A Sensitivity Model for Energy Consumption in Buildings Part I. Effect of Exterior Environment

F. L. Lansing
DSN Engineering Section

A simple analytical model is developed for the simulation of seasonal heating and cooling loads of any class of buildings to complement available computerized techniques which make hourly, daily, and monthly calculations. An expression for the annual energy utilization index, which is a common measure of rating buildings, having the same functional utilization, is derived to include about 30 parameters for both building interior and exterior environments. This article, the first of two parts, reports on the sensitivity of a general class building to either controlled or uncontrolled weather parameters. A hypothetical office-type building, located at the Goldstone Space Communication Complex, Goldstone, California, is selected as an example for the numerical sensitivity evaluations. Several expressions of variations in local outside air temperature, pressure, solar radiation and wind velocity are presented. Further study is planned to cover in the ensuing parts the effects of the major parameters of interior building environment.

I. Introduction

Efforts to reduce energy consumption in residential and nonresidential buildings have been addressed quantitatively in enough detail in the literature. Several solutions have been proposed in the form of energy conservation regulations, standard measures, or procedures. Local government offices at city, county, and state levels, together with utility companies, private industries, and other institutions have been updating these regulations to suit the general need. Common modifications such as changing room temperatures in offices down to 18.33°C (65°F) in winter and up to 25.56°C (78°F) in summer, adding or increasing insulation materials to exterior walls, preventing air leakage by weather-stripping, reducing domestic hot water temperatures in boilers down to 40.56°C (105°F), etc., are a few examples to save the building or facility owner some maintenance and operation costs. The impact, however, of implementing these modifications and others on different types of buildings will not be the same since the savings depend on local weather conditions, building size, occupancy density, building activity, lighting levels, internal electrical loads, type of automatic controls, utility cost structure, etc. Taking different weather conditions, for instance, a 1% percentage change in the site heating degree days (HDD) in a selected winter month, around its design or average profile, will cause different percentage changes in the energy consumption for that month if an office building, a central control building, a hospital, a theatre, or a hotel are the buildings under observation.

Other building simulation computer programs have also been developed by various utility companies, architect/engineering offices, consulting engineers and other private profit making and nonprofit organizations. These computer programs are considered valuable tools for the hourly, daily, monthly and yearly simulations of a given building. The latter could be either in design stage or under a retrofit. To develop a general sensitivity model using these computer programs, many different buildings have to be tested. This proves to be a costly and a cumbersome task, although the technique is advantageous when only a single building is studied. An analytical model of less complexity and use cost thus needs to be established instead.

At the Deep Space Network communication complexes, the above energy conservation measures, in addition to other unique measures, have been studied, designed and implemented as part of a NASA-wide energy conservation program. Also, a computer model for energy analysis has been developed as a tool (Ref. 1) to facilitate the energy computations for over 200 buildings throughout the Network. Although the Network buildings can be grouped into several categories of similar functions, such as control buildings, office buildings, cafeterias, etc., the building size, orientation, energy loads, and environment are different. This will result in different responses to the same energy measures when implemented. Because it is known that these differences are small if the savings are expressed per unit gross floor area, grouping of buildings that have the same function is made to be compared on a uniform basis. Though the computer-aided design tools are more accurate and produce results that are specific to a particular building, general but simpler techniques are also sought for a quick evaluation and screening of measures and the understanding of the physical meaning of both building and equipment behavior under varied operating conditions.

The model should be structured to enable analysts to group analogous buildings which have the same function and occupancy activity but are of different size and floor area for handling on a similar basis when modifications or measures are proposed. The objective of this study is directed toward developing this analytical model for a general-class building. Various simplifying assumptions are made and presented in Section II in order to reduce the complexity of the multivariate problem at hand and to identify the key parameters affecting the total energy consumption and cost. The sensitivity analysis and the parameterization of these key variables are presented next for an example building to give numerically the relative magnitudes and trends of changes. Exterior weather patterns are studied first and presented in this part of the study. The rest of the operating system parameters will be addressed in a following report. While some parts of the present study are described, in some detail, to be viewed by a wide audience spectrum, the study is not intended to review the technical performance at the components level but to cover the whole building treated here as the system for a complete sensitivity analysis.

II. Energy Model and Assumptions

Although buildings operate differently in general, some classifications could still be made according to their function and internal loads. Many differences, however, still exist within same function buildings regarding the size of air-conditioned zones (or spaces); the capacity, and type of fan-coil units, air-handlers, coolers, boilers, furnaces, and heaters; the wattage, and type of lighting equipment; the wattage and type of motors, machines, electrical and electronic equipment; the number and activity of occupying people; the number and type of automatic temperature, pressure, humidity and flow controls; the hours of building operation; the design inside temperatures, etc. Different buildings may look different on the inside and the outside. However, from an energy consumption viewpoint, there are several common features that could be shown. These features are essential in forming the basis for the present model. Figure 1 depicts an interior building envelope with boundaries encompassing the building structure, occupants, and all internal functional equipment such as lighting, machinery, etc., except the heating, ventilation and air conditioning (HVAC) equipment. The HVAC equipment, although physically placed either inside or on the roof of the building, is treated in the model as external equipment to the building envelope. The selected control volume boundary hence encloses the building interior conditioned space, heating and cooling coils, while it leaves in the surroundings the HVAC equipment prime movers (such as boilers, chillers, electric motors for driving fans, motorized dampers, circulating pumps for cold and hot fluids etc.) in addition to the external outdoor equipment (such as external lighting, heated pools, etc.).

In air conditioning practice, a fraction ϵ of the total air flow (which ranges between 0 and 90%) leaves the air-conditioned space as exhaust air, and the balance returns to the fan-coil unit for reutilization in air conditioning. This percentage of the circulated air that is vented to the atmosphere as exhaust air is replaced equally by fresh outside air as a ventilation air. The ventilation process causes an additional burden on the air-conditioning system, depending on the fraction ϵ. The selection of the control surface, as illustrated in Fig. 1 to include this air circulation loop, is more suitable for energy analysis than when the building interior alone, without the air circulation loop, acts as the control volume.

The following assumptions and idealizations are made to simplify the analysis:

(1) A general-class building, with its multiple air-conditioned zones, air handlers, heaters, coolers, mechanical and electrical equipment, is treated as a

single macro-air-conditioned space with two air streams – air supply and air return streams. The average inside design temperature T_i, at any period, is taken as the weighted average of all the air-conditioned zones, with each zone represented by its share of the air flow rate. The volume of the equivalent air-conditioned space Ω is the sum of individual zone volumes, and the equivalent air flow rate V is assumed the sum of individual zone rates.

A general-class building could also be divided into several major sections where heating and cooling requirements are always at odds with each other throughout a given season. This case, however, should be distinguished from different intrinsic zone requirements during a given heating or a given cooling season. To illustrate this point further, consider for instance an industrial building which is divided into an office section and a machinery section. Within the office section, several air-conditioned zones may be present, with each having a minor variation of internal loads. All the zones within the office section may require either cooling or heating energies at a given time period. Also, the machinery section could be divided into several air-conditioned zones, with each generating internally an excessive amount of heat due to the operating machines. Subject to the internal load magnitude, the machinery section may require only cooling energy throughout the year, regardless of the exterior weather conditions. Hence, in the winter season, for instance, the two building sections will be in a different mode of air-conditioning. Therefore, for this type of building, subject to the size and energy requirement and mode of air-conditioning of each internal building section, the analyst should select either the simple approach of averaging the building conditions as one macrozone is appropriate or otherwise divide the building into a number of sections. The study model will proceed assuming the building is represented by a single macrozone composed of several analogous zone energy requirements.

(2) The exterior outside air temperature variations throughout the year are assumed for energy analysis to be categorized into only two seasons: summer and winter. Spring and fall periods will be merged as appropriate such that the summer and winter seasons occur for M_s and M_w months, respectively, which are not necessarily equal, but total 12. Heating energy is commonly consumed in the winter season, and cooling energy is commonly consumed in the summer season, with possible overlapping depending on the type of air handler and the internal load profile. Heating and cooling modes will not necessarily occur in winter and summer, respectively and their energy requirements are allowed in this general model to occur simultaneously, to suit different air-handling mechanisms. Hence, a distinction should be made between summer and winter seasons and cooling and heating modes.

(3) Variations of outside air temperature, throughout a given time period, are generally not predictable and uncontrollable. However, statistical averages obtained from weather bureaus for a particular hour, day, or month, for instance, are found satisfactory for the analysis. The longer the statistical period, the better the expectation of energy consumption will be. Available monthly or yearly data of cooling degree days (CDD) or heating degree days (HDD) for many locations (Refs. 2-6) could be converted, by simple expressions, to give the monthly and seasonal average outside air temperature. In the present model, only two average outside air temperatures are assumed to represent the weather pattern for the two seasons: $T_{o,s}$ for the summer season and $T_{o,w}$ for the winter season. These two temperatures are obtained by averaging the daily or monthly temperatures over the season period. To avoid the transient hourly heat transfer mechanism to and from a building, the minimum period of the system study is taken as 24 hours or one day.

(4) The internal heat gain to (or loss from) a building space Q_i is then taken as the sum of three major quantities. The first, the internal heat generated Q_g, is itemized into the following five categories: (a) due to occupants Q_p, (b) due to lighting Q_l, (c) due to electrical and electronic equipment Q_e, (d) due to mechanical equipment with electrical drives Q_m, and (e) due to internal fuel-fired appliances Q_f, such as ovens, ranges, etc. The second quantity is the heat transmitted to and from the building structure Q_t, which is divided into two categories: (a) a solar heat portion Q_r, due to direct solar radiation incidence upon fenestration areas and to indirect solar radiation falling on exterior walls and roofs (the solar heat is assumed independent of outside air temperature variations, although it takes into consideration reradiation to the sky), and (b) the heat transmission due to varying outside air temperatures on walls, roofs, floors, and glazing areas Q_a. Note that the heat exchanged and transmitted through interior walls or partitions inside the control volume, separating specially controlled zones, which are kept at specific temperature patterns different from their neighbors, is not considered in the model. The reason is shown by the first assumption, which considers a building

section as a single air-conditioned space kept at an equivalent "average" inside temperature, representing the weighted average of the various internal zones.

The third quantity of Q_i is the undesirable heat gain or loss due to infiltration or exfiltration Q_x, which is caused by the repetitive opening and closing of doors and windows, buoyancy losses, and air leakages through various cracks as a result of differences in the pressure between the inside and the outside environments. With only sensible heat calculations made, the energy sum is written as:

$$Q_g = Q_p + Q_l + Q_e + Q_m + Q_f \qquad (1)$$

$$Q_t = Q_r + Q_a \qquad (2)$$

$$Q_i = Q_g + Q_t + Q_x \qquad (3)$$

(5) The latent heat gain to or loss from a space is mainly due to three components: (a) humidity differences associated with moving infiltrated or exfiltrated air, (b) water vapor from the occupants due to their rate of metabolism, and (c) steam generated by internal sources such as food, coffee machines, etc.

In general, the latent heat portion of the total heat to a space is very small although it could be significant in certain building types. The assumption of eliminating the latent heat from direct calculations is made to simplify the analysis but will be incorporated later as the methodology is developed. A detailed itemization of the heat quantities that appear in Eq. (3) is given next.

