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Publication Date
1982-12-01

LBL-14026 c.2

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UNIVERSITY OF CALIFORNIA

ENERGY & ENVIRONMENT DIVISION

Presented at the International Conference on Systems Simulation in Buildings, Liege, Belgium, December 6-8, 1982

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December 1982

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Prepared for the U.S. Department of Energy under Contract DE-AC03-76SF00098

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LBL-14026
EEB-DOE-2 82-2

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Summary

General considerations and models used by the DOE-2 program for the simulation of residential and commercial HVAC equipment are discussed. The use of transfer functions and linear controller models allows strong coupling between the envelope and the equipment models while the simulation of envelope and equipment configurations is performed sequentially, resulting in significantly reduced computation time. Empirical formulations of equipment models allows fast calculations without sacrificing sensitivity to primary physical parameters. A semi-iterative (up to three iterations) approach can produce accurate simulations of interacting control systems without a significant increase in computation time.

Introduction

The DOE-2 building energy simulation program was intended to be used by the building design community to assist in the design of more energy efficient buildings. To meet this goal, a program must be relatively easy to use, inexpensive to obtain and operate, but sufficiently comprehensive and accurate for the intended use. Given these limitations, many trade-offs must be considered: speed of computation vs detail of simulation; completeness of model vs simplicity of use.

DOE-2 is composed of two major segments, each of which is made up of several primary components. The first major segment is the Building Description Language (BDL). This segment allows the user to translate the building description into a structured language with simple syntax rules (commands to define components and keywords to describe component parameters). This description of the problem is then organized and passed on to the second major segment of DOE-2. The second segment is the simulator. This segment is composed of three main parts: the LOADS program simulates the building envelope heat transfer including gains from walls, windows, people, lights and equipment; the SYSTEMS program simulates the envelope-equipment coupling, terminal airside equipment, fans, coils, furnaces and direct expansion units; the PLANT program simulates central plant equipment such as boilers, chillers, cooling towers, storage tanks electrical generation equipment, and active solar energy collection systems. The level of model detail and general formulation is kept relatively constant between these components of the program to allow reasonable comparative studies of alternative envelope and equipment/control configurations.

For each of the simulation programs a time-step of one hour is used. Any dynamics within this time-step are assumed to be well represented as linear changes. Dynamics between time-steps are handled by two methods: transfer functions and/or semi-iterative calculations. Two types of transfer functions are used, response factors and weighting factors. The response factors are used to calculate the instantaneous heat gain due to conduction through external walls. Two types of weighting factors are used, heat gain and air-temperature. The heat gain weighting factors are used in the LOADS program to calculate the heating or cooling load, at a constant reference temperature, due to instantaneous gains from solar radiation through windows, conduction through walls, lights, people or equipment (one set of heat gain weighting factors for each of these sources). The air-temperature weighting factors are used in the SYSTEMS program to relate the reference temperature heating and cooling loads along with equipment operation characteristics to room temperature and equipment heat extraction/addition rates. Semi-iterative calculations are performed by the SYSTEMS program to allow the simulation of interacting or multiple control point systems. This method utilizes up to three estimation iterations to provide for a transition from one time-step to the next as well as within time-step dynamics convergence. Obviously, however, processes which cannot be well represented by linear changes within a one-hour time-step may not be accurately simulated.

This paper will deal primarily with the SYSTEMS program component of DOE-2. The PLANT program follows a similar approach.

Overview of the SYSTEMS Program

The SYSTEMS program simulates equipment which provides heating, ventilation and/or air-conditioning to the thermal zones and the interaction of this equipment with the building envelope. This simulation is composed of two major parts:

1) Since the LOADS program calculates the "load" at constant space air temperature, it is necessary to correct these calculations to account for equipment operation.

2) Once the net sensible exchange between the thermal zones and the equipment is solved, the heat and moisture exchange between equipment, heat exchangers, and the building can be completely calculated and the resultant primary equipment or utility "loads" can be calculated.

The constant air temperature calculation in LOADS has two major advantages. First, it greatly reduces the computation time of this part of the calculation, although introducing some approximations which preclude accurate calculation of certain configurations. Second, and more important, it allows tight coupling between the envelope and equipment calculation. This coupling is very important since the equipment operation in response to the control system actuation is most often a nonlinear process. This results in the energy input to the equipment not always being proportional to the envelope "load". Stated another way: the operation of and energy input to the HVAC equipment quite often can mask out the base envelope load.

The dynamics of the interaction between the equipment and the envelope are calculated by the simultaneous solution of the room air-temperature weighting factor relation with the equipment controller relation. To prevent the necessity of solving the interactions of all the zones simultaneously, zone histories are used to approximate the heat exchange across internal walls. Likewise, to eliminate the necessity of iterating until all temperatures in the equipment loop converge, temperature histories and estimation iterations are used in the calculation of equipment capacities. The equipment capacities are then used in establishing the relationship between the equipment output and the controller signal.

Once the supply and thermal zone temperatures are known the return air temperature can be calculated and the outside air system and other controls can be simulated. Thus the sensible exchange across all coils can be calculated.

