# Calibrating AHU Models Using Whole Building Cooling and Heating Energy Consumption Data

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## ABSTRACT

Calibrated building simulation models can be used to identify operational problems, and optimize system operation and control schedules. It is important that the model is capable of modeling the departures from ideal behavior which occur in real systems, and several common model deficiencies are noted. Model calibration is generally a time consuming task at the daily or hourly level needed for these tasks. A set of graphic signatures which can be used to assist engineers in determining a physically correct calibrated model is presented in this paper. The graphic signatures can be used to rapidly identify the input parameter(s) which need to be changed to improve the agreement between the simulation output and measured consumption. A two-level graphic procedure is presented for calibrating building AHU heating and cooling energy consumption models.

## Introduction

Comprehensive building simulation software packages were developed in the 1970's [APEC, 1967; Kusuda, 1971; Henninger, 1975; Bennett, 1977; Hittle, 1977; LBL, 1980]. These programs have very sophisticated dynamic models of building heat transfer and energy systems (lighting, HVAC, chillers). Consequently, these programs require comprehensive inputs of building information, system information and weather information.

Even the best simulation program will not predict the measured consumption of a building when incorrect input parameters are used. Model calibration is a process which attempts to use measured data as feedback to modify the inputs until they are "correct". To be useful in practical application for determining energy savings of retrofits [Katipamula and Claridge, 1993], identifying operating problems [Katipamula and Claridge, 1992; Liu, 1993; Liu and Claridge, 1995], and optimizing system operation [Liu and Claridge, 1995], the calibration process must be relatively fast. This suggests the use of simulation models which include only input parameters which impact energy use in a measurable way at the whole building level. The simplified models of HVAC systems and building heat transfer developed by Knebel [1983] are simple enough to permit rapid calibration, but detailed enough to be useful for the applications noted above. The authors have found that energy signatures, which show the overall sensitivity of energy use to individual parameters can be very helpful in identifying input parameters which require change to achieve an accurately calibrated simulation.

Even the most sophisticated and detailed models are generally based on idealized models of building and system performance. These idealizations are another important factor in the discrepancies which are often seen between simulation results and measured performance. For example, the normal dual duct terminal box model assumes zero hot air flow when the maximum cooling is needed. In fact, the authors have observed that hot air leakage in an actual terminal box may be as high as 20% under peak cooling conditions. A simulation model must be able to include the departures from ideal behavior which occur in real systems if it is expected to accurately portray system performance. This issue is addressed in the next section.

**Modeling Issues**

Heating and cooling energy consumption depends on building characteristics, occupancy, operational schedules, type of HVAC system, weather and other parameters. Even if a simulation model is capable of using highly detailed information about a building and its systems, it is impossible to obtain all of this information accurately. When "estimated" parameters are used, "estimated" results are obtained. Model calibration can improve precision, but is laborious. Model simplification, which limits input detail to items which have a detectable impact on the measured energy use, is highly desirable from our perspective. Model calibration should also recognize difficulty of estimating internal gains, and the imperfections of real-world devices such as mixing boxes and control valves.

**Zones**

A building may have many rooms or spaces. Each room or space can be treated as a separate zone. However, it may make no difference if two rooms are treated as separate zones when these rooms are served by the same terminal box.

A zone may be selected as the area served by each terminal box. However, there may be many boxes which serve similar areas with the same load requirements and air flow requirements. This suggests that zoning can be further simplified without sacrificing modeling accuracy.

Knebel [1983], Katipamula and Claridge [1993] and others have found that buildings can often be adequately treated as two zones: interior and exterior zones. A case study presented by Liu and Claridge [1995], showed very accurate results. An air side simulation program [Liu, 1997] has been developed using the two-zone model. The simulation program has been used to calibrate the system model, identify system operational problems and optimize system operation by the authors and their research group, since 1993. *This experience indicates that the two-zone model works well provided the interior and exterior zones are properly determined.*

It is a common practice to identify the exterior zone as the area which is less than 20 feet from the exterior walls or windows. This is a good approximation. However, the zoning can be improved significantly if the exterior zone is taken as the areas which can be affected by the weather directly. For example, if a conference room has a depth of 40 feet from its windows to its back exterior wall, this entire room should be treated as part of the exterior zone. If an exterior wall is one side of a corridor, everything to the inside of the corridor should be treated as part of the interior zone.

