor item. In heat balance calculations, however, they cannot be overlooked. If estimates of such gains are to be consistent with solar gain calculations for windows, opaque surface areas may be conveniently converted to equivalent glass areas of the same transmittance as used for windows at any given orientation. The "design glass area" ($\overline{DGA}$) is the aggregate of the actual and equivalent glass areas.

The conversion of opaque surfaces to equivalent glass areas may be based on any recognized design procedure. The ASHRAE Handbook[5] provides a method for such calculations on the basis of Sol-air temperatures, Cooling Load Temperature Differences (CLTD) and Solar Heat Gain Factors (SHGF). While the tabulated CLTD values apply only to cooling season conditions, the suggested procedure uses simple ratios between opaque surface and glass gains extrapolated from such values to winter conditions in accordance with the CLTD design method. Then:

$$q_{wall} = U \times CLTD \tag{9}$$

where

q_{wall} = unit heat gain rate for opaque wall surface, $Btu/ft^2 \cdot h$

U = thermal transmittance factor for that surface, $Btu/ft^2 \cdot h \cdot °F$

CLTD = Cooling Load Temperature Difference for winter conditions, °F

$$q_{glass} = SHGF \times SC \tag{10}$$

where

q_{glass} = unit heat gain rate for vertical glazed area, $Btu/ft^2 \cdot h$

SHGF = Solar Heat Gain Factor, appropriate averaged value, $Btu/ft^2 \cdot h$

SC = Shading Coefficient for the given glass, a dimensionless factor

Calculation of heat gains for opaque surfaces with different thermal properties (such as walls and roof) but at the same orientation (such as the vertical projection of a sloping roof) may be streamlined by considering averaged surface area and thermal transmittance ($A_o U_o$) values. The given opaque surface area is equated with an equivalent glass area as:

$$\overline{EGA} = \overline{OSA}\,\frac{q_{wall}}{q_{glass}} \tag{11}$$

where

$\overline{EGA}$ = equivalent glass area, ft^2

$\overline{OSA}$ = opaque surface area with the same orientation as glass, ft^2

The resulting design glass area ($\overline{DGA}$) is:

$$\overline{DGA} = \overline{AGA} + \overline{EGA} \tag{12}$$

where

$\overline{AGA}$ = actual glass area, ft^2

Solar Acceptance Factor ($\overline{SAF}$). The solar acceptance factor is a dimensionless ratio of useful-to-nominal solar heat gains:

$$\overline{SAF} = \frac{Q_{SU}}{Q_{SG}} \tag{13}$$

For the purposes of this discussion, nominal solar heat gain (Q_{SG}) is defined as measured or calculated solar radiation received through glass (including equivalent glass area), and useful solar heat gain (Q_{SU}) as the heat expended in maintaining a floating indoor-outdoor temperature differential ($t_{\bar{f}} - t_{\bar{o}}$) attributable to solar gain effects. The Q_{SU}/Q_{SG} ratio for a test structure may be established on the basis of floating temperature measurements and energy balance calculations, or it may be estimated by rigorous computer simulation.

As the efficiency of solar collectors is inversely proportional to their operating temperatures, the magnitude of the solar acceptance factor applicable to a house as a collector/storage system varies not only with such physical conditions as dispersion of sunlight over interior surfaces and presence of sufficient thermal mass for effective absorption of this energy, but also with the associated room temperature swings. Room temperature rise, furthermore, is also a function of the coincident indoor-outdoor temperature differential. As a result of such interactions, the solar acceptance factor is a variable quantity.

About 5 percent of the solar radiation received through glass is lost by reflection from interior surfaces. While the heat received through opaque surfaces is not subject to reflectance losses, that fraction is not large enough to materially affect the high limit of possible $\overline{SAF}$ values, which therefore may be estimated at 0.95. This value also may be applicable only where no further heat is supplied as a condition for full utilization of the solar heat received. Analysis of data from a few available studies of floating temperatures in light-frame wood structures suggests $\overline{SAF}$ values at about 0.85 for normal room temperatures. In design with large window areas but insufficient thermal ballast for room temperature control, this factor may be reduced to 0.65 or still lower to exclude from consideration that portion of solar gains which is associated with excessive temperature rise.

For practical design purposes $\overline{SAF}$ values would be selected from appropriate references, when available. Such references may be prepared in the form of nomograms with multiple entries (similar to psychrometric charts), or in a more simplified form of a triangular matrix. The coordinates of such matrix may be scaled in terms of variations in construction (thermal mass effects), design (dispersion of sunlight over interior surfaces) and loading environment (allowable room temperature swing centered on a reference temperature, such as

68 ± 5° F). Different climate zones may require different $\overline{SAF}$ matrices (similar to different solar gain tables for different latitudes) for satisfactory modeling of relationships between solar gain utilization and average indoor-outdoor temperature differentials for the period under consideration.