The moisture content of the air is calculated at three points in the system: the supply air, return air, and mixed air. These values are calculated assuming a steady state solution of a system moisture balance will closely approximate the problem. The return air humidity ratio is used as the input to the controller activating a humidifier in the supply airflow or resetting a cooling coil controller to maintain maximum space humidities. The moisture condensation on cooling coils is simulated by characterizing the coils by their bypass factors and solving the bypass relation simultaneously with the system moisture balance.

Once the above sequence is complete, all coil loads are known. These values are then either passed on to the PLANT program as heating and cooling loop loads or, in the case of direct-expansion and non-hot water and steam coils, equipment required to handle these loads is simulated in SYSTEMS.

Room Air Temperature and Extraction Rate Calculation

The net heat input or extraction for a space is related to the air temperature through the air temperature weighting factors:

$$\sum_{n=0,2} p^j_n \, Qnet^j_{\tau-n} = \sum_{n=0,3} g^j_n \, \Delta t^j_{\tau-n} \qquad 1)$$

where

p^j_n, g^i_n are the transfer functions calculated to relate the net heat extraction rate to a pulse in room air temperature for room j at time τ-n (n=0 is the current simulation hour);

$Qnet^j_{\tau-n}$ is the net heat extraction in response to the temperature deviation in space j at time τ-n;

$\Delta t^j_{\tau-n}$ is the deviation in space j air temperature from the LOADS calculation temperature.

$$\Delta t^j_{\tau-n} = t^{Loads,j} - t^j_{\tau-n} \qquad 2)$$

where

$t^{Loads,j}$ is the LOADS calculation temperature for space j;

$t_{\tau-n}$ is the air temperature at time τ-n for space j.

Let us now consider the details of the terms contained in $Qnet^j_{\tau-n}$. As this term appears in 1) it represents the deviation in heat input from that calculated by LOADS due to the air temperature deviation from that used in LOADS. Thus $Qnet^j_{\tau-n}$ must contains terms to correct the LOADS calculated value for space temperature deviations. The main terms which are space temperature dependent are the infiltration as well as external and internal surface heat transfer. The correction to the infiltration is simply proportional to the deviation of the space temperature (except in the case where the infiltration flowrate is a function of space temperature). The correction for internal and external walls is approximated using overall U-values.

$$Qnet^{j}_{\tau-n} = ER^{j}_{\tau-n} - Q^{Loads,j}_{\tau-n} - C_1 * \dot{V}inf^{j}_{\tau-n} * \Delta t^{j}_{\tau-n} - UA_E * \Delta t^{j}_{\tau-n} - \sum_{k=1}^{nattach} (UA)_k * (\Delta t^{j}_{\tau-n} - \Delta t^{k}_{\tau-n}) \qquad 3)$$

where

$ER^{j}_{\tau-n}$ is the equipment heat input to space j at the time τ-n;

$Q^{Loads,j}_{\tau-n}$ is the LOADS constant temperature load for space j at time τ-n;

C_1 is the thermodynamic conversion factor $\left[\dfrac{J}{\frac{m^3}{s}\ {}^{o}C}\right]$

$\dot{V}inf^{j}_{\tau-n}$ is the infiltration flowrate to space j at time τ-n;

UA_E is the overall external wall conductance;

$(UA)_k$ is the overall conductance between space j and space k;

nattach is the number of spaces adjoining space j.

Note that the last term in 3) only corrects for spaces j and k temperature deviations from their respective LOADS-calculated temperatures. The term $Q^{Loads,j}_{\tau-n}$ contains the net transfer due to unequal LOADS temperatures for the spaces. It can be seen from this equation that a solution for one space temperature would require a simultaneous solution for all the space temperatures. To simplify this problem, we substitute τ-n-1 for τ-n in the space terms for the internal transfer. This is a good approximation if the derivative of the zone temperatures is not rapidly changing. If we substitute 3) into 1) and collect terms, solving for t^{j}_{τ}, we get

$$t_\tau^j = \frac{F - P_o^j * ER_\tau^j}{G_o} \qquad 4)$$

where

$$F = P_o^j * Q_\tau^{Loads,j} + \sum_{n=1,3} G_n^j * \Delta t_{\tau-n}^j + \sum_{n=0,2} \sum_{k=1,nattach} P_n (UA)_k * \Delta t_{\tau-n-1}^k$$

$$G_n^j = g_n^j + P_n^j * (k_t + C_1 * \dot{V}inf_{\tau-n}^j) \text{ (note } P_3 = 0)$$

$$k_t = k_E + \sum_{k=1,nattach} (UA)_k$$

This is further simplified when it is remembered that $P_0 = 1.0$.

The equipment heat input, ER_τ^j, takes different forms depending upon the type of zone and the type of equipment. Simplest is the unconditioned zone, since the value is zero. Next simplest is a return air plenum in which ER_τ^j is calculated from the temperature of air entering the plenum. The heat gain from the return air is expressed as

$$ER_\tau^j = C_1 * \dot{V}_r * \left[\frac{t_{\tau-1}^j + t_\tau^j}{2.0} - t_r \right] \qquad 5)$$

where

$\dot{V}_r$ is the return air flowrate

t_r is the entering air temperature

Lastly, for equipment controlled by the zone thermostat action, we assume a linear relation to describe the thermostat, space temperature, and equipment output relationship.

$$ER_i^j = W^j + S^j * t_\tau^j \qquad 6)$$

where

W^j and S^j are intercept and slope of this linear relationship.