(6) The heat given off by the building occupants Q_p depends mainly on the number of occupants and the type of occupant activity. The average sensible heat generated per person s_p ranges from 60 W/person (about 205 Btu/hr person) to 150 W/person (about 510 Btu/hr person). Hence,

$$Q_p = s_p N$$

where N is the number of people in the building. For comfortable human spacing, the population density D_p (=N/A) is usually designed within certain limits. Therefore, per unit floor area A, the sensible heat gain from people is written as:

$$Q''_p = s_p D_p \qquad (4)$$

(7) For the lighting equipment, electrical energy is converted first into both radiation and convection parts in different proportions depending on the type of light bulb used. The radiation energy part is then converted by multiple reflections and absorptions with the building walls, materials and furnishings, to both convection and conduction parts at a later time. The result of this transient heat transfer is only a reduction of the peak lighting load as "seen" by the air-conditioning system. Note that the illumination part, which is later converted to convection, is still affecting the air-conditioning load even after the lighting equipment is turned off. The heat transfer to the air could be assumed in quasi-steady state if a long period (preferably not less than 24 hours) is considered. In the present model, it is assumed for simplicity, that the heat generated from both incandescent and fluorescent lighting fixtures Q_l is instantaneous and averaged over a given day.

$$Q_l = E_l$$

where E_l is the electrical wattage of all types of light bulbs in the interior air-conditioned space. In practice, illumination engineers use energy intensity limits E''_l typically between 10-60 W/m^2 (1-6 W/ft^2). Therefore, per unit floor area A, the intensity of lighting heat gain Q''_l is:

$$Q''_l = E''_l \qquad (5)$$

(8) The heat gain to the space, as sketched in Fig. 1, from the interior electronic and electrical appliances which have no motor drives (such as computers, TV sets, electric irons, radios, etc.) also is assumed to be instantaneously convected to the air stream. According to the equipment wattage value E_e, the heat gain to the space Q_e, due to electrical and electronic equipment excluding equipment motor drives, is expressed as

$$Q_e = E_e$$

or per unit floor area,

$$Q''_e = E''_e \qquad (6)$$

(9) For internal electromechanical appliances and machines with motor drives, various configurations arise where both the motor and the driven machine are located either in the same space or one is inside and the other is outside the space. If E_m is the input

wattage of the motor that drives the machine, then the heat generated inside the space Q_m is expressed by any one of the following equations:

$$Q_m = E_m$$

if motor and driven machine are both inside the space, or

$$Q_m = \eta_m E_m$$

if motor is outside the space and the driven machine is inside, or

$$Q_m = (1 - \eta_m) E_m$$

if motor is in the space and the driven machine is outside, where η_m is the average motor efficiency. In general, the above relationships between Q_m and E_m can be written as

$$Q_m = \lambda_m E_m$$

where λ_m is a fraction ranging from $(1 - \eta_m)$ to 1 according to the physical location of motor-driven appliances. With common motor efficiencies ranging from 60% for small-size motors to 90% for large-size motors, the fraction λ_m will range from 0.1 to 1. In addition, if the power density of the electromechanical equipment E_m is specified by the designer, then per unit floor area, the heat rate Q_m is written as

$$Q''_m = \lambda_m E''_m \tag{7}$$

(10) The solar portion of the transient heat transmission to a building Q_r is independent of the outside air temperature variations. It depends on fenestration surface area, orientation, optical transmissivity, emissivity and absorptivity properties; absorptivity and emissivity of exterior opaque walls; and wall orientation and surface area. Under the cyclic variation of ambient air and sky temperatures, a building would have a net heat transmission Q_a out or into its conditioned space according to whether the average daily inside temperature is kept higher than or lower than the average daily outside air temperature, respectively. During a particular day, the calculations of the hourly heat transmission require the solution of a complex transient heat transfer problem involving the building interior thermal mass. The quasi-steady-state approximation, given by Appendix A, on a daily basis, is still, however, a common practice in air-conditioning calculations. The net daily solar heat gain Q_r is treated as a combined building-site characteristic parameter while the net daily ambient heat transmission Q_a is in linear proportionality to the difference between daily average inside and outside temperatures. This building approximation has been used also in several solar collector studies (Ref. 7) which produced good results, since by analogy, a building can be treated as a low-efficiency solar collector. The daily ambient heat transmission Q_a is written as:

$$Q_a = UA\,(T_o - T_i)$$

where U is a building characteristic parameter representing the equivalent coefficient of heat transmission, lost or gained from the building envelope and is given as an area-weighted average of all the individual exterior walls, roof, glazing and floor surface areas A_j. By using Eq. (A-9) in Appendix A,

$$U = \sum_j A_j U_j / A \tag{8}$$

Accordingly, per unit gross floor area, the ambient heat gain to space Q''_a is written in the form

$$Q''_a = U(T_o - T_i) \tag{9}$$

On the other hand, the solar heat transmission part Q_r is written using Eqs. (A-10) and (A-13) in Appendix A as the summation over all the building envelope areas A_j

$$Q_r = AR''$$

where

$$R'' = \frac{1}{A} \sum_j \left[\left(\alpha_j A_j R''_j \frac{U_j}{\theta_{o,j}} \right)_{\substack{\text{opaque}\\ \text{exterior}\\ \text{walls}}} + (\tau_j A_j R_j)_{glazing} \right] \tag{10}$$

where R'' is the daily solar intensity for the building envelope representing the area-weighted average of net solar heat gain to space due to both absorption by opaque walls and direct transmission by glazing, $\theta_{o,j}$ is the outside heat transfer coefficient as obtained from Eq. (A-4) in Appendix A, and R_j'' is the average daily solar radiancy falling on an area A_j at a given orientation. Per unit gross floor area, the solar heat portion Q_r'' is written as:

$$Q_r'' = R'' \tag{11}$$

(11) The heat gain to a space due to the undesirable infiltration or exfiltration Q_x is determined following the air change method assuming that infiltrated air replaces the air in the space volume by several times each hour. Thus

$$Q_x = \gamma \Omega \, n_x (T_o - T_i)$$

where n_x is the number of infiltration air changes per hour, Ω is the volume of the building space, and γ is a conversion factor representing the air specific heat times the air density product. For air, γ is calculated at normal temperature and pressure (NTP), [15°C (59°F) temperature and 101 KPa (29.92 in. Hg) pressure] as 0.342 Wh/m^3°C. For a unit gross floor area, the heat gain by infiltration Q_x'' is written as

$$Q_x'' = \gamma h n_x (T_o - T_i) \tag{12}$$

where h is the average height of the building macrozone. The value of n_x usually ranges from 0.5 hr^{-1} in residential to 3 hr^{-1} in most industrial buildings. Weather stripping is one measure in reducing the effect of infiltration losses.

(12) Ventilation air is allowed in all building designs for hygienic purposes as well as for dust, moisture and odor removal. The amount of ventilation air can be computed by any one of the following four methods: (a) a number of air changes per hour n_v, (b) a specific volume rate supplied per occupant, (c) an air rate required per unit floor area of space or (d) a percentage ϵ of the total circulated air flow to the space. These four computational methods are alternates, and only one method should be selected as appropriate. Because the introduction of outdoor air is an energy-consuming and an expensive process in either winter or summer seasons, the amount of such air should always be kept at a minimum unless it is otherwise specified by health standards. By arbitrarily selecting the first computational method using the number of air changes per hour, n_v, the amount of heat gain associated with the ventilation air, is calculated analogous to Eq. (12) for the infiltration flow. Hence,

$$Q_v'' = \gamma h n_v (T_o - T_i) \tag{13}$$

where the temperatures of both return and exhaust air streams in Fig. 1 are taken the same as the average inside space temperature T_i for a given day. Typical values of n_v for instance range from 1-12 in office buildings, 4-20 in restaurants, and 1-5 in apartments.

(13) A simple heat rate balance of the selected control volume in Fig. 1 on a given day and per unit floor area, gives

$$Q_i'' + Q_v'' = Q_c'' - Q_h'' \tag{14}$$

or by using Eqs. (1), (2), (3),

$$(Q_g'' + Q_r'') + (Q_a'' + Q_x'' + Q_v'') = Q_c'' - Q_h''$$

where Q_c'' is the cooling load of the air handler cooling coils (energy extracted from the control volume) and Q_h'' is the heating load of the air handler heating coils (energy added to the control volume). Note that simultaneous heating and cooling energies could be expended in a given period, such as the case in some multizone air handling units as described in Appendix B. In the summer season where $T_o > T_i$, the sum $(Q_i'' + Q_v'')$ is always positive and it becomes most economical if the proper selection of the cooling setpoint is made to set Q_h'' to zero. On the other hand, during the winter season where $T_i > T_o$, the sum $(Q_i'' + Q_v'')$ could be, from Eq. (14), a negative, a zero or a positive quantity depending on whether the combined internal heat generation and the solar gain $(Q_g'' + Q_r'')$ is less than, equal to, or greater than the combined transmission, infiltration, and ventilation losses $(Q_a'' + Q_x'' + Q_v'')$, respectively. The sum $(Q_g'' + Q_r'')$ is treated from Eqs. (4), (5), (6), (7), and (10) as a positive parameter, independent of the ambient air temperature fluctuations while the sum $(Q_a'' + Q_x'' + Q_v'')$ depends on the average temperature difference $(T_o - T_i)$ using Eqs. (9), (12), and (13). Therefore, for most economical space conditioning in the heating season, it is essential to minimize or equate to zero the cooling energy expenditure Q_c''. This could be achieved by properly selecting the heating coil setpoint. Note that the relative magnitudes of Q_c'' and Q_h'' are obtained only by detailed examination of the temperature controls of each air handler as described in Appendix B. In general, for a unit gross floor area, the gross sensible heat gain $(Q_i'' + Q_v'')$ from Eq. (14) is written using Eqs. (4), (5), (6), (7), (9), (11), (12), and (13) as

$$Q_i'' + Q_v'' = (s_p D_p + E_l'' + E_e'' + \lambda_m E_m'' + Q_f'' + R'') + [U + \gamma h(n_x + n_v)](T_o - T_i) \quad (15)$$

A plot of the relationship between the net sum $(Q_i'' + Q_v'')$ and the temperature difference $(T_i - T_o)$ from Eq. (15) gives, as sketched in Fig. 2, a straight line relationship in the form

$$Q_i'' + Q_v'' = \overline{A} - B(T_i - T_o)$$

where the intercept $\overline{A}$ is given as

$$\overline{A} = s_p D_p + E_l'' + E_e'' + \lambda_m E_m'' + Q_f'' + R'' \quad (16a)$$

The parameter $\overline{A}$ is always a positive quantity which represents the gross heat gain to the building space in the absence of any temperature difference between the inside and outside ambient temperatures. The building parameter $\overline{A}$ represents only the summation of heat gain due to people, lighting equipment, electronic equipment, electrical-powered machines, fuel-fired appliances and the direct solar heat gain through glazing areas and the indirect solar heat absorption by exterior walls. The slope B of the straight line relation in Fig. 2 is obtained also from Eq. (15) as

$$B = U + \gamma h(n_x + n_v) \quad (16b)$$

The parameter B represents the overall heat transfer coefficient due to thermal and fluid convection, conduction, and radiation between the building interior and exterior environments. The larger the slope B the more losses to the ambient the building will have for a given temperature difference $(T_i - T_o)$. A comparison of Eq. (16) and the known Hottel, Whillier and Bliss straight line performance equation for solar collectors (Ref. 7) indicates the resemblance in thermal behavior between solar collectors and buildings. By analogy, any building will act basically like an air solar collector but has two additional features: (a) internal heat generation rate, obtained from Eq. (16a) to equal $(A - R'')$, to augment the absorbed solar heat portion R'', and (b) heat losses due to air leakages, whether controlled or uncontrolled, to or from the ambient air due to infiltration and ventilation. This heat loss is represented by $\gamma h(n_x + n_v)$ in addition to the transmission part U.