The application of the solar acceptance factor is illustrated by the following relationships representing a composite of equations (5), (6), (7), (8), and (13):

$$Q_{HH} = q_{h1}\left[t_{\bar{t}} - t_{\bar{o}} + \frac{Q_{IG} + (Q_{SG} \times \overline{SAF})}{q_{h1} \times T}\right]T \tag{14}$$

Energy Management Factor ($\overline{EMF}$). The actual heating load for an occupied house varies with the living habits of its occupants. Occupancy considerations have remained the largest unquantified design factor because they are regarded as a condition falling beyond the designer's control. The possible impact of different occupancy patterns, however, may be modeled by the introduction of an energy management factor. This factor constitutes a dimensionless ratio of the energy efficiency of an occupied house against that of the same house assumed to be maintained at "design" indoor temperatures under otherwise static or unoccupied conditions. In terms of variations in heating load such relationships may be expressed as:

$$\overline{EMF} = \frac{Q_{HH} \text{ occupied}}{Q_{HH} \text{ unoccupied}} \tag{15}$$

Energy management practices include night thermostat setback, protection of glass areas at night with insulated covers, and control of smoking to minimize ventilation requirements. Window management also extends to daytime protection with shades or blinds for control of excessive glare and room temperature variations, and to ventilation for dissipation of unwanted solar heat gains. Because of the wide range of variables entering into consideration, occupancy factors can have either positive or negative effect on the resulting net heating load. If a neutral management effort is represented by an $\overline{EMF}$ value of 1.0, likely variations in heating load attributable to better or poorer management practices may lead to $\overline{EMF}$ values in the range of 0.5 to 1.5. When occupancy conditions are disregarded, as in conventional thermal design and also in this analysis, the $\overline{EMF}$ value reverts to 1.0.

As present design and construction standards make no further distinction between estimated heating loads (Q_{HH}) under occupied or unoccupied conditions, such distinction is also overlooked in this analysis. When the energy management factor is introduced in calculations, it leads to performance estimates under "occupied" conditions; when it is not considered, estimates apply to "unoccupied" conditions. The application of the energy management factor is illustrated:

$$Q_{HH} = q_{h1}\left[t_{\bar{t}} - (t_{\bar{o}} + \frac{Q_{IG} + (Q_{SG} \times \overline{SAF})}{q_{h1} \times T})\right]T \times \overline{EMF} \tag{16}$$

Variable Temperature Design Procedure

The key to reliable variable temperature design is sufficiently accurate quantification of useful solar heat gains for the period under consideration. While such estimates may be based on any method offering the desired level of accuracy and sophistication, the proposed design procedure and the variable temperature design examples show applications of the $\overline{SAF}$ concept. The variable temperature design sequence, therefore, can be outlined as follows:

1. Determine the applicable solar acceptance factor ($\overline{SAF}$) value from references when available, or, if such references are not available (as for this analysis), on the basis of best judgment.

2. Calculate design glass areas ($\overline{DGA}$) for the various exposures if solar heat gains through opaque surfaces are to be recognized, or introduce another appropriate mechanism to allow such recognition.

3. Calculate solar radiation received through glass by any recognized method. As the procedure requires averaging of design data, it favors use of averaged solar radiation data[6]. Solar gains based on incident radiation data are estimated as:

$$H_{SG} = \overline{I}_v \times \overline{DGA} \times \tau \tag{17}$$

where

H_{SG} = solar heat gain received through glass, Btu/h

$\overline{I}_v$ = average solar radiation intensity received on vertical surface (or intensity of vertical radiation on average days), Btu/ft·h

τ = glass transmittance factor

Depending on design objectives, the average solar gain rate ($\overline{H}_{SG}$) may represent an average for a winter daylight period of approximately 8 hours, or a 24-hour average. To minimize such ambiguities, equation (6) provides for calculation of heat gain credit temperatures on the basis of the total gains received during the period under investigation. The procedure, however, remains flexible to allow calculation of this temperature credit (t_c) on the most meaningful basis for any given design task. Where both solar and internal heat gains are expressed in terms of Btu/h, the remaining calculations take the following form:

4. Calculate recognizable heat gains for the period under consideration. Such gains consist of internal and useful solar heat gains, and the procedure can be derived from equations (5) and (13) as:

$$H_{HG} = H_{IG} + (H_{SG} \times \overline{SAF}) \tag{18}$$

where

H_{HG} = total heat gain rate, Btu/h

H_{IG} = internal heat gain rate, Btu/h

5. Calculate nominal heat loss rate (q_{h1}) for average conditions as (c.f. eq. (4)):

$$q_{h1} = h_t + h_s \tag{19}$$

6. Calculate heat gain temperature credit (t_c) as (c.f. eq. (6)):

$$t_c = \frac{H_{HG}}{q_{h1}} \tag{20}$$

7. Calculate floating room temperature ($t_{\overline{f}}$) from equation (7). Average outdoor temperatures may be derived from degree-day numbers or weather data. This step also permits adjustments for microclimatic conditions, if warranted by the available data.