It must be noted that the values of ER_τ^j in Eqn. 6) are restricted to a certain range due to the capacity of the equipment. Thus the equipment capacity must be estimated before the above equations can be solved. This capacity estimate is made using the dry and wetbulb temperature from the end of the previous time step. Also, there are three distinct regions of 6) with different slopes and intercepts. These are the heating, deadband, and cooling regions. The minimum, maximum and resulting slope and intercept for each of these regions must be calculated.

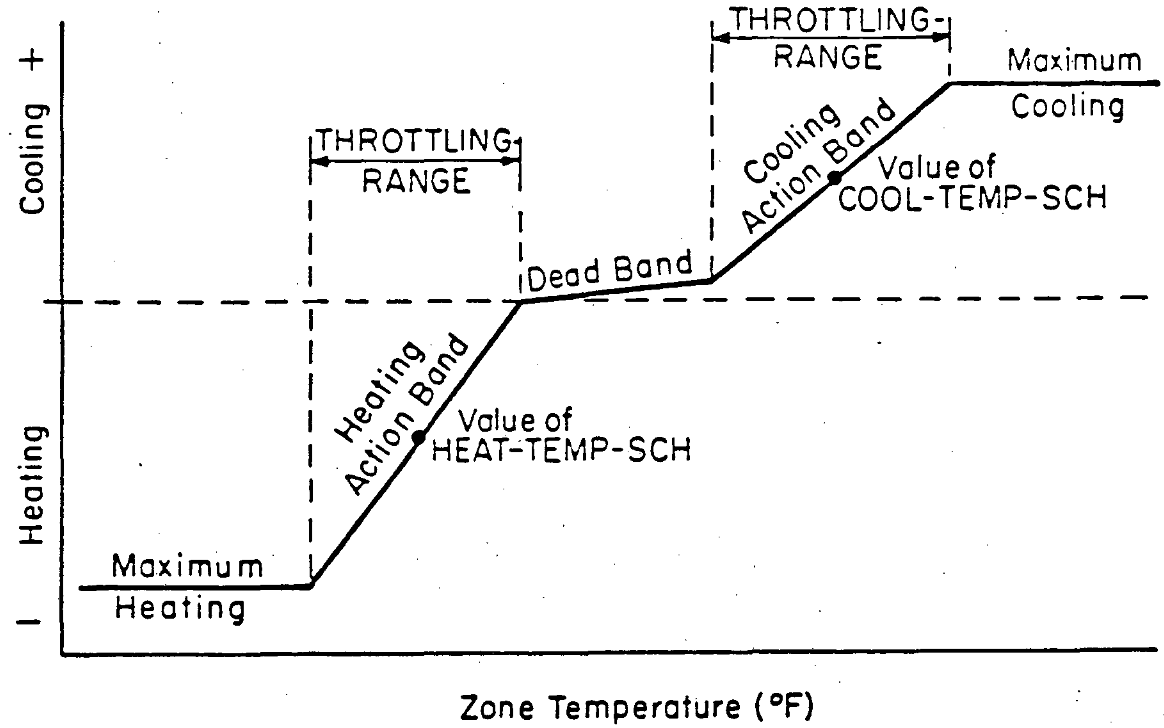

Fig. 1.
Five Possible Regions of Thermostatic Action

If the system being simulated is an air system, the supply air temperature is assumed to be constant (at full capacity) during the hour. Thus, in addition to 6), an equation of similar form to 5) arises to describe the fluctuations in extraction simply due to the fluctuation of space temperature during the hour, with respect to the supply air temperature.

It should be noted that the above equations are somewhat complicated in the case when the supply air temperature is also a function of space temperature and also when the air/flowrate is a function of space temperature. In these cases it is necessary to introduce more equations of the same form as 6) and first find the equilibrium supply temperature by iterating through the zones and solving these additional relationships to find the average controller signals during this time step.

Simulation of Heat and Moisture Exchange with HVAC Equipment

The first step to simulating equipment performance is to characterize the equipment in terms of the primary performance parameters. In most cases, this means we need to know the variation of equipment capacity and energy consumption as a function of various parameters. In this program we have chosen to express both capacity and energy input as the product of a "rated" value and modifier functions. Capacity, for example,

$$CAP_{t_1,t_2} = CAP_{rated} * f(t_1,t_2) \qquad 7)$$

is usually specified in the manufacturers literature in terms of a rated value at certain conditions. Often times this information is accompanied by off rated values. As seen in 7), we assume that the capacity at an off rated point can be expressed as the product of the rated capacity and a function normalized with respect to this rated capacity.

$$f(t_{1,rated}, t_{2,rated}) = 1.0 \qquad 8)$$

Very often more than two parameters are of importance to equipment operation. In this case we assume the this can be well approximated by the product of multiple modifier functions.

$$CAP_{t_1,t_2,t_3} = CAP_{rated} * f_1(t_1,t_2) * f_2(t_3) \qquad 9)$$

The program is capable of handling functions of one or two independent variables to produce linear, quadratic, cubic, bi-linear, and bi-quadratic curves.