The temperature difference at which the sum $(Q_i'' + Q_v'')$ changes its sign from positive (cooling mode) to negative (heating mode) is denoted by the characteristic building temperature difference ΔT^*. This is obtained from Eq. (15) by equating $(Q_i'' + Q_v'')$ to zero.

$$\Delta T^* = \frac{\overline{A}}{B}$$

For a fixed interior temperature T_i, the temperature T_o^* represents the characteristic outside air temperature below which heating the building is required and above which cooling is required. Note that T_o^* is always less than or equal to T_i since

$$T_i - T_o^* = \Delta T^* = \frac{\overline{A}}{B} \quad (17a)$$

Accordingly, in terms of T_o^*, the net heat (or loss) from the control volume is written as

$$Q_i'' + Q_v'' = B(T_o - T_o^*) \quad (17b)$$

Another interpretation of ΔT^* can be made for a given outside air temperature T_o, where the characteristic interior temperature T_i^* is defined as the "equilibrium" temperature at which the building should be, to result in zero heat gain or loss to the ambient. Note that T_i^* is always larger than or equal to T_o since

$$T_i^* - T_o = \Delta T^* = \frac{\overline{A}}{B} \quad (17c)$$

Accordingly, in terms of T_i^*, the net heat gain (or loss) from the control volume is written as:

$$Q_i'' + Q_v'' = B(T_i^* - T_i) \quad (17d)$$

Differentiation between heating and cooling modes or the winter and summer seasons is, therefore, a necessary step once the temperature T_o^* (or T_i^*) is determined. The first two modes represent the times when the outside air temperature T_o differs from T_o^*. If $T_o > T_o^*$, cooling will be the major mode of operation, and if $T_o < T_o^*$ heating will be the major mode of operation instead. Since the summer and winter seasons are taken in this model as the only two representative seasons of the year, the seasons could be distinguished by whether the daily average temperature T_o differs from an arbitrary selected temperature.[1] When T_o^* differs from T_o, several possibilities can be shown in Fig. 3. In the winter season, both heating or cooling modes can exist, depending on the size of internal heat gain compared to the ambient transmission losses. The situation in summer is the same but with a minor difference. Summer days [with average outside temperature $T_o > 18.33°C$ (65°F)] will require cooling if $T_o > T_o^*$, which is the most common mode of operation. Less frequently is the heating mode in summer which could occur during mild summer days when T_o is higher than 65°F if the temperature T_o^* (which always has to be less than the inside temperature T_i) falls above T_o. In summary, a check of the relative magnitudes of T_i, T_o^* should be made in order to determine the major mode of HVAC equipment operation.

(14) The design of the air circulation rate throughout the building is commonly made based on peak heat gain or peak heat loss as determined by $Q_{i,max}$ or $Q_{i,min}$ during extreme hours. Two-speed fans are sometimes installed to discharge one air rate in summer and another rate, usually lower, in winter. Per unit gross floor area A, the air circulation rate V'' has been found (Ref. 2) to depend on the building function and normally lies within the range: 10-60 $m^3/(m^2 hr)$ [0.5-3 $ft^3/(ft^2 min)$].

To check on the design value of V'', the following summer and winter flow rates V''_s and V''_w should be compared:

$$\left.\begin{aligned} Q''_c &= (Q''_i + Q''_v)_s = \gamma V''_s (T_i - T_{cp})_s + Q''_{v,s} \\ -Q''_h &= (Q''_i + Q''_v)_w = \gamma V''_w (T_i - T_{hp})_w + Q''_{v,w} \end{aligned}\right\} \qquad (18)$$

where T_{cp} is the cooling setpoint [which varies from 10°C (50°F) to 15.6°C (60°F)] and T_{hp} is the heating setpoint [which varies from 21.1°C (70°F) to 32.2°C (90°F)]. If only a single air flow rate is used throughout the year, the larger V'' value from Eq. (18) is selected subject to meeting the other ventilation requirements.

(15) The heating, ventilation and air-conditioning equipment is assumed in the model to operate during the year continuously, regardless of the transient behavior of internal load sources. In practice, the HVAC equipment controls will be intermittent in operation with some cycling on and off, depending on the internal loads and the desired inside temperature. Several energy conservation measures call for shutting down HVAC equipment during some predetermined periods such as on weekends, holidays, or during any other periods when the building is unoccupied. These measures claim to reduce the energy consumption and allow the inside space temperature to drift either higher or lower than what is normally set at. This latter temperature drift is not undesirable since the building is unoccupied. However, depending on the magnitude of the heat capacity (or thermal mass) of the building walls, furniture, equipment, etc., and depending on the shutdown duration period, these energy measures may or may not be very effective. The reason is due to the additional energy expenditure either in heating or in cooling needed to bring the building space back to its desired inside temperature after a given shutdown period. This additional energy consumption may or may not offset the energy savings incurred. However, one needs to distinguish between the difference of turning off an air handler during unoccupied periods as mentioned above, and the installation of timers, for instance, to lighting equipment or to any other non-HVAC related equipment for the purpose of energy conservation. The first affects the building energy content while the second is considered part of the internal load fluctuations or transient behavior. Alternating cloudy and sunny days are analogous to the second type of controls by timers which are already accounted for in the model as a reduced daily value of the parameter $\bar{A}$ in Eq. (16).

Since the hourly fluctuations of internal loads will be damped and averaged out if longer periods are studied, the building model assumes the quasi-steady-state case if a minimum of 24-hour modeling period is selected. The fluctuations that could take place due to the building thermal mass are minimized. The model also excludes measures of shutting down HVAC equipment and assumes their operation at all times is uninterrupted. Hence, the heating or cooling loading Q_h or Q_c will represent the equivalent quasi-steady-state loading for the day, month, or season under study.

(16) Regarding the cooling equipment, the type of input energy used could be either in electrical form (e.g., mechanical-driven vapor compression refrigerators) or thermal (e.g., absorption refrigerators driven by fossil fuel combustion, waste heat, solar energy, etc.).

[1] Heating degree days and cooling degree days are based in most weather bureaus on 18.33°C (65°F) reference temperature.

Similarly, the heating equipment could be either electrically powered (e.g., electric-resistance heaters and heat pumps) or thermally powered (e.g., fuel fired heaters or solar heaters, etc.). To simplify the energy accounting, the building space under consideration is assumed to be powered by only two types of energy – electrical and thermal. If more than one fossil fuel is employed in fuel-fired equipment, or if different nonfuel sources of heat are consumed, a unified thermal conversion should be used to unify all thermal energies or fuel rates into one appropriate thermal unit (e.g., kWh(t), therm, Btu, joule). One electrical meter connected to the building would indicate the energy sum of: (a) lighting equipment E_l, (b) internal electronic and electrical equipment which have no motor drives, E_e, (c) mechanical equipment with motor drives E_m, (d) cooling equipment E_c, if electrically driven, (e) heating equipment E_h, if electrically driven, and (f) all electrically powered accessories. The latter accessories are excluded from all internal heat gain calculations although some may have a minor effect. Accessories are divided in the analysis into two groups only as follows:

(a) Accessories that are related to space air conditioning and air circulation such as supply fans, return fans, exhaust fans, boiler circulating pumps, chilled water circulating pumps, condenser fans, cooling tower fans, etc. These accessories consume energy E_{ax1}, proportional to the load of their mother air-handling equipment. The air temperature rise in supply and return fans is assumed negligible with respect to the internal load Q_i. However, the fans' power is still computed as part of the accessories and it is not neglected.

(b) Accessories that are unrelated to HVAC equipment. These are likely to be physically located outside the building control volume, such as the external lights, water pumps for pools, etc. This type of accessory consumes electrical energy E_{ax2} in direct relation only to the building's external activity.

Accordingly, the total electrical meter reading E can be written as the sum:

$$E = E_l + E_e + E_m + E_c + E_h + E_{ax1} + E_{ax2} \tag{19}$$

A similar discussion of a single electrical meter could be made for a single "thermal" meter connected to the building. The thermal meter indicates the sum of all thermal energies consumed in (a) internal fuel-fired appliances G_f, (b) cooling equipment G_c, if heat driven, (c) heating equipment Q_h, if heat driven, and (d) all heat-powered accessories G_{ax}. The thermal accessories are commonly unrelated to HVAC equipment and located outside the building conditioned space such as external heating equipment in swimming pools or in domestic hot water boilers. For the above thermal meter, the total reading G will be

$$G = G_f + G_c + G_h + G_{ax} \tag{20}$$

Note that G_f and G_{ax} are the thermal energies supplied to devices inside the conditioned space and to space-external accessories, respectively, taking into account their conversion efficiency.

(17) To accommodate the simultaneous existence of both types of electrical and thermal-driven coolers or heaters, the fraction of load which is satisfied by each energy type needs to be defined. If β is the fraction of load that is provided by electricity powered devices, then after accounting for the component conversion efficiencies, the cooling equipment consumption will be

$$\left.\begin{aligned} E_c &= \beta_c Q_c / \eta_{ec} \\ G_c &= (1 - \beta_c) Q_c / \eta_{fc} \end{aligned}\right\} \tag{21}$$

and for the heating equipment, the consumption will be

$$\left.\begin{aligned} E_h &= \beta_h Q_h / \eta_{eh} \\ G_h &= (1 - \beta_h)\, Q_h / \eta_{fh} \end{aligned}\right\} \tag{22}$$

where η_{ec}, η_{fc} are the coefficients of performance of electrical-powered and heat-powered coolers, respectively, and η_{eh}, η_{fh} are the efficiencies of electrical- and heat-powered heaters, respectively.

Once the cooling load Q_c and heating load Q_h are computed, as in Appendix B, subject to the constraints of the temperature-control mechanism of air handlers, Eqs. (19) and (20) are used to give the total electrical and thermal meter readings E and G for the period under investigation.

(18) The energy utilization index is a common measure for rating buildings which have the same functional utilization. The index divides the total energy consumption for the season by the gross floor area of the

building. Since there are two types of energy provided, two energy indices are possible – one for electrical and the other for thermal. A modified representation of the annual energy consumption is to convert the electrical index to an equivalent thermal index by taking into consideration the conversion efficiency at the electrical power plant η_{pp} and add it to the thermal index. Hence, in equivalent thermal units, the energy utilization index representing the energy consumption per unit floor area is written as:

$$I = G'' + E''/\eta_{pp} \tag{23}$$

(19) The operating costs of energy are computed in the analysis based on a simple uniform tariff throughout the year, independent of demand charges, power factors or other seasonal charges. This uniform rate structure could be obtained by dividing the annual cost of energy by the annual energy consumption. If C_e and C_f are the costs of unit electrical and fuel energies, respectively, then the total energy cost per unit floor area C'' is written as:

$$C'' = C_e E'' + C_f G'' \tag{24}$$

The above 19 points constitute the assumptions and idealizations needed for the sensitivity analysis to be developed for any 24-hour period in any season. On a yearly basis, the annual energy index I_y, which is the sum of summer and winter effects, is computed taking into consideration an average of 730 hours per month as

$$I_y = 730(M_s I_s + M_w I_w) \tag{25}$$

where I_s and I_w are the summer and winter season utilization indices, obtained from Eq. (23). For most buildings, the annual energy index I_y ranges from 126 kWh/m^{2}yr (about 40,000 Btu/(ft^{2}yr)) to 1260 kWh/m^{2}yr (400,000 Btu/(ft^{2}yr)) depending on the building usage, size, internal loads, etc.

Before a general expression for modeling air handling equipment could be made, an investigation of various temperature controls was carried out (Appendix B). Different air-handler designs affect to a great extent the determination of the equipment heating and cooling loads. Q_h and Q_c, respectively. A simplified approach to this equipment loading is needed to enable the engineers to make a reasonable assessment of energy conservation measures without being lost in the specific design details.