8. Calculate heating load (Q_{HH}) at any given thermostat setting or targeted indoor temperature ($t_{\overline{t}}$) from equation (8) or, if adjustments are to be made for energy management effects, from equation (16).

VARIABLE TEMPERATURE DESIGN EXAMPLES

Applications of the variable temperature design concept are illustrated by two examples which allow a comparison of the relative simplicity of the proposed design procedure against conventional design methods. The references were chosen at random from current literature in this field. They illustrate two commonly used procedures that allow heating load calculations with somewhat greater accuracy than the basic degree-day method--a modified degree-day method (example 1) and the bin method (example 2). The corresponding variable temperature analyses are streamlined to take advantage of the data given by these references. It may, however, be noted that the given design data are still more detailed than those likely to be considered in most house designs.

The variable temperature calculations of these examples are done in English rather than SI units to emphasize the derivation of this method from customary thermal design procedures, and to facilitate comparison with other data, also in English units, found in the two reference publications. The applicable conversion factors from English to SI units are:

$$\text{Btu} = 0.293 \times 10^{-3} \text{ kWh}$$

$$\text{Btu/h} = 0.293 \text{ W}$$

$$\text{Btu/h}\cdot{}^\circ\text{F} = 0.528 \text{ W/}^\circ\text{C}$$

Design Example No. 1

Data for Example No. 1, hereafter referred to as the "Brookhaven" example, were derived from a report[7] which discusses the thermal performance of an actual house, calculated on the basis of measured electric heat input and recorded temperature differentials during an unoccupied 42-day study period. Solar radiation levels were not measured but estimated on the basis of data from the closest weather station (Schenectady, N.Y.).

The Brookhaven analysis introduces a number of concepts not common to conventional design procedures. The nominal heat loss rate (q_{h1}) is identified as the "normalized heating requirement" (NHR). The calculated energy dissipation rate (measured electrical heat input divided by the average indoor-outdoor temperature differential) is referred to as the "Measured NHR". After adjustments for estimated solar heat gain and storage effects, the resulting value is identified as the "Adjusted NHR". As this factor is quantified in units of Btu/h·°F, such adjustments have the same effect as modifications to U values. On the basis of the estimated solar and internal heat gains the report shows a calculated break-even temperature of 51° F at a thermostat setting of 68° F. No further attempt, however, is made to reconcile differences between performance estimates and measurements through adjustments in temperatures.

The Brookhaven analysis is based on a conservatively low glass transmittance factor (0.68 for double glazing), but makes no allowance for solar gains through opaque surfaces other than assuming that such gains for the south wall would be sufficient to offset daytime heat losses. Solar heat gains for windows with other than southerly exposure are disregarded.

The variable temperature analysis, by contrast, uses a higher glass transmittance factor (0.78) consistent with its solar data source[6], even if this value ultimately should prove to be too high for the diffuse radiation component. It also recognizes solar gains through opaque surfaces with southerly orientation and gains through glass at any orientation. It may be noted that the magnitude of opaque surface gains estimated by the $\overline{\text{DGA}}$ procedure falls below the daytime south-facing wall losses (and thus below the quantity assumed by Brookhaven), but that the total estimated solar gains still show reasonable agreement. Such agreement suggests that any remaining differences between the predicted and observed performance of the study house may be attributable to differences between assumed and actual solar radiation levels, rather than to differences in data processing between the two analytical methods.

The design data used in the variable temperature analysis are derived from the Brookhaven report with a minimum of processing or reinterpretation.

Summary of Brookhaven Data. Location: Near Schenectady, N. Y.

Period of investigation: 42 days (January 18 to February 28, 1978).

Thermostat setting during the test period: 60° F.

Measured 42-day energy consumption (Q_{HH}): 2,320 kWh.

Average energy consumption: (2,320 kWh/42 days) x 3,413 Btu/kWh = 188,528 Btu/day.

Internal heat gain (H_{IG}): none.

Estimated average useful solar heat gain (H_{SU}): 208,540 Btu/day.

Calculated break-even temperature: 38.2° F (at $t_{\bar{t}}$ = 60° F).

Number of degree days below 38.2° F break-even temperature: 979.