For all modifier functions used in the program, there are built-in default performance curves that the user can easily replace (CURVE-FIT instruction in BDL). The rated point values can be calculated by the program or the user may choose to specify them. Although the specification of default-overriding performance data is done in the SYSTEMS input, the calculation of the new performance curve is done in BDL.

For cooling equipment we need to be able to calculate the capacity, both sensible and latent, and the energy input to handle a load. The sensible capacity, at a particular operating point, is calculated from three modifier functions. The first modifier function is used to calculate the total capacity and the second and third are used to modify the rated sensible capacity; then we insure that the sensible does not exceed the total. In this way we can get an accurate transition from a wet to dry coil surface condition. Since this transition is not a smooth one, we needed to have multiple functions to describe these regions.

$$QCT_{t_1,t_2} = \text{COOLING-CAPACITY} * \text{COOL-CAP-FT}(t_1,t_2) \qquad 10)$$

$$QCS_{t_1,t_2,t_3} = \text{COOL-SH-CAP} * \text{COOL-SH-FT}(t_1,t_2) - f_{dx1}(t_3)$$

or QCT_{t_1,t_2}, whichever is smaller.

where

t_1 = entering evaporator wetbulb

t_2 = entering evaporator drybulb for water coils

= entering condenser drybulb for direct expansion

t_3 = entering evaporator drybulb for direct expansion

f_{dxl} = 0.0 for chilled water systems

$= C_1 * \dot{V} * (1.0\text{-COIL-BF}) * (t_{ri} - t_3)$ for DX

where

t_{ri} is the rating point entering evaporator dry bulb temperature

$\dot{V}$,COIL-BF are the airflow rate and coil bypass factors

From the above expressions, it can be seen that at full load we also know the amount of latent load or moisture removal. Since the number of hours at full load operation are small and other factors which effect moisture condensation on the coil can vary, we have developed a slightly more complex method of calculating the latent load. Using only the expressions in 10) would also make it necessary to iterate during a single time step in order to get the supply and mixed air moisture levels accurately. This is due to the fact that the supply and mixed air moisture levels are coupled. One way to avoid this problem would be to ignore it by using the previous hour's entering evaporator wetbulb to calculate the capacity and resultant supply and new (for the next hour) mixed air wetbulb. We have avoided these problems by using an accurate but simple relationship that can be solved simultaneously with a system moisture balance to produce steady state values for the moisture levels at the important points in the system. Thus we introduce the coil bypass factor concept.

The coil bypass factor (CBF) model characterizes the air exiting the coil as being composed of two major streams: the air which has not been influenced by the coil and the air which leaves at the coil surface condition. The coil bypass factor is the fraction of air which exits unaffected by the coil. Thus, we have relations for the exit drybulb temperature and humidity ratio in terms of the entering conditions and the coil bypass factor.

$$t_{exit} = t_{entering} * CBF - (1.0 - CBF) * t_{surf} \qquad 11)$$

$$W_{exit} = W_{entering} * CBF - (1.0 - CBF) * W_{surf} \qquad 12)$$

The coil bypass factor is a function of both physical and operation parameters of the coil. Since the physical characteristics are constant, we express the coil bypass factor as a product of the design or rated value and two modifier functions. The most important variable is

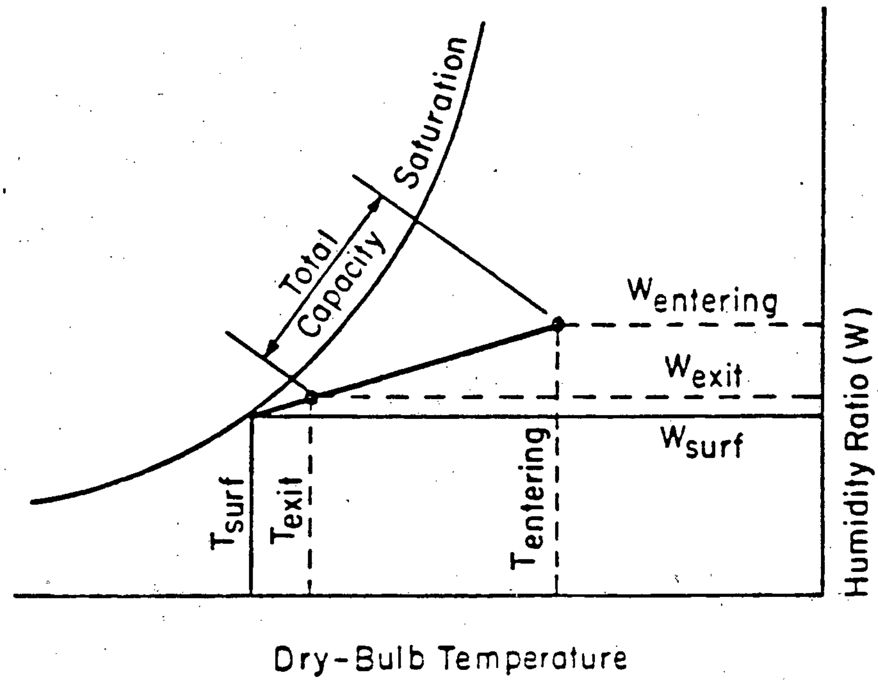

Fig. 2
Cooling Coil Performance

the coil surface air velocity, which is directly proportional to the unit flowrate, CFM. Of secondary importance are the entering coil wet-bulb and dry bulb temperatures.

