Simulation of the Post-Retrofit Thermal Energy Use for the Perry–Castaneda Library Building with the Use of Simplified System Models

Report

Srinivas Katipamula, P.E., Ph.D.
David E. Claridge, P.E., Ph.D.

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ENERGY SYSTEMS LABORATORY
Department of Mechanical Engineering
Texas Engineering Experiment Station
Texas A&M University

College Station, Texas 77843
(409) 845-6402, or
(409) 845-1251

INTRODUCTION

Several state owned buildings with dual-duct constant volume (DDCV) systems are being retrofitted with energy efficient variable air volume (VAV) systems as part of 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.

Unfortunately, in the Perry-Castaneda Library (PCL) building, the retrofits were completed before the monitoring instrumentation was installed. Therefore, no pre-retrofit monitored data are available for this building. Hence another method to estimate savings is needed. Such a method was developed and tested (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

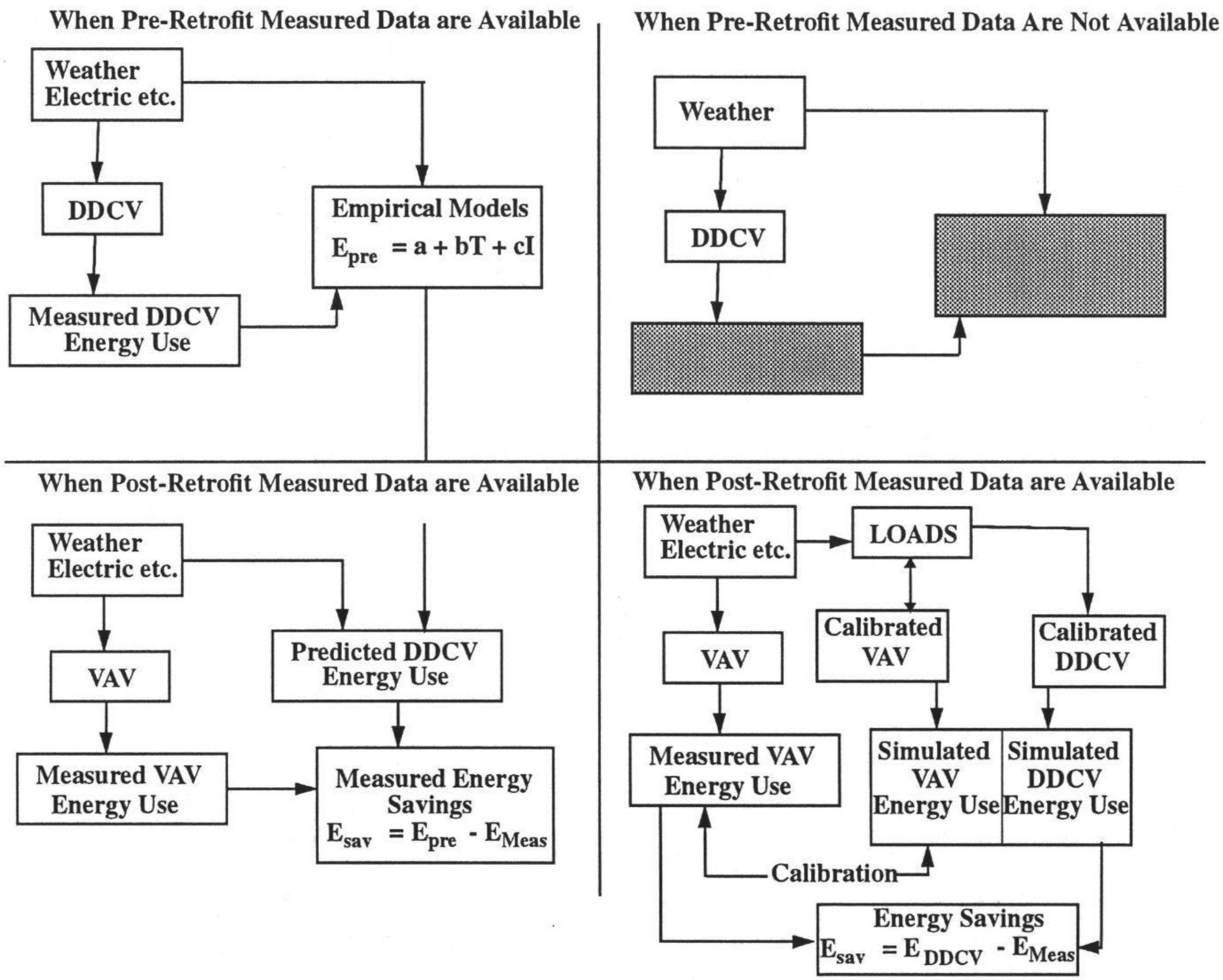

Figure 1 - Schematic of Saving Estimation Procedure.

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 PCL located on University of Texas Campus (UT) is a six-story building constructed in 1975, with a conditioned floor area of 450,000 sf. It includes offices, study rooms, and stack area. The typical weekday/weekend operating hours of the PCL are shown in Table 1. This schedule changes during exams week. The PCL is a heavy structure with six-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. In November 1990, all the DDCV systems and the single duct reheat systems were converted to variable volume systems (VAV). In addition, the dual-duct systems were converted from single fan to dual-fan systems.

Table 1 – Typical Weekday/Weekend Operating Schedule.

Day of Week	Operating Hours
Monday-Friday	8 a.m. to 10 p.m.
Saturday	9 a.m. to 10 p.m.
Sunday	12 noon to 10 p.m.

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 PCL (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 PCL) 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 PCL has four identical dual-duct, dual-fan VAV systems serving the perimeter zones, eight variable air volume single duct reheat (VAVSR) systems serving the interior zones and twelve VAV return air fans. The supply fans are rated between 25 and 60 kW and the return air fans are rated between 10 and 12 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 PCL 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, 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 PCL (Figure 4) is about 75,000 sf. The total external wall area is about 119,000 sf of which 14,000 sf is glass area and the area of the roof is about 75,000 sf.

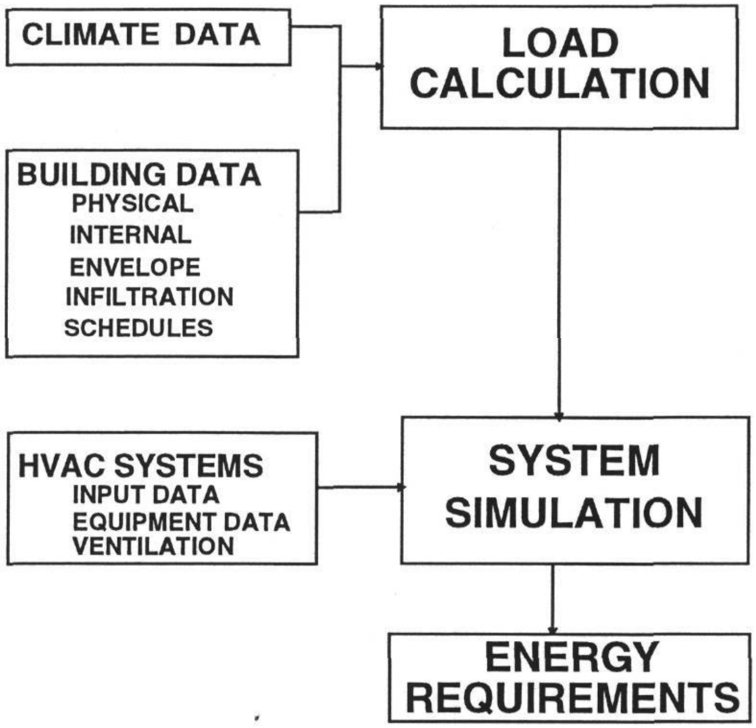

Figure 2 – Schematic of the Simulation Procedure.