Estimated NHR for test conditions: 399 Btu/h·°F.

Resulting 42-day heating load (at NHR = 399): 2,747 kWh.

Adjusted NHR to recognize solar gain effects: 361 Btu/h·°F.

Resulting 42-day heating load (at NHR = 361): 2,485 kWh

Average outdoor temperature calculated on the basis of above data: 14.9° F.

The average outdoor temperature of 14.9° F was calculated from Brookhaven data of a 42-day period, 38.2° F break-even temperature and 979 degree-days below the given break-even temperature. It must be noted that this break-even temperature is different from the initial Brookhaven reference noted above, as the two apply to different heat gain levels and thermostat settings. Adjustments to NHR values were made by Brookhaven and are explained in that report[7].

Variable Temperature Design Calculations.

1. Select $\overline{SAF}$ value.

The stipulated 60° F thermostat setting allows effective solar heat collection without excessive indoor temperature rise. Allowing for 5-percent loss of incident solar energy by reflection from interior surfaces, the $\overline{SAF}$ value may be estimated at 0.95.

2. Calculate $\overline{DGA}$ value.

The study house is built with a cathedral ceiling, so that both south-facing wall and roof surfaces receive solar heat. The aggregate area of the south wall and the vertical projection of the sloping roof is 1,400 ft^2. At a south glass area of 250 ft^2 the remaining opaque surface area can be estimated at 1,050 ft^2. As the house is built in post-and-beam construction with wood plank decking and sandwich panel exterior walls with approximately equal thermal resistance values for both roof (R-39) and walls (R-36), both elements can be reduced to an equivalent glass area by a single calculation (eq. (9)). Generalized calculations of glass and opaque surface heat gain rates under winter conditions suggest the use of an opaque surface-to-glass transmittance ratio of 0.01 (as compared to a value of 0.02 that may be applicable to conventional wood frame construction at typical insulation levels and double glazing). The lack of more detailed design data does not warrant closer investigation of this ratio for the present purposes. The desired values, therefore, can be estimated as:

Equivalent glass area of opaque south surface $\overline{EGA}$: 1,050 x 0.01 = 10.5 ft^2

South-facing design glass area $\overline{DGA} = \overline{AGA} + \overline{EGA}$: 250 + 10.5 = 260.5 ft^2

3. Calculate solar radiation through glass.

Solar gains for this example are estimated on the basis of average day data[6] for Schenectady, N.Y., by summation of one-third of January and two-thirds of February values. At a glass transmittance factor of 0.78 for all orientations, the calculated nominal solar gains (H_{SG}) are:

Solar radiation received through glass: $\bar{H}_{SG} = I_v \times \overline{DGA} \times$

South:	907 Btu/ft^2.day	x 260.5 ft^2	x 0.78 =	184,293 Btu/day
East and west:	452	x 129.0	x 0.78 =	45,480
North:	73	x 8.0	x 0.78 =	456
Total:			H_{SG} =	230,229 Btu/day
			÷ 24 h/day =	9,593 Btu/h

(17)

4. Calculate recognizable heat gains.

As there is no internal heat gain, the calculation is reduced to:

$$H_{HG} = H_{IG} + (H_{SG} \times \overline{SAF}) \qquad (18)$$

$$= 0 + (9{,}593 \times 0.95) = 9{,}113 \text{ Btu/h}$$

5. Calculate nominal heat loss rate.

q_{h1} = 399 Btu/h·°F as the basic reference used in Brookhaven calculations

6. Calculate heat gain temperature credit.

$$t_c = \frac{H_{HG}}{q_{h1}} \qquad (20)$$

(20)

$$= 9{,}113 \div 399 = 22.84\ °F$$

7. Calculate floating room temperature.

$$t_{\bar{f}} = t_{\bar{o}} + t_c \qquad (7)$$

$$= 14.9 + 22.84 = 37.74°\ F$$

8. Calculate heating load for the period under investigation.

$$Q_{HH} = q_{h1}(t_{\bar{t}} - t_{\bar{f}})\ T$$

$$= 399 \text{ Btu/h·°F} \times (60 - 37.74)\text{°F} \times 24 \text{ h/day} \times 42 \text{ days}$$

$$= 8{,}952{,}794 \text{ Btu} \qquad (8)$$

$$\div\ 3{,}413 \text{ Btu/kWh} = 2{,}623 \text{ kWh}$$

Comparison of Calculations. Although the above variable temperature performance calculation is based on a nominal heat loss rate q_{h1} or NHR = 399 Btu/h·°F, it does take into consideration solar heat gains and storage effects, and therefore corresponds to Brookhaven calculations at the "adjusted NHR" value of 361 Btu/h·°F. The Brookhaven analysis establishes this calculation as the reference for other calculations and performance measurements. The results of the variable temperature calculations, therefore, are also compared with this Brookhaven reference:

Brookhaven estimate at NHR = 361 Btu/h·°F: 2,485 kWh = 100 percent

at initial NHR = 399 Btu/h·°F: 2,747 kWh = 111 percent

Variable temperature estimate at

$\overline{SAF}$ = 0.95 and q_{hl} = 399 Btu/h·°F: 2,623 kWh = 106 percent

Measured 42-day energy consumption: 2,320 kWh = 93 percent

The streamlined variable temperature design estimate falls within the range of the two Brookhaven estimates based on considerably more lengthy calculations. All heating load estimates exceed the measured energy consumption, and this disagreement may be attributable to either overestimated heat loss rate (q_{hl} or NHR), overestimated room temperature (the heater is said to have had a relatively wide dead band and plotted indoor temperatures do fall below 60° F level), or underestimated solar heat gains. Differences in solar radiation level received at the test site and predicted from averaged solar data indeed may be considerable. Nevertheless, the differences between any two heating load calculations fall within the accepted tolerance range.

Design Example No. 2

Data for Example No. 2, a theoretical design example here referred to as "NBSIR"[8], represent application of the ASHRAE Bin Method.

The NBSIR analysis is very generalized and provides no detailed information on such factors as U values for walls and roof, or wall-to-window area ratios. The NBSIR solar data source is different from the average day data source[8] used in variable temperature analysis; consequently, differences in solar gain estimates are attributable not only to differences in design procedures but also those in data sources.

The full NBSIR analysis also makes adjustments for heating system efficiency and night thermostat setback. The net heating load before such adjustments is shown as "heating demand". The variable temperature heating load estimate before consideration of energy management factors is comparable to the NBSIR heating demand estimate, and the comparison is limited to that portion of analysis.

Design data are derived from NBSIR report[8] with a minimum of processing or reinterpretation.

Summary of NBSIR Data. Location: Columbus, Ohio.

Period of investigation: 5,878 hours (245 days, mid-September through mid-May).

Indoor design temperature: 70° F.

Outdoor design temperature: 4° F.

Heat loss rate at 66° F design temperature differential: 80,000 Btu/h.

Resulting nominal heat loss rate (q_{hl}): 80,000 Btu/h ÷ 66° F = 1,212 Btu/h·°F.

Estimated internal heat gain rate (H_{IG}) = 7,830 Btu/h.

Averaged solar heat gain rate (H_{SGY}) = 3,883 Btu/h (24-h basis).

Estimated heating load (subtotal of "Heating Demand" column (Q_{HH}): 120.11 x 10^6 Btu for the season.

Variable Temperature Design Calculations.

1. Select $\overline{SAF}$ value.

As no daytime temperature control is stipulated (after night setback the thermostat is returned to the indoor design temperature level), solar heat collection efficiency could suffer from excessive room temperature rise during the midday hours, suggesting the use of an $\overline{SAF}$ of about 0.75. The stipulated shading coefficients, on the other hand, suggest relatively low solar heat impact and may warrant the use of $\overline{SAF}$ = 0.85 as for average conditions.

2. Calculate $\overline{DGA}$ value.

NBSIR data[8] show actual south glass area of 90 ft^2, but provide no information on south facing wall area. The total given window area of 310 ft^2 suggests a glass-to-opaque wall ratio of approximately 0.3. The opaque wall-to-glass solar energy transmittance ratio may be estimated at 0.02. The south-facing design glass area, therefore, may be estimated as:

Estimated opaque south wall area: $90 \div 0.3 = 300$ ft^2

Equivalent glass area $\overline{EGA}$: $300 \times 0.02 = 6$ ft^2

South-facing design glass area $\overline{DGA} = \overline{AGA} + \overline{EGA}$: $90 + 6 = 96$ ft^2

3. Calculate solar radiation through glass.

NBSIR data show glass shading coefficients as 0.83 for south windows, 0.59 for east and west windows, and 0.56 for north windows. These values can be translated into glass transmittance factors on the basis of established relationships[5]. On the basis of such transmittance factors and NBS BSS 96[6] average day radiation data for mid-September through mid-May in Columbus, Ohio, the nominal solar heat gain (Q_{SG}) can be estimated as:

Solar radiation received through glass: $H_{SG} = \bar{I}_v \times \overline{DGA} \times \tau$

South: 191,858 Btu/ft^2 - season x 96 ft^2 x 0.70 = 12,892,857 Btu/season

East and West: 133,039 x 140 x 0.38 = 7,077,675 (17)