$$CBF = COIL\text{-}BF * COIL\text{-}BF\text{-}FCFM(PLRCFM) * COIL\text{-}BF\text{-}FT(t_1,t_2) \qquad 13)$$

where

t_1 = entering evaporator wetbulb

t_2 = entering evaporator drybulb for chilled water systems

= entering condenser drybulb for dx systems

PLRCFM = ratio of instantaneous flowrate to rated flowrate

The values for the coil bypass factor can be easily calculated from manufacturers' data by plotting the entrance and exit condition and drawing a line through them to intersect the saturation line. This intersection is the apparatus dewpoint. Using this point along with 11), a series of CBF values can be determined and the rated value and modifier functions generated.

In the previous section it was described how the various supply and space temperatures are calculated. During this process it was necessary to know the range of temperatures of supply air that could be available. This required the estimation of capacity before the entire problem was solvable. To avoid iteration, we first use the previous hour's mixed air wetbulb temperature to estimate the sensible capacity using 10). Then we can calculate the minimum supply temperature.

$$t_{exit} = t_{entering} - \frac{QCS}{C_1 * \dot{V}} \tag{14}$$

where

$t_{entering}$ = the estimated entering air temperature using an extrapolation of return air temperature along with a simulation of outside air controls.

Thus we know the capacity limits from this estimate. Now, based upon the supply air temperature control method (fixed, scheduled, reset or zone control), we can calculate the actual supply air temperature and all the zone temperatures. From the resulting return and mixed air temperatures, we know both the coil entering and exit drybulb temperatures and the sensible cooling load can be calculated.

We can then use 11) to calculate t_{surf} and then calculate the saturation humidity at this temperature, WSURF.

To solve the entire moisture problem we start with a moisture balance on the system.

$$\dot{V} * W_r + \dot{V}inf*W_r = \dot{V} * W_{exit} + \dot{V}inf * W_0 + \Delta W_r \tag{15}$$

Solving for W_r and letting $F = \frac{\dot{V}inf}{\dot{V}}$, we get

$$W_r = \frac{W_{exit} + F * W + \Delta W}{1 + F} \tag{16}$$

We also know the relation for the mixed air humidity in terms of the return and outdoor conditions.

$$W_{entering} = P_0 * W_0 + (1 - P_0) * W_r \tag{17}$$

where

W_0 = outside air humidity ratio

P_O = fraction of outside in mixed air

Combining 16) and 17) plus the dry coil assumption $W_{exit} = W_{entering}$, we get:

$$W_{entering} = W_O + \left[\frac{1 - P_O}{F + P_O} * \Delta W\right] \qquad 18)$$

or $W_r = W_O + \frac{\Delta W}{F + P_O}$ if the coil entrance condition calculated in 18) is greater than the coil surface saturation humidity, our dry coil assumption is incorrect. In this case, we combine 16), 17), and 12) to get:

$$W_r = \frac{CBF * P_O * W_O + (1-CBF) * W_{surf} + \Delta W + F*W_O}{1 + F - CBF * (1 - P_O)} \qquad 19)$$

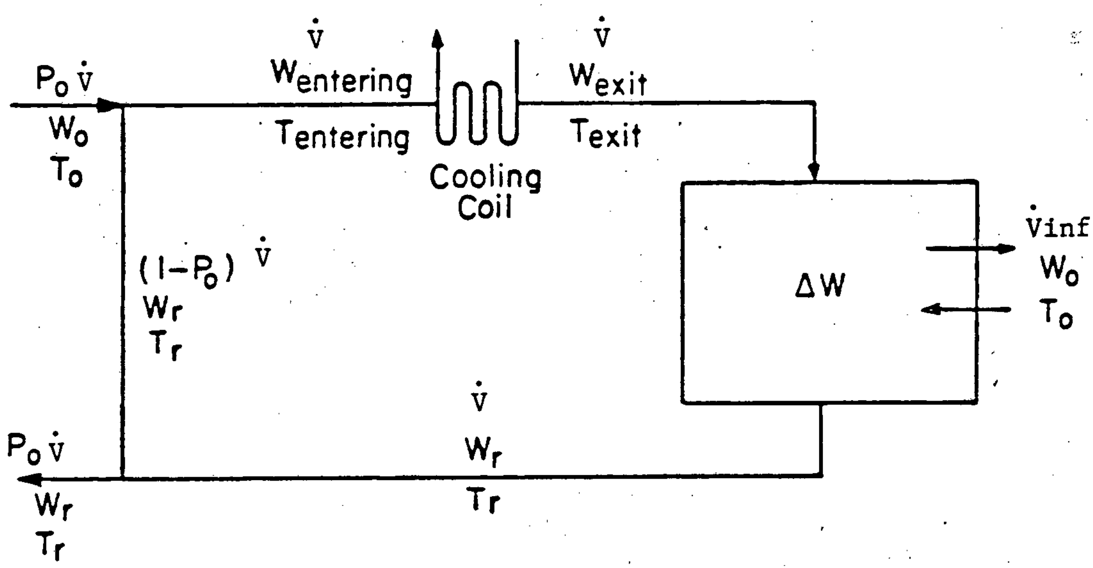

Fig. 3.