**Internal Gain**

The internal gain is often estimated based on nameplate data. This approach significantly over-estimates the cooling requirement since all of the equipment does not operate at the same time and the actual consumption is typically less than the nameplate rated power consumption. In one case study [Wilkins, 1998], the nameplate electrical gain was 3.5 W/ft$^2$. When it was adjusted to account for the disparity between nameplate data and the measured maximum consumption of each piece of equipment, the internal gain was estimated to be 1.75 W/ft$^2$. The measured total peak power consumed by the equipment was 0.8 $W/ft^2$. This suggests that accurate values of internal loading require the use of a diversity factor .

The internal gain is sometimes estimated by using measured whole building electricity consumption. This approach may also significantly over-estimate the heat gain since a large fraction of whole building electricity use, such as that used by pumps, exhaust fans, elevator motors, and air compressors, may be converted to heat in non-conditioned spaces, such as mechanical rooms, basements, and penthouses.

Even the heat generated in the conditioned space may not become cooling load since some of the lighting energy is carried out directly to the outside by the exhaust air.

Thus the estimation of internal gains is a complex issue. When there is no measured data available, the internal gains may be evaluated based on equipment nameplate data provided the following factors are considered properly: diversity factor, ratio of maximum consumption to rated consumption; and the fraction of energy carried out directly. When the measured whole building electricity consumption data is available, the following factors must considered: fraction of electrical power consumed in non-conditioned space; and fraction of power carried out from the conditioned space to outside and return air. *It appears that it is very hard to accurately determine the internal heat gain due to the electrical devices in a building.*

**HVAC Systems**

System models have been developed for terminal boxes, dampers, control valves, and coils. Most of these models are idealized models which assume that the systems perform as designed, but system performance should be modeled to match real system operation. The discussion which follows considers key factors which must be included in accurate AHU simulation models.

**Terminal Boxes:**

**Single Duct Constant Air Volume (CAV) Terminal Boxes:** The box maintains constant air flow regardless the static pressure. The idealized models, perform well when the entire AHU has only CAV terminal boxes. When an AHU provides air to both variable air volume (VAV) and CAV terminal boxes, the actual flow through the CAV boxes may be higher than the design values under partial load. Under partial load, the total flow rate is reduced due to the partially closed VAV boxes. If the static pressure is maintained constant at a point 2/3 of the down stream length of the main duct, the static pressure is increased before the CAV terminal boxes. In certain types of terminal boxes, the actual flow is increased due to the increased pressure.

**Single Duct VAV Terminal Box:** The VAV box modulates the air flow from the minimum to the maximum value to maintain room temperature and/or minimize the reheat. The ideal model works fine until the flow is close to the minimum air flow. The minimum air flow may never reach the designed minimum air flow due to poor damper quality and high static pressures at the terminal boxes.

The minimum primary air flow is often set to zero in office buildings. Field measurements show that the minimum flow will never be zero even if the application requires the zero minimum air flow. In the absence of measured data, a minimum 5% to 10% air flow should be used for model calibration. When a variable frequency drive (VFD) or inlet guide vane (IGV) is used to reset the static pressure based air flow, 5% minimum air flow may be reasonable. If a VFD or IGV are used to maintain a constant static pressure, a minimum 6% air flow may be used. If no VFD or IGV are used, a minimum 10% air flow should be used. If the actual leakage, or minimum air flow, is measured in the field, this value should be used.

Excessive pressure at the terminal box damper may result in the minimum air flow being higher than the design minimum flow. In most applications, the static pressure is maintained at a fixed value, for example, 1.0 $inH_2O$. When the air flow is reduced from 100% to 30%, the pressure drop from the pressure sensor to the terminal box is reduced by 91%. The static pressure at the terminal box may be increased from 0.4 inH2O to 0.8 inH2O. Due to this pressure increase, the terminal box may actually deliver more than 30% of the total flow. The flow bias is strongly dependent on the quality of the box.

If the static pressure is reset based on air flow rate and the minimum air flow is higher than 30% of the maximum flow, the idealized model works well.

**Dual Duct Constant Air Volume Terminal Boxes:** The idealized model states that the box modulates the hot and cold air flow to maintain constant flow through the box. Under full cooling conditions, the minimum hot air flow is zero. In the full heating mode, the minimum cold air flow is zero or minimum. Some terminal boxes require a non zero minimum cold air flow because of fresh air requirements.

In real systems, most terminal boxes may maintain constant total flow. However, even when constant total flow is maintained, the model has significant error in either full cooling or full heating modes. The following correction must be made.

Under the full cooling model, the minimum hot air flow may vary from 5% to 10% depending on the system. The static pressure on the hot air damper inside the box can be close to the static pressure at the fan outlet, which is often in a range of 1.5 $inH_2O$ to 5 $inH_2O$. If we assume the damper leakage is 5% with a differential pressure of 1 $inH_2O$, the leakage will be 11% when the static pressure is increased to 5 $inH_2O$. Under full heating mode, the minimum cold air flow may vary from 5% to 10% due to over-pressurization of the cold air duct.