III. Air Handler Model

Four different types of air handlers are briefly described in Appendix B, to demonstrate their differences in responding to heating and cooling needs. The variation of the quantities Q_c and Q_h takes place according to the type of temperature control, coil setpoints and air flow rate. For a sensitivity analysis to be complete, the specific features of the building air handler(s) dealing with the major loads have to be given. The analysis, in turn, will be narrowly directed toward a specific configuration rather than the general case originally intended. A generalized but simple air handler model is, therefore, sought for the present sensitivity study. While the selected methodology is still applicable to any specific air handler configuration, minor alterations may be necessary. In the simplified air handler configuration, the sum $(Q_i'' + Q_v'')$ is always positive in the cooling mode where only cooling energy is assumed expended upon the proper adjustment of cooling set points. Hence,

$$\left.\begin{aligned} Q_h'' &= 0 \\ Q_c'' &= Q_i'' + Q_v'' \end{aligned}\right\} \tag{26}$$

Similarly, in the heating mode, the net sum $(Q_i'' + Q_v'')$ is always negative where only heating energy is assumed consumed,

$$\left.\begin{aligned} Q_c'' &= 0 \\ -Q_h'' &= Q_i'' + Q_v'' \end{aligned}\right\} \tag{27}$$

Combining Eqs. (26) and (27) with Eqs. (19), (20), and (23), we can write the energy utilization index for cooling and heating modes as

$$\begin{aligned} I_{cm} = {} & (G_f'' + G_c'' + G_{ax}'')_{cm} + \frac{1}{\eta_{pp}} (E_l'' + E_e'' + E_m'' \\ & + E_c'' + E_{ax1}'' + E_{ax2}'')_{cm} \end{aligned} \tag{28}$$

$$\begin{aligned} I_{hm} = {} & (G_f'' + G_h'' + G_{ax}'')_{hm} + \frac{1}{\eta_{pp}} (E_l'' + E_e'' + E_m'' \\ & + E_h'' + E_{ax1}'' + E_{ax2}'')_{hm} \end{aligned} \tag{29}$$

Furthermore, by using Eqs. (21) and (22) and regrouping of terms, the indices I_{cm} and I_{hm} give the following

At $T_o > T_o^*$ (cooling mode)

$$I_{cm} = (G_f'' + G_{ax}'')_{cm} + \frac{1}{\eta_{pp}} (E_l'' + E_e'' + E_m'' + E_{ax1}'' + E_{ax2}'')$$

$$+ \left(\frac{1 - \beta_c}{\eta_{fc}} + \frac{\beta_c}{\eta_{ec}\, \eta_{pp}} \right) (Q_i'' + Q_v'')_{cm} \tag{30a}$$

At $T_o < T_o^*$ (heating mode)

$$I_{hm} = (G_f'' + G_{ax}'')_{hm} + \frac{1}{\eta_{pp}} (E_l'' + E_e'' + E_m'' + E_{ax1}'' + E_{ax2}'')$$

$$- \left(\frac{1 - \beta_h}{\eta_{fh}} + \frac{\beta_h}{\eta_{eh}\, \eta_{pp}} \right) (Q_i'' + Q_v'')_{hm} \tag{30b}$$

where the sum $(Q_i'' + Q_v'')$ at any particular mode is obtained from Eq. (15) as

$$Q_i'' + Q_v'' = (Q_p'' + E_l'' + E_e'' + \lambda_m E_m'' + Q_f'' + R'')$$

$$+ [U + \gamma h (n_x + n_v)]\ (T_o - T_i) \tag{31}$$

Equations (24), (25), (30), and (31) constitute the basic equations of the simplified model needed to obtain the sensitivity relations.

IV. Building Parameters

It is beneficial to the building designer, or owner, to observe, both in magnitude and direction, the effects of small or large changes imposed on the variables and parameters affecting the building energy consumption and operation costs. The sensitivity S, which is the measure of the dependency of a system output Y on variations of a particular input element or parameter X, keeping all other input parameters unchanged, is expressed analytically as:

$$\left.\begin{aligned} S &= \frac{\Delta Y/Y}{\Delta X/X} \\ \text{or} \qquad S &= \frac{dY}{dX} \cdot \frac{X}{Y} \\ \text{or} \qquad S &= \frac{d \ln Y}{d \ln X} \end{aligned}\right\} \tag{32}$$

where Δ represents a differential change in either X or Y, as sketched in Fig. 4. Equation (32) is computed at a reference operating condition of the entire system. A system operating at an optimum value of one of its elements should have zero sensitivity with respect to this element. The concept of sensitivity has been widely used in studying many physical systems, and its present application to energy consumption in buildings is a useful tool in identifying key parameters for optimum energy utilization. The input element, or parameter X, could be any one of the following 30 elements.

Weather parameters

- Outside air temperature T_o
- Outside air pressure p
- Local solar radiation R''
- Wind velocity v_d

Building architecture

- Average height of conditioned space h
- Equivalent heat transmission coefficient U
- Exterior wall absorptivity α
- Glazing transmissivity τ
- Glazing area/floor area ratio
- Opaque wall surface area/floor area ratio

Occupants

- Population density D_p
- Activity or sensible heat generated per person s_p

Internal equipment

- Lighting power density E_l''
- Power density of electronics and electrical equipment without motor drives E_e''
- Power density of electromechanical equipment with motor drives E_m''

Heat rate of fuel-fired equipment Q''_f

Hours of operation of each piece of equipment H

Environment control

Inside space temperature T_i

Total air circulation rate V''

Infiltration air change rate n_x

Ventilation air change rate n_v

Type of air-handler temperature control

Accessories

Power density of electrical accessories E''_{ax}

Heat rate of heat-powered accessories G_{ax}

Equipment performance

Coefficient of performance of coolers η_{ec}, η_{fc}

Efficiency of heaters η_{eh}, η_{fh}

Fraction of electrically powered coolers or heaters β_c, β_h

Electrical power plant efficiency η_{pp}

Costs

Cost of a thermal energy unit C_f

Cost of an electrical energy unit C_e

The building output Y could be any one of the following: (1) the energy utilization index I, (2) the total energy cost C, or (3) any suitable output from the above analysis. Denoting the energy index sensitivity to an input parameter X by S and the total cost sensitivity by $\overline{S}$, the analysis is carried out for the above input parameters using the equations described in Sections II and III. Accordingly, the sensitivity expressions for S and $\overline{S}$ using Eqs. (23), (24), and (32) will be written as:

$$S = \frac{X}{I}\left[\frac{\partial G''}{\partial X} + \frac{\partial (E''/\eta_{pp})}{\partial X}\right] \tag{33}$$

$$\overline{S} = \frac{X}{C''}\left[\frac{\partial (C_e E'')}{\partial X} + \frac{\partial (C_f G'')}{\partial X}\right] \tag{34}$$

Furthermore, if the unit electrical energy cost C_e is directly related to the unit thermal energy cost C_f, through the power efficiency η_{pp} as

$$C_e = \frac{C_f}{\eta_{pp}} \tag{35}$$

then, the sensitivity expressions S and $\overline{S}$ become identical, and in this case, Eqs. (23) and (24) are reduced to

$$C'' = C_f I \tag{36}$$

Since Eq. (35) is considered in practice a good approximation to the unit energy costs, it results that only the sensitivity S needs to be determined for all parameters involved.

The sensitivity expressions are also beneficial in determining the total system variation when more than one input parameter are simultaneously changing. Since the building output Y is a function of all input independent variables, X_1, X_2,. . . (i.e., $Y(X_1, X_2, \ldots)$ it can be proved that

$$\frac{\Delta Y}{Y} = S_1\left(\frac{\Delta X_1}{X_1}\right) + S_2\left(\frac{\Delta X_2}{X_2}\right) + \cdots \tag{37}$$

where S_1, S_2,. . . are sensitivities of the output Y to each of the parameters X_1, X_2,. . ., respectively.

When computing the changes in the annual energy index I_y with respect to an input parameter X due to changes in the summer only, in the winter only, or in both seasons, Eq. (25) gives

$$\Delta I_y = 730\left[M_s\left(\frac{\Delta I}{\Delta X}\right)_s \Delta X_s + M_w\left(\frac{\Delta I}{\Delta X}\right)_w\right]\Delta X_w \tag{38}$$

where I_s, I_w are the energy utilization indices obtained from Eqs. (30) and (31) for the cooling and heating modes of each season, and $\Delta X_s, \Delta X_w$ are the expected summer and winter changes in the input parameter X, respectively.

V. Data of a Hypothetical Office Building

The weather, architectural, internal and external loads data and operating conditions of a hypothetical office building are grouped to form a baseline configuration for the study. Typical office building data and design values are utilized to the maximum extent to represent closely actual conditions except for the artificial inclusion of some energy-consuming equipment that is not necessarily available in all office-type buildings. Hence, the chosen example building represents neither an existing office building nor an "ideal" configuration from energy conservation viewpoint. The building data are hypothetically constructed this way to illustrate the wide spectrum of energy modification measures and to assist in the numerical evaluation of the sensitivity expressions. The methodology that is followed in analyzing this building, however, could be applied to any other type of office or non-office building. The building site is arbitrarily selected to be at the Deep Space Network Communication Complex, Goldstone, California. Other data are categorized as follows:

Site	
Location	Goldstone, California, USA
Latitude	35° north
Elevation	About 610 m (2000 ft)
Barometric pressure p	93.91 kPa (27.82 in. Hg)
Weather	
Heating degree days, HDD[2]	1549.4°C (2789°F) day
Heating days d_w [2]	212
Heating months M_w [2]	7 (Oct – Apr)
Average outside air temperature during the winter[3] season $T_{o,w}$	51.84°F (11°C)
Sensible-cooling degree days, SCDD[4]	1143.9°C (2059°F) day
Cooling days[4] d_s	153
Cooling months[4] M_s	5 (May – Sept)
Average outside air temperature during the summer season[5] $T_{o,s}$	78.46°F (25.81°C)
Wind velocity v_d	16 km/hr (10 mph)
Building envelope	
Gross floor dimensions[6]	30.5 × 30.5 m (100 × 100 ft)
Space height h	3.1 m (10 ft)
Orientation	Sides are facing compass, E, W, N, and S directions
Occupants	
Gross floor area[7] allocated/person $1/D_p$	10.3 m^2/person (111 ft^2/person)
Sensible heat dissipation rate[8] s_p	117 W/person (400 Btu/(hr person))
Occupancy duration[9] H_p	2260 hr/yr
Internal lighting	
Design intensity D_l	32.3 W/m^2 (3.0 W/ft^2)
Lighting hours[10] H_l	2760 hr/yr
Internal electrical equipment	
Installed power density D_e [11]	16.1 W/m^2 (1.5 W/ft^2)
Operating hours[12] H_e	1000 hr/yr

[2]The site heating degree days per day is the difference between 18.33°C (65°F) and the daily mean temperature when the latter is less than 65°F. Data are taken from average weather statistics over a 10-year period for the Goldstone Space Communication Complex at Goldstone, California. For major cities or locations, HDD values appear in Ref. 2 or from weather bureaus.

[3]Obtained from the formula

$$T_{o,w} = \left(65 - \frac{\mathrm{HDD}}{d_w}\right) \quad , \,^\circ\mathrm{F}$$

[4]The site cooling degree day is defined in Ref. 2 by a complex expression compared to the heating degree day, since the energy consumption for a cooling process is affected not only by the dry bulb temperature, but also by the air relative humidity. Since latent heat calculations are assumed negligible in the present simplified model, an analogous definition to the heating degree day is adopted. The number of sensible cooling degree days per day is the difference between 18.33°C (65°F) and the daily mean temperature when the latter is higher than 65°F. Data are obtained from weather records of the site under investigation.

[5]Obtained from the formula

$$T_{o,s} = \left(65 + \frac{\mathrm{SCDD}}{d_s}\right), \,^\circ\mathrm{F}$$

[6]Building dimensions do not enter into the calculations since all the expressions are evaulated per unit floor area.

[7]Assuming 90 occupants in the building. The occupation density is selected to fit closely densities in office-type buildings.

[8]Assumed the same in both summer and winter seasons

[9]Assuming 9 hr/day (from 8 a.m. to 4 p.m.) for 251 working days per year. This excludes weekends and 10 holidays/year.

[10]Based on 11 hr/day (7 a.m. to 5 p.m.) for 251 working days/year. Evenly distributed over each season.

[11]An assumed figure which is obtained by summing the power of copying machines, TV sets, typewriters, CRT displays, vending machines, water fountains, and other appliances.

[12]Based on an average of 4 hr/day for 250 work days/yr.