North: 72,321 x 80 x 0.33 = 1,909,274

Total: H_{SG} = 21,879,806 Btu/season

÷ 5,878 h/season = 3,722 Btu/h (24-h basis)

4. Calculate recognizable heat gains.

$$H_{HG} = H_{IG} + (H_{SG} \times \overline{SAF}) \tag{18}$$

$$= 7{,}830 + (3{,}722 \times 0.85) = 7{,}830 + 3{,}164 = 10{,}994 \text{ Btu/h}$$

5. Calculate nominal heat loss rate.

q_{h1} = 1,212 Btu/h·°F derived from NBSIR calculations

6. Calculate heat gain temperature credit.

$$t_c = \frac{H_{HG}}{q_{h1}} \tag{20}$$

$$= 10{,}994 \div 1{,}212 = 9.07° \text{ F}$$

7. Calculate floating room temperature.

NBSIR data show heating requirements below 65° F outdoor temperature and yield a degree-day number (mean temperature x heating hours ÷ 24) of 5,194 (°F-days). These data allow the following calculation of average outdoor and floating room temperatures for the 245-day period of investigation:

Average daily temperature differential $(65 - t_{\bar{o}}) = 5{,}194 \div 245 = 21.2° F$

Average outdoor temperature $t_{\bar{o}} = 65 - 21.2 = 43.8° F$ (7)

Floating room temperature $t_{\bar{f}} = t_{\bar{o}} + t_c$

$= 43.8 + 9.07 = 52.9° F$

8. Calculate heating load for the period under investigation.

$$Q_{HH} = q_{h1}(t_{\bar{t}} - t_{\bar{f}})\, T$$

$$= 1{,}212 \text{ Btu/h·°F} \times (70 - 52.9)\text{°F} \times 5{,}878 \text{ h} \quad (8)$$

$$121.82 \times 10^6 \text{ Btu}$$

Comparison of Calculations. Seasonal heating load calculated by:

NBSIR using the bin method: 120.11×10^6 Btu = 100 percent

Variable temperature method: 121.82×10^6 Btu = 101 percent

EVALUATION OF VARIABLE TEMPERATURE DESIGN EXAMPLES

The variable temperature design procedure demonstrated by the above examples allows markedly more rapid seasonal heating load calculations than the procedures used in the reference analyses. Its potential validity as a procedure for demonstrating compliance with performance-oriented construction standards can be assessed on the basis of the following considerations.

Significance of Agreement

The above examples use the same data, except for solar radiation references. The relative importance of accuracy in solar gain calculations varies with the portion of the total heating requirement satisfied by solar gains, known as the "solar heating fraction". This fraction for the two houses, based on the reference analysis data, may be estimated as:

$$\text{Solar heating fraction} = \frac{\text{Useful solar heat gain}}{\text{Useful solar heat gain + Net heating load}}$$

As the reference analyses do not distinguish between "useful" and "nominal" solar heat gains, calculations are based on the data given:

Brookhaven example: $\text{Solar heating fraction} = \frac{208{,}540}{208{,}540 + 188{,}528} = 0.525$

NBSIR example: $\text{Solar heating fraction} = \frac{3{,}883 \times 5{,}878}{(3{,}883 \times 5{,}878) + (120.11 \times 10^6)} = 0.160$

The above range suggests that the procedure may be applicable to all houses not specifically designed for a higher solar heating fraction. The 1-percent difference between NBSIR and variable temperature calculations can be considered insignificant, and the 6-percent difference between the basic Brookhaven and variable temperature calculations also is acceptable for all practical design purposes, particularly when the magnitude of the solar heating fraction and differences in solar radiation data are considered.

For the Brookhaven house, the magnitude of the solar heating fraction (0.525) and solar gain temperature credit (22.84° F) suggest not only a high level of internal heat utilization made possible by heavily insulated construction, but also marked temperature variations above the thermostat setpoint between cloudy and sunny days. Recorded room temperatures for the 42-day study period indeed show a low solar gain credit of 0.2° F on January 21, and a high of 17.4° F on February 9, while a 15° F temperature rise was exceeded on 6 days and a 10° F rise on 15 days. Temperature fluctuations above 10° F may be considered acceptable only at thermostat settings below 65° F because of overheating problems. Consequently, the Brookhaven house may actually represent a case of somewhat oversized south glass areas for its insulating value and solar heat storage arrangements (or thermal ballast). Under typical occupancy conditions of higher room temperatures, its solar acceptance factor, heat gain credit temperature, and solar heating fraction can be expected to fall below those estimated under the given test conditions.