**Dual Duct Variable Air Volume Terminal Boxes:** The use of DDVAV terminal boxes extends the full heating and full cooling modes over a wide load range since the total air flow is allowed to modulate to a lower value. However, the air leakage through terminal dampers can be lower than with DDCV terminal boxes since the VFD or IGV often decreases the static pressure at the fan outlet under partial load.

The air leakage can be controlled at the design level for the following conditions: (1) a hot air damper in the main hot air duct resets the hot air static pressure according to heating load; or (2) both hot and cold air static pressure are reset according to heating and cooling loads for dual fan dual duct systems.

It appears that an air leakage factor must be introduced into the terminal box model. The air leakage at minimum flow varies from 5% to 10% of the maximum flow rate. The exact value of leakage can be determined through the model calibration. The minimum air flow may be 10% to 30% higher than the design value. The actual value should be determined during the model calibration.

**Coils and Control Valves:**

Most air-side simulation programs assume that coils and control valves can maintain the temperature reset schedules. This assumption is good when the coil load is 20% or higher. When the coil load is lower than 20%, the water leakage may actually determine the discharge air temperature.

Most control valves have a turndown range of 20:1 to 40:1; the minimum predictable flow varies from 2.5% to 5%. According to ASHRAE, the valve can control the load no lower than 10% to 20% of range. In real systems, leakage always exists. Under high cooling loads, hot water leakage is increased due to increased differential pressure across the hot water control valve. During high heating loads, the chilled water leakage is high due to increased chilled water differential pressure across the chilled water valve.

Pre-heat coils often heat air up by 3°F or more during summer months if hot water or steam is supplied to the coil. The re-heat coil may also warm air 3°F or more during full cooling mode due to leakage.

For dual duct systems, the hot air flow may be as low as 10% of the total flow during full cooling mode. The hot air temperature can be as high as 100°F due to hot water leakage based on the authors' experience. This has significant effects on the total building energy cost.

During the winter season, the cold air temperature may be as low as 45°F due to chilled water leakage even when the set-point is 55°F or higher. The chilled water requirement is much lower in winter. Excessive differential pressure is often applied to the control valves due to inappropriate control of the chilled water pumps. It can lift the control valve seat and increase the normal leakage as well.

The simulation engineer must know the water side operational schedule and the control schedules in order to calibrate the heating and cooling energy consumption. A leaking control valve can be simulated by using artificial temperature reset schedules. For example, hot air temperature may be at 90°F or higher to simulate a leaking hot water valve during summer months.

**Graphic Signatures**

It is impossible to obtain perfect inputs for the system simulation. Modification of the inputs is always necessary to calibrate a model with measured heating and cooling energy consumption. Even the simplest simulation program requires dozens of input variables. For different types of AHUs, the variation of the same input variable may have a totally different impact on the building energy consumption pattern. Energy "signatures" are needed to help choose the proper system parameters to change in the model.

Energy signatures are sensitivity plots. They show computed response of cooling or heating energy use to a step change in a particular parameter, versus outside air temperature. Change in energy use is shown as percent of a maximum or baseline value. The step change is a plausible user-selected change to the parameter of interest. For example, the first plot in Figure 1a shows change in chilled water energy use for a step change (2°F reduction) in cold deck temperature over a 20°F to 100°F outdoor drybulb range. These signatures can help simulation engineers identify the parameters to adjust.

The signatures have been developed for typical AHUs by using AirModel [Wei, Liu and Claridge, 1998]. The parameters include the cold deck temperature, total supply air flow rate, floor area, pre-heat deck temperature, hot deck temperature, internal heat gain, outside air intake, room temperature, heat transfer value of the envelope, and economizer operation. Table 1 summarizes the parameter changes.

**Table 1**. Description of input variable changes.

| Input variables | Changes made |
|---|---|
| Cold deck temperature set point $T_c$ | decreased from 55 °F to 53 °F |
| Supply air | raised from 1.2 cfm/ft$^2$ to 1.3 cfm/ft$^2$ |
| Floor area | raised from 120,000 ft$^2$ to 130,000 ft$^2$ |
| Preheat coil temperature set point $T_{ph}$ | raised from 45 °F to 55 °F |
| Hot deck temperature set point $T_h$ | increased by 2 °F |
| Internal heat gain | decreased from 0.8 W/ft$^2$ to 0.4 W/ft$^2$ |
| Outside air flow | increased from 0.1 cfm/ft$^2$ to 0.15 cfm/ft$^2$ |
| Room temperature $T_{room}$ | raised from 73 °F to 74 °F |
| U-value | decreased from 0.1 to 0.08 Btu/h-°F-ft$^2$ |
| Economizer cycle | from none to temperature economizer cycle |

To help visualize the impacts of each parameter, the cooling and heating energy consumption are compared to their respective baseline. The ratio of the differences to the maximum baseline values is plotted in a percentage manner. Figure 1 presents the graphic signatures of a dual duct constant volume system.