Internal fuel-fired equipment	
Installed capacity[13] D_f	15 W/m^2
Operating hours H_f	1000 hr/yr
Internal thermal load Q_f''	1.71 W/m^2
Inside temperatures	
Summer[14] $T_{i,s}$	23.89°C (75°F)
Winter[14] $T_{i,w}$	23.89°C (75°F)
Architecture	
U factors[15]	
Single glass pane	1.13 Btu/(hr ft^2°F) or 6.42 W/m^2°C
Walls	0.3 Btu/(hr ft°F) or 1.70 W/m^2°C
Roof	0.18 Btu/(hr ft°F) or 1.02 W/m^2°C
Floor	Insulated
Area ratios	
Roof/floor	1.0
Gross wall/floor	0.1 for each orientation
Glazing/wall	0.30 for each orientation
Glazing/floor	0.03 for each orientation
Opaque wall/floor	0.07 for each orientation
Optical properties	
Glazing transmissivity, τ	0.80 (all glazing)
Exterior wall absorptivity, α	0.60 (all walls)
Ventilation air	
Ventilation air[16]	25 ft^3/(person min) or 0.71 m^3/(person min)
Number of air changes per hour n_v[17]	1.35
Infiltration air	
Number of air changes per hour n_x	1.2
Primary equipment	
Power plant thermal efficiency n_{pp}	0.33
Coefficient of performance[18] of electrical-driven chillers n_{ec}	2.8
Coefficient of performance[18] of thermal-driven chillers n_{fc} (absorption refrigeration)	0.65
Efficiency of electrical[18] resistance heaters/boilers n_{eh}	0.9
Efficiency of fuel-fired[18] boilers n_{fh}	0.6
Fraction of building cooling load[19] that is provided by electrical powered chillers β_c	1.0
Fraction of building heating load[20] that is provided by electrical resistance heaters β_h	0.0
Accessories	
Supply fan	
Air rate	1 ft^3/(ft^2 min) floor area
Operating hours H_{fan}	8760 hr/yr
Static pressure Δp_{fan}	3 in. water
Electric power per gross floor area[21] E''_{ax1}	5.75 W/m^2 (0.53 W/ft^2)

[13] A mini cafeteria is assumed located in the building providing some snacks for 90 persons. Cooking appliances using natural gas are assumed.

[14] Before the present mandatory energy conservation measures.

[15] Assumed the same in summer and winter seasons.

[16] This will amount to 2250 ft^3/min (63.71 m^3/min) for the building with 90 occupants. The ventilation rate is also equivalent to 0.225 ft^3/(ft^2 min) of gross floor area.

[17] Calculated by dividing 2250 ft^3/min (63.71 m^3/min) of ventilation air rate by the volume of space.

[18] Includes the accessories load such as condenser fans, cooling pumps, boiler pumps, etc.

[19] In this example building, all chillers are assumed electrically powered.

[20] In this example building, all heaters are assumed fuel-fired.

[21] Using the formula: Fan power = $\dot{V}$ cfm × ΔP in. water/ηfan × 8.507, watts, and assuming a 66% fan efficiency.

Domestic hot water heater	
Operating period	250 days/yr
Operating temperature[22]	40.56°C (105°F)
Type of heater	Fuel-fired (natural gas)
Energy consumption[23] G''_{ax}	0.53 W/m^2 (1473 Btu/yr ft^2)
External lighting	
Design intensity D_{ex2}	5.4 W/m^2 (0.5 W/ft^2)
Lighting hours[24] H_{ex2}	4015 hr/yr
Average external lighting power E''_{ax2}	2.48 W/m^2
Energy cost	
Yearly average cost of an electrical unit C_e	0.06 $/kWh(e)
Yearly average cost of a thermal unit C_f	0.02 $/kWh(t)

To manipulate Eq. (15), additional information representing the hypothetical office building at the configuration given is calculated, per unit gross floor area as follows:

People internal load Q''_p[25]	2.92 W/m^2
Internal lights $Q''_l = E''_l$	10.18 W/m^2
Internal electrical and mechanical applicances load[25] $Q''_e = E''_e$	1.84 W/m^2
Daily average solar intensity[26] R''	
For the summer season	20.38 W/m^2
For the winter season	15.05 W/m^2
Specific heat • density product[27] γ	0.318 Wh/m^3°C
Overall heat transfer coefficient[28] U	2.269 W/m^2°C (0.4 Btu/hr ft^2°F)
Building intercept parameter[29] $\overline{A}$	
In summer	37.03 W/m^2
In winter	31.70 W/m^2
Building slope parameter B	4.783 W/m^2°C
Characteristic temperature T_o^*	
In summer	16.15°C (61.07°F)
In winter	17.26°C (63.07°F)
Modes of operation during each season	
Since in summer $T_{o,s} > T^*_{o,s}$	All cooling
Since in winter $T_{o,w} < T^*_{o,w}$	All heating
Seasonal energy utilization index	
Summer I_s	113.614 W/m^2
Winter I_w	113.337 W/m^2
Annual EUI I_y	993.8 kWh/m^2 (315,100 Btu/ft^2 yr)

VI. Sensitivity Results

In the previous building example, deliberate inclusion of some energy-consuming equipment was made to show later on the impact of future energy conservation measures. Most parameters used in the sensitivity analysis are expressed in both the metric units (SI) and the common English units. Each parameter is analyzed, for convenience, separately to assess its individual weight on the energy utilization index. This first part of the study is directed toward the understanding of the exterior environment parameters. The rest of the building parameters are examined later in part 2 of this article. Sensitivity derivations are generally avoided for they are self-explanatory from Sections III and IV.

A. Outside Air Temperature

By neglecting the effect of outside air temperature variations on the efficiency of air-conditioning equipment and their associated accessories, the rate of change of I with respect to the average outside temperature T_o is found from:

[22]Following present energy conservation measures.

[23]Domestic water is assumed heated from 50 to 105°F. The energy consumed in Btu/yr ft^2 is obtained from:

$$\frac{\text{gallon}}{\text{day person}} \times \Delta T^\circ \text{ F} \times 8.33 \frac{\text{lb}_m}{\text{gallon}} \times \frac{\text{operating days}}{\text{yr}} \times \frac{D_p}{\eta_{boiler}}$$

taking $1/D_p$ as 111 ft^2/person, η_{boiler} as 70%, and 1 gallon/day per person.

[24]Based on 11 hr/day (7 to 5 pm) for full year.

[25]Using Eqs. (4), (5), and (6).

[26]Using Eq. (10) and Table 1.

[27]Obtained from the relation $\gamma = \gamma^* (P/P^*)$, where P^* is the standard atmospheric pressure (29.92 in. Hg or 101 KPa), γ^* is calculated at NTP as 0.342 Wh/m^3°C. See Ref. 4 for the variation of P^* with site elevation.

[28]Using Eq. (8) and Appendix A where

$$\theta_o = \left(1 + \frac{\nu_d}{3}\right) \text{ Btu/hr ft}^2\,{}^\circ\text{F}$$

where ν_d is the wind velocity in mile/hr. For 10 mph, θ_o is about 4.33 Btu/hr ft^2°F (24.59 W/m^2°C).

[29]Using Eq. (16).

$$\left.\begin{aligned} \Delta I_{cm} &= \zeta_c B \, \Delta T_{o,s} \\ \Delta I_{hm} &= \zeta_h B \, \Delta T_{o,h} \end{aligned}\right\} \quad . \tag{39}$$

where B is the building characteristic thermal loss rate given by Eq. (16b) and ζ_c, ζ_h are the cooling and heating coupling coefficients, respectively, given by

$$\begin{aligned} \zeta_c &= \frac{1-\beta_c}{\eta_{fc}} + \frac{\beta_c}{\eta_{ec}\,\eta_{pp}} \\ \zeta_h &= \frac{1-\beta_h}{\eta_{fh}} + \frac{\beta_h}{\eta_{eh}\,\eta_{pp}} \end{aligned} \tag{40}$$

Any change in the outside air temperature could take place because of possible yearly variations, due to a different location for the example building, or simply due to an error in measurement. An increase in the yearly average outside temperature will cause an increase in the summer index or a decrease in the winter index. Referring to the numerical example, for instance, a 5°F (2.78°C) increase in the average outside air temperature during the summer will represent an increase of the cooling degree days from 2059 to 2824 for the same number of days, or a 37% increase. This will increase the summer index by 14.38 W/m^2, which represents a 12.7% increase of the reference summer index. On the other hand, the winter reference index will decrease by 22.15 W/m^2 if the average outside air temperature in winter increases by 5°F (2.78°C). The latter represents a decrease of the heating degree days (HDD) from 2789 to 1730 for the same number of days, (i.e., a decrease of 38% of HDD). The change in the winter index, thus represents a 19.5% of the reference winter index. The change in the annual index I_y follows from the relation

$$I_y = 730\,(M_s \zeta_c B \; \Delta T_{o,s} - M_w \zeta_h B \; \Delta T_{o,w}) \tag{41}$$

The maximum effect of ambient temperature changes is felt when there is a drift to simultaneous higher temperature in summer and lower temperature in winter, than the reference weather given. If the average daily summer and winter outside temperatures are 83.46°F (28.59°C) and 46.84°F (8.24°C), respectively, (i.e., a change of +5°F during the summer and -5°F during the winter[30] or an amplitude change of 38%).

[30]The annual average outside air temperature is calculated as 17.2°C (62.93°F) based on 5 summer months at an average of 25.8°C (78.46°F). and 7 winter months at an average of 11°C (51.84°F). The changed annual average outside air temperature becomes 16.7°C (62.1°F).

The differential increase in the yearly index I_y becomes +165.7 kWh/(yr m^2), i.e., an increase of 16.7 compared to the reference conditions. This gives an EUI sensitivity to outside air temperature variations in the order of 0.44, which could have a significant effect on consumption.

A further examination of Eq. (41), shows that the dominant parameter in ΔI_y is the building characteristic slope B. Building measures which reduce the overall transmission U-factor, infiltration and ventilation rates n_v and n_x will reduce the magnitude of B, thus reducing the role of extreme weather fluctuations on the annual energy consumption.

B. Outside Air Pressure

Variations of the outside air barometric pressure for the example building could take place due to daily weather changes caused by the movement of earth-air boundary layer, or due to designing the building at a different elevation above the sea level, or due to some errors in pressure measurements. The specific heat-density product γ depends mainly on pressure changes (Ref. 4) and is proportional to the pressure P. For instance, if the example building is located at an elevation of 4000 ft instead of 2000 ft above sea level, the barometric pressure P will change to 25.84 in. Hg, i.e., a decrease of 7.1% from the given site pressure. The new value of γ becomes 0.295 Wh/m^3°C, and the slope $\Delta I/\Delta\gamma$ is obtained from Eqs. (30) and (31) as:

$$\left.\begin{aligned} (\Delta I/\Delta\gamma)_s &= h\,\zeta_c\,(n_x + n_v)\,(T_o - T_i)_s \\ (\Delta I/\Delta\gamma)_w &= -h\,\zeta_h\,(n_x + n_v)\,(T_o - T_i)_w \end{aligned}\right\} \tag{42}$$

This means that the lower the outside air pressure, the lower the consumption will be in both seasons. The result of the above elevation change is a decrease in the summer index by 0.4 W/m^2 and a decrease in the winter index by 3.9 W/m^2, which reduces the annual index by 3.9 W/m^2, and reduces the annual index, from Eq. (38), by only 21.4 kWh/m^2. yr. (i.e., -2% of the annual index). Hence, the effect of changing the outside air pressure or having different site elevation is not insignificant, with a sensitivity S in the order of +0.28.

C. Solar Radiation

Given that all other parameters are the same, the seasonal variation of solar radiation falling on a given building site would be affecting directly the building solar characteristic R'' for all wall orientations. Although there is an intrinsic relationship between the solar radiation on a given area and the earth-boundary layer outdoor air temperature, the two are assumed independent of each other. The slope $\Delta I/\Delta R''$ will be obtained from Eqs. (30) and (31) as

$$\left.\begin{aligned} (\Delta I/\Delta R'')_x &= \zeta_c \\ (\Delta I/\Delta R'')_w &= -\zeta_h \end{aligned}\right\} \quad (43)$$

which shows the direct correspondence between solar radiation and energy utilization index. For instance, an increase in R'' by 10% in both summer and winter due to either a change in R''_j from Eq. (10), or even an error in computations, will cause an increase in the summer index by 2.21 W/m^2 and a decrease in the winter index by 2.51 W/m^2. This results in a yearly index change of only −0.5%. The percentage change represents a sensitivity S in the order of 0.05, which could be larger if the summer increase of R'' is accompanied by a simultaneous decrease in winter. In this case, an increase of 10% of solar radiation in summer and a decrease of 10% in solar radiation in the winter will cause the index I_y to increase by 2.9%. The solar radiation sensitivity S is, therefore, in the order of 0.3, which is also a significant part of the exterior weather effects.