System Moisture Balance

We can then reapply 17) and 12) to calculate the coil entering and exit condition, thus being able to calculate the enthalpy change across the coil.

For direct expansion packaged equipment, we continue on to calculate the energy input to the compressor-condenser section. DOE-2 characterizes devices of this type in terms of energy input ratios: the ratio of energy input to load handled. In the case of electric direct expansion cooling equipment we use the electric input ratio, EIR. All the energy input ratios are defined in terms of the capacity.

$$Q_{elec} = QCT_{op} * EIR_{op} \tag{20}$$

where

QCT_{op} is the operating capacity or the capacity at the current operating conditions;

EIR_{op} is the operating electric input ratio

The EIR_{op} is calculated as a product of the rated EIR and two modifier functions. The first modifier function accounts for off design temperatures and the second for part load operating characteristics.

$$EIR_{op} = \text{COOLING-EIR*COOL-EIR-FT}(t_1, t_2)\text{*COOL-EIR-FPLR(PLRC)} \tag{21}$$

where

t_1, t_2 are as described above;

PLRC is the total cooling part load ratio (cooling load/operating capacity)

Actually the EIR_{op} is a bit more complex than described above. Generally, there are three regions of equipment operation: a range below full load within which the compressor can unload, a range within which a hot gas bypass is engaged, and a lower range within which the compressor cycles. Only the upper range is meant to be described by the COOL-EIR-FPLR curve.

As can be seen in Fig. 4, it is assumed that within the bypass region the electric energy input is constant and equal to the value defined by the curve COOL-EIR-FPLR evaluated at PLRC = MIN-UNL-RATIO. It can further be seen that within the cycling region the energy input is assumed to follow a linear relationship through zero from the value within the bypass region.

For the simulation of heating equipment we have used similar concepts. The main difference is that, except in the case of heatpumps, the capacity is constant independent of operating conditions. Hot water coil loads are passed directly to the PLANT program. Gas and oil furnaces, electric resistance heat and heatpumps are simulated in SYSTEMS, passing only the utility load to PLANT.

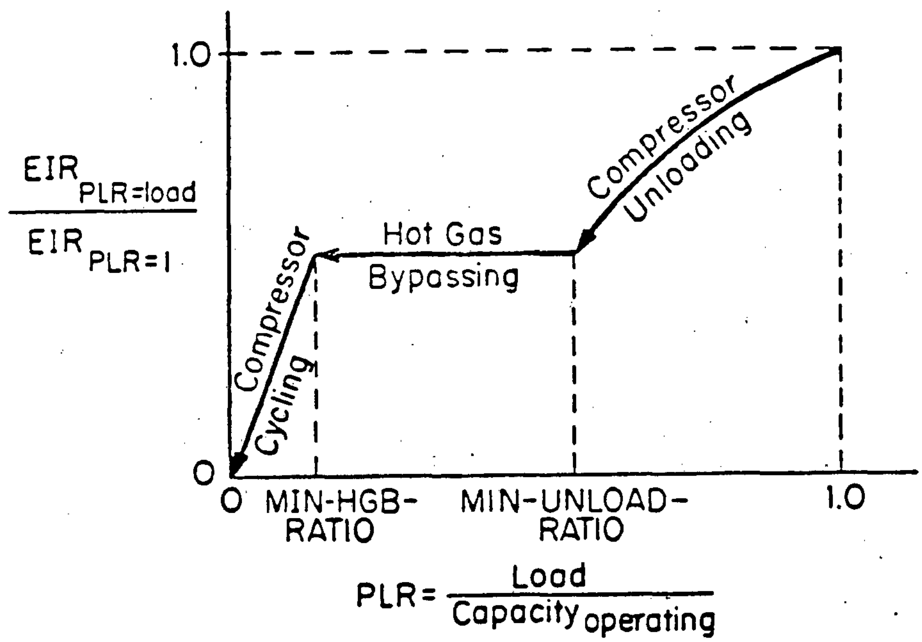

Fig. 4.

Electric input ratio vs part-load ratio for direct expansion cooling units

$$QHT_{op} = \text{HEATING-CAPACITY} * \text{HEAT-CAP-FT}(t_1, t_2) \qquad 22)$$

where

HEAT-CAP-FT(t_1,t_2) = 1.0 except for heatpumps

t_1 = outdoor temperature for air-to-air, water temperature for water to air

t_2 = indoor temperature

For gas and oil furnaces the energy input is calculated in terms of the heat input ratio, HIR. The HIR is the ratio of heat input to load handled.

$$Q_{fuel} = QHT_{op} * HIR_{op} \qquad 23)$$

The operating heat input ratio is a product of the full load value and a part load modifier.

$$HIR_{op} = \text{FURNACE-HIR} * \text{FURNACE-HIR-FPLR(PLRH)} \tag{24}$$

where PLRH is the ratio of furnace load to capacity.