**Cold Deck Temperature Set Point $T_c$**: Lowering $T_c$ increases both cooling and heating energy consumption. To maintain room conditions, the cold deck uses less air and the hot deck uses more. Since the hot deck temperature set point does not change, heating energy increases and cooling energy use increases. The shapes of these two signatures are not identical.

**Supply Air:** Increasing the amount of supply air results in an increase in both the heating and cooling energy consumption. Due to the higher hot deck set point at lower outside air dry bulb temperatures, the penalty decreases as outside air dry bulb temperature increases.

**Floor Area:** The effect of increasing the floor area is almost the same as that of increasing the amount of supply air.

**Preheat Coil Temperature Set Point $T_{ph}$**: Raising the preheat coil temperature set point increases both heating and cooling energy consumption in the temperature range where the preheat coil functions.

**Hot Deck Temperature Set Point $T_h$**: An increase in the hot deck temperature set point results in redistribution of the cold air and the hot air. In order to maintain the room conditions, less air flows through the hot deck and more air flows through the cold deck. The dramatic differences in these two signatures are again due to the better humidity control when more moisture is condensed as more air flows through the cold deck.

**Internal Heat Gain:** A reduction in internal heat gain calls for more heating and less cooling. To maintain the room conditions, less air flows through the cold deck and more air flows through the hot deck, resulting in an increase in heating and a decrease in cooling energy consumption.