D. Wind Velocity

Changes in the local wind currents will affect the computations of the following quantities: (a) the building overall heat transmission coefficients U, (b) the infiltration rate n_x, due to air leakage, and (c) the exterior wall heat transfer coefficient θ_0 which appears in the solar parameter R''. An increase in the wind velocity by Δv_d affects each of the above quantities differently, for it will enlarge the coefficients θ_0 and U, but will reduce the parameter R''. The slope $\Delta I/\Delta v_d$ is obtained, by differentiation, as:

$$\left.\begin{aligned} \left(\frac{\Delta I}{\Delta v_d}\right)_s &= \zeta_c\,[\Delta R'' + (T_0 - T_i)(\Delta U + \gamma h\,\Delta n_x)]_s \\ \left(\frac{\Delta I}{\Delta v_d}\right)_w &= -\zeta_h\,[\Delta R'' + (T_o - T_i)(\Delta U + \gamma h\,\Delta n_x)]_w \end{aligned}\right\} \quad (44)$$

Note that the wind velocity builds up a pressure on the windward side of the building and a slight vacuum on the leeward side. The outdoor pressure buildup causes air to infiltrate in the windward side and exfiltrate on the leeward side. Offsetting the infiltration/exfiltration air is usually made, though not completely, by weather stripping and pressurizing the air circulation loop. For the example building, one finds that by increasing the average wind velocity to 20 mph instead of 10 mph (an increase of 100%), and assuming a negligible change Δn_x, then the change in the coefficient U, (using Eq. A-9) will be from 1.13 to 1.275 Btu/(hr ft^2°F) for glazing, from 0.3 to 0.309 Btu/(hr ft^2°F) for opaque walls, and from 0.18 to 0.183 Btu/(hr ft^2°F) for the roof. The new overall heat transfer coefficient will be 0.423 Btu/(hr ft^2°F) or 2.4 W/m^2°C. The solar parameter R'' will also change to 16.79 W/m^2 in the summer and to 12.77 W/m^2 in the winter. The changes in the energy index will be −3.61 W/m^2 in the summer and +6.63 W/m^2 in winter. The yearly index will change, therefore, by +20.7 W/m^2, i.e., increases by 2.1% percentage. The wind velocity effect, as a result, produces a sensitivity S in the order of +0.02. Though the effect of wind speeds in summer is advantageous to air conditioning for the wind acts as a cooling medium, this function in winter becomes undesirable. However, the sensitivity to wind currents is generally of a negligible magnitude.

VII. Summary

A simple analytical model is developed for the simulation of seasonal heating and cooling loads of any class of buildings. The model complements other computerized techniques, previously developed also at the Deep Space Network Engineering Section, to act as a design tool in screening and evaluating energy conservation measures. A general expression for the annual Energy Utilization Index (EUI) is given to include about 30 parameters for both building interior and exterior environments.

The first part of the study is focused on the EUI sensitivity to exterior weather parameters. A hypothetical office type building located at the Goldstone Space Communication Complex, Goldstone, California, is selected for the numerical analysis. The effects of variations in outside air temperature, pressure, and solar radiation are found to be of the same order of magnitude with a sensitivity range from 0.3 to 0.5. Wind currents have shown a negligible effect on consumption. Part 2 of the study, which covers the effects of the interior environment parameters, is being developed and will be presented in a future *TDA Progress Report*.

Acknowledgement

The author would like to acknowledge S. N. Higgins who assisted in the initial phase of the model and who made a number of helpful suggestions in the study preparation.

References

1. Lansing, F. L., et al., *A Computer Model Simulating Heating, Cooling and Energy Loads in Buildings*," Publication 78-76, Jet Propulsion Laboratory, Pasadena, Calif., Sept. 1978.
2. Stamper, E., and Koral, R. L., editors., *Handbook of Air Conditioning, Heating and Ventilating*, Industrial Press. Inc., New York, N.Y., 1979.
3. *Handbook of Fundamentals*, American Society for Heating, Refrigeration and Air Conditioning Engineers (ASHRAE), New York, N.Y., 1972.
4. *Carrier System Design Manual*, Carrier Air Conditioning Co., Syracuse, N.Y., Part 1-12, 1972.
5. Kusuda, T., and Ishii, K., *Hourly Solar Radiation Data for Vertical and Horizontal Surfaces on Average Days in the United States and Canada*, NBS Building Science Series 96 NBS, U.S. Dept. of Commerce, April 1977.
6. Threlkeld, J. L., *Thermal Environmental Engineering*, Prentice Hall, Inc., 1962.
7. Whillier, A., "Design Factors Influencing Solar Collector Performance," *Low Temperature Engineering Application of Solar Energy*, ASHRAE Publication, 1967, Chap. III, pp. 27-40.

Table 1. Solar radiation and outside air temperature of the example building

Month	Wall orientation: E	W	N	S	H	Daily average[2] outside air temperature,[b] °C (°F)
	Average daily solar radiation, Btu/ft^2, Ref. 5					
Jan	569	569	253	1249	942	6.4 (43.54)
Feb	778	778	334	1308	1296	11.1 (51.92)
Mar	970	970	415	1221	1703	12.1 (53.79)
Apr	1053	1053	513	911	1907	14.3 (57.67)
May[a]	1112	1112	627	747	2081	19.8 (67.57)
Jun[a]	1180	1180	700	711	2258	26.1 (78.96)
Jul[a]	1257	1257	650	797	2394	29.7 (85.42)
Aug[a]	1162	1162	511	1000	2147	29.1 (84.33)
Sep[a]	1017	1017	410	1294	1809	24.4 (75.96)
Oct	770	770	329	1326	1298	17.4 (63.33)
Nov	574	574	252	1276	953	10.7 (51.33)
Dec	544	544	225	1345	875	5.3 (41.50)
Daily average intensity in 5 summer months, Btu/hr ft^2 (W/m^2)	47.73 (150.5)	47.73 (150.5)	24.15 (76.2)	37.91 (119.6)	89.08 (280.9)	
Daily average intensity in 7 winter months, Btu/hr ft^2 (W/m^2)	31.30 (98.7)	31.30 (98.7)	13.82 (43.6)	51.40 (162.1)	53.42 (168.5)	

[a]Summer months.

[b]For Goldstone, California.

Definition of Symbols

$\overline{A}$	building characteristic thermal gain
A	gross floor area of building
B	building characteristic heat loss rate
C	energy cost
D	population or equipment density (per unit floor area)
E	electrical power input
e	emissivity
G	input fuel heat rate
H	duration of operation
h	average height of building zones
I	energy index
k	thermal conductivity of walls
M	months per season
N	number of occupants
n	number of air changes
P	atmospheric pressure
Q	heat rate
R	average solar radiation power
S	sensitivity
s	sensible heat generated
T	temperature
U	effective heat transfer coefficient for walls
V	air flow rate
v	wind velocity
X	general input parameter

x thickness

Y general output parameter

α absorptivity of exterior walls

β load fraction provided by electric powered heaters or coolers

γ specific heat times density product

Δ change

ϵ ventilation air/total circulation air ratio

η efficiency

θ combined heat transfer coefficient by convection and radiation

λ fraction

τ transmissivity of glazing

σ Stefan-Boltzmann constant

Ω building volume

Suffixes

a transmission by ambient air

ax accessories

c cooling effect

cm cooling mode

$conv.$ convection part

cp cooling setpoint

d wind currents

e electrical device

ec electrical-powered coolers

eh electrical-powered heaters

eq equivalent

f fuel-fired device

fc	heat-powered coolers
fh	heat-powered heaters
g	generated inside a space
h	heating effect
hm	heating mode
hp	hot setpoint
i	inside environment or internal to space
j	index
l	lights
m	mechanical motor-driven machines
o	outside air
p	people
pp	power plant
r	solar radiation effect
rad.	radiation heat transfer part
s	summer season
t	transmitted through walls
v	ventilation
w	winter season
x	infiltration/exfiltration
y	yearly profile