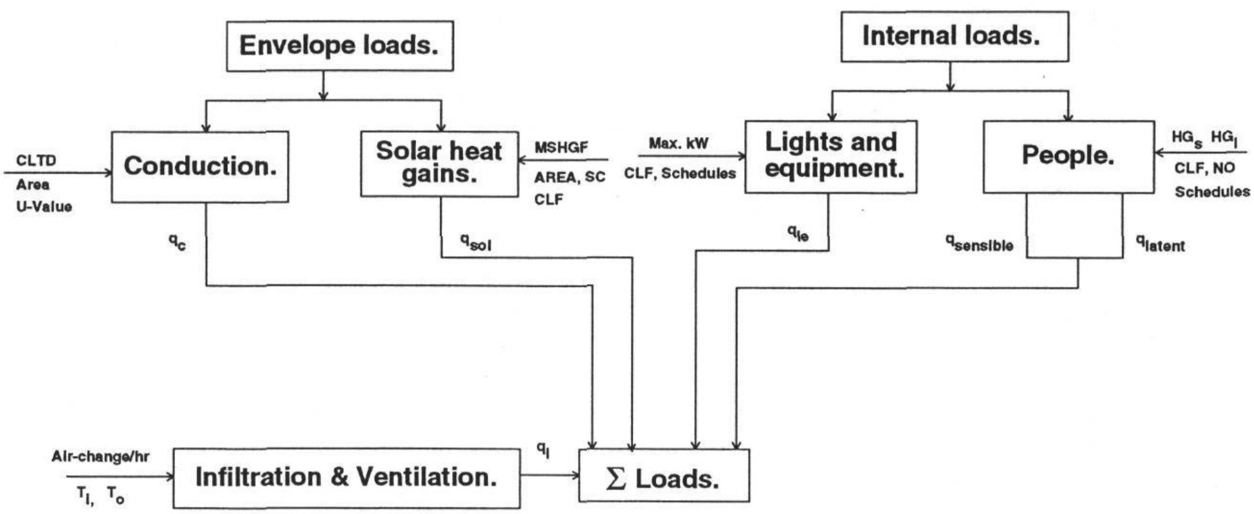

Figure 3 – Schematic of the Load Calculation.

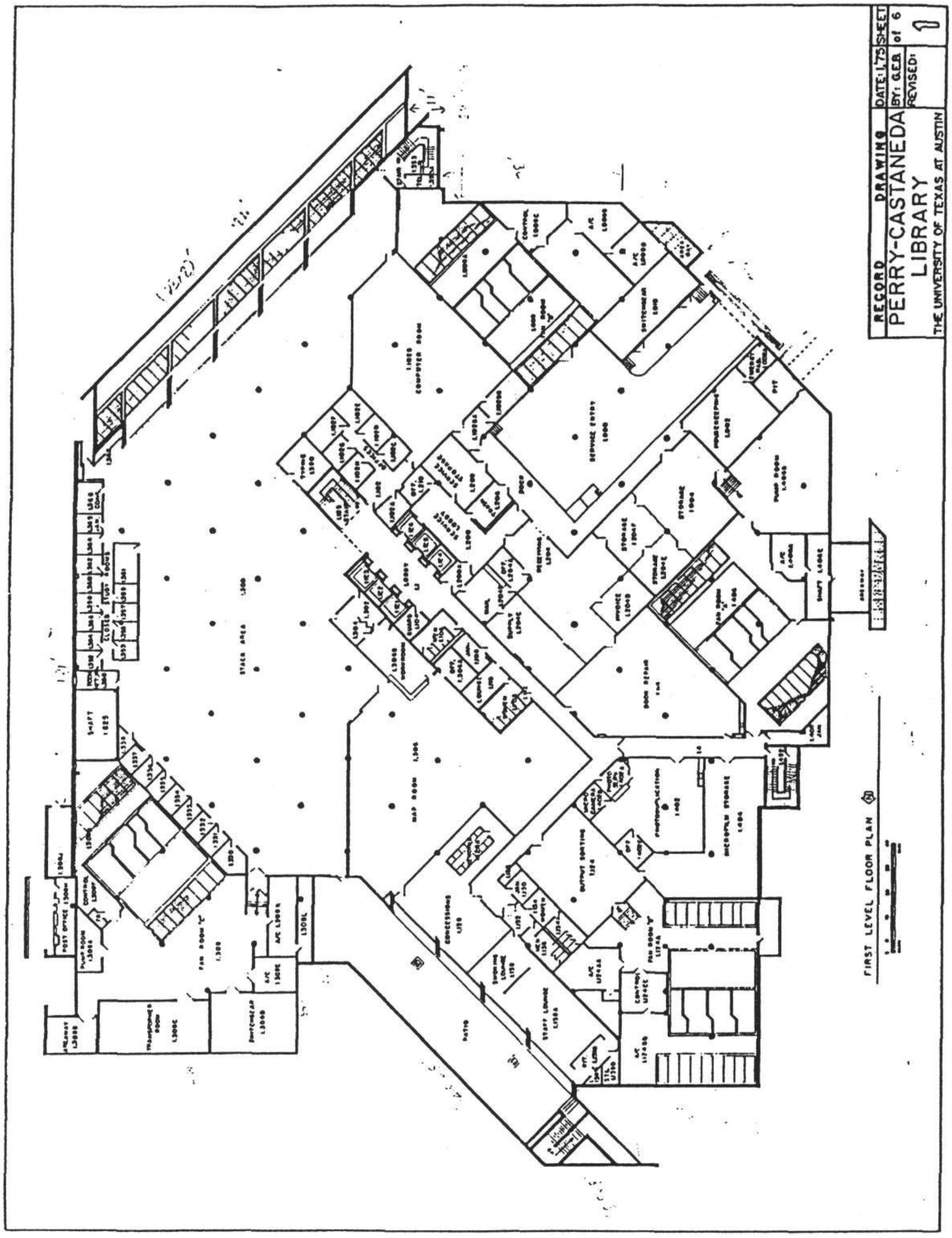

Figure 4 – Typical Floor Layout of the PCL Building.

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 PCL 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 $\frac{Btu}{hFsf}$.

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 5.42 and 2.71 $\frac{Btu}{hsf}$, respectively, based on hourly post-retrofit monitored data. Eighty-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.). The profiles for Tuesday through Friday are similar while the profiles for Monday, Saturday, and Sunday are different because of change in the operating schedules. 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 1000 compared to 400 on weekends and the rest of the weekday hours. The sensible and latent heat gain per person is assumed to

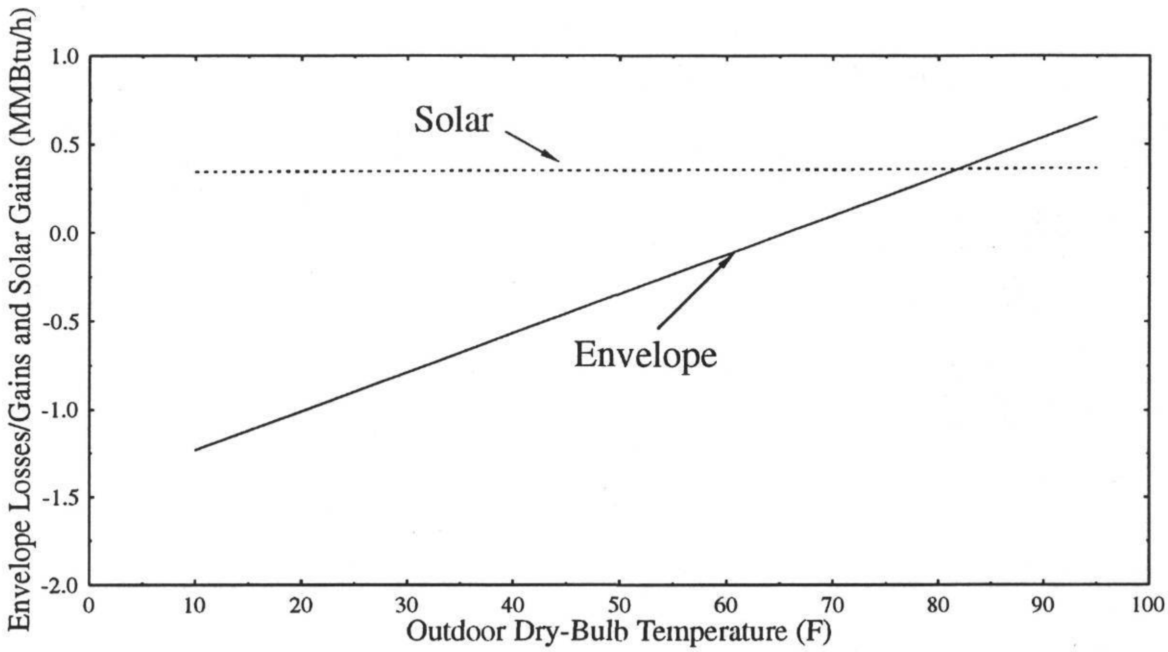

Figure 5 – Conduction Heat Losses/Gains and Solar Heat Gains as a Function of Outdoor Dry-Bulb Temperature.

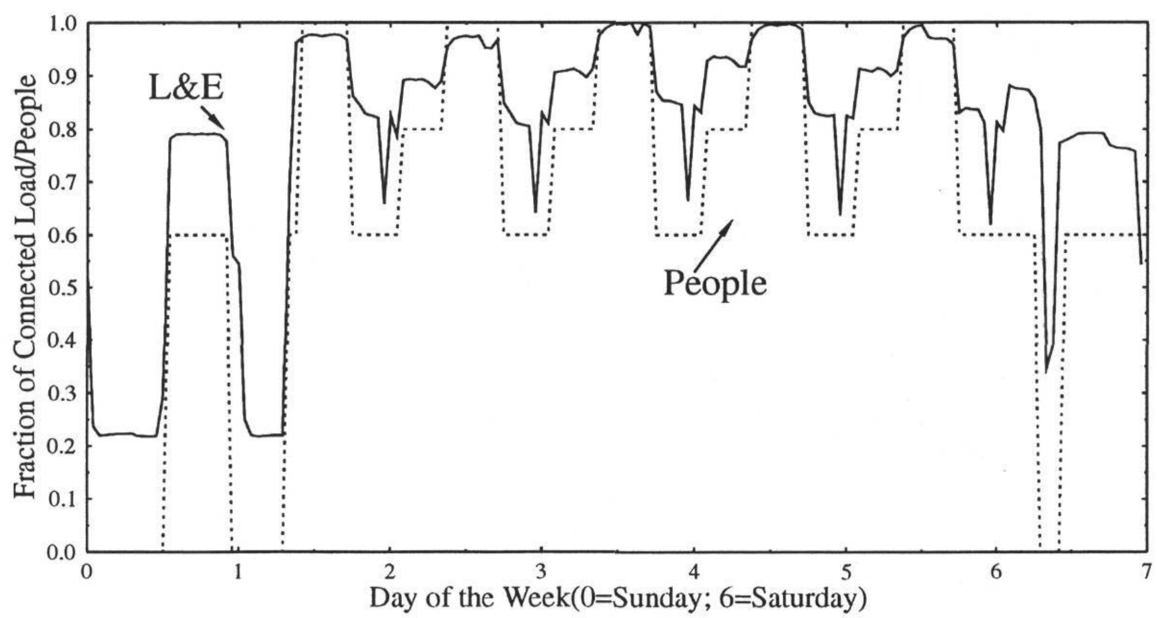

Figure 6 – People, Lighting and Equipment Schedules.

be 250 Btu each. The people schedule is shown in 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 10,000 cfm will be assumed for base case. This turns out to be 10 cfm/person at peak occupancy. The ASHRAE recommended ventilation rate is 10-15 cfm/person.

HVAC SYSTEM SIMULATION

Since the building has two different types of HVAC systems the building was divided into three zones: external, intermediate, and core. The external and the intermediate zones are served by dual-duct, dual-fan VAV system and the core zone is served by VAVSR (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 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 outdoor air is mixed with the return air and the mixture enters the cooling coil, whereas only the return air enters the heating coil. 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 load.

The existing controls show that the air leaving the cooling coil is fixed at 55 F and the air leaving the heating coil is reset based on the outside air (Table 2). At outdoor dry-bulb temperatures below 55 F the outdoor air intake is modulated so as to provide a mixed air

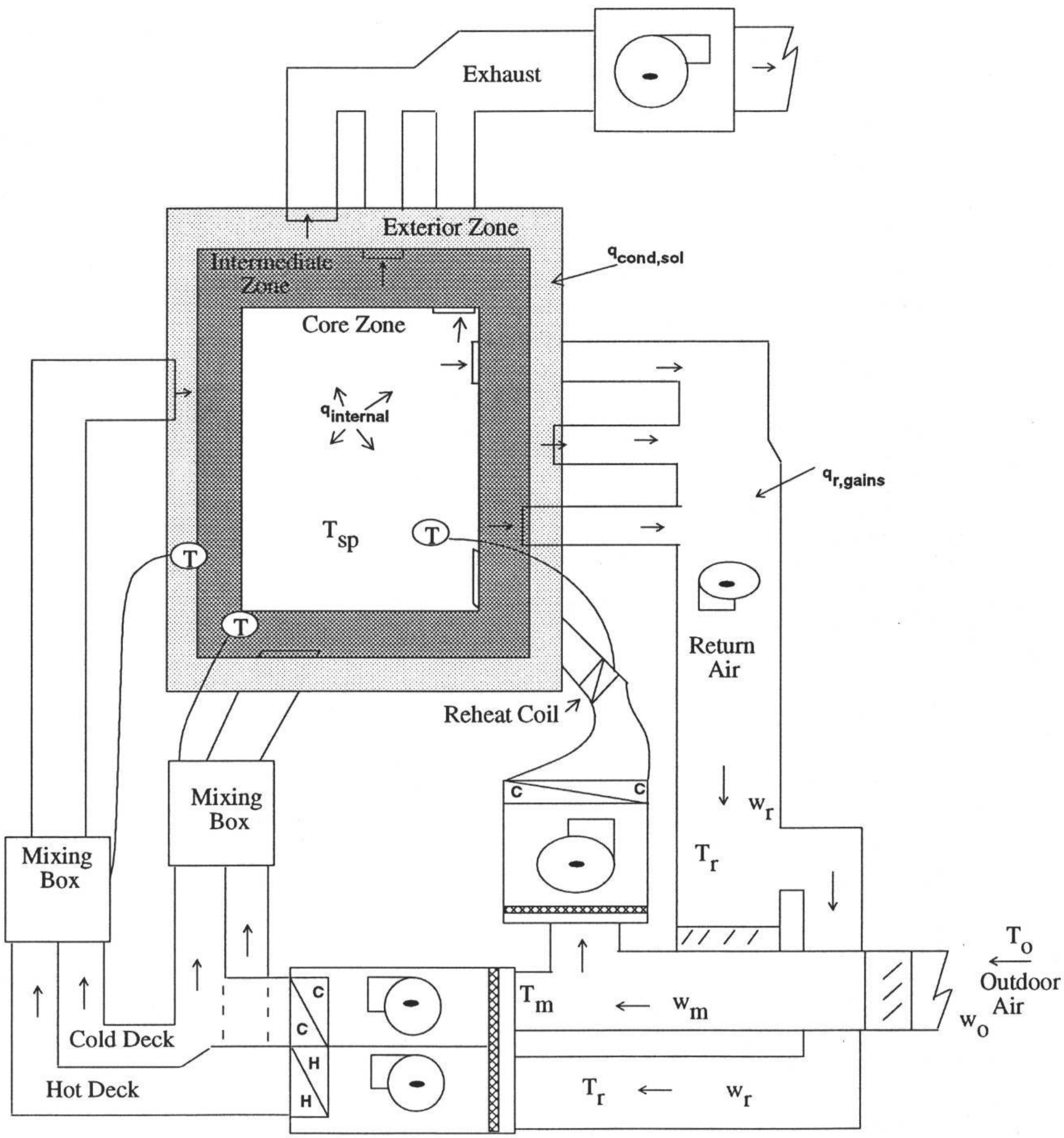

Figure 7 – Schematic of the HVAC System in the PCL.

temperature of 55 F to the cooling coil. For example, if the outdoor air temperature is 55 F the air entering the cooling coil is 100% outside air.

Table 2 – Hot Deck Reset Schedule.

Outdoor Air Temperature (F)	Hot Deck Discharge Air Temperature (F)
20	140
80	80

The VAVSR is a single duct air system consisting of a central air handling unit with a fan, cooling coil, reheat coil and terminal box. The air leaving the cooling coil passes through the reheat coils which are supplied with low pressure steam (5 psig). When the supply fan is running at the minimum speed and the zone thermostat calls for heating, the reheat coil heats the cold air to a necessary zone supply temperature to meet the zone load. However, if the supply fan is running above the minimum speed the volume of air is modulated to meet the zone load. The existing controls show that the air leaving the cooling coil is fixed at 55 F.

The perimeter zone is divided into two zones: external and intermediate with 10% and 25% of the total conditioned area, respectively. The energy use of these two zones is simulated with a dual-duct, dual-fan VAV system. The energy use of the rest of the conditioned area (65%) is simulated with a VAVSR. The intermediate and core zones are assumed to be insulated from the envelope 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 all three 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%. This is true even of the dual-duct system, 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. However, on several site visits to the PCL it was noticed that the hot deck fans were operating at 40 - 50% of full speed even when the outdoor temperature was 95 F.

For the dual-duct system, the total rated flow (at full speed) for the hot deck fans and the cold deck fans is $\approx$ 139,000 cfm each. For the single duct reheat systems the total flow is $\approx$ 250,000 cfm. 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 10,000 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.

The base case conditions are: (i) the indoor temperature set point of 78 F, (ii) a constant outdoor air intake of 10,000 cfm for outdoor dry-bulb temperatures above 55 F, and (iii) the 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 July through October 1991. Prior to July 1991 there was an excessive use of steam in the reheat coils due to malfunction in the HVAC controls.

Figure 8 shows a time series plot of the simulated chilled water consumption for the PCL (July - October 1991). For the same time period the chilled water consumption is also plotted as a function of outdoor dry-bulb temperature (Figure 9). The minimum chilled water consumption which occurs during the unoccupied hours of the weekend is between 1.0 and 1.5 MMBtu/h. The maximum consumption during the occupied hours is about 5.5 MMBtu/h. The chilled water consumption between the outdoor dry-bulb temperatures of 55 F and 70 F is almost flat. Below 55 F outdoor dry-bulb temperatures there is no chilled water consumption.

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 use for outdoor temperatures above 55 F. The maximum simulated chilled water consumption also compares well with the measured consumption (about 5.5 MMBtu/h). However, the minimum simulated consumption for outdoor temperatures above 55 F is 1.5 MMBtu/h compared to 3.5 MMBtu/h for the measured consumption. The simulated chilled water consumption below 55 F outdoor temperature is zero, whereas the measured chilled water consumption is between 2.0 to 3.5 MMBtu/h.

Since the internal heat gains from lights, equipment and people constitutes a major portion of the loads, the simulated consumption during the unoccupied hours is far less

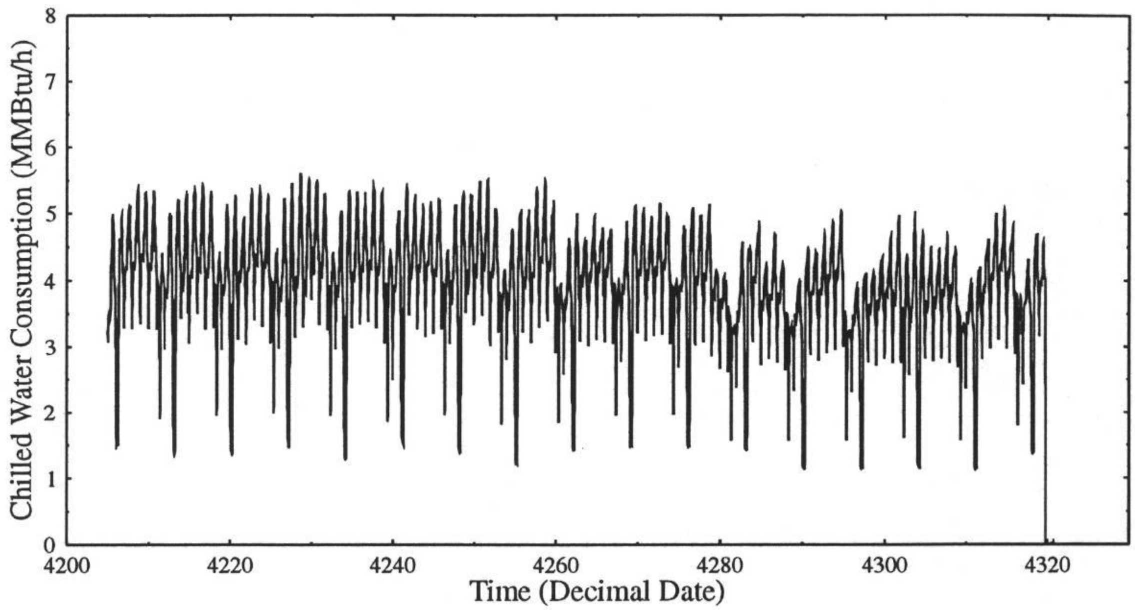

Figure 8 – Simulated Chilled Water Consumption for PCL With Minimum Fan Speed of 30% (July – October 1991).

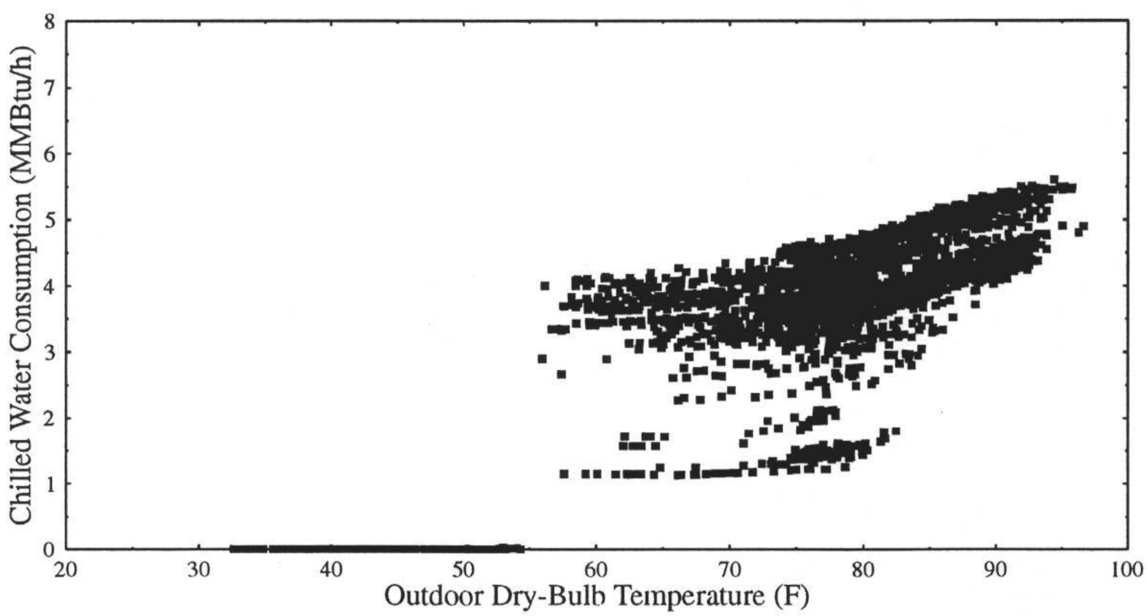

Figure 9 – Simulated Chilled Water Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature With Minimum Fan Speed of 30% (July – October 1991).

than during the occupied hours. However, the measured consumption, for outdoor dry-bulb temperatures above 55 F, does not show much difference between occupied and unoccupied hours.

Figure 12 shows the maximum, the minimum and the difference between the maximum and the minimum consumption of lights and equipment in the PCL during the least occupied periods of the weekends (Sunday midnight to Monday 6 a.m.). The maximum and the minimum consumption is constant for several weekends between July and Oct. 1991. The average difference between the maximum and the minimum consumption is about 2.7 MMBtu. If the HVAC system is functioning as a true VAV system, the difference between the maximum and the minimum chilled water consumption should correspond to the difference in internal loads. However, the average difference between maximum and minimum chilled water consumption is about 1.5 MMBtu/h (Figure 13).

The simulated steam consumption from July through October 1991 is shown in Figures 14 and 15. The measured steam consumption for the same period is shown in Figures 16 and 17. The measured steam consumption is greater than the simulated consumption. The minimum speed for the hot deck fans was set at 30% as per the design specification. However, as mentioned earlier, on several site visits to the PCL the hot deck fans were in fact running between 40 - 50% of the full speed. Therefore, the measured steam consumption is higher than the simulated.

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) minimum speed of the supply fans, (ii) adjustments for mass effects (CLFs), (iii) zone envelope loads, (iv) zone set point temperature, and (v) ventilation and infiltration rate. The calibration process is down in two stages: (i) for outdoor dry-bulb temperatures above 55 F and (ii)

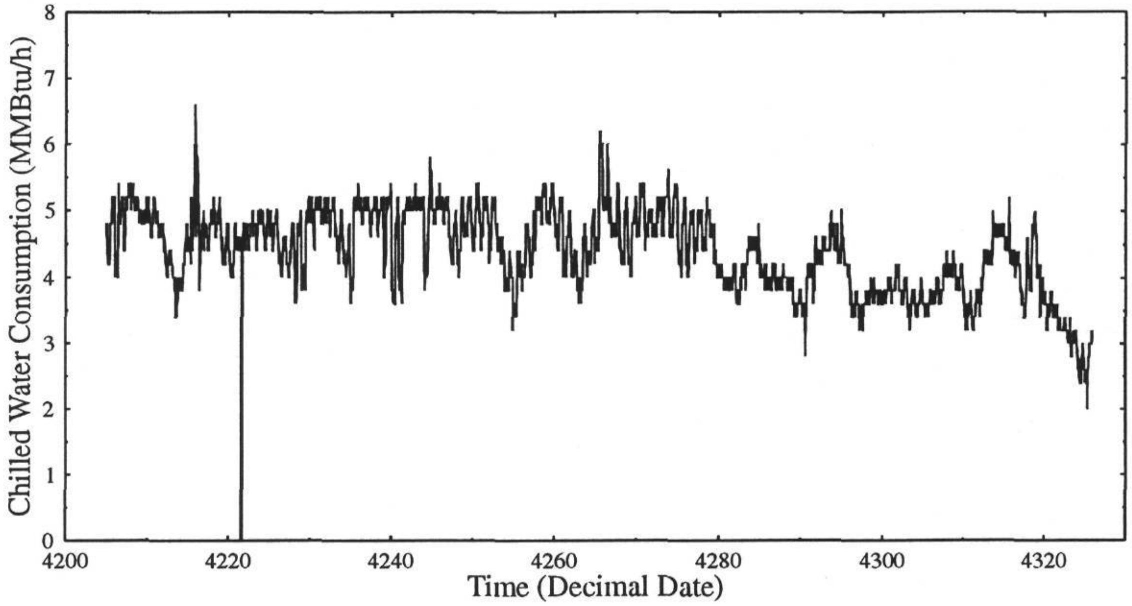

Figure 10 – Measured Chilled Water Consumption for PCL (July – October 1991).

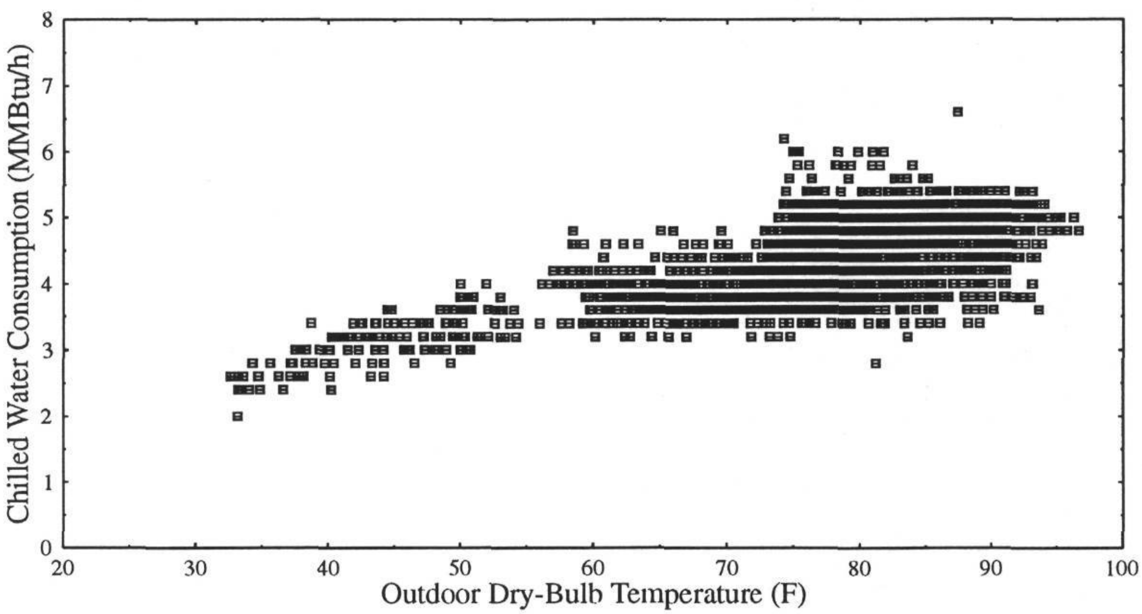

Figure 11 – Measured Chilled Water Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature (July – October 1991).

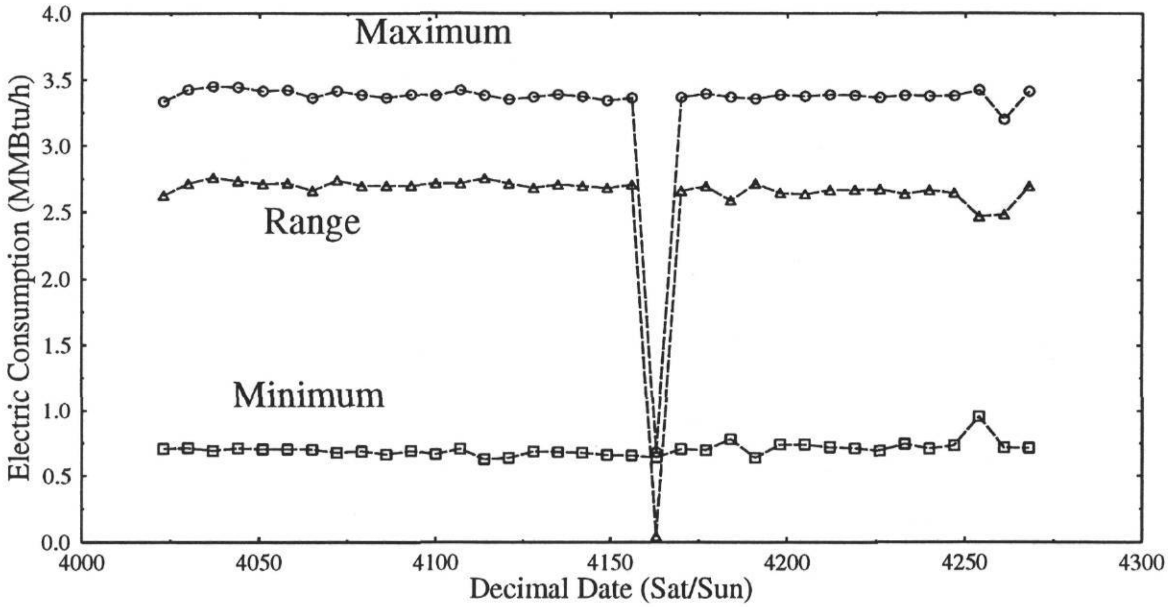

Figure 12 – Maximum, and Minimum Lights and Equipment Consumption During Weekends.

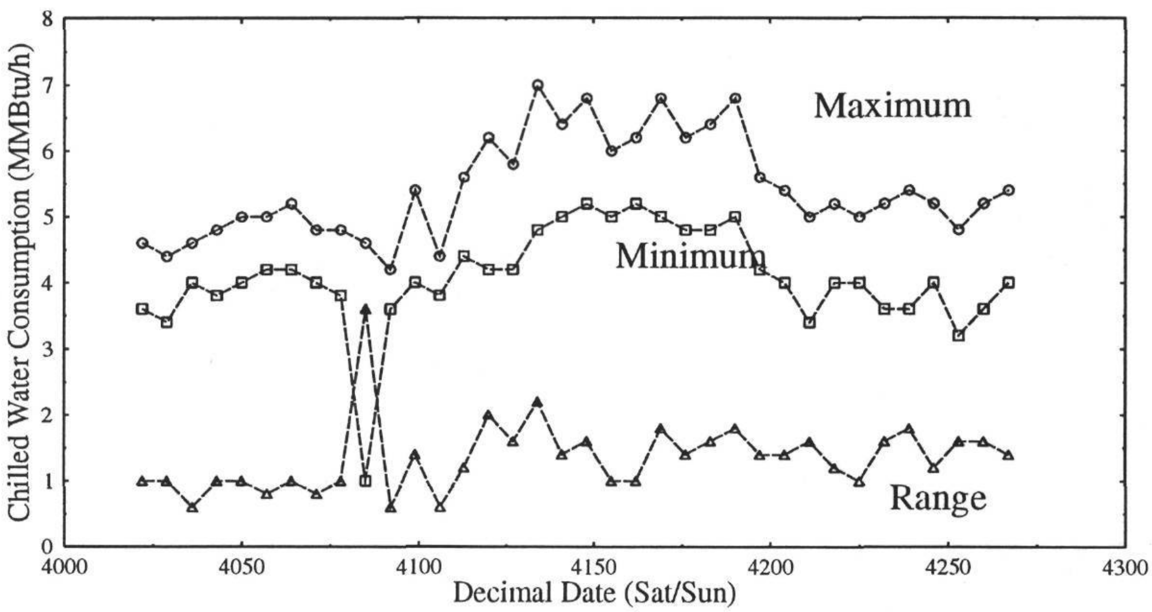

Figure 13 – Maximum, and Minimum Chilled Water Consumption During Weekends.

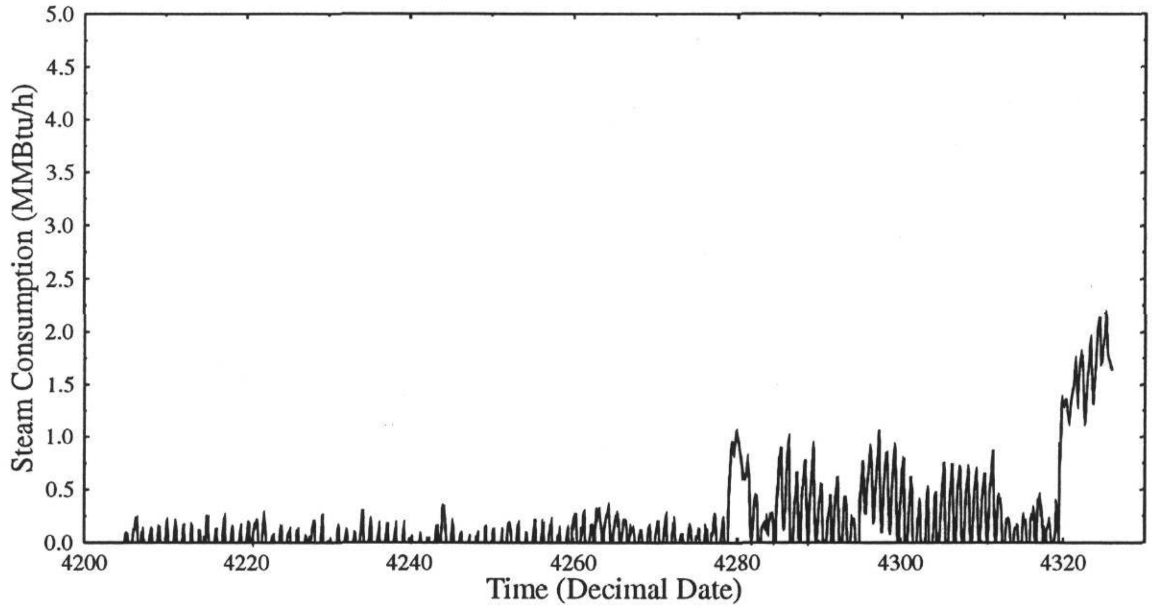

Figure 14 – Simulated Steam Consumption for PCL With Minimum Fan Speed of 30% (July - October 1991).

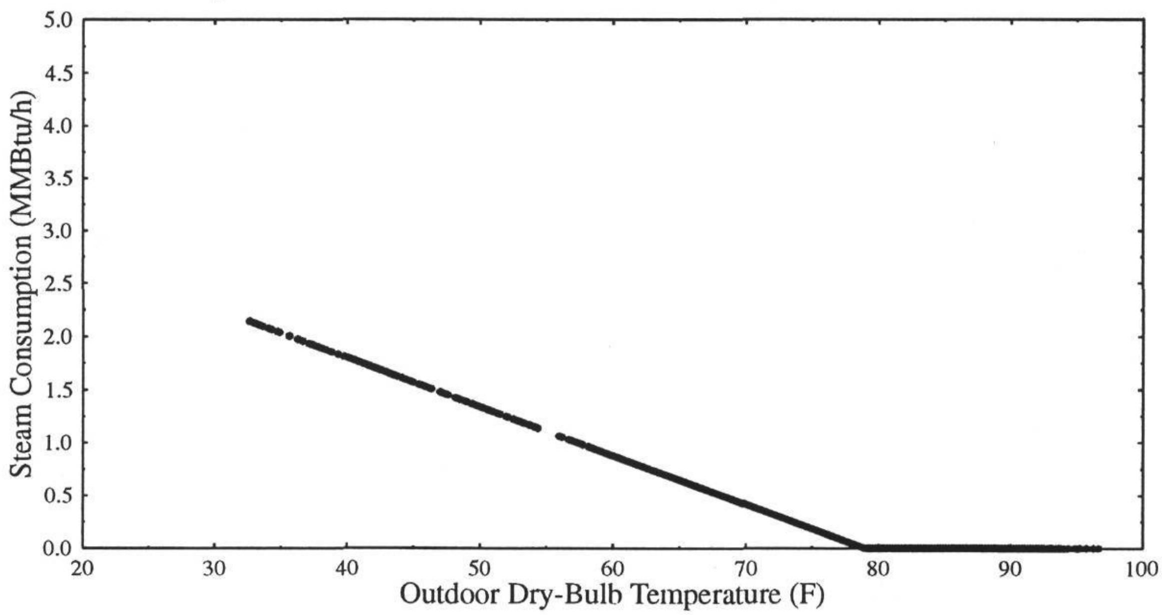

Figure 15 – Simulated Steam Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature (July - October 1991) With Minimum Fan Speed of 30%.

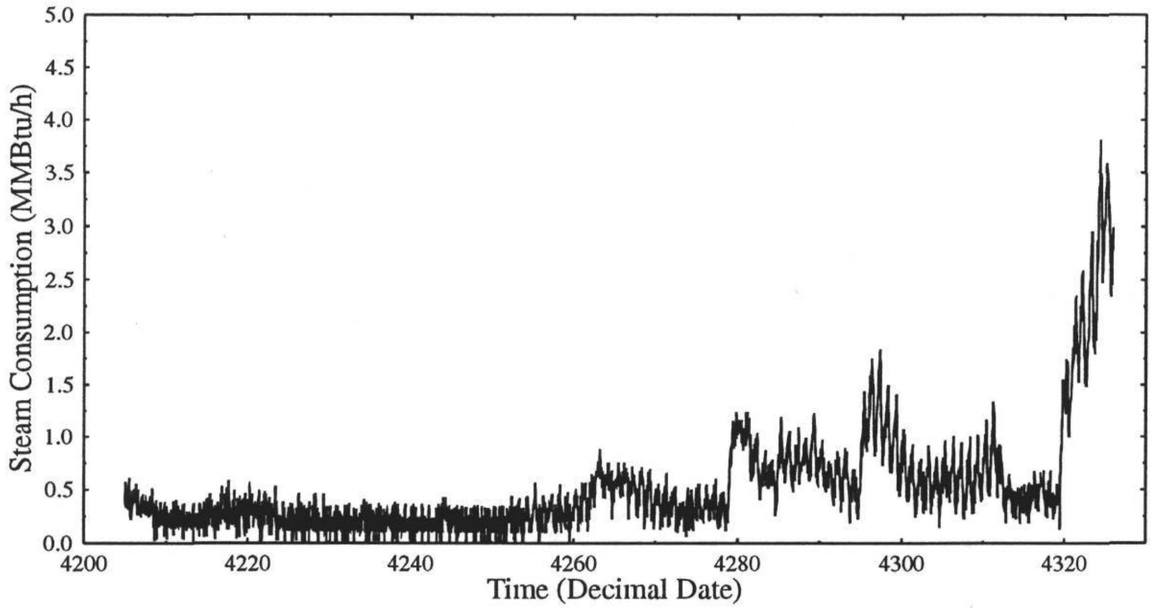

Figure 16 – Measured Steam Consumption for PCL (July - October 1991).

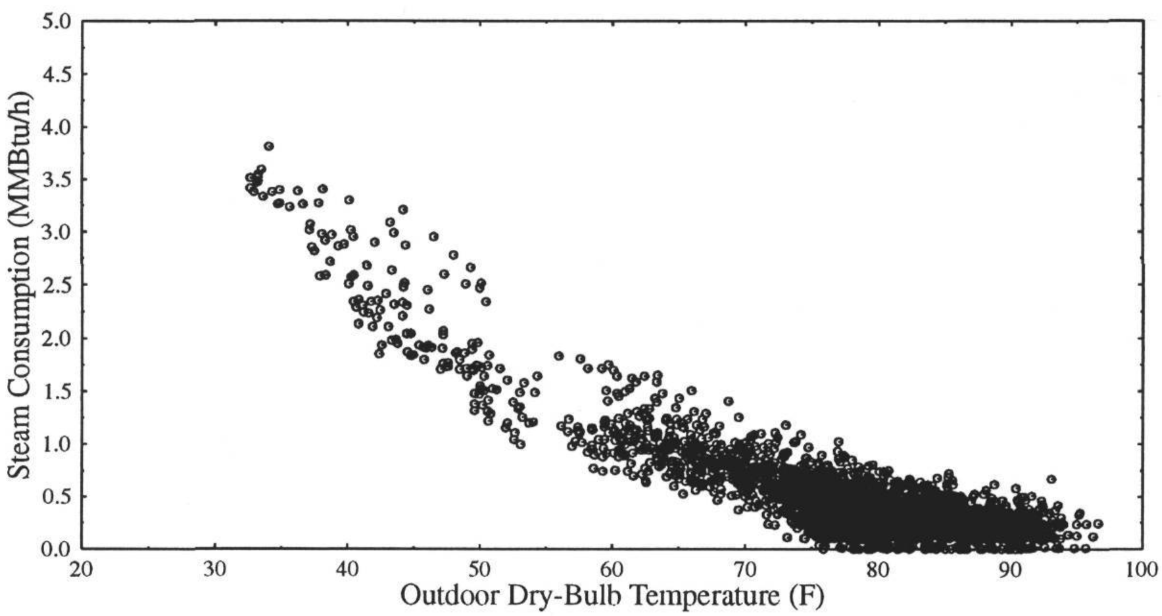

Figure 17 – Measured Steam Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature (July - October 1991).

for outdoor dry-bulb temperatures below 55 F.

Calibration of the VAV Model: O/A Above 55 F

The general trends of the base case results for outdoor dry-bulb temperatures above 55 F, discussed in the previous section, compare well with the measured chilled water and steam consumption. However, the simulated chilled water consumption during the unoccupied hours of the weekends is significantly less than the measured consumption (2.5 MMBtu/h). Also the simulated steam consumption in general is lower than the measured steam consumption. Therefore, the first step in the calibration process is to force weekday internal gain profiles on weekends and increase the minimum supply fan speed from 30% to 40% of full speed.

The simulated chilled water and the residuals (simulated - measured) are shown in Figure 18. The residuals are almost evenly distributed except around 80 F outdoor dry-bulb temperature. The simulated steam consumption and residuals are shown in Figure 19. Above 80 F outdoor temperature the simulated steam consumption is zero, whereas the measured consumption is between 0 and 0.5 MMBtu/h, which reflects the domestic hot water use (30 - 50 gallon/h). Below 80 F outdoor temperature the residuals are evenly distributed. The simulated steam consumption is linear without much scatter, whereas the measured steam consumption shows some scatter. The model simulated hot deck air flow was always less than the minimum hot deck flow (40%).

Since it is difficult to assume hourly CLF values for the internal gains, they were assumed to be one for the base case. One way of estimating the CLF is to compare the typical simulated hourly chilled water consumption with the measured. However, in the PCL the measured values of chilled water consumption were discrete (200 kBtu increments); therefore, comparison of the typical hourly chilled water consumption was not possible. This will lead to a slight increase in the magnitude of the hourly residuals.

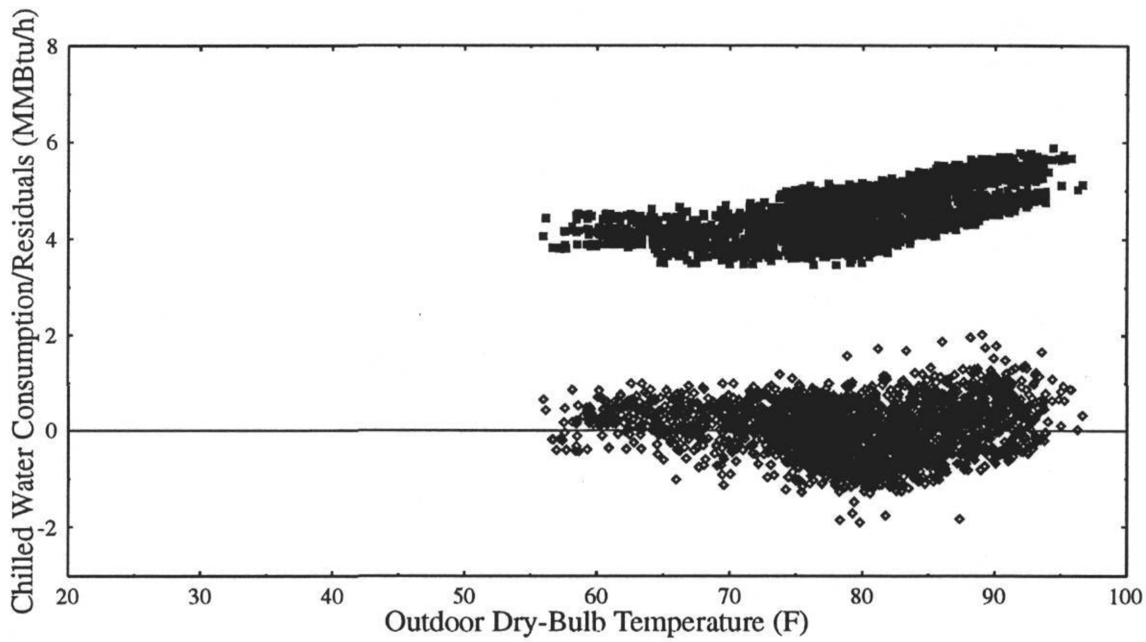

Figure 18 – Simulated Chilled Water Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (With Weekday Schedules for Weekends and Minimum Supply Fan speed at 40%: July - October 1991.)

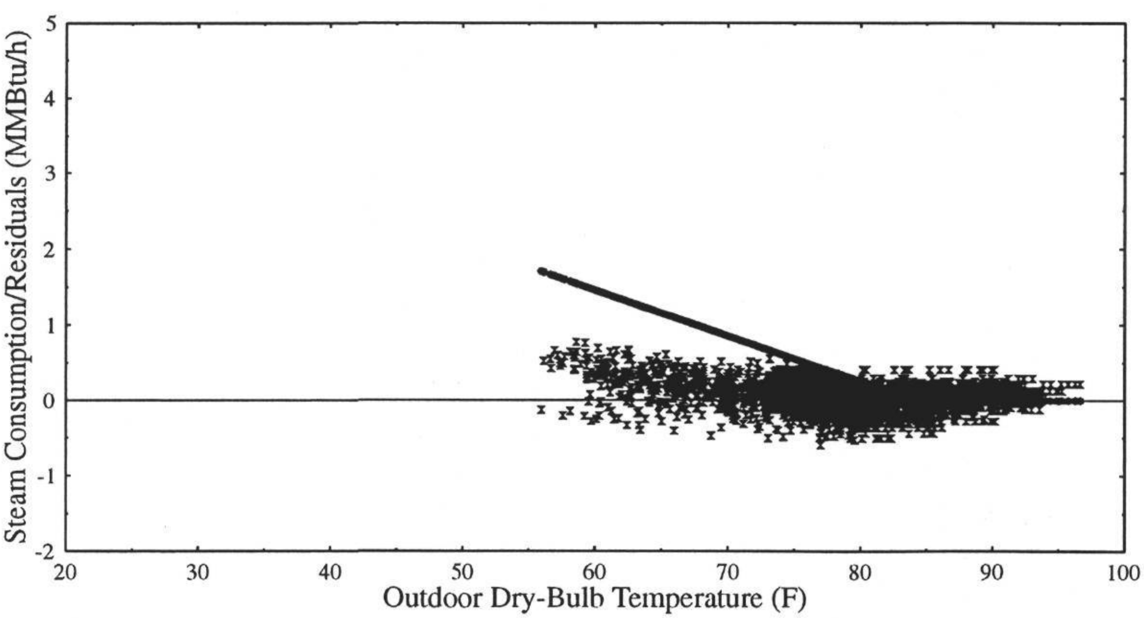

Figure 19 – Simulated Steam Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (With Weekday Schedules for Weekends and Minimum Supply Fan speed at 40%: July - October 1991.)

However, comparisons at the daily level should not have any effect.

Since the hourly residuals (Figure 18 and 19) did not show any systematic bias with outdoor dry-bulb temperature, the estimated envelope heat gains/losses are probably close to the actual. Therefore, no calibration is needed for the envelope loads. The next step in the calibration process is to determine the zone set point temperature and the ventilation rate. One way of estimating them is by minimizing the root mean square error (RMSE):

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

where n is the total number of data points. Table 3 and Figure 20 show the RMSE for various zone set point temperatures. The use of a zone temperature of 76 and outdoor airflow rate of 15,000 cfm minimized the cooling load RMSE.

Calibration of the VAV Model: O/A Below 55 F

According to the design specifications the chilled water supply to the cooling coils is cut-off when the outdoor dry-bulb temperature goes below 55 F (vent cycle). The zone cooling load is met by mixing the return and the outdoor air to get a mixed air temperature of 55 F. Therefore, at 55 F outdoor dry-bulb temperature, 100% of the air flow through the cooling coil is drawn from outside.

The measured chilled water consumption is shown in Figures 10 and 11. There is significant chilled water consumption (2.0 - 3.5 MMBtu/h) below 55 F outdoor dry-bulb temperature. Since the vent cycle is not being implemented properly in the PCL, the simulated consumption with the design specification will not match the measured consumption. The only parameter that can be adjusted to make the simulated consumption comparable with the measured is the outdoor air intake.

The simulated consumption, below 55 F outdoor dry-bulb temperature, compared

Table 3 – Comparison of RMSE Cooling Load at Various
Zone Set Point Temperatures and Outdoor Air Flow Rates.

Zone Temperature (F)/ Outdoor Air Intake cfm	RMSE (MMBtu/h)
72/10,000	0.857
73/10,000	0.743
74/10,000	0.650
75/10,000	0.584
76/10,000	0.555
77/10,000	0.566
78/10,000	0.601
79/10,000	0.650
76/5,000	0.586
76/7,500	0.568
76/10,000	0.555
76/12,500	0.548
76/15,000	0.546
76/20,000	0.556
76/40,000	0.742

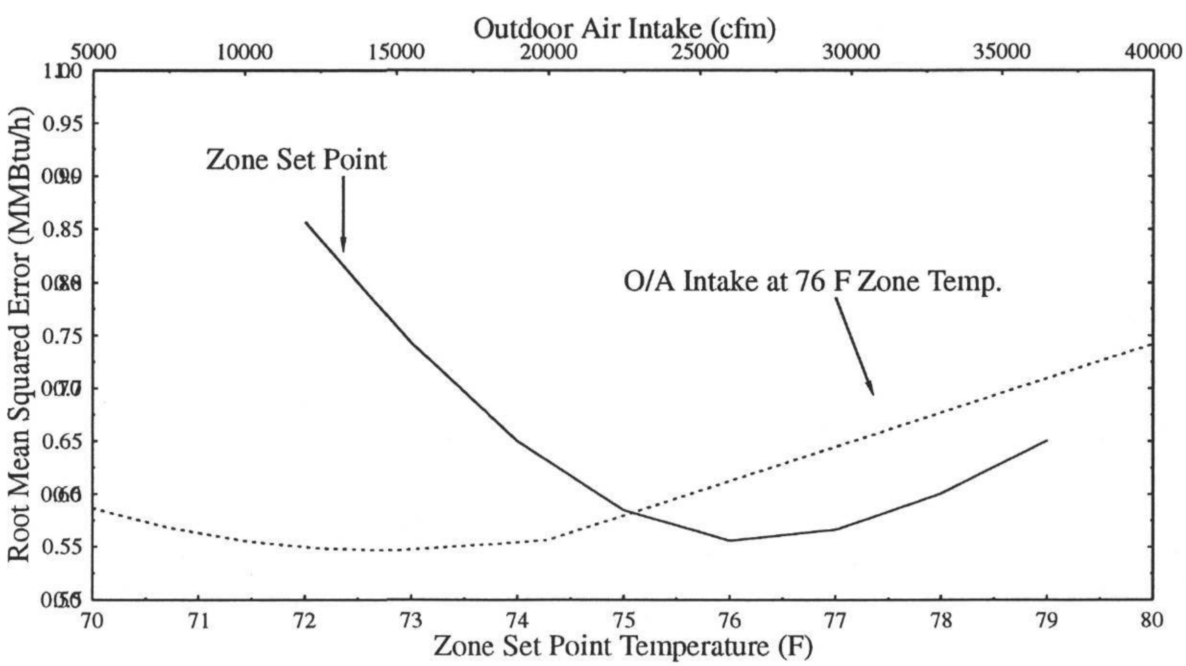

Figure 20 – RMSE as a Function of Zone Set Point (solid line) and Outdoor Air Intake (dashed line).

well with the measured when the outdoor air intake was raised from 15,000 cfm to 45,000 cfm. All the other parameters remained as described in the earlier section (O/A above 55 F).

The calibrated chilled water and steam consumption and residuals (simulated - measured) are shown in Figures 21 and 22. Although at some hours the simulated consumption is significantly different from the measured, the average hourly chilled water and steam deviation from the measured (July - October 1991) is about 0.5 MMBtu/h and 0.25 MMBtu/h, respectively. The hourly simulated values are summed up to daily and compared with the daily measured consumption in Figures 23 and 24. The simulated daily chilled water consumption compares well with the daily measured consumption ($\pm$ 10%) with an average deviation of 0.72 MMBtu/day. The simulated daily steam consumption did not compare as well as the chilled water. However, the average deviation is about 1.2 MMBtu/day.

PRE-RETROFIT ENERGY USE FOR THE PCL BUILDING

In the absence of pre-retrofit data the retrofit savings of the PCL building were estimated by predicting the pre-retrofit system behavior with the use of a calibrated simplified hourly simulation model. The PCL building (pre-retrofit) had four DDCV systems, eight single duct constant volume reheat (SDCVR) systems and 12 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 SDCVR system has an air handler unit with a fan, cooling coil and a reheat coil. The pre-retrofit controls in the PCL showed that the air leaving the cooling coil is fixed and the air leaving the heating coil (reheat coil) is reset based on the outdoor air dry-bulb temperature. The preheat coil and the economizer cycle were not used in the PCL before the retrofit.

Since there was no change in the envelope of the building it is reasonable to assume