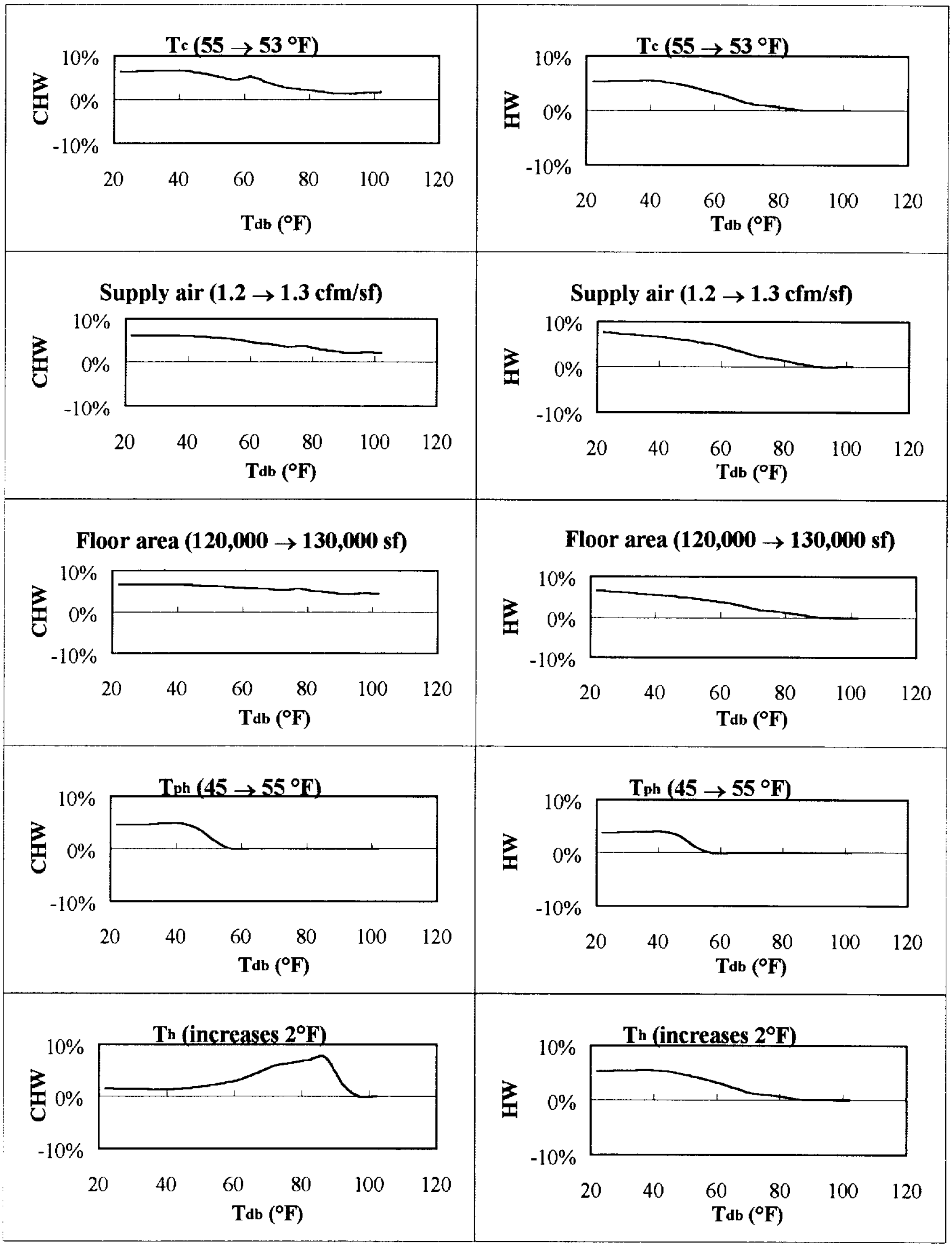


Figure 1a. Signatures of heating and cooling energy consumption for constant volume dual-duct AHUs

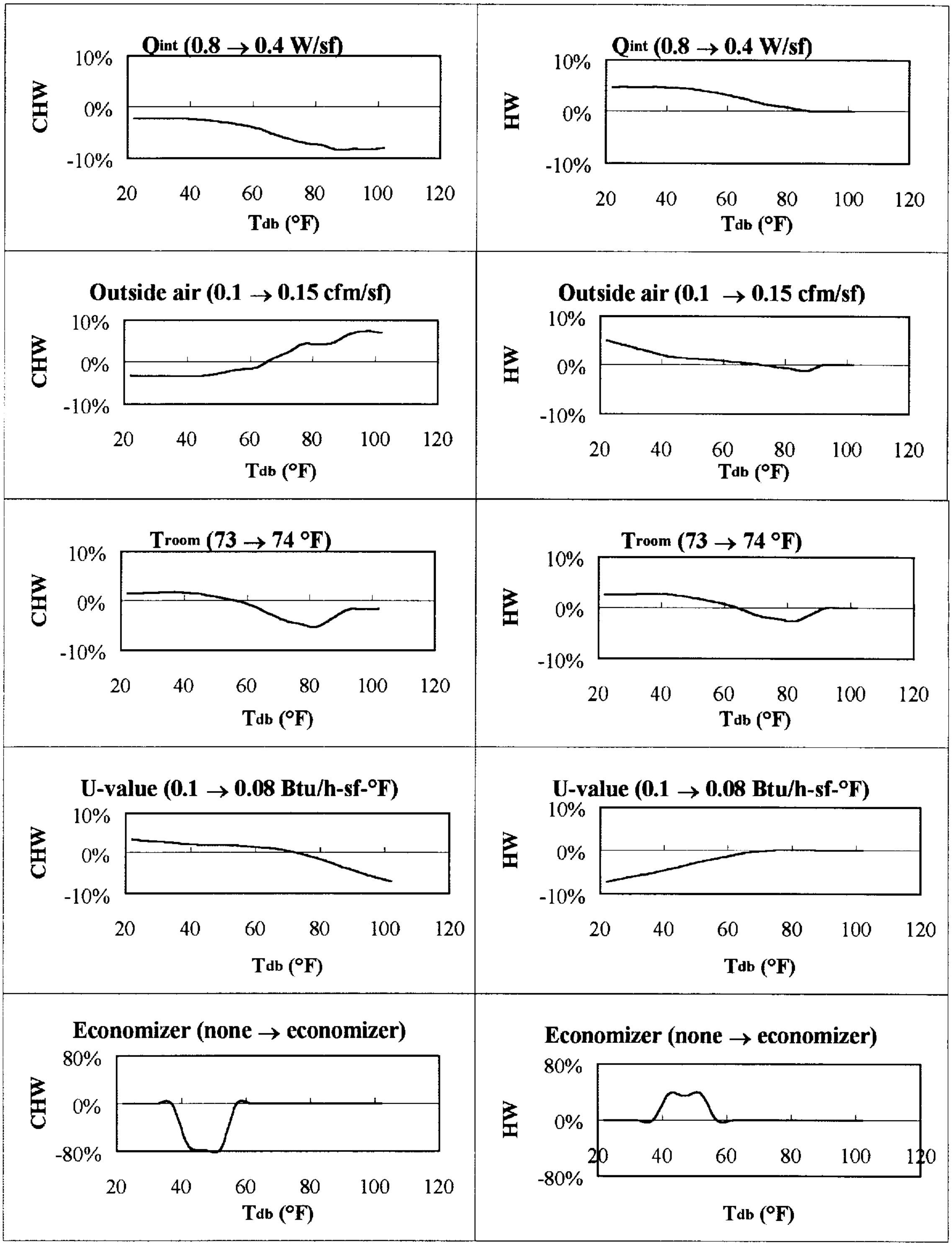


Figure 1b. Signatures of heating and cooling energy consumption for constant volume dual-duct AHUs (continued)