The furnace load during a partial operation hour may be effected by off-cycle induced infiltration. This effect, if present, can be simulated through the use of the keyword FURNACE-OFF-LOSS. This function expresses the induced load as a fraction of unused capacity.

For heatpumps the electrical energy input is calculated as the product of the operating capacity and electric input ratio.

$$Q_{elec} = QHT_{op} * EIR_{op} \tag{25}$$

As for cooling, the operating EIR is a product of the design value and two modifier functions: one for off design temperatures, and the other for part load effects.

$$EIR_{op} = \text{HEATING-EIR} * \text{HEAT-EIR-FT}(t_1, t_2) * \text{HEAT-EIR-FPLR(PLRH)} \tag{26}$$

For air-to-air heatpumps, the addition of a defrost system and auxilliary electric resistance heat can also be simulated. Through the use of the DEFROST-T and DEFROST-DEGRADE keywords, the user may specify the outdoor temperature below which the defrost system is activated and the amount of defrost time as a function of outdoor conditions. Though the use of the ELEC-HEAT-CAP, MAX-ELEC-T, and MIN-HP-T keywords, the user can specify the capacity of the auxilliary heat and the outdoor temperature below which it can operate as well as the outdoor temperature below which the heatpump can no longer operate.

In addition to the types of heating devices already discussed, the user may augment or replace the default system heating equipment with baseboards. Baseboard output may be controlled either as a function of outdoor temperature (independent of zone temperature) or by the zone thermostat (in response to zone temperature). Thermostatically controlled baseboards are sequenced on first in response to a zone heating load.

Interactions of Equipment Control Systems

Although the user selects the generic type of equipment with the SYSTEM-TYPE keyword, the details of the hourly simulation and resultant energy calculation can be greatly affected by choices for keywords describing control options. Some of these effects have already been mentioned with regard to control of part load and other operational parameters for compressor-condenser units. Space thermostat, cooling coil, mixed air and other control systems can have similar dramatic effects on the energy estimates. In general, there are keywords that allow the user to describe the setpoints and sequencing of most of the various commonly available control systems.

As earlier described, the user-supplied heating and cooling thermostat setpoint schedules (HEAT-TEMP-SCH and COOL-TEMP-SCH keywords) together with the THROTTLING-RANGE keyword defines the three action bands of the physical space thermostat. The HEAT-TEMP-SCH defines the midpoint of the heating action range and the COOL-TEMP-SCH defines the midpoint of the cooling action range. If these two values are separated by more then one THROTTLING-RANGE, a deadband has been defined within which the equipment action is the same as the bottom of the cooling action band. If these two values are separated by less than a THROTTLING-RANGE, the program assumes a mistake has been made and constructs two values centered on the midpoint of the users values but separated by a THROTTLING-RANGE.

The actual actions within the heating and cooling ranges vary based on the type of equipment. The cooling range is used to cycle on cooling equipment in zonal systems or control volume and temperature in air-handler systems. If no equipment affected by action in one of these ranges is present, that setpoint schedule may be omitted. If multiple types of equipment controlled in a single range are present they are sequenced in the following manner:

Heating Range (in response to a fall in temperature)
1. Increase supply temperature
2. Increase baseboard flow (only if thermostatically controlled)
3. Increase reheat coil flow
4. Increase volume flow (only if a reverse acting thermostat)

Cooling range (in response to a rise in temperature)
1. Decrease Supply temperature
2. Increase volume flow

Thus it can be seen that a dual duct or reheat system does not use the COOL-TEMP-SCH unless zone controlled supply temperature and/or a variable flow system is used. If both these options are selected the temperature of supply air will remain at the minimum until all zones are in the bottom half of the THROTTLING-RANGE. In a similar manner, a variable volume system has no need for a HEAT-TEMP-SCH unless a reverse action thermostat, a reheat coil, and/or zone-controlled baseboards have been specified. The actions within these ranges for the various types of systems will differ.

The MAX-HUMIDITY and MIN-HUMIDITY keywords place relative humidity control on the return air stream. If the return air relative humidity falls below the MIN-HUMIDITY setpoint, steam or hot-water is injected into the supply air. The resultant load is passed on to PLANT as a steam or hot-water load. If the return air humidity goes above the MAX-HUMIDITY setpoint, the cold supply temperature is reset towards the minimum until the correct level is maintained or coil capacity (or MIN-SUPPLY-T) is reached. The reset of the supply temperature to dry the air can often defeat the action of a zone controlled cooling coil. The simulation of this control scheme is accomplished by use of an estimation interaction which predicts the return air moisture level and, thus, the required supply air temperature necessary to hold the setpoint. The reset of the supply temperature can cause a very different space thermostat and terminal unit performance from the initial calculation. Since the detailed zone calculation is not redone significant errors may result.