Superscript

$''$	per unit gross floor area

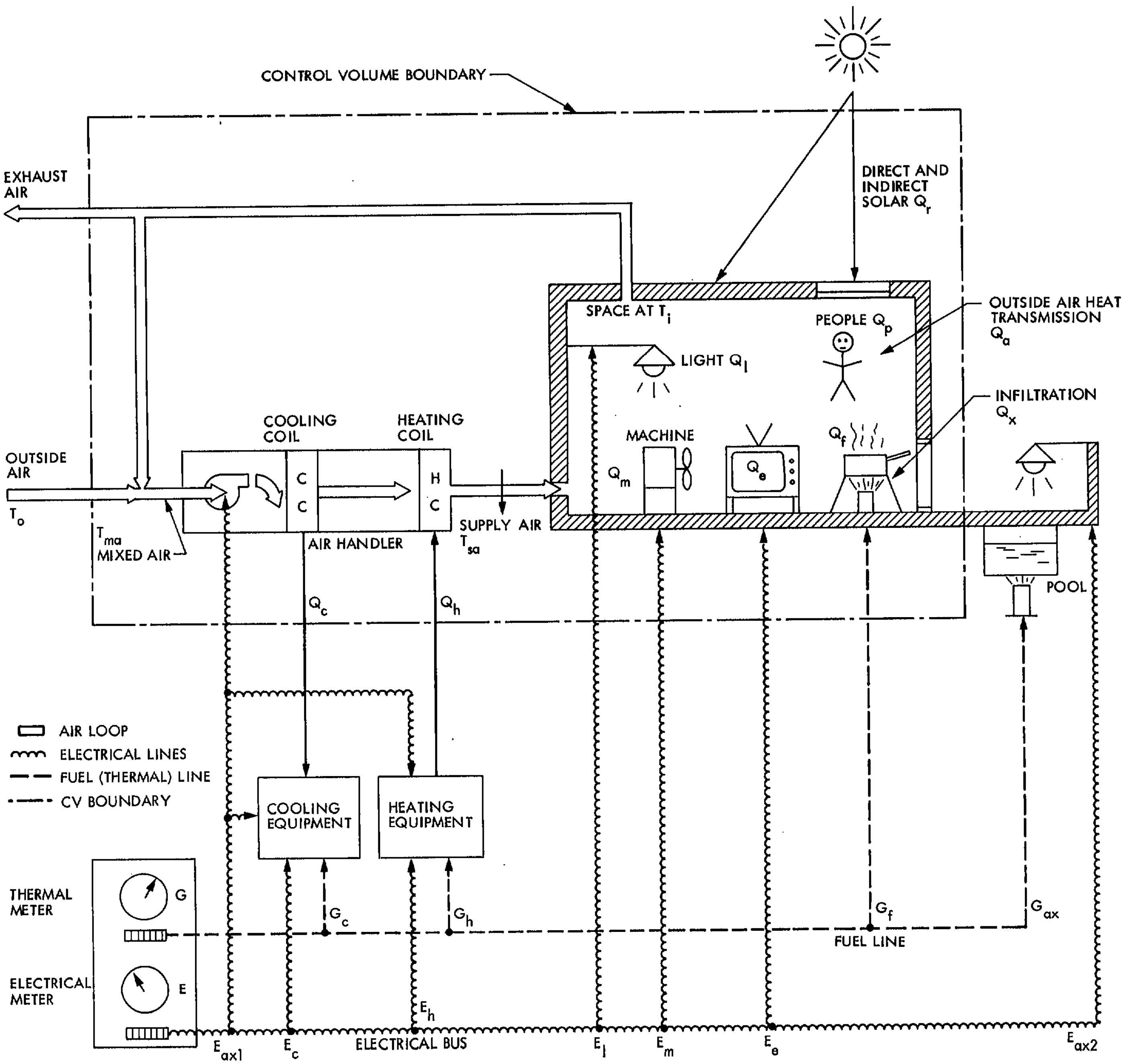

Fig. 1. Energy flows for the building model

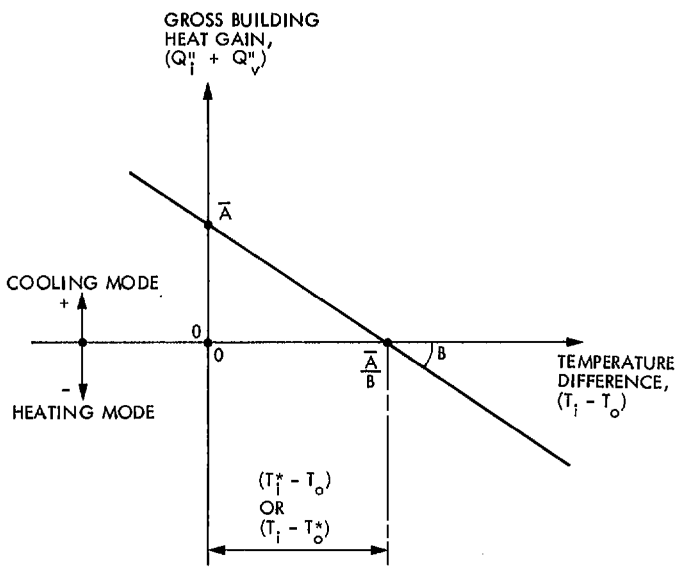

Fig. 2. Building characteristic temperatures T_i^*, T_o^*

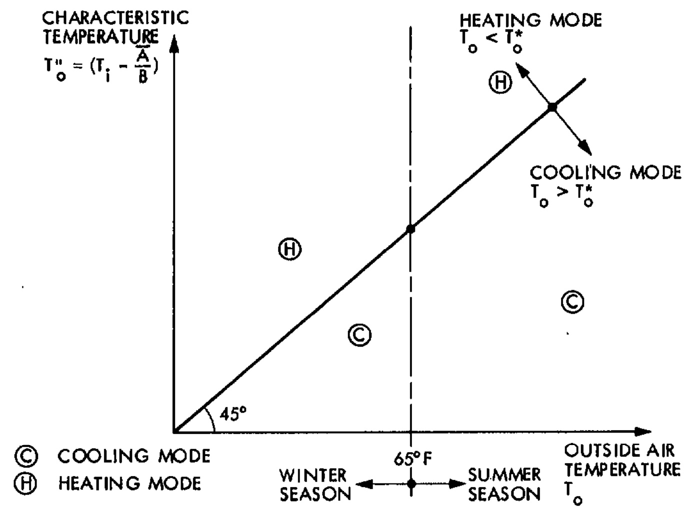

Fig. 3. Heating and cooling modes in different seasons

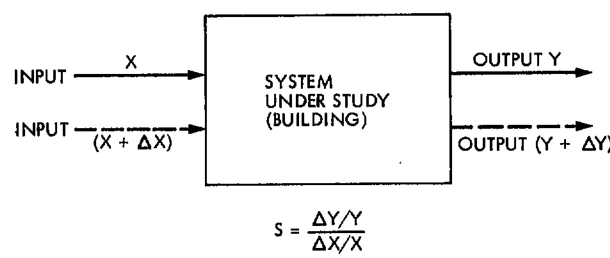

Fig. 4. Sensitivity definition

Appendix A

Heat Transfer Through Solid Walls, Roofs, Floors, and Windows

Because of fluctuating changes in outside air temperature and in incidence of solar radiation, the external thermal environment is constantly changing. Thus steady-state heat transmission seldom occurs in a building structure if short periods (of the order of an hour) are studied. However, the assumption of quasi-steady-state still provides a convenient procedure for some longer periods in the order of 24 hours or more. The following cases are examined assuming quasi-steady-state heat transmission:

I. Heat Transmission Through Walls, Roofs, and Floors

Consider the one-dimensional plane wall shown in Fig. A-1, representing a composite wall, roof, or floor where the wall height or width are large compared to the thickness x. The interior and exterior heat transfer coefficients θ_i and θ_o, respectively, are combining the effects of convection and radiation heat transfer. For the inside wall surface, the heat transfer from the outside Q_t'' through the wall interior at $T_{w,i}$ to the inside environment is written as

$$Q_t'' = \theta_i (T_{w,i} - T_i) \qquad \text{(A-1)}$$

where

$$\theta_i = \theta_{i,\,conv.} + \frac{\sigma e_i (T_{w,i}^4 - T_{s,i}^4)}{(T_{w,i} - T_i)} \qquad \text{(A-2)}$$

where σ is Stefan Boltzmann constant (0.1714×10^{-8} Btu/hr ft^2°R^4) and e_i is the emissivity of the interior wall.

On the other hand, the heat transfer through the outside wall layer, at steady-state, is the sum of the absorbed portion of direct solar radiation where the absorptivity is α_0, and the combined convection and sky radiation effects.

$$Q_t'' = \alpha_0 R'' + \theta_o (T_o - T_{w,o}) \qquad \text{(A-3)}$$

where θ_o is written as

$$\theta_0 = \theta_{o,\,conv.} + \sigma e_o \frac{(T_{s,o}^4 - T_{w,o}^4)}{(T_o - T_{w,o})} \qquad \text{(A-4)}$$

In Eq. (A-3), R'' represents the total incidence of solar radiation (direct, diffuse and reflected) upon the exterior wall surface, and $T_{s,o}$ represents mostly the sky temperature although it includes the effect of neighboring surfaces. The rate of heat transfer Q_t'' may also be expressed, from Eq. (A-3), as

$$Q_t'' = \theta_o (T_{eq.} - T_{w,o}) \qquad \text{(A-5)}$$

where T_{eq} is an equivalent outdoor temperature called the sol-air temperature, which combines the effects of solar radiation and outside air temperature T_o. Equations (A-3) and (A-5) give the fictitious temperature T_{eq} as

$$T_{eq} = T_o + \frac{\alpha_o R''}{\theta_o} \qquad \text{(A-6)}$$

The heat transfer from the outside environment through the composite wall Q_o'' will cause a differential in temperature $(T_{w,o} - T_{w,i})$, where

$$Q_o'' = (T_{w,o} - T_{w,i}) \Big/ \sum_j \frac{X_j}{K_j} \qquad \text{(A-7)}$$

where K_j and X_j are the thermal conductivity and the thickness, respectively, for the jth layer constituting the composite site.

Combining Equations A-1, A-5, and A-7 at steady-state, an overall heat transfer coefficient U is obtained where the end temperatures T_{eq} and T_i are utilized:

$$Q_o'' = U(T_{eq} - T_i) \qquad \text{(A-8)}$$

and

$$1/U = \frac{1}{\theta_i} + \frac{1}{\theta_o} + \sum_j \frac{x_j}{K_j} \qquad \text{(A-9)}$$

the determination of the coefficients, θ_i, θ_o, and U are usually made either by direct calculations from Eqs. (A-2), (A-4), and (A-9) or from the literature such as Refs. 2, 3, 4, and 6 for a variety of building materials. The coefficient θ_o is particularly dependent on the exterior wind velocity and the outside wall surface emissivity. Substitution of the sol-air temperature expression from Eq. (A-6) into Eq. (A-8) yields for a solid, wall, roof, or floor

$$Q_t'' = U(T_o - T_i) + \frac{\alpha_o R'' U}{\theta_o} \qquad \text{(A-10)}$$

The first term in the right-hand side of Eq. (A-10) represents the quasi-steady-state heat transmission to the space interior Q_a'' due to the ambient air temperature, while the second term $(\alpha_o R'' U/\theta_o)$ represents the solar portion Q_r''. The latter is due to the effect of solar radiation intensity as it falls on the outside surface with the modifier ratio (U/θ_o).

II. Heat Transmission Through Windows

The same discussion presented in Section I applies for the heat transmission through fenestration areas. Equations (A-2), (A-4), and (A-9) are also applied using the glass properties in place of the wall properties. However, the total heat transmission to the interior space will be somewhat different from that given by Eq. (A-10) since it should allow for the directly transmitted solar radiation portion $\tau R''$. Accordingly, the modified expression Q_t'' for the glazing areas (Ref. 6) will be:

$$Q_t'' = U(T_o - T_i) + \left[\frac{\alpha R'' U}{\theta_o} + \tau R''\right] \qquad \text{(A-11)}$$

The first term in the right-hand side in Eq. (A-11) is similar to that in Eq. (A-10), representing the heat transmission portion Q_a'' due to the temperature difference between internal and external environments, while the second bracketed term represents the solar portion Q_r''. For common window glass, the absorptivity is in the order of 3% and the thermal resistance portion in Eq. (A-9), $(\sum_j x_j/K_j)$ could be neglected. Therefore,

$$U_{glass} \cong 1 \Big/ \left(\frac{1}{\theta_i} + \frac{1}{\theta_o}\right) \qquad \text{(A-12)}$$

The term $\alpha R'' U/\theta_o$ is reduced to $\alpha R''/(1 + \theta_o/\theta_i)$ which could be entirely neglected from Eq. (A-11). Hence, for low absorptivity glass

$$Q_{t,glass}'' = U(T_o - T_i) + \tau R'' \qquad \text{(A-13)}$$

Equations (A-10) and (A-13) constitute the basic equations needed to determine Q_t'', combining all walls, floors, roofs and glazing areas.

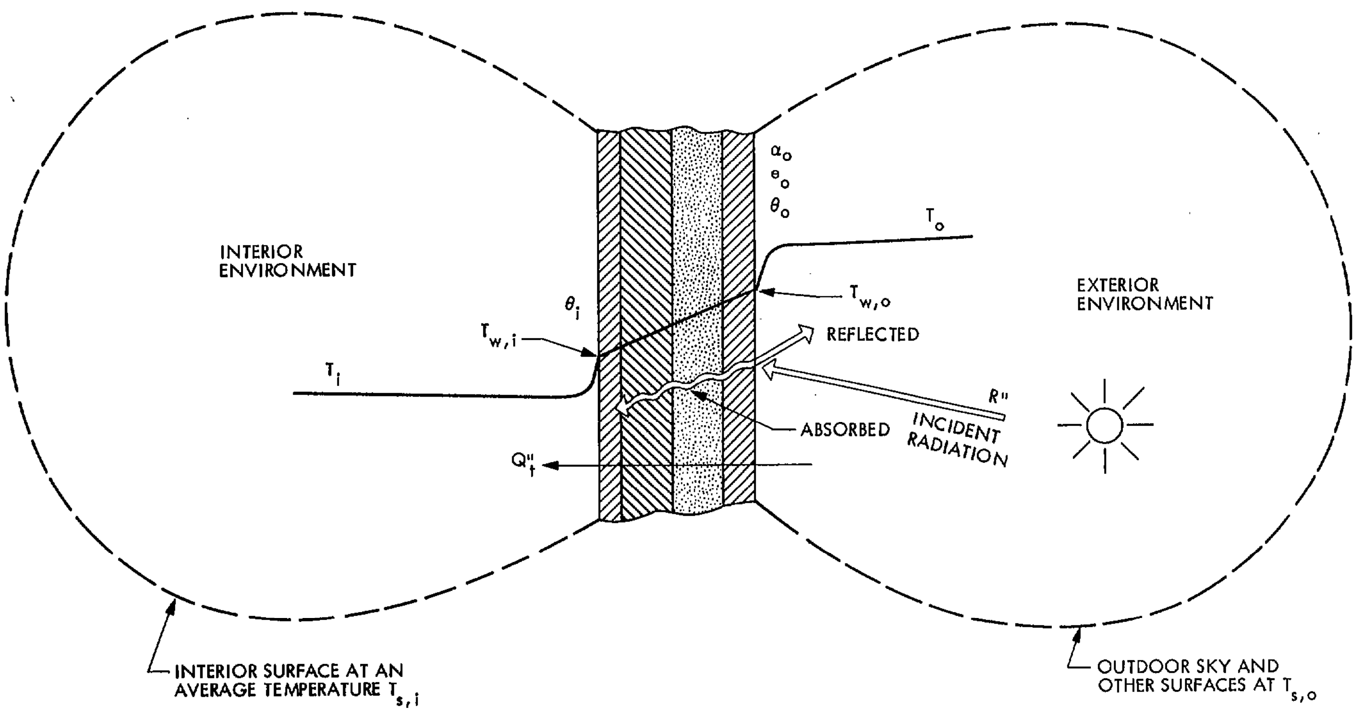

Fig. A-1. Temperature profile across a solid wall separating two environments

Appendix B
Major Air Handling Systems

This appendix is an extension of the building model presented in Section II, where four major types of air-handling systems and their temperature controls are presented. The cooling and heating loads Q_c, and Q_h for each of the four air handlers are analyzed to show their differences.