**Outside Air Flow:** Increasing the amount of outside air reduces the demand for cooling and increases the need for heating during the heating season. The situation reverses during the cooling season when more cooling is needed and less heating is required.

**Room Temperature $T_{room}$**: When the outside air temperature is low, an increase in room temperature set point results in an increase of air flow through the hot deck and a reduction in air flow through the cold deck. However, due to the higher return air temperature, the heating energy consumption tends to be reduced and the cooling energy consumption to be increased. The net effect in this case is an increase in both heating and cooling energy consumption. However, when the outside air temperature is high, the impacts of less cold air through the cold deck is greater than that of higher mixed air temperature due to higher returned air temperature, which results in a reduction in cooling energy consumption. On the other hand, the impacts of higher mixed air temperature is greater than that of the increased air flow through the hot deck, causing a reduction in heating energy consumption as well.

**U-Value:** A lower U-value results in less heat loss in the heating season and less heat gain in the cooling season through the building envelope. The result is that less air flows through the hot deck and more air flows through the cold deck during the heating season, thus reducing the heating energy consumption and increasing the cooling energy consumption. During the cooling season, less air is required to flow through the cold deck and more air can flow through the hot deck, thus reducing the cooling energy consumption. Notice that there is almost no change in the heating energy consumption since the hot deck temperature set point comes very close to the mixed air temperature.

**Economizer cycle:** The introduction of the economizer cycle reduces the cooling energy consumption significantly within the temperature range in which it functions. However, since the temperature of outside air entering the hot deck is lower than the mixed air temperature when there is no economizer cycle, the economizer results in a heating energy penalty.

## Two-Step Calibration Method

The input parameters can be listed in three groups: (1) system type; (2) parameters which affect weather dependent consumption; and (3) occupancy and schedule parameters. The system type is the most important input parameter. Fortunately, this input can be determined properly for most cases. The weather dependent parameters include deck temperature reset schedules (pre-heat, pre-cool, heating, cooling), air flow rates (total, minimum, outside air), building characteristics (heat transfer value, floor area, room temperature, and internal gain), and economizer. The occupancy and schedule parameter refers to the hourly load profiles.

Since the system type is generally selected properly, the model calibration process focuses on weather dependent and occupancy dependent parameters. If the model needs to be calibrated fully, an hourly model must be used. When half a year or more of hourly data is used, it is often difficult to manage the output of the simulation. Therefore, model adjustments are made in two steps.

First, the weather dependent parameters are calibrated using daily average data. During the first step of calibration, both simulated and measured energy consumption are plotted versus the ambient temperature. The graphic signatures are used to identify the appropriate parameters for adjustment. Second, a daily load profile needs to be introduced. The load profile varies from 0 to 2 with an average value of 1 for 24 hours. This profile should improve the hourly fit between measured and simulated heating and cooling energy consumption.

## Application

The signatures of heating and cooling energy consumption for AHUs can help HVAC simulation engineers calibrate models, identify malfunctions in HVAC components, and develop optimized HVAC operating and control schedules.
These signatures have been successfully applied to the calibration of HVAC models developed by the authors. One example is a 12-story building located at Galveston, Texas [Liu and Claridge, 1998]. This is an in-patient hospital facility with a total conditioned floor area of 298,500 ft$^2$. There are four constant volume dual-duct AHUs, which supply 302,000 CFM to the building with about 30% outdoor air intake. An Energy Management and Control System (EMCS) controls the pre-cooling deck discharge air temperature at 60 °F and the main cold deck discharge air temperature at 55 °F. Hourly whole-building cooling and heating consumption (chilled water and steam) are being measured by the LoanSTAR program [Turner, 1990].

Figure 2 compares the measured and first-cut simulated heating and cooling energy consumption. Simulated heating and cooling energy consumption totals were 26% and 16% lower than the measured values, respectively.