Validity of $\overline{SAF}$ Values Used

Because the solar acceptance factor has been defined as a ratio of useful to nominal solar heat gains, it is a specific physical parameter, even if its design values still cannot be obtained by analytical or empirical methods. Analysis of available heat balance calculations suggests values in the range of 0.95 to 0.65, and the relative significance of possible error within that range of values may be judged from the position of the $\overline{SAF}$ factor in equation (16) where it appears as only one of a number determinants of equal importance:

$$Q_{HH} = q_{h1}\left[t_{\bar{t}} - t_{\bar{o}} + \frac{Q_{IG} + (Q_{SG} \times \overline{SAF})}{q_{h1} \times T}\right] T \times \overline{EMF}$$

The solar acceptance factor enters only into heat gain credit temperature calculations. At a 50-percent solar heating fraction the credit temperature equals the thermostat-to-outdoor temperature differential, and any error in the determination of the heat gain temperature credit through choice of inappropriate $\overline{SAF}$ values also leads to a comparable error in net heating load (Q_{HH}) calculations. At a lower solar heating fraction the effects of such error diminish proportionately.

Heat gain credit temperature and heating load calculations are also dependent on the accuracy of nominal heat loss (q_{h1}) estimates, which enter into consideration twice. While errors in $\overline{SAF}$ and q_{h1} in temperature credit calculations could be mutually compensating, the accuracy of net heating load estimates is dependent on the accuracy of both q_{h1} and $(t_{\bar{t}} - t_{\bar{f}})$ factors entering into this calculation (eq. (8)), where inaccuracies in q_{h1} are not self-compensating (an underestimated heat loss rate also leads to overrated heat gain temperature credit, increased floating indoor temperature $t_{\bar{f}}$ and underestimated design temperature differential $(t_{\bar{t}} - t_{\bar{f}})$).

At current construction standards one-third or more of the total heat loss is attributable to air leakage and ventilation effects that vary with indoor-outdoor temperature differentials and are difficult to quantify. Differences between estimated heating load and measured energy consumption in Brookhaven analysis may be attributable more to inaccuracies in heat loss estimates than to any other single factor. Common design tolerances may exceed 10-percent variation between calculated and actual average heat loss rates. In view of such considerations, the $\overline{SAF}$ values assigned to this analysis appear to fall within the limits of acceptable tolerance.

Applicability of Averaged Design Data

Difficulties in quantifying the transient response of buildings to variations in thermal environment do not permit streamlining design procedures for rapidly changing conditions. The minimum period of investigation to which the variable temperature design method may be applicable has already been identified at 5 days. While averaging indoor temperatures over a period of time allows more accurate approximation of the response of the structure, averaging outdoor temperatures and solar gain levels tends to mask temporary overheating and distort the actual heating load distribution pattern.

The basic degree-day method relies on simple temperature averaging, while the modified degree-day procedure allows some compensation for distortion in heating load pattern through introduction of correction factors that vary with outdoor temperatures. The Bin Method greatly minimizes this error and permits consideration of such short-term conditions as night thermostat setback but requires multiple calculations. If thermostat setbacks were to be considered in simple indoor temperature averaging, they would have to be accompanied by a weighting factor to maintain the correct relationship with actual outdoor temperature variations, which are also averaged.

Readily available averaged solar radiation data[6] are easier to use than clear day data derived from typical solar tables, which still require corrections for the prevailing local sky conditions. While averaged day data may tend to increase the predicted solar heating fraction by masking overheating on clear days, such errors may be minimized by the choice of an appropriate glass transmittance coefficient applicable to average day conditions in a given locality. For investigation of clear day conditions (or a period of clear days) the procedure also permits entry of clear day solar data.

While there are obvious limitations to the use of averaged day data, the agreement between the above variable temperature and reference calculations suggests that such data may find valid applications in the analysis of the seasonal thermal performance of residential buildings.

SUMMARY AND CONCLUSIONS

This paper introduces a proposed method for residential heating load calculations that is based on recognition of variable reference temperatures instead of traditionally fixed "design" temperatures. This method allows rapid consideration of credits for the use of solar energy, without adjustments to well-established and generally understood design coefficients, such as U values.

The proposed method allows calculation of solar heat gains through opaque surfaces on the basis of the same data as used for glass areas. It permits exclusion of solar gains that cannot be considered "useful" because of excessive room temperature fluctuations. It also provides a mechanism for further adjustments attributable to energy management practices commonly overlooked in the design process. These considerations are introduced in the form of three new design concepts based on recognized engineering design practices: "design glass area" ($\overline{DGA}$), "solar acceptance factor" ($\overline{SAF}$), and "energy management factor" ($\overline{EMF}$). This procedure allows more rapid solar heat gain and net heating load calculations than the processing of comparable data by conventional longhand design methods.