Although no keyword is available to the user to specify a mixed air controller setpoint, the program simulates this equipment. The mixed air controller is simulated as having the same setpoint as the cold duct controller compensating for heat gain due to a blowthrough fan arrangement. This controller modulates the movable outside air dampers (if available) to open in response to a rising value of mixed air temperature. Thus, if the outside temperature is above the setpoint, it may not be possible to obtain the desired value. In fact, we may be inducing an extra cooling load if the return temperature is less than the outside air temperature. For this reason, there is a high limit override which specifies the outdoor temperature at which the dampers are forced back to their minimum position. This limit is specified by the ECONO-LIMIT-T keyword. Additionally, there may be an enthalpy controller which also resets the dampers to minimum if the outdoor enthalpy is greater than the return air enthalpy (otherwise action is similar to that already described).

A variable volume supply and/or return air fan is simulated assuming the existence of a pressure control system. This results in a constant pressure being maintained across the outside air dampers, thus the minimum outside air fraction is relative to design supply flowrate. As the total flowrate drops, the constant pressure across the outside air dampers insures a constant volume of outside air--thus a larger fraction relative to total flow.

Design Calculations

As described in previous sections, many equipment design parameters must be known before the hourly simulation can proceed. Most of these parameters may be specified by the user in the definition of a thermal ZONE or HVAC SYSTEM through the use of keywords. To make the program easier to use in the early stages of analysis we have developed a set of procedures to calculate most design parameters if the user has not provided enough information. Before the simulation can start, all air flowrates, equipment capacities, and off-design performance modifier functions must be known. The off-design performance modifier functions are retrieved from a library of such values. These defaults were calculated once for what we considered typical equipment for each type of system. If, upon examination, the user finds these curves to be undesirable, they may be replaced with better data using the CURVE-FIT command.

Air flowrates and coil capacities, however, cannot be precalculated. These values depend, usually, entirely upon heating and cooling requirements. Again, the user may specify the values of all air flowrates and coil capacities using the keywords provided for this purpose. If any flowrate or capacity is left unspecified, the program calculates the value using whatever information has been supplied plus values calculated itself or by the LOADS program.

Basically, the relationship

$$Q = C_1 * \dot{V} * \Delta t \tag{27}$$

where

Q is the sensible load

$\dot{V}$ is the air flowrate

Δt is the temperature difference required

must hold. Usually the Δt is known from user specified values. The DESIGN-HEAT-T and DESIGN-COOL-T are required for all conditioned zones. These values together with MAX-SUPPLY-T and MIN-SUPPLY-T (or REHEAT-DELTA-T for reheat coil systems) define the zone temperature and supply temperature values for Δt. If the user has specified the flowrate, the Q can be directly calculated. Similarly, if the user has supplied Q, the flowrate can be calculated. If neither value has been supplied the Q value is taken from the peak calculated by LOADS (found in LS-B) and flowrate is calculated. If both the flowrate and the Q values have specified, they take precedence over the Δt.

Once 27) has been applied for each zone, taking into account exhaust, we can calculate the return and mixed air temperatures for both the heating and cooling mode. Then the sensible coil capacities for the main air-handler can be calculated. The latent cooling load is calculated as already discussed in a previous section. Following these general techniques all coil capacities are calculated. Once the capacities have been calculated at the peak condition, we can calculate the rated capacities (entering, for U.S. applications, indoor 80°F drybulb, 67°F wetbulb, and outside 95°F drybulb for cooling; entering 70°F drybulb and outside 47°F drybulb for heatpumps in the heating mode) can be calculated.

For systems with variable flowrate capabilities, the flowrate and capacity calculations can be a bit more complex. In the zone by zone application of 27) usually the cooling load will define the flowrate. If a minimum flowrate has not been specified, it will be calculated from either the heating or ventilation requirements. Additionally, the fan in the main air handler will be sized on the building coincident peak load calculated by the LOADS program instead of the sum of the zone design flowrates, unless otherwise requested. This can lead to problems if a night setback or setup is used, since the morning load may be too large for the available air flowrate.

Verification of Accuracy

A significant amount of work as been undertaken by various groups to verify the accuracy of the DOE-2 program and its components. The results of this work have been mixed but generally favorable. Whole building simulations have been compared to metered building data producing total energy consumption estimates accurate to less than 1% variance or up 12% to difference. (See Ref. 1.) Hourly, monthly, and yearly demand, as well as consumption values, have been compared to other similar purpose programs producing a similar range of differences as mentioned above, (Refs. 1 and 2). Comparisons against laboratory test data for HVAC components and room weighting factors have produced average deviations between 2% and 12% with the bulk of the differences in the range of 4% to 9% (Ref. 3).

A large "real world" user community has provided guidance for bug fixing as well as ongoing user-responsive model development. A significant verification of the program's accuracy is inherent in the widespread use and trust that has been developed in the program. Much more laboratory test quality data, however, is needed by all developers of this type of program to continually check simulation results.

Conclusions

A general approach to simulating HVAC equipment in buildings has been presented. This approach has been used to develop the SYSTEMS simulation program component of the DOE-2 building energy use analysis program. This program has compared well to whole building component level measured data. DOE-2 has also gained widespread acceptance in the building design and research community.

Acknowledgment

This work was supported by the Assistant Secretary for Conservation and Renewable Energy, Office of Building Energy Research and Development, Buildings Systems Division of the U. S. Department of Energy under Contract No. DE-AC03-76SF00098.

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