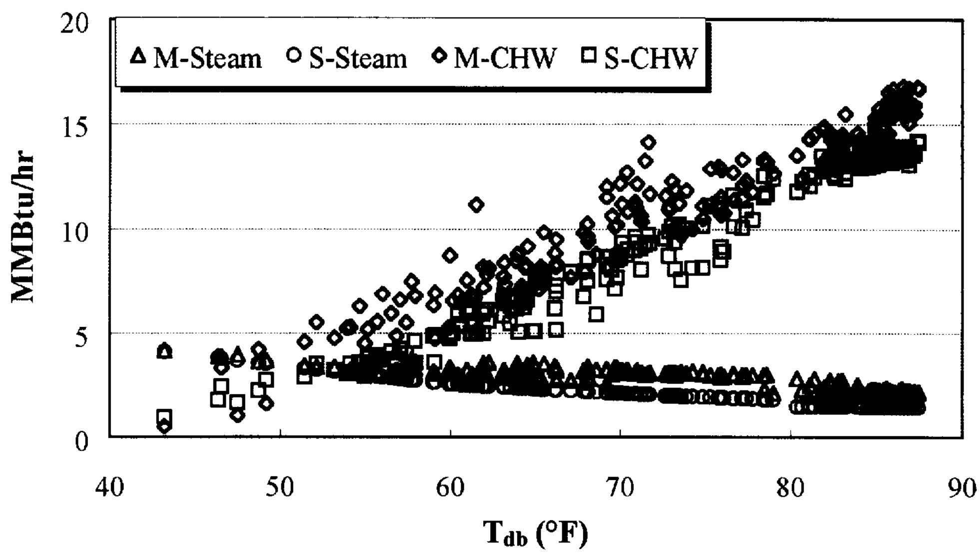


Figure 2. Comparison of measured and initial simulated heating and cooling energy consumption (M stands for measured values and S stands for simulated values)

Simulated heating and cooling energy consumption should match the measured energy consumption. The graphic signatures helps narrow the choice of parameters to change in the model: decrease the cold deck temperature set point; increase the amount of total supply air; increase the floor area; or increase the hot deck temperature set point.

Since total supply air and floor area values are believed to be accurate, it is speculated that the actual cold deck discharge air temperature might be lower than the set point value due to malfunctioning control components or temperature sensors. Consequently, the simulated pre-cooling deck discharge air temperature, main cold deck discharge air temperature, and hot deck discharge air temperature were adjusted to match simulated and measured cooling and heating energy consumption.

It was found that the simulated cooling and heating energy consumption matched measured values within 5% when the pre-cooling deck discharge air temperature was decreased by 8 °F to 52 °F, the main cold deck discharge air temperature was decreased by 3 °F to 52 °F, and the hot deck air temperature was assumed to be 5 °F higher than the set point.

To verify this, site measurements of these parameters were made [Liu and Claridge, 1995]. The cold and hot deck discharge air temperatures of all four AHUs were simultaneously measured using portable thermometers and the EMCS. It was found that the average main cold deck discharge air temperature was 2.4 °F lower than the EMCS measured value, and the average pre-cooling deck discharge air temperature was 3.2 °F lower than the EMCS measured value. The higher temperature values measured by the EMCS were due to the short probe sensors used by the EMCS. Normally, an averaging sensor is required to sample the entire cross-sectional area.

The measured pre-cooling and cold deck discharge air temperatures were 7.2 °F and 3.5 °F lower than the set point values, respectively, while the measured hot deck discharge air temperature was 5 °F higher than the set point. This confirmed the speculated sensor problem.

When the measured results were introduced into the model, the predicted total cooling and heating energy consumption values were only 3% and 0.3% lower than the measured values, respectively. Figure 3 shows a scatter plot of measured and calibrated simulated consumption versus temperature, while Figure 4 presents a time series comparison of these values.