Comparison of heating load calculations yielded by the two design examples leads to the following conclusions:

1. The streamlined variable temperature design method can yield heating load calculations that fall within the accepted accuracy tolerance limits when compared to similar calculations by more lengthy procedures.

2. The averaged design data used in the variable temperature analysis led to heating load estimates that compare favorably with reference calculations based on more specific and detailed design data.

The variable temperature design method may allow sufficiently accurate demonstration of compliance with energy budget codes and performance standards to minimize the need for more rigorous analysis of most residential structures. Its validity remains to be verified by experimental field data. The concept, nevertheless, appears to permit a consistent and simplified approach to a complex design problem. Demonstrated agreement with calculations by other methods using comparable data seems to warrant further development of the proposed variable temperature design method.

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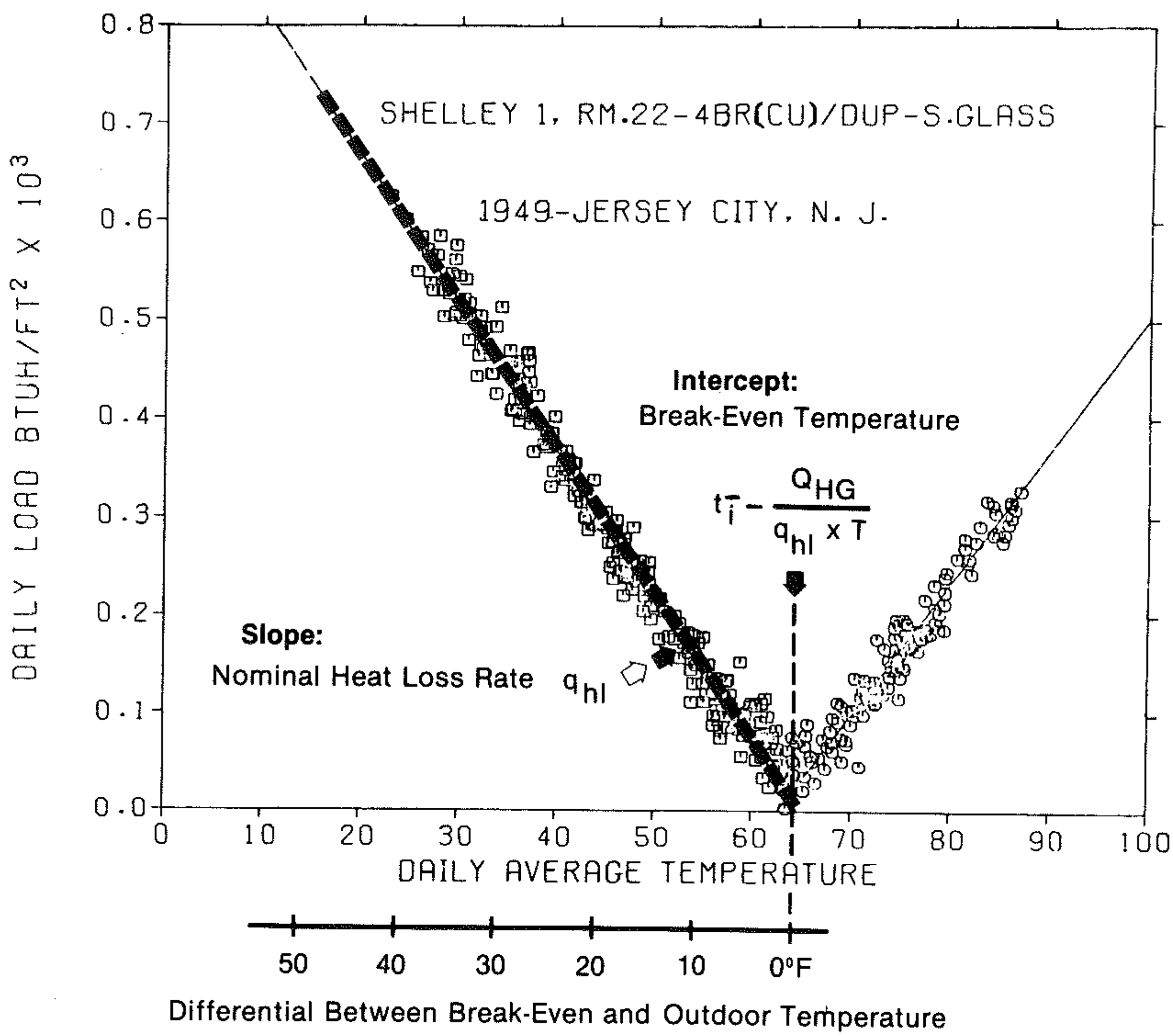

Fig. 1.--Assumed relationships in conventional thermal design, superimposed over a graph (4) of calculated daily total thermal loads for an apartment.