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ENERGY SYSTEMS LABORATORY TECHNICAL REPORT

Simulation of the Post-Retrofit Thermal Energy Use for the University Teaching Center (UTC) Building with the Use of Simplified System Models

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INTRODUCTION

Several state owned buildings with dual-duct constant volume (DDCV) systems have been retrofitted with energy efficient variable air volume systems (VAV) as part of the Texas LoanSTAR Program. One method of determining the energy savings resulting from energy conserving retrofits relies on the use of a model for the daily whole building consumption, E_{pre}, in the pre-retrofit configuration. E_{pre} is typically a function of primary influencing parameters such as ambient temperature, humidity, building internal gains and others (Figure 1). Following the retrofit, the energy saved, E_{sav} is determined using measured daily consumption, E_{meas} as shown in Figure 1. This method is being used in the Texas LoanSTAR monitoring and analysis program for buildings that have adequate pre-retrofit monitored data (Kelly et al., 1992).

Unfortunately, in the University Teaching Center (UTC) the retrofits were completed before the monitoring instrumentation was installed. Therefore, no pre-retrofit monitored data are available. Hence another method to estimate savings was needed. Such a method was developed and tested on a large engineering center (Katipamula and Claridge 1991). This method was based on the use of the ASHRAE TC 4.7 simplified energy analysis procedure (SEAP). It involved developing one model each for the VAV (post-retrofit system) and the DDCV (pre-retrofit system) systems.

Since load calculation is independent of the type of HVAC system, the module that estimates the loads is common for both models. First, the VAV model is calibrated by comparing the simulated energy use with the measured post-retrofit energy use. The parameters that are adjusted in the calibration process are: (i) zone envelope loads, (ii) zone set point temperature, (iii) ventilation and infiltration rate, (iv) adjustments for mass effects (CLFs), and (v) minimum speed of the supply fans. The calibration of the VAV model also implies calibration of the loads module. Therefore, the loads module can

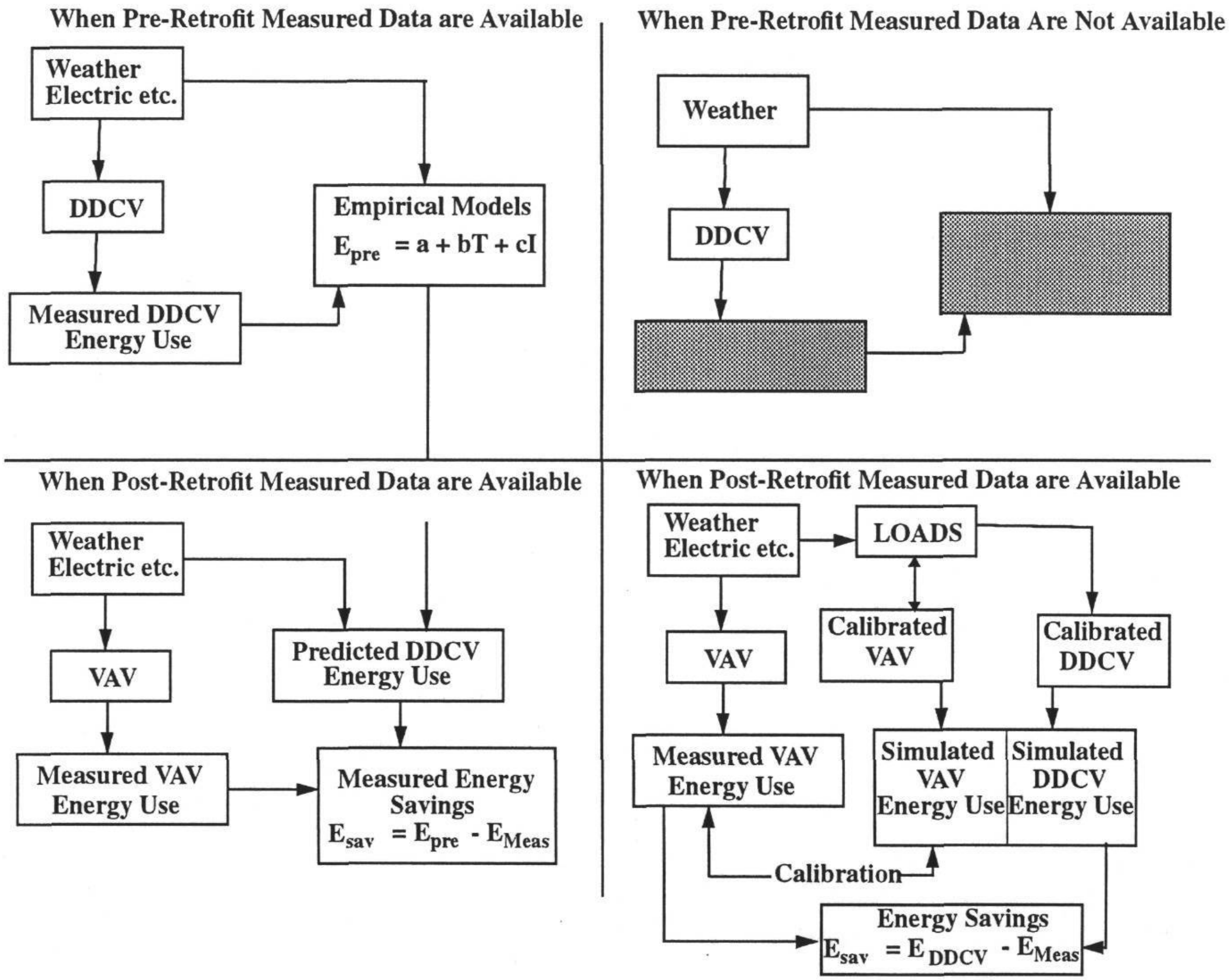

Figure 1 - Schematic of Saving Estimation Procedure.

be used with the DDCV system model to predict the energy use of the building in the pre-retrofit condition.

DESCRIPTION OF THE MONITORED BUILDING

The UTC located on University of Texas Campus (UT) is a six-story building constructed in 1984, with a conditioned floor area of 150,000 sf. A large portion of the conditioned area consists of classrooms, and a few offices, auditoriums and study areas. The typical operating hours of UTC are 7 a.m. to 11 p.m., Monday through Friday, with very little occupancy during weekends. The UTC is a heavy structure with 6 inch concrete floors and insulated concrete walls. The exterior walls consist of limestone panels on concrete blocks. The windows consist of 1/4 inch, single pane tinted glass. The windows are all recessed 3 feet into the walls and receive very little direct sunlight. In November 1990, all the single-fan DDCV systems were converted to dual-fan VAV systems.

The HVAC systems are supplied with high pressure steam, chilled water and electricity from the central campus plant. The campus does not individually meter buildings, but a data logger was installed in the UTC (in October 1990) to collect hourly post-retrofit consumption data. Whole building data collected include electricity use, air handler electricity, chilled water load (Btu), and steam load (Btu). A weather station on a different LoanSTAR site (close to UTC) collects outdoor dry-bulb temperature, relative humidity, horizontal solar radiation and wind velocity data. In addition, hourly dry-bulb and dew-point temperatures from the National Weather Service (NWS) (Austin airport) are also recorded.

After the retrofit, the UTC had eight dual-duct, dual-fan VAV systems serving the conditioned area and eight VAV return air fans. The supply fans are rated between 5 and 25 kW and the return air fans are rated between 3 and 8 kW. A CO_2 sensor is used to control the outdoor air intake by maintaining the CO_2 concentration in the return air below

2000 ppm.

MODEL DEVELOPMENT

The compliance with or deviation from expected performance of the system can best be determined if the measured data are compared with the predictions of a calibrated simulation model that uses measured values of weather variables and system parameters. A VAV model, based on the ASHRAE TC 4.7 simplified energy analysis procedure (SEAP), was developed to simulate the energy use of the UTC building. The TC 4.7 SEAP uses hourly bins to estimate the energy use whereas the current model calculates energy use hourly. For details refer to the attached paper by Katipamula and Claridge 1991. The flow chart of the simulation process and load calculations are shown in Figures 2 and 3.

Load Calculations

Climate Data The monitored hourly outdoor dry-bulb temperature and relative humidity are used in the simulation process.

Physical Data The building dimensions, construction materials, percent glass area, orientation of the building, number of zones, area of each zone, number of people, and peak electric consumption (equipment and lighting) are used to estimate the loads. The typical floor layout of the UTC (Figure 4) is about 30,000 sf. The total external wall area is about 50,000 sf of which 5,000 sf is glass, and the area of the roof is about 30,000 sf.

Envelope Loads The envelope loads include conduction losses/gains and solar heat gains. The CLTD method is used to estimate envelope loads at 95, 60 and 20 F outdoor dry-bulb temperature (McQuiston and Spitler, 1992). Figure 5 shows these loads for the UTC as a function of outdoor dry-bulb temperature. The overall U-value for the external walls and the glass is assumed to be 0.1 and 1.0 $\frac{Btu}{hFsf}$, respectively. The U-value of the roof is about 0.088 $\frac{Btu}{hFsf}$.

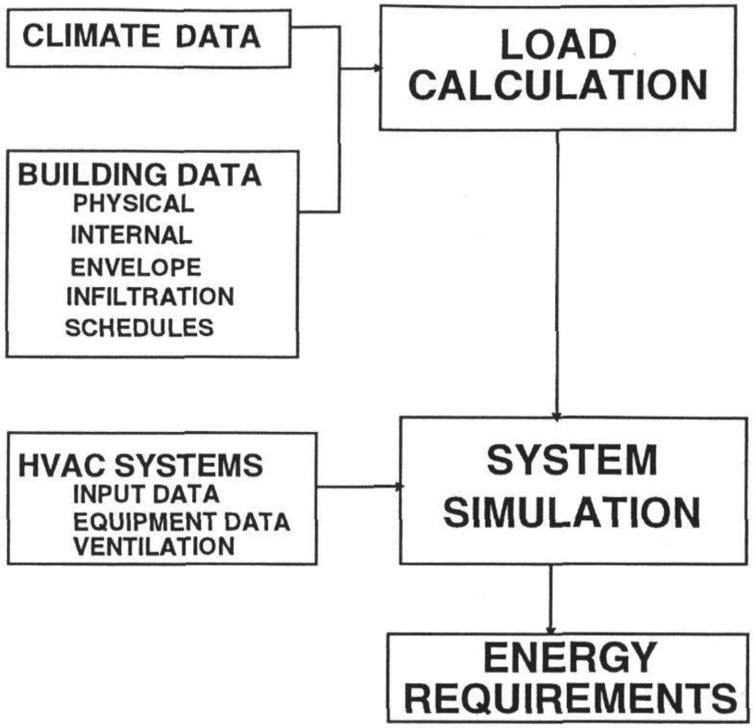

Figure 2 – Schematic of the Simulation Procedure.