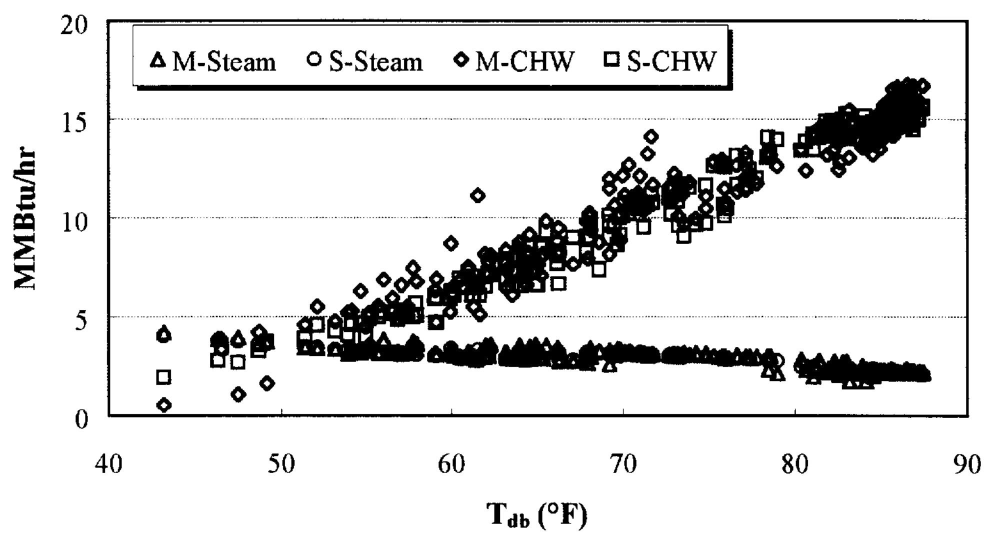


Figure 3. Comparison of measured and calibrated simulated heating and cooling energy consumption

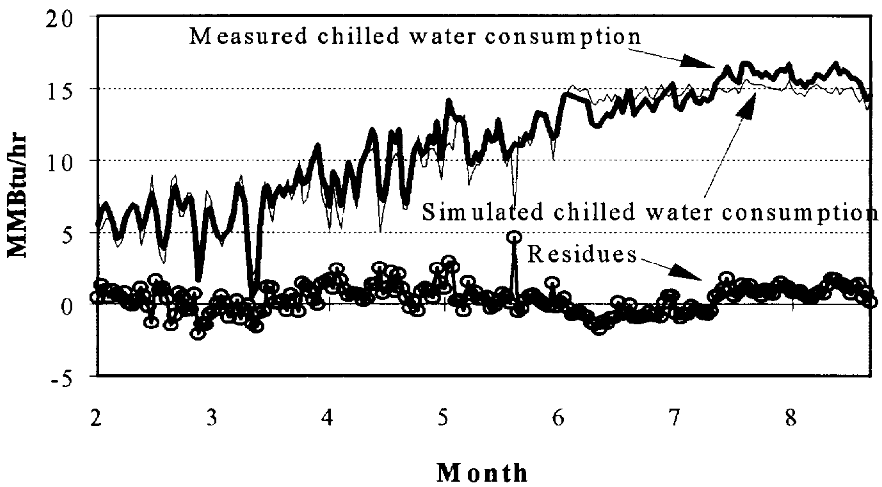


Figure 4a. Comparison of daily average calibrated simulated and measured cooling energy consumption

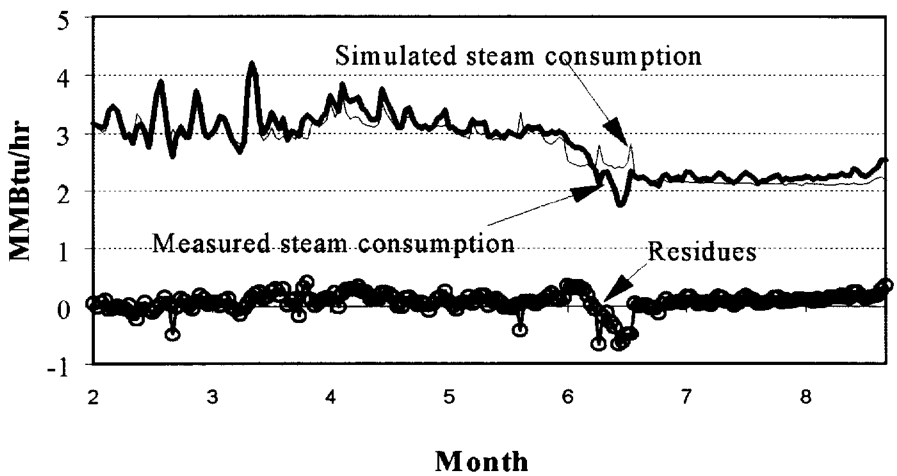


Figure 4b. Comparison of daily average calibrated simulated and measured heating energy consumption

## Conclusions

Accurate calibration of simulation models requires that the model is capable of modeling the departures from ideal behavior which occur in real systems, and several common model deficiencies are noted. Model calibration is generally a time consuming task at the daily or hourly level. A set of graphic signatures which can be used to assist engineers in determining a physically correct calibrated model has been presented. The graphic signatures can be used to rapidly identify the input parameter(s) which need to be changed to improve the agreement between the simulation output and measured consumption. A two-step graphic method is recommended to simplify the calibration process.

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