I. A Dual-Duct System

The dual-duct air handler system is used for multizone applications where two parallel ducts are employed – one carrying cold air, the other warm air. The air flow rate to each zone is constant at all times. In each conditioned zone or space, a mixing box actuated by a room thermostat mixes the warm and cold air in the proper proportions to satisfy the varying space temperature needs. In all seasons, regardless of the net internal heat gain or loss, simultaneous cooling and heating of the air are needed at the air handler. Figure B-1 shows the temperature of air in the various stations. Simple mass and heat balance equations are written for the control volume encompassing the space, which is kept at T_i and has an internal load Q_i together with the mixing box. The mixing box mixes two air streams: the cold stream with a flow rate V_c at temperature T_{cp} and the hot stream with a flow rate V_h at temperature T_{hp}. Hence,

$$V_c + V_h = V$$

$$\gamma V_c T_{cp} + \gamma V_h T_{hp} + Q_i = \gamma V T_i$$

Solving for V_c and V_h gives

$$V_c = [Q_i + \gamma V(T_{hp} - T_i)] / \gamma (T_{hp} - T_{cp}) \qquad \text{(B-1)}$$

$$V_h = [\gamma V(T_i - T_{cp}) - Q_i] / \gamma (T_{hp} - T_{cp}) \qquad \text{(B-2)}$$

Denoting the mass ratio of ventilation air to total circulation air by ϵ, the mixed air temperature T_{ma} is obtained from a heat balance of the mixing process as:

$$T_{ma} = \epsilon T_o + (1 - \epsilon) T_i$$

or

$$T_{ma} = \epsilon (T_o - T_i) + T_i \qquad \text{(B-3)}$$

The cooling load of the cooling coil Q_c will be written as:

$$Q_c = \gamma V_c (T_{ma} - T_{cp})$$

By substituting for V_c and T_{ma} using Eqs. (B-1) and (B-3) and using the definition of the ventilation heat gain Q_v from Eq. (13), then

$$Q_c = [Q_i + \gamma V(T_{hp} - T_i)] [Q_v + \gamma V(T_i - T_{cp})] / [\gamma V(T_{hp} - T_{cp})] \qquad \text{(B-4)}$$

Similarly, the heating load of the heating coil Q_h will be expressed as

$$-Q_h = \gamma V_h (T_{ma} - T_{hp})$$

or

$$-Q_h = [\gamma V(T_i - T_{cp}) - Q_i] [Q_v - \gamma V(T_{hp} - T_i)] / [\gamma V(T_{hp} - T_{cp})] \qquad \text{(B-5)}$$

If the cold setpoint T_{cp} or the flow rate V are adjusted in the cooling mode such that on the average

$$Q_{i,s} = \gamma V(T_i - T_{cp})$$

then, it follows from Eq. (B-5) that no heating is needed and the average cooling load will be equal to

$$Q_c = Q_i + Q_v \qquad \text{(B-6)}$$

On the other hand, if the hot deck temperature T_{hp} or the flow rate V are adjusted in the heating mode such that on the average

$$-Q_i = \gamma V(T_{hp} - T_i)$$

Then, it follows from Eq. (B-4) that the cooling load Q_c will be zero, and the average heating load will be equal to

$$-Q_h = Q_i + Q_v \qquad \text{(B-7)}$$

In practice, the temperatures T_{cp} and T_{hp} are about 50°F and 80°F in the cooling season, respectively, and 60 and 100°F in the heating season. Consequently, simultaneous heating and cooling energies are always required to condition the space according to Eqs. (B-4) and (B-5) for Q_c and Q_h computations in any season. However, a heat balance of the control volume encompassing the building and the air handler would still yield at all times:

$$Q_c - Q_h = Q_i + Q_v \tag{B-8}$$

Equation (B-8) can also be verified by summing the equipment loads for the general case, which are represented by Eqs. (B-4) and (B-5), and also for the seasonal average case, as represented by Eqs. (B-6) and (B-7).

II. Face and Bypass Damper System

The second common air-handling system is sketched in Fig. B-2, where either heating energy or cooling energy is expended at one time in a constant air flow. The air temperature modulation is made by using bypassed mixed air dampers to suit the varying space internal loads. Cooling only takes place if the sum $(Q_i + Q_v)$ is positive (or $T_o > T_o^*$), while heating is only taking place if the sum $(Q_i + Q_v)$ is negative (or $T_o < T_o^*$). A simple heat balance of the control volume encompassing both the building and the air-handling system gives for the first case when $(Q_i + Q_v)$ is positive:

$$\left.\begin{aligned} Q_h &= 0 \\ Q_c &= Q_i + Q_v \end{aligned}\right\} \tag{B-9}$$

If $(Q_i + Q_v)$ is negative, then

$$\left.\begin{aligned} Q_c &= 0 \\ -Q_h &= Q_i + Q_v \end{aligned}\right\} \tag{B-10}$$

For this type of air handler, either Eq. (B-9) or Eq. (B-10) is applied to determine the equipment loads at a given hour. The equipment controls operate the heating or cooling coils at one time according to the algebraic sign of the sum $(Q_i + Q_v)$. This type of air handler is found advantageous for buildings, such as theaters, supermarkets, central control buildings, etc., with large positive internal heat gain Q_i which requires a cooling energy expenditure all year around.

III. Variable Air Volume System

In a variable air volume (VAV) system, as sketched in Fig. B-3, the supply air temperature is held constant (either at the cold setpoint or the hot setpoint) while the amount of air flowrate is changed to satisfy the varying space load requirements. The VAV system controls are thus in contrast with the constant volume air handler systems where the air flow rate to a space remains the same while the supply air temperature is varied to suit the varying space loads. Multizone dual ducts with cold and hot air streams and VAV controls are used to supply either cooling or heating energies according to different zone needs. Subject to a zone internal load, either heating or cooling is supplied at one time. Similar to the case of face and bypass damper system, the cooling mode will only take place if the sign of the sum $(Q_i + Q_v)$ is positive and heating will only take place if the sum $(Q_i + Q_v)$ is negative. Accordingly, the above Eqs. (B-9) and (B-10) are also applicable to VAV air-handler systems.

IV. Single Cold Duct and Terminal Reheat System

The temperature control in this type of air-handler system, as sketched in Fig. B-4, is achieved by two different mechanisms. First, in the cooling mode, the mixed air stream is cooled to a fixed cold setpoint temperature T_{cp}, independent of the load variations, but this is followed by a variable-degree heating process up to the desired supply air temperature. Second, for the zone heating mode, the cooling equipment is turned off and only the terminal zone heater is operated to raise the mixed air temperature T_{ma} up to the desired supply air temperature. Cooling or heating modes depend on the sign of the net heat gain $(Q_i + Q_v)$. A simple heat balance of the control volume, including the building space and the heating coil, as sketched in Fig. B-4, yields the following:

(1) When $(Q_i + Q_v)$ is positive, the system is in the cooling mode where,

$$\left.\begin{aligned} -Q_h &= Q_i - \gamma V\,(T_i - T_{cp}) \\ Q_c &= Q_v + \gamma V\,(T_i - T_{cp}) \end{aligned}\right\} \tag{B-11}$$

which satisfies the energy balance equation for the building and the air handler:

$$Q_c - Q_h = Q_i + Q_v$$

Note that if T_{cp} is adjusted such that the average conditions of the cooling mode is

$$Q_i = \gamma V (T_i - T_{cp})$$

no heating will be required, and the cooling load Q_c is reduced to

$$Q_c = Q_v + Q_i \quad \text{(B-12)}$$

(2) If $(Q_i + Q_v)$ is negative, the system is in heating mode.

$$Q_c = 0$$
$$-Q_h = Q_i + Q_v \quad \text{(B-13)}$$

Therefore, if the average building conditions for cooling and heating modes are assumed, it will enable the separation of equipment loads and facilitate the computations of the energy consumption during each season.

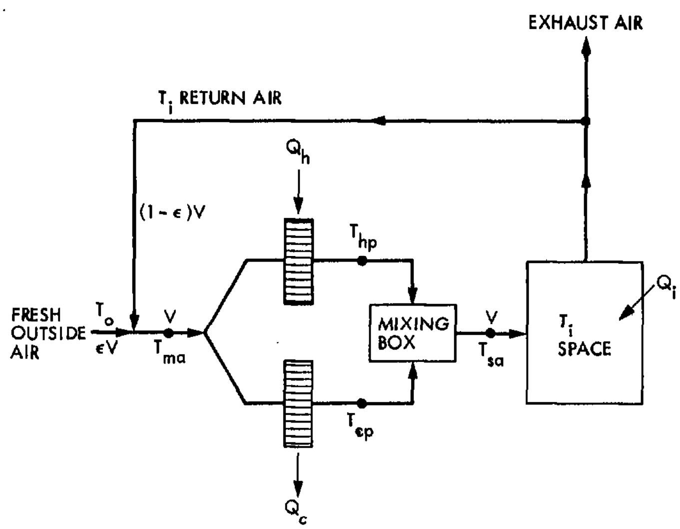

Fig. B-1. Dual duct system

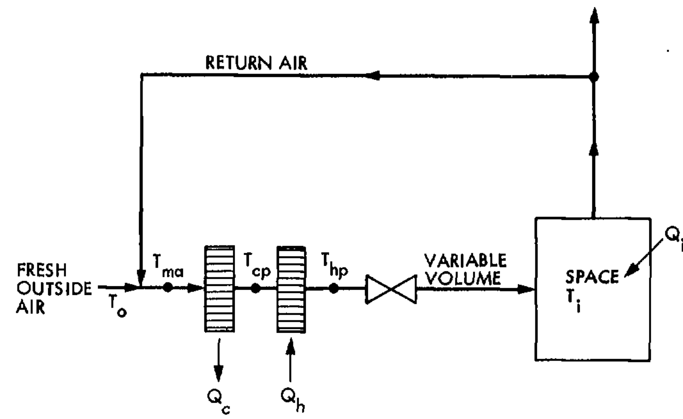

Fig. B-3. Variable volume control system

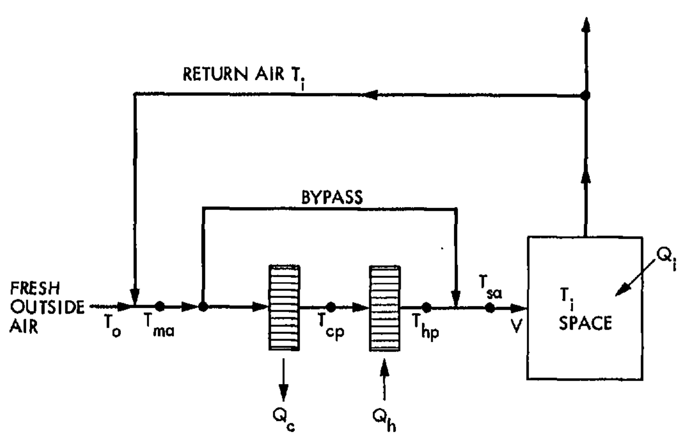

Fig. B-2. Face and bypass damper system

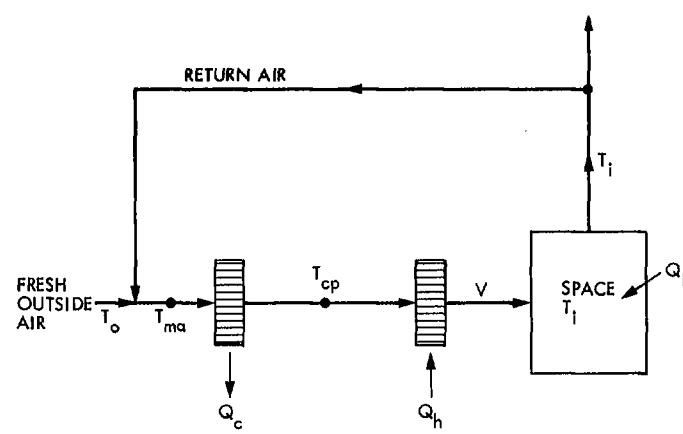

Fig. B-4. Single cold duct and terminal reheat system