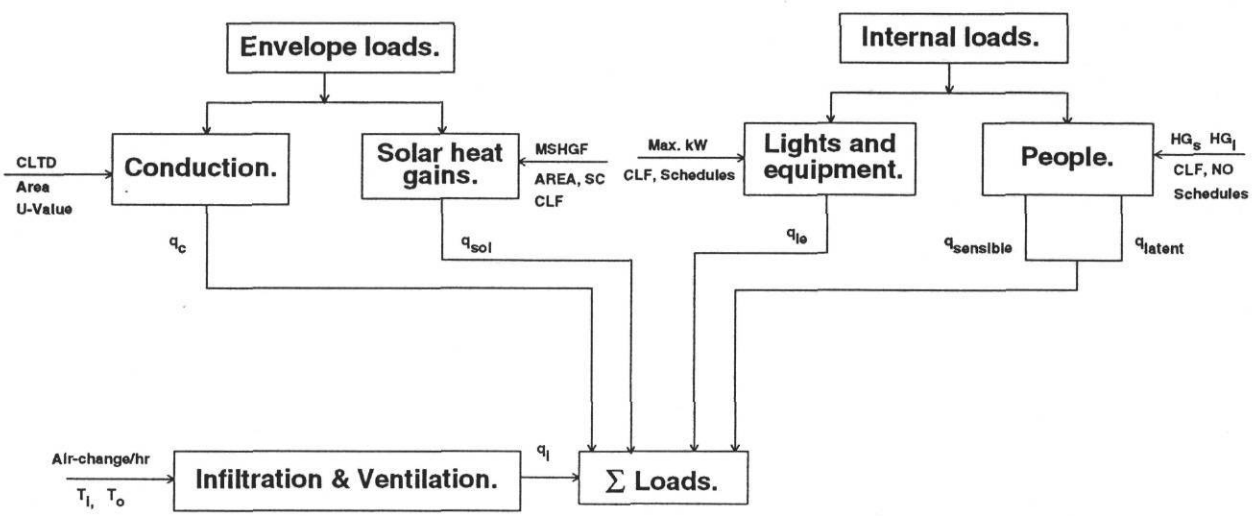

Figure 3 – Schematic of the Load Calculation.

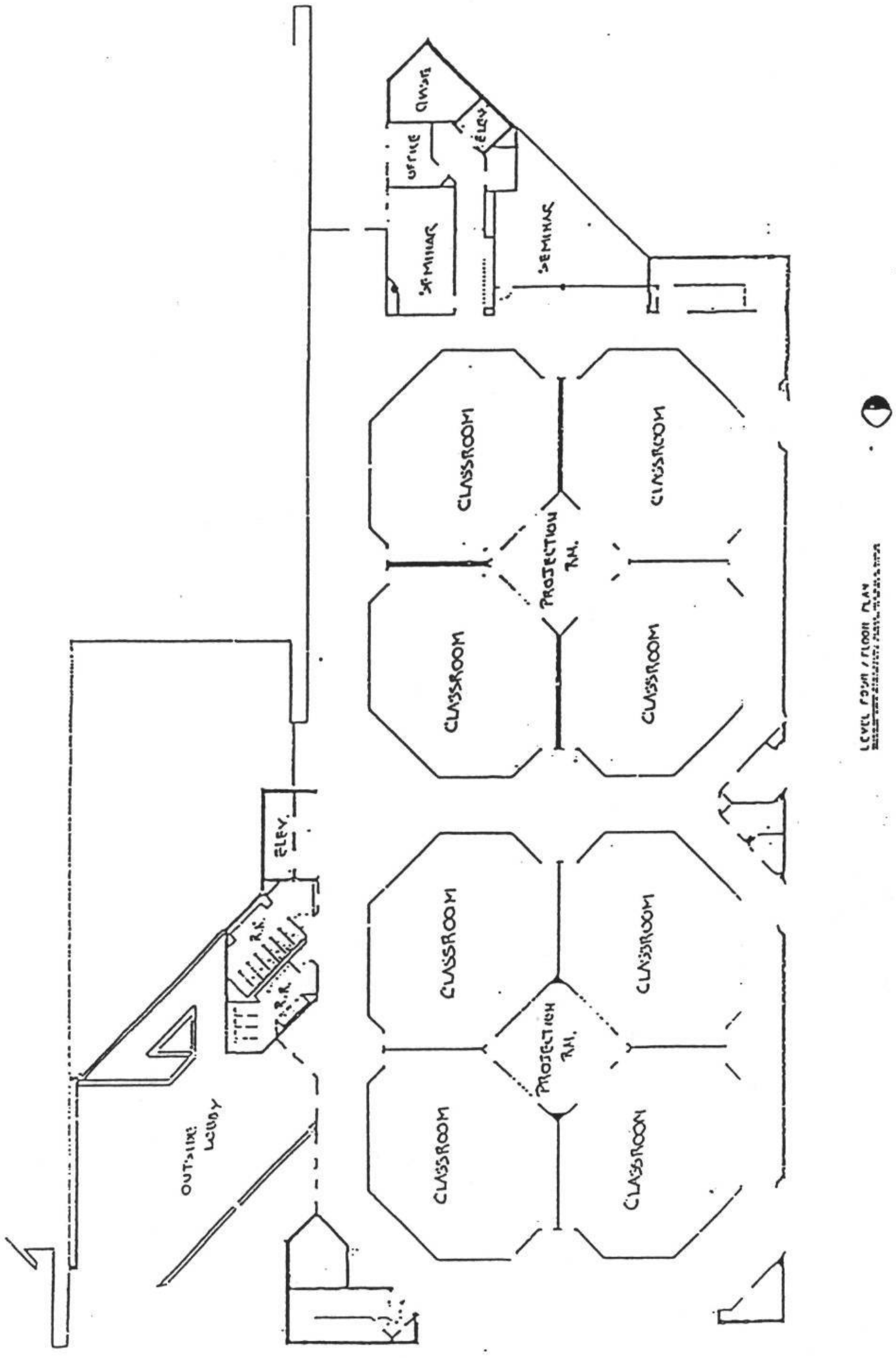

Figure 4 – Typical Floor Layout of the UTC Building.

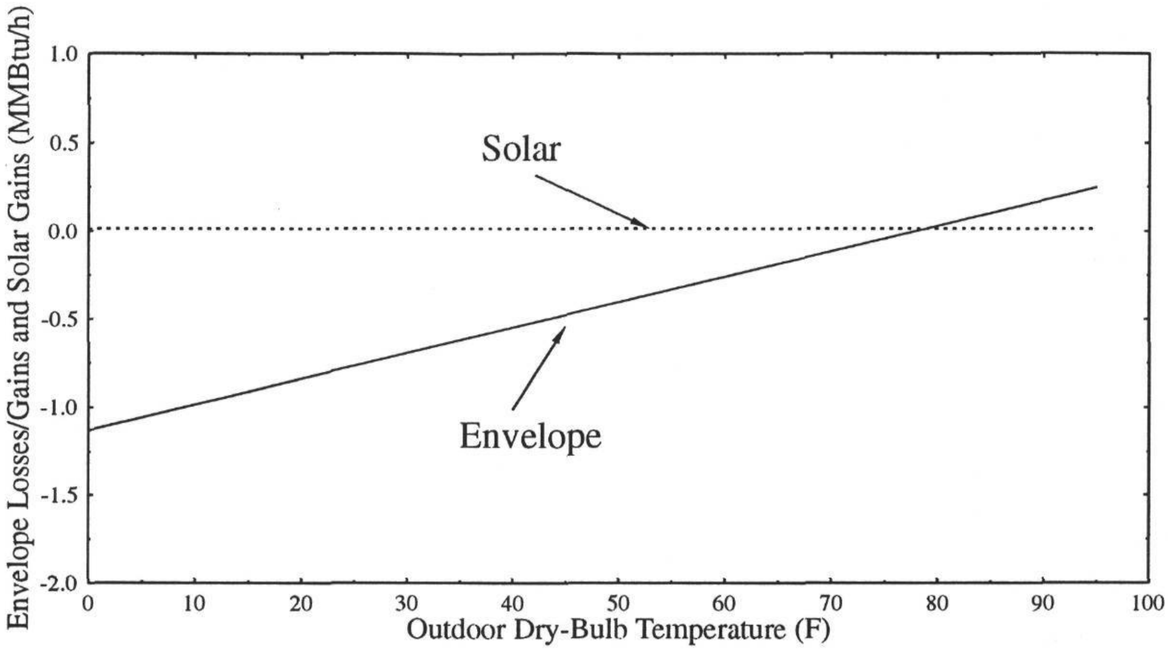

Figure 5 – Conduction Heat Losses/Gains and Solar Heat Gains as a Function of Outdoor Dry-Bulb Temperature.

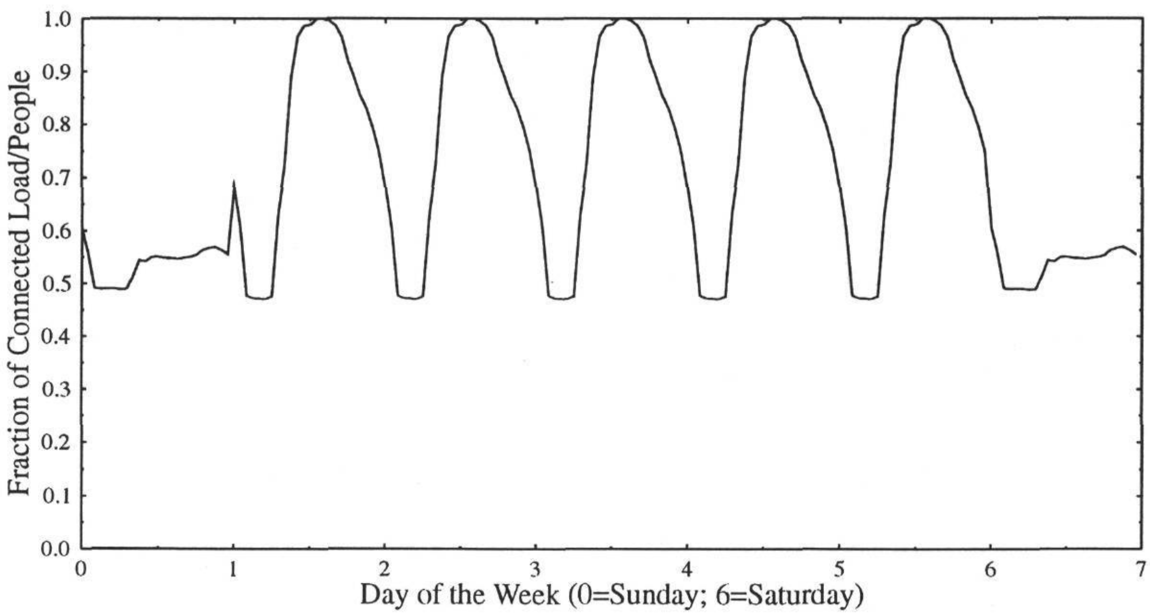

Figure 6 – People, Lighting and Equipment Schedules.

Lights and Equipment In commercial buildings the internal loads constitute a major portion of the total cooling load. The cooling load q_{le} due to internal loads varies with time of day and day of week. The maximum lighting and equipment use for the building are 1.3 and 2.5 $\frac{Btu}{h\ sf}$, respectively, based on hourly post-retrofit monitored data. Ninety-five percent of the maximum lighting and equipment use is directly attributed to the conditioned area.

The typical hourly profile for lights and equipment (whole building electric minus the air handler and pumps) is shown in Figure 6. These profiles are based on the monitored hourly consumption and are generated using the methodology developed by Katipamula and Haberl [1991], which is based on statistical analysis of monitored hourly non-weather dependent loads (lights, equipment etc.). Estimating CLF values to account for the mass effects is difficult. Initially, the CLF of one will be assumed and actual CLFs will be derived by matching typical simulated hourly cooling load profiles with the measured cooling load profiles. This procedure is described in the calibration section of this paper.

People The maximum number of people occupying the building on weekdays between 8 a.m. and 5 p.m. is assumed to be 500 and 100 on weekends and rest of the weekday hours. The sensible and latent heat gain per person is assumed to be 250 Btu/h each. The people schedule is same as the lights and equipment schedule (Figure 6).

Infiltration and Ventilation In estimating the infiltration load, the mass flow rate is assumed to be equivalent of 0.2 air changes per hour. Since the outdoor air intake is based on the CO_2 concentration, the exact volume of the outdoor air at any given time is an unknown. Therefore, a constant volume of 8,000 cfm will be assumed for base case. This turns out to be 16 cfm/person at peak occupancy. The ASHRAE recommended ventilation rate is 10-15 cfm/person.

HVAC SYSTEM SIMULATION

Since the building has just one type of HVAC system the building was divided into two zones: external and core (Figure 7).

The dual-duct, dual-fan VAV system has a central air handling unit with two fans (one each for hot and cold ducts), cooling coil, heating coil and mixing box (Figure 7). Air leaving the cooling coil and heating coil may be controlled as fixed set point, outside air reset, or zone controlled reset. Air leaving the unit is delivered to mixing boxes that modulate the zone air flow rates in response to the zone thermostat. When the air flow is at a minimum and there is a call for heating, the zone thermostat opens the hot dampers in the mixing box. Since the air flow is modulated to meet the zone load, fan power consumption will also be modulated. The outside air may be set at a fixed amount or an economy cycle may be used to increase the outside air quantity to reduce the cooling coil load.

The existing controls show that the air leaving the cooling coil is fixed at 50 F and the air leaving the heating coil is reset based on the outside air (Table 1). The steam supply to the building is usually turned off during the summer months (June through October). The outdoor air is mixed with the return air and the mixture enters the cooling coil, whereas only the return air enters the heating coil.

Table 1 – Hot Deck Reset Schedule.

Outdoor Air Temperature (F)	Hot Deck Discharge Air Temperature (F)
20	120
80	110

The external zone is assumed to be 35% and the core zone to be 65% of the total conditioned area. The energy use of the two zones is simulated with a dual-duct, dual-fan VAV system. The core zone is assumed to be insulated from the envelope

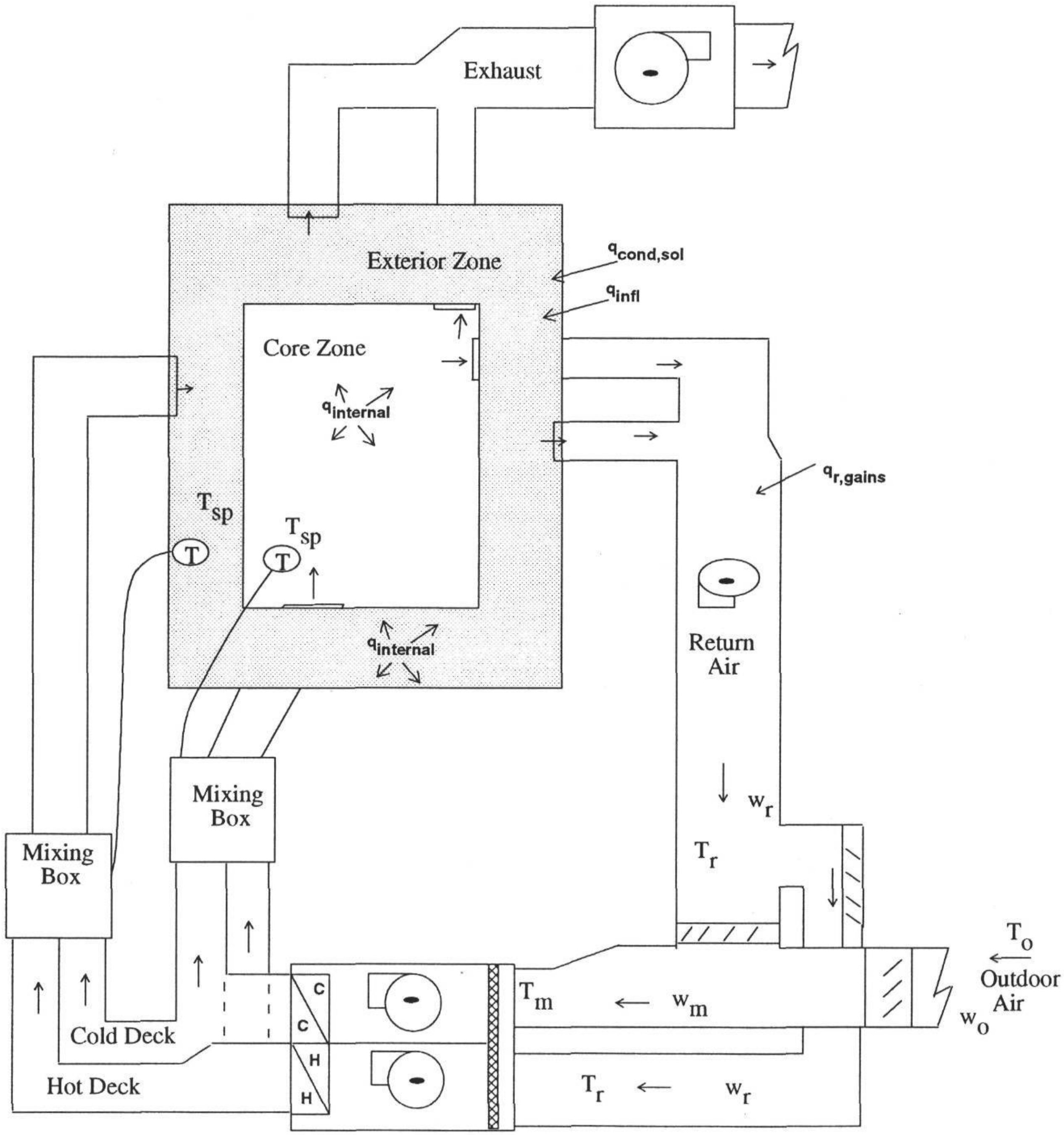

Figure 7 – Schematic of the HVAC System in the UTC.

heat losses/gains, solar heat gains and infiltration heat loss/gain. The roof conduction losses/gains from the intermediate and core zones are lumped with the external zone. The envelope loads (conduction losses/gains and solar gains), for the external zone, for a given outdoor temperature are linearly interpolated. The internal loads for the two zones are estimated based on the hour of the day and the day of the week. The infiltration load is based only on the outdoor temperature and 0.2 air changes/hour. The zone temperature is assumed to be constant at 78 F for the base case.

According to the design specifications the minimum speed for the supply fans is 30%. Because the hot ducts have to be maintained at the same static pressure as the cold ducts to prevent cold air from coming back through the hot ducts. The total rated flow (at full speed) for the hot deck fans and the cold deck fans is $\approx$ 55,000 cfm and 80,000 cfm, respectively. All flows are estimates based on the audit reports and HVAC specifications provided by the UT physical plant.

Since the outdoor air intake is based on the CO_2 concentration, the exact volume of the outdoor air at any given time is an unknown. Therefore, a constant volume of 8,000 cfm is assumed for the base case. Since the simplified VAV model does not simulate a heat exchanger, the specific humidity of the air leaving the cooling coil cannot be calculated. Therefore, the cooling coil leaving condition is approximated by assuming a supply air relative humidity of 85%.

BASE CASE RESULTS

The inputs to the VAV model were outdoor dry-bulb temperature, outdoor relative humidity, and decimal date. The day of the week and the hour of the day is extracted from the decimal date. The conduction heat gains/losses, solar heat gains, and infiltration heat gains/losses are calculated for the given outdoor condition as described in the earlier section. The internal heat gains are calculated for the given day of the week and hour of the day.

For the base case the following assumptions are made: (i) indoor temperature set point of 78 F, (ii) a constant outdoor air intake of 8,000, (iii) minimum air flow in the cold duct of 25,000 cfm, (iv) minimum air flow in the hot duct of 15,000 cfm, (v) cold deck supply temperature of 50 F, and (vi) CLF of one (for internal gain calculation). First, simulated energy use (base case) from the VAV model will be compared with the measured energy use. The base case comparisons will be for the time period October 1991 through June 1992.

Figure 8 shows a time series plot of the simulated chilled water consumption for the UTC. For the same time period the chilled water consumption and residuals (simulated - measured) are also plotted as a function of outdoor dry-bulb temperature (Figure 9). The minimum chilled water consumption which occurs during the unoccupied hours is 0.5 MMBtu/h. The maximum consumption during the occupied hours is about 1.75 MMBtu/h. The measured chilled water consumption is shown in Figures 10 and 11. The general trend of the simulated consumption appears to be similar to the measured chilled water. However, the residuals above 70 F outdoor dry-bulb temperature are higher than the residuals below 70 F outdoor dry-bulb temperature.

The simulated steam consumption, for the base case, is shown in Figures 12 and 13. The minimum simulated steam consumption is about 0.5 MMBtu/h and the maximum is about 1.75 MMBtu/h. The measured steam consumption for the same period is shown in Figures 14 and 15. The general trends appear to be same as that of the simulated steam consumption. However, there is slightly more variation in the measured steam consumption. The hot deck and the cold deck controls were adjusted on March 17, 1992; therefore, there is change in the steam usage beyond March 17, 1992 (4459).

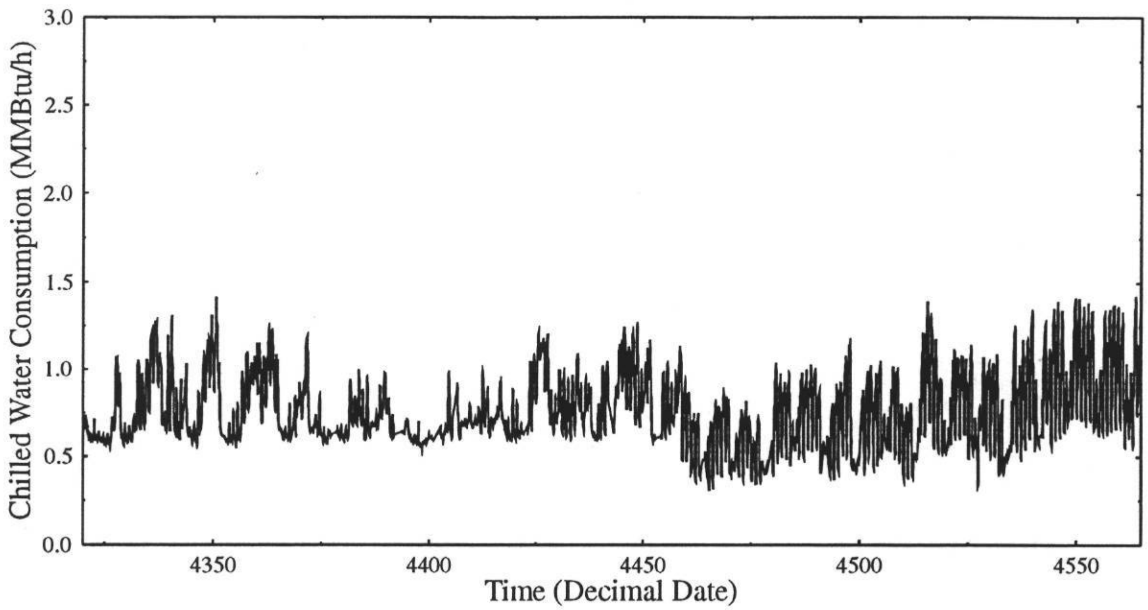

Figure 8 – Simulated Chilled Water Consumption for UTC (October 1991 – June 1992).

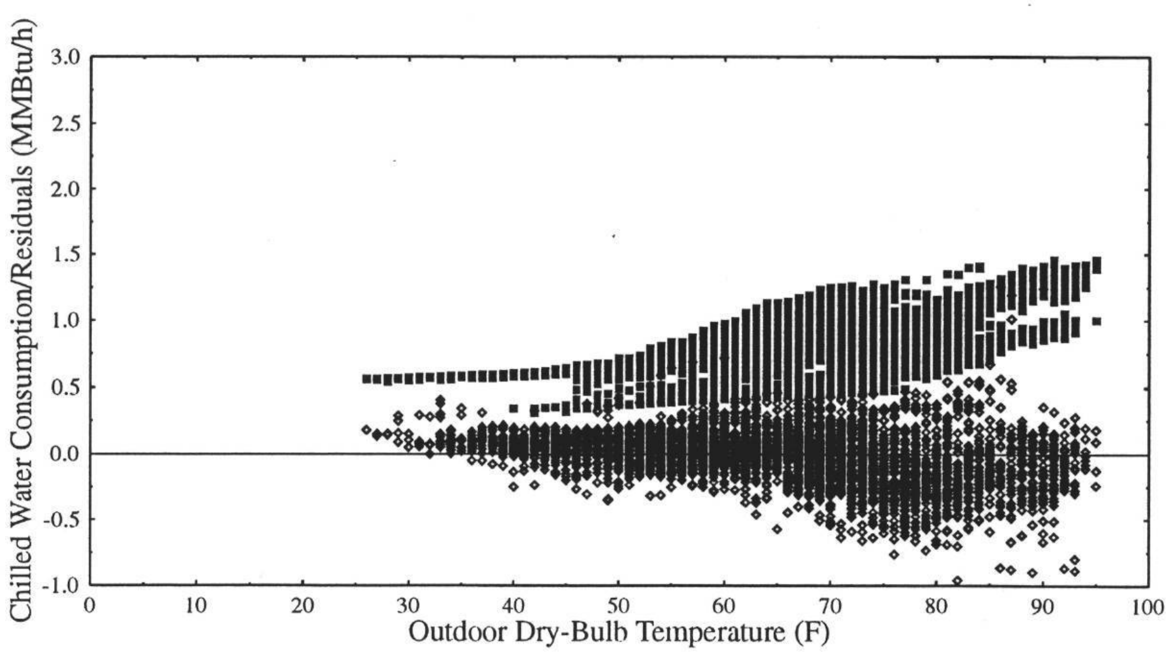

Figure 9 – Simulated Chilled Water Consumption and Residuals for UTC as a Function of Outdoor Dry-Bulb Temperature (October 1991 – June 1992).

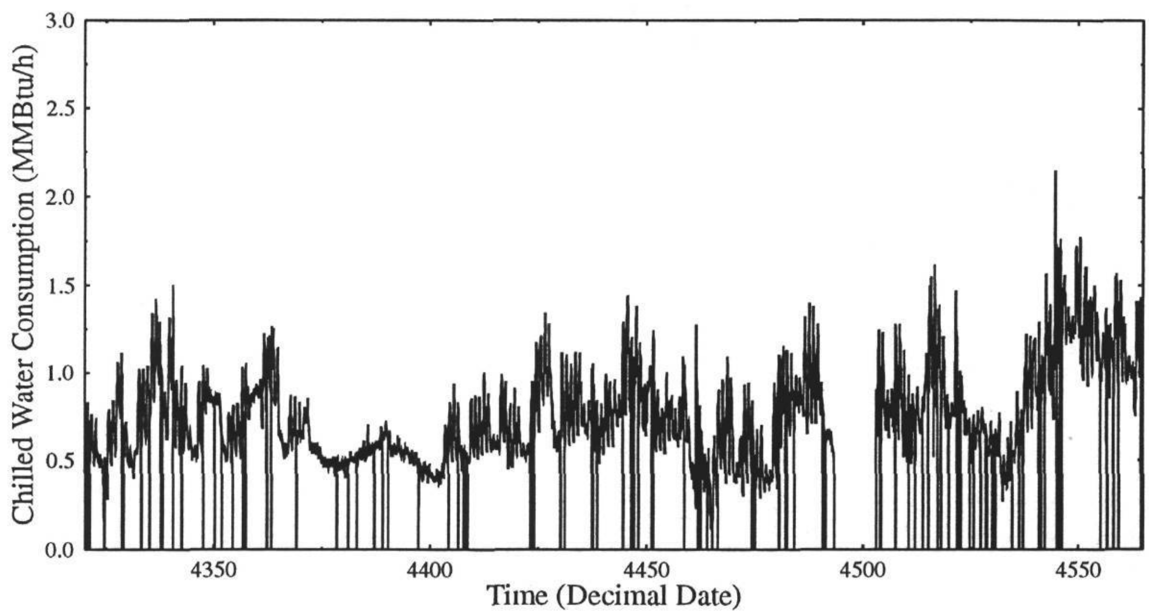

Figure 10 – Measured Chilled Water Consumption for UTC (October 1991 – June 1992).

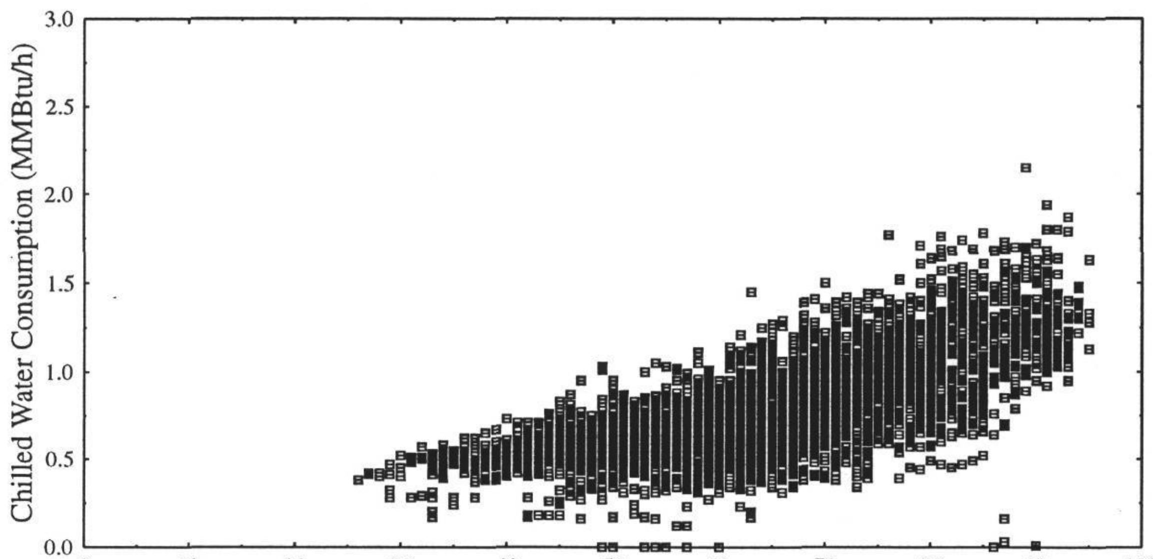

Figure 11 – Measured Chilled Water Consumption for UTC as a Function of Outdoor Dry-Bulb Temperature (October 1991 – June 1992).

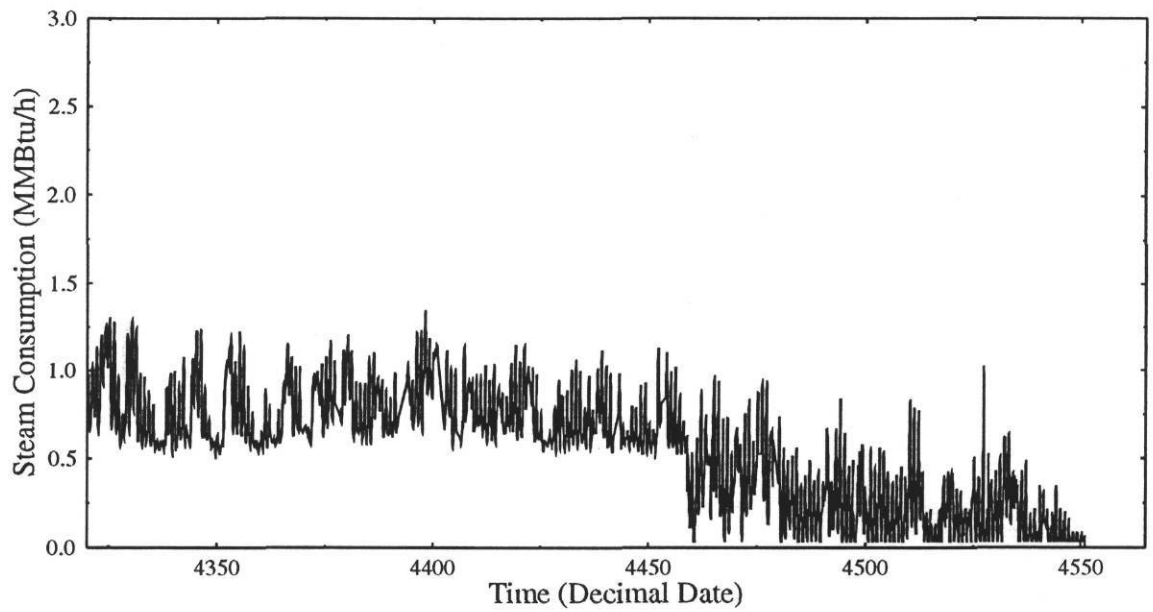

Figure 12 – Simulated Steam Consumption for UTC (October 1991 – June 1992).

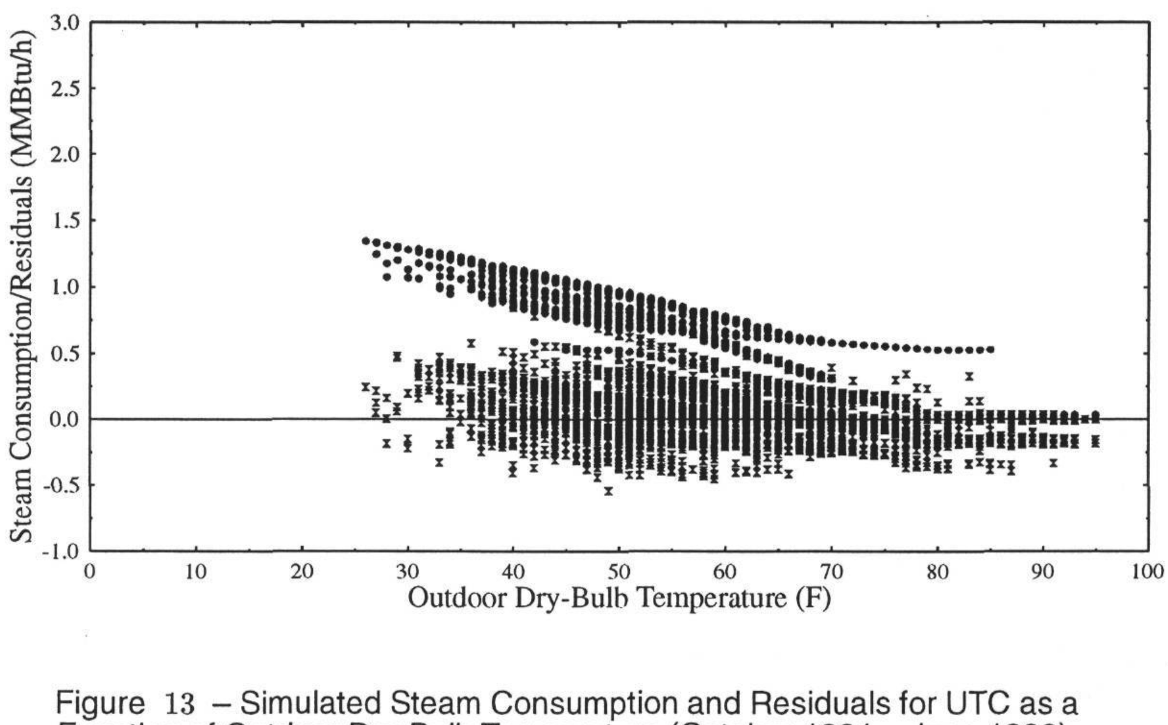

Figure 13 – Simulated Steam Consumption and Residuals for UTC as a Function of Outdoor Dry-Bulb Temperature (October 1991 – June 1992).

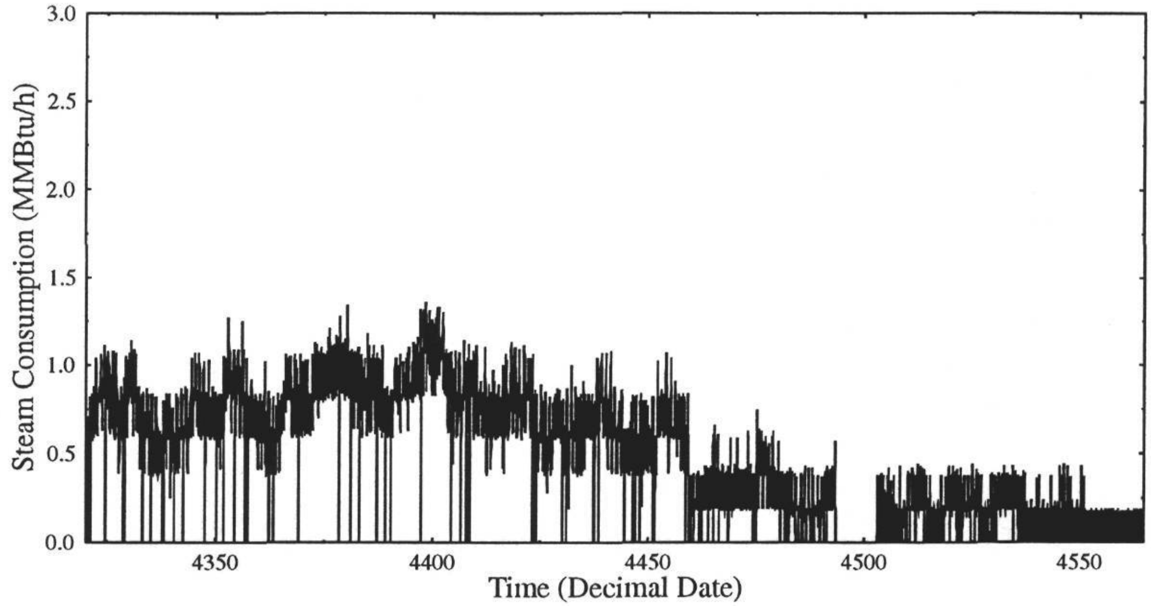

Figure 14 – Measured Steam Consumption for UTC (October 1991 – June 1992).

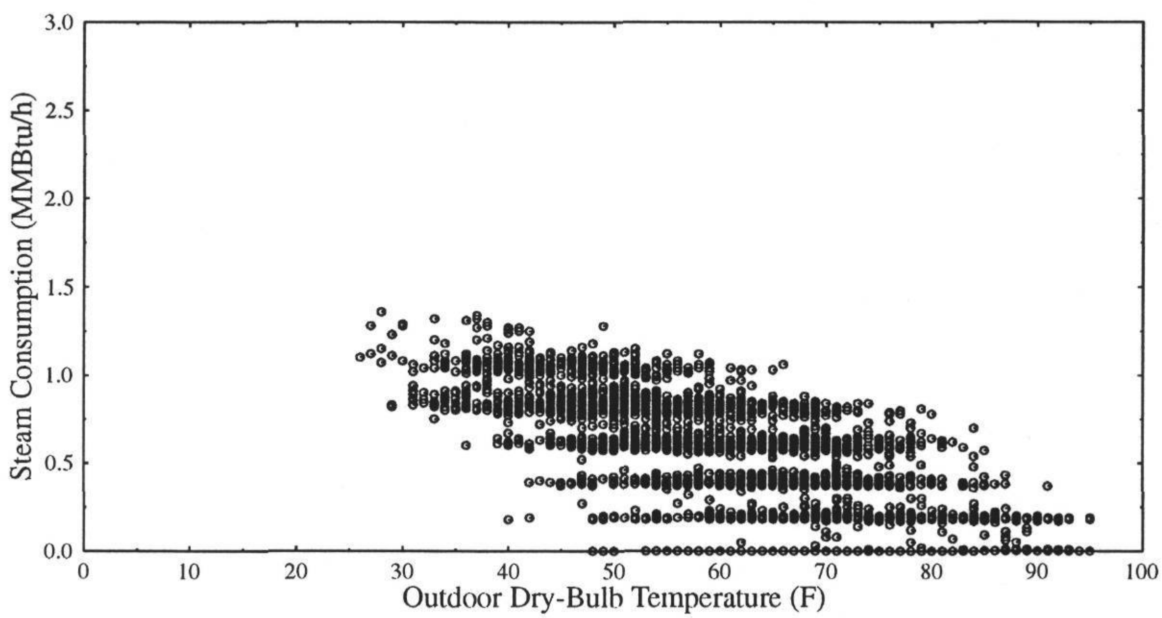

Figure 15 – Measured Steam Consumption for UTC as a Function of Outdoor Dry-Bulb Temperature (October 1991 – June 1992).

CALIBRATION OF THE VAV MODEL

The general trends from the base case VAV model are comparable to the measured consumption. However, several parameters of the model need calibration: (i) adjustments of controls for consumption after March 17, 1992, (ii) adjustments for mass effects (CLFs), (iii) zone set point temperature, (iv) ventilation rate, and (v) minimum speed of the supply fans.

In the middle of March 1992 the hot deck and cold deck controls were tuned. According to the UT physical plant personnel the cold deck supply temperature was raised from 50 F to 60 F and the hot deck reset schedule was also changed as shown in Table 2. Therefore, the cold deck and the hot deck supply temperatures were changed accordingly during simulation.

Table 2 – Hot Deck Reset Schedule.

Outdoor Air Temperature (F)	Hot Deck Discharge Air Temperature (F)
20	120
80	80

For the base case, as mentioned earlier, the CLFs for internal gains were assumed to be 1. This produced large errors between 7 a.m. and 10 a.m. and between 8 p.m. and 11 p.m. The actual CLFs for the internal gains were estimated by regressing the measured chilled water consumption with the simulated:

$$CWm_i = clf_i \times CWs_i$$

where CWm_i and CWs_i are the measured and simulated chilled water consumption at hour i (i=1,2,... 24), clf_i is the cooling load factor at hour i. The profile for the new clf_i is shown in Figure 16.

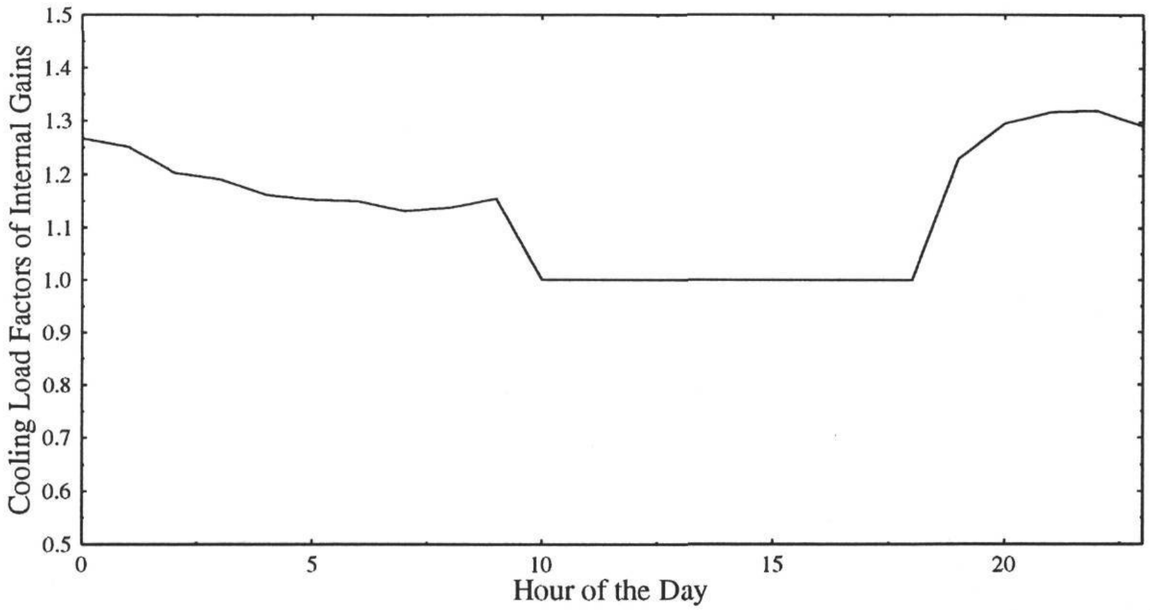

Figure 16 – Cooling Load Factor for Internal Gains.

The optimum zone set point temperature, ventilation rate, and minimum speed of the hot deck supply fans were estimated by minimizing the root mean squared error (RMSE):

$$\text{RMSE} = \sqrt{\frac{\sum_{i=1}^{n}(CWs_i - CWm_i)^2}{n}} \tag{6}$$

where n is the total number of data points. Table 3 and Figures 17 and 18 show the RMSE for various zone set point temperatures, ventilation rates, and minimum hot deck supply. First, the RMSE was minimized by optimizing the zone set point temperature. Then the RMSE was minimized by optimizing the ventilation rate, and finally, minimum hot deck supply was optimized. The use of a zone temperature of 76, ventilation rate of 12,000 cfm and minimum hot deck supply of 12000 cfm minimized the RMSE of the cooling and heating energy use.

The calibrated chilled water and steam consumption and residuals (simulated - measured) are shown in Figures 19 and 20. Although at some hours the simulated con-

Table 3 – Comparison of RMSE Cooling Load at Various Zone Set Point Temperatures, Ventilation Rates and Minimum Hot Deck Supply.

Zone Temperature (F)/ Ventilation Rate/Minimum HD Supply cfm	RMSE CW (MMBtu/h)	RMSE HW (MMBtu/h)
72/8,000/15,000	0.31	0.17
74/8,000/15,000	0.28	0.18
76/8,000/15,000	0.22	0.18
78/8,000/15,000	0.21	0.19
80/10,000/15,000	0.21	0.20
76/10,000/15,000	0.22	0.18
76/12,000/15,000	0.22	0.18
76/14,000/15,000	0.23	0.18
76/12,000/10,000	0.21	0.20
76/12,000/11,000	0.21	0.19
76/12,000/12,000	0.21	0.18
76/12,000/13,000	0.21	0.18
76/12,000/14,000	0.21	0.18

sumption is significantly different from the measured, the RMSE is 0.21 and 0.19 MMBtu/h for chilled water and steam, respectively. The hourly simulated values are summed up to daily and compared with the daily measured consumption in Figures 21 and 22. The simulated daily chilled water and steam consumption compare well with the daily measured consumption (RMSE 2.48 MMBtu/day and 2.36 MMBtu/day).

PRE-RETROFIT ENERGY USE FOR THE UTC BUILDING

In the absence of pre-retrofit data the retrofit savings can be estimated by predicting the pre-retrofit system behavior with the use of hourly simulation model. The UTC building had eight dual-duct constant volume (DDCV) systems with eight return air fans.

The DDCV system has an air handling unit with a fan, cooling coil, heating coil, and mixing box at each zone. The pre-retrofit controls in the UTC showed that the air leaving the cooling coil was fixed and the air leaving the heating coil was reset based on the outdoor air dry-bulb temperature. The preheat coil and the economizer cycle were not

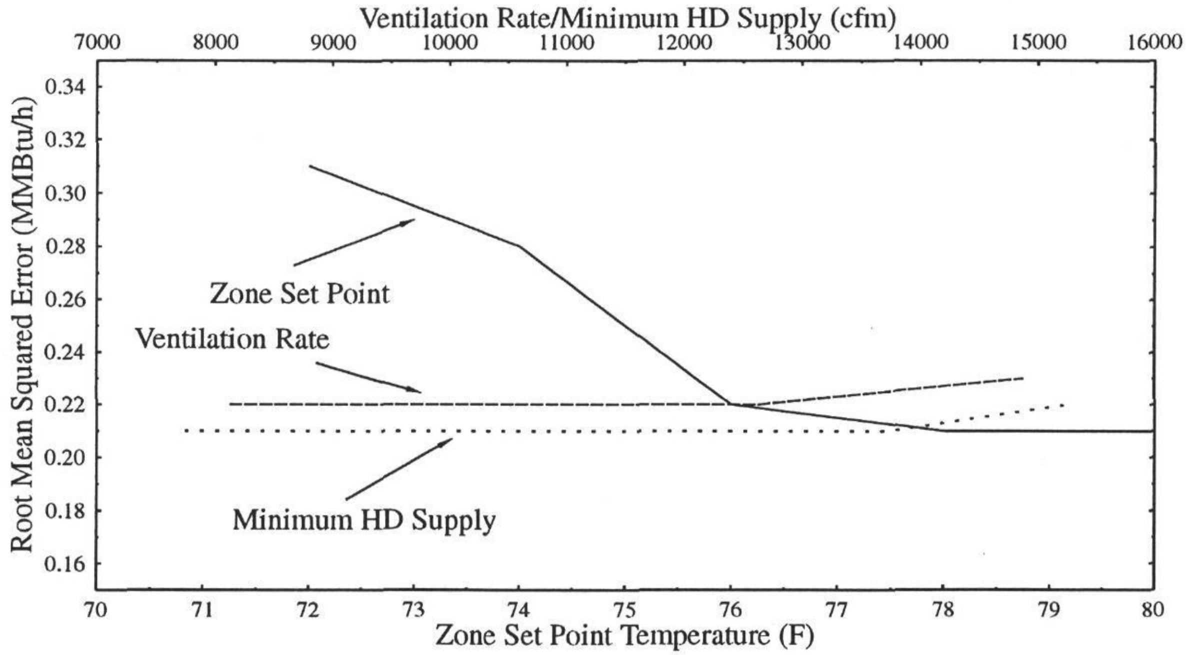

Figure 17 – Chilled Water RMSE as a Function of Zone Set Point, Ventilation Rate, and Minimum Hot Deck Supply.

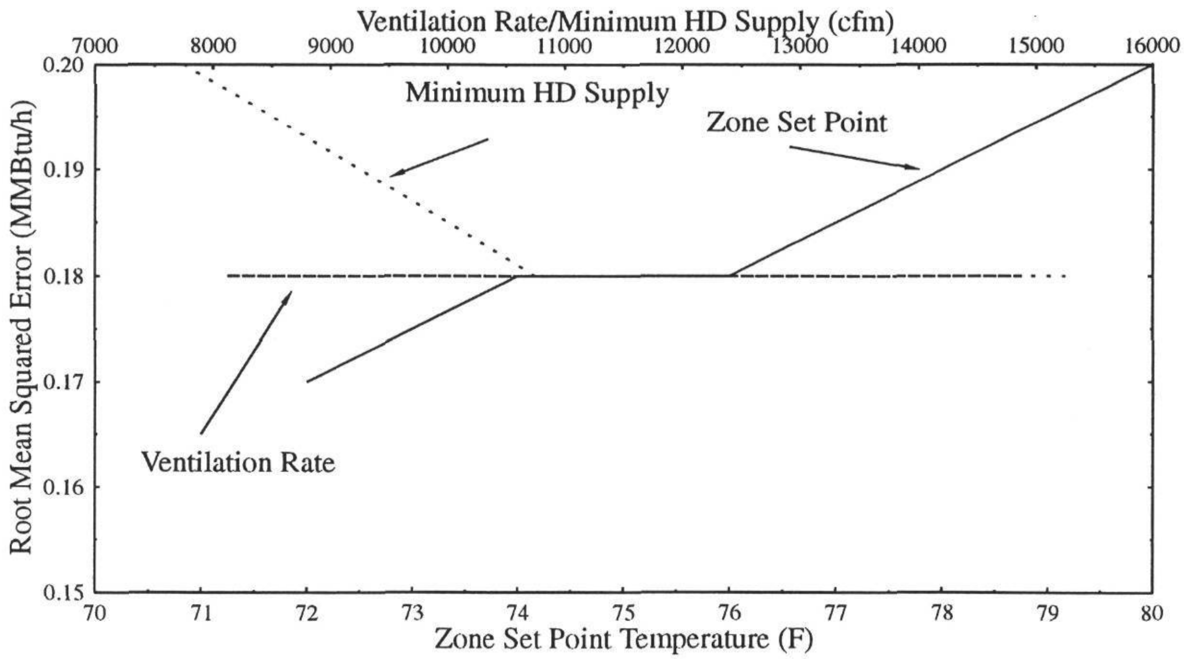

Figure 18 – Steam Consumption RMSE as a Function of Zone Set Point, Ventilation Rate, and Minimum Hot Deck Supply.

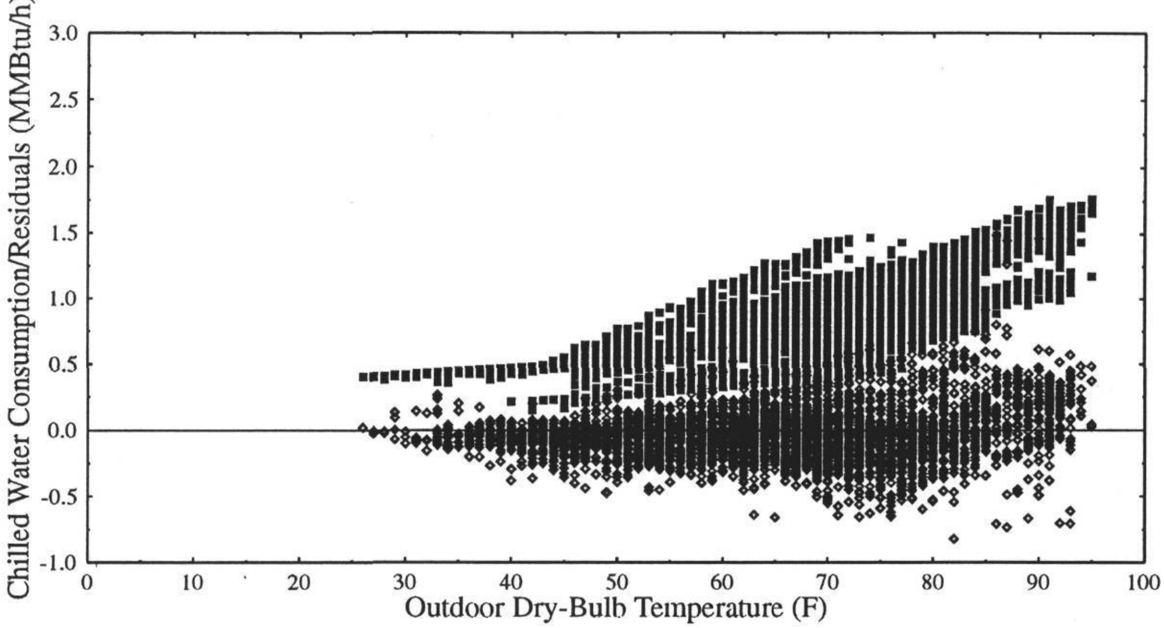

Figure 19 – Simulated Chilled Water Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (Calibrated).

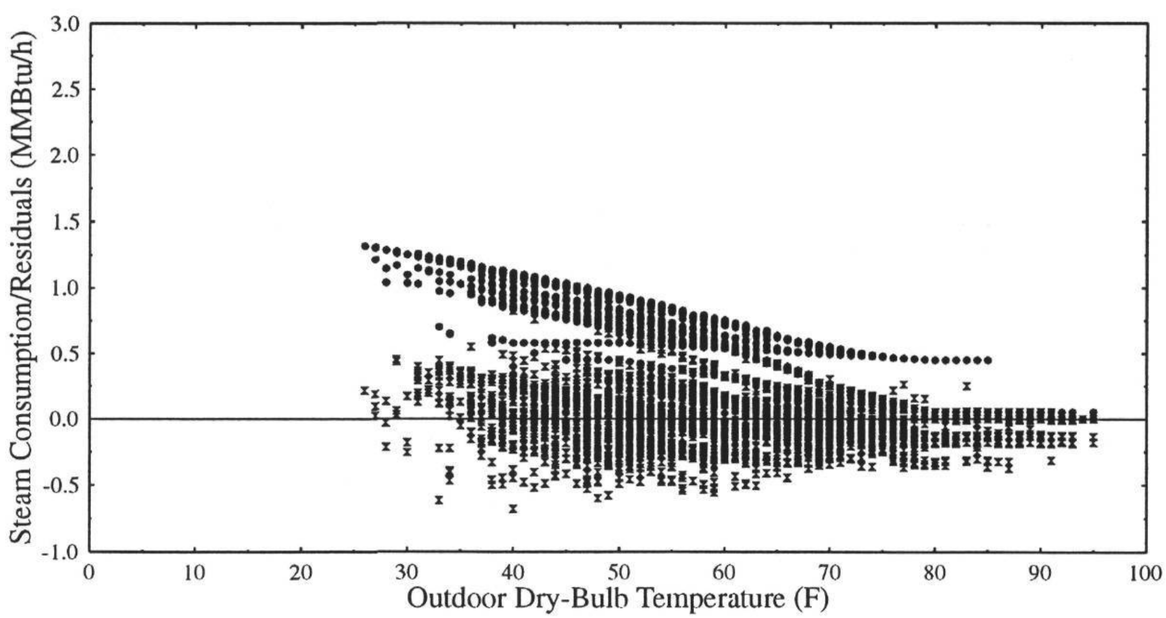

Figure 20 – Simulated Steam Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (Calibrated).

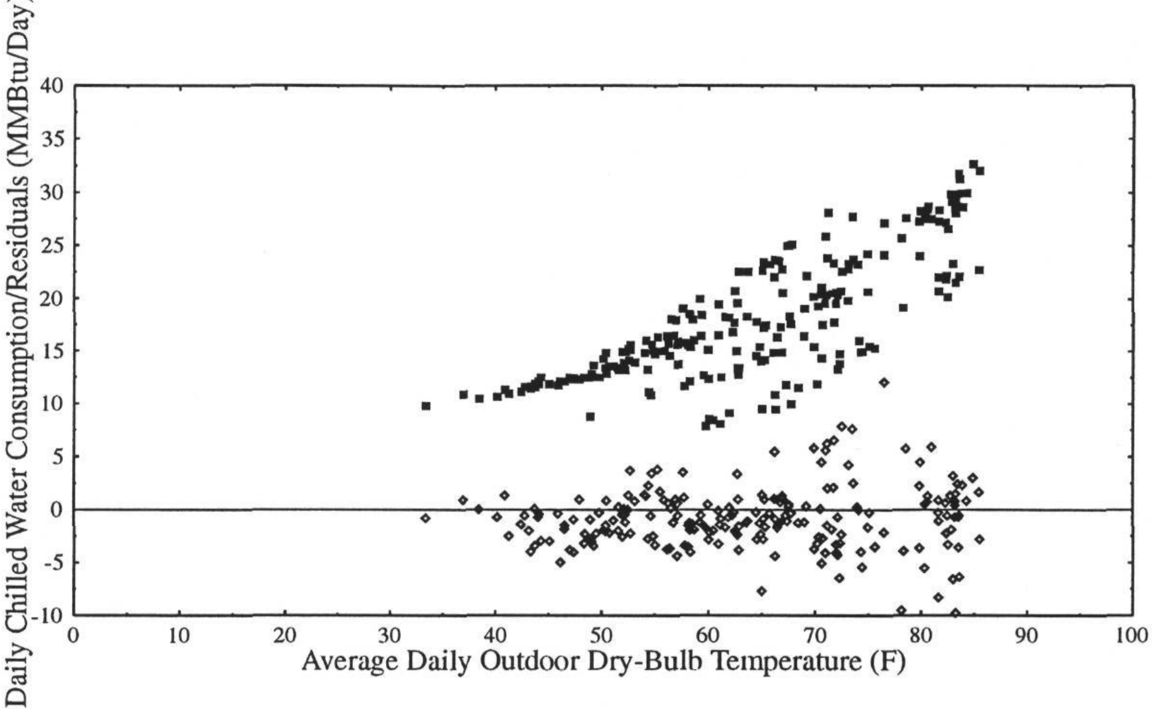

Figure 21 – Simulated Daily Chilled Water Consumption and Residuals as a Function of Average Daily Outdoor Dry-Bulb Temperature (Calibrated).

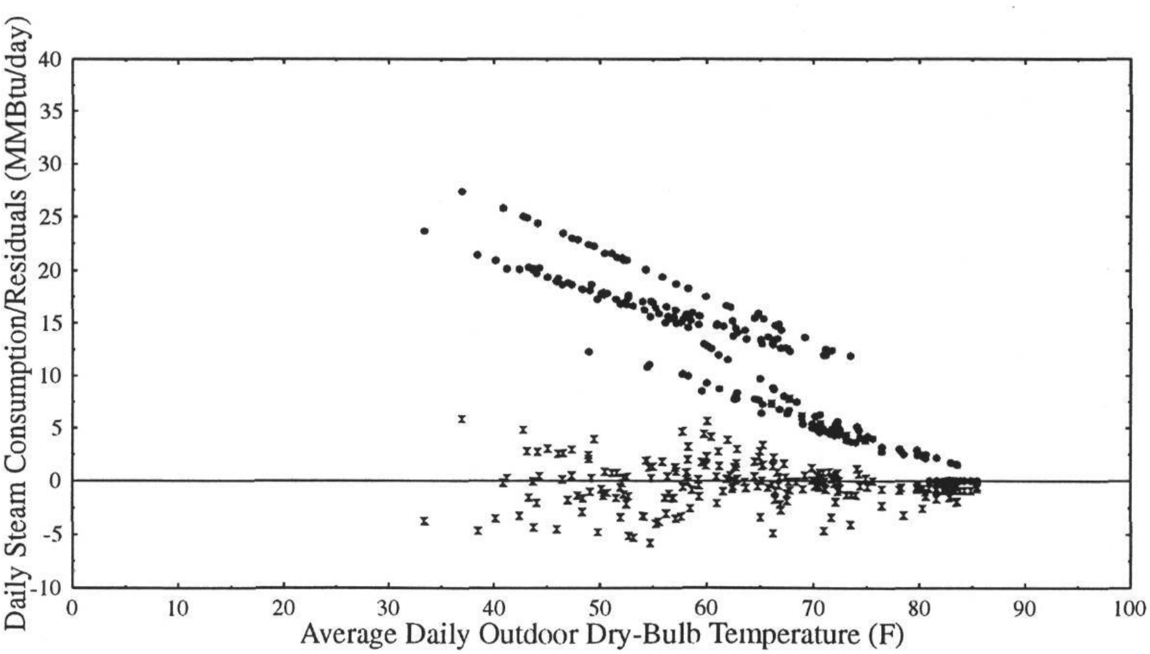

Figure 22 – Simulated Daily Steam Consumption and Residuals as a Function of Average Daily Outdoor Dry-Bulb Temperature (Calibrated).

used in the UTC.

Since there was no change in the envelope of the building it is reasonable to assume the loads on the UTC building before retrofit were same as after the retrofit. Therefore, the loads (envelope, solar, and internal) from the calibrated post-retrofit model were used in simulation of the pre-retrofit energy use for the UTC building. However, several of the system parameters for the DDCV system were needed to simulate the pre-retrofit energy use, including: (i) cold and hot deck supply temperatures, (ii) ventilation rate, and (iii) total air flow rate.

According to the UT physical plant personnel the cold deck supply temperature was fixed at 55 F and the hot deck supply temperature was fixed at 100 F all year, for the pre-retrofit of the DDCV system. The ventilation rate used in the simulation of the post-retrofit energy use was used for the simulation of pre-retrofit energy use.

The total air flow rate for the DDCV system was estimated based on 5" of static pressure in the ducts, fan efficiency of 65% and the power consumption of the fan motors (taken from the Audit reports). The total field measured kW of the eight supply fans was 90 kW. Based on these assumption the total flow for the dual-duct system was 100,000 cfm.

The inputs to the DDCV model were outdoor dry-bulb temperature, outdoor relative humidity, and decimal date. The day of the week and the hour of the day is extracted from the decimal date.

Figure 23 shows the pre-retrofit simulated chilled water consumption for the UTC building as a function of the outdoor door dry-bulb temperature. The consumption shows two regions, one that varies from 0.8 MMBtu/h at 30 F outdoor dry-bulb temperature to about 2.4 MMBtu/h at 95 F outdoor dry-bulb temperature. In the second region the consumption varies from about 0.4 MMBtu/h at 70 F outdoor dry-bulb temperature to about

1.6 MMBtu/h at 95 F outdoor dry-bulb temperature. According to the UT physical plant personnel the steam to UTC was normally cutoff during summer months (June - October). Since the hot deck temperature during the summer months dropped from 100 F to about 80 F (mixed air temperature), the chilled water consumption also decreased during this period. The measured post-retrofit consumption for the same period is shown in Figure 11. The consumption varies from 0.25 MMBtu/h at 30 F outdoor dry-bulb temperature to 1.6 MMBtu/h at 95 F outdoor dry-bulb temperature. There appear to be no saving during summer months. However, during non-summer months there appears to be significant savings due to use of variable air volume system.

Figure 24 shows the pre-retrofit simulated steam consumption for the UTC as a function of the outdoor dry-bulb temperature. The consumption varied from an high of 1.8 MMBtu/h to about 0.6 MMBtu/h. The measured post retrofit consumption for the same period is shown in Figure 15. The consumption varied from 1.4 MMBtu/h to about 0. Again the reduction is due use of variable air volume system.

CONCLUSIONS

The energy use of the UTC building was simulated with the use of a VAV model which was based on the ASHRAE TC 4.7 SEAP methodology. The model had two parts:(i) one to simulate the loads and (ii) another to simulate the HVAC systems. The parameters that were adjusted in the calibration process include: (i) cold deck and hot deck supply temperature controls for consumption after March 17, 1992, (ii) adjustments for mass effects (CLFs), (iii) zone set point temperature, (iv) ventilation rate, and (v) minimum speed of the supply fans.

Although the hourly residuals (simulated - measured) were high, the simulated daily consumption compared well with the measured daily consumption (0 - 15%). The calibrated loads were used to simulate the energy use of the UTC building with the pre-retrofit

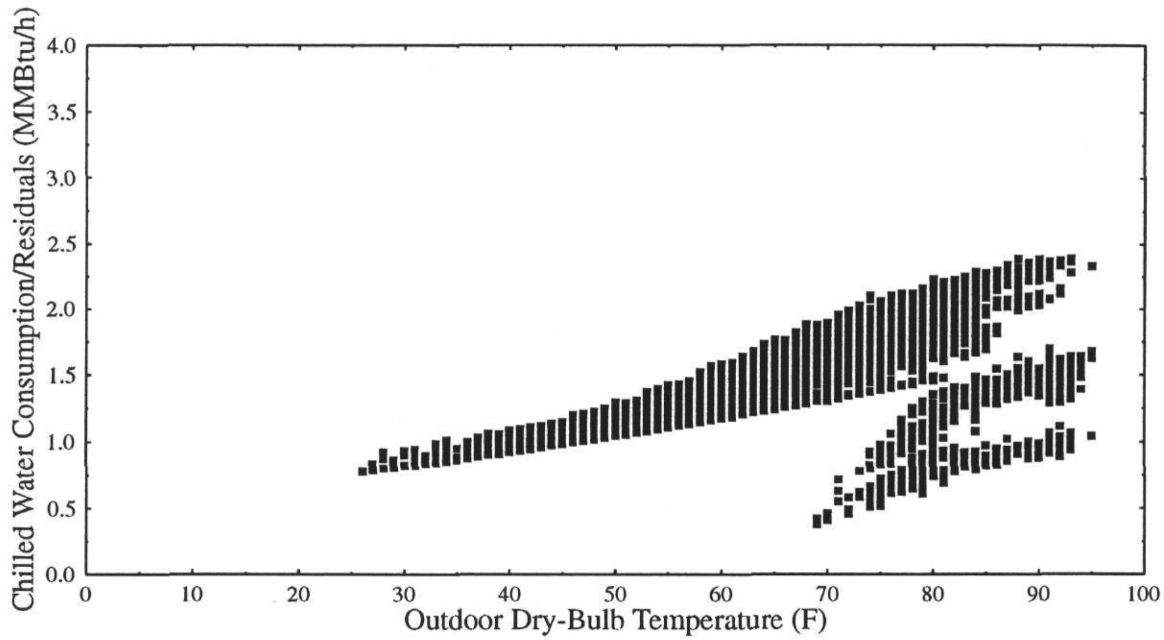

Figure 23 – Chilled Water Consumption for UTC as a Function of Outdoor Dry-Bulb Temperature With Pre-Retrofit HVAC System (October 1991 through June 1992).

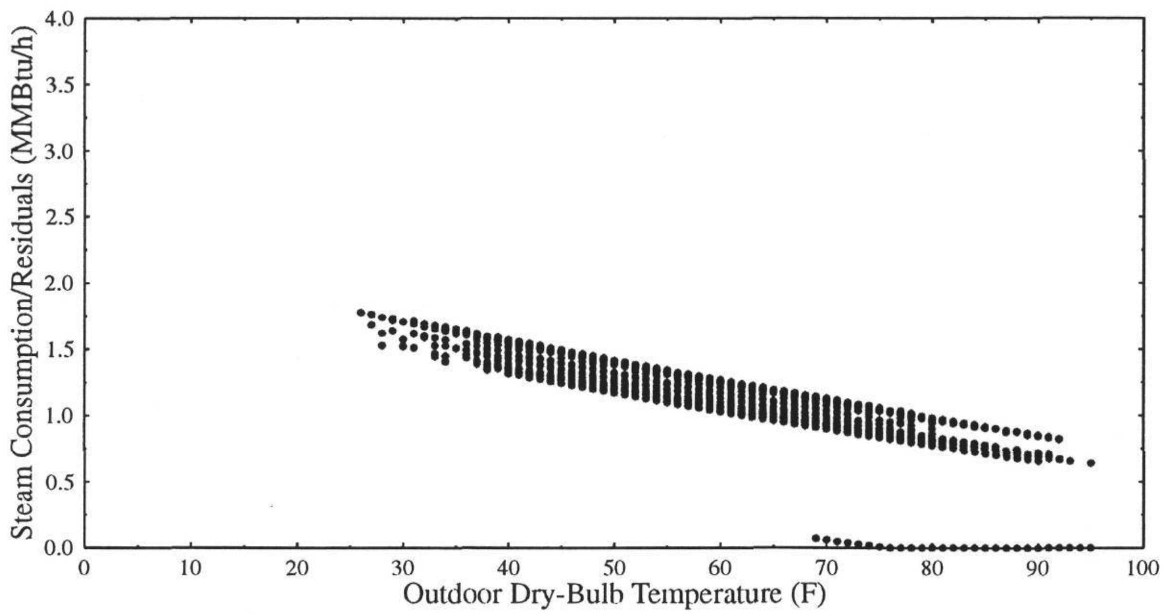

Figure 24 – Steam Consumption for UTC as a Function of Outdoor Dry-Bulb Temperature With Pre-Retrofit HVAC System (October 1991 through June 1992).

HVAC system (DDCV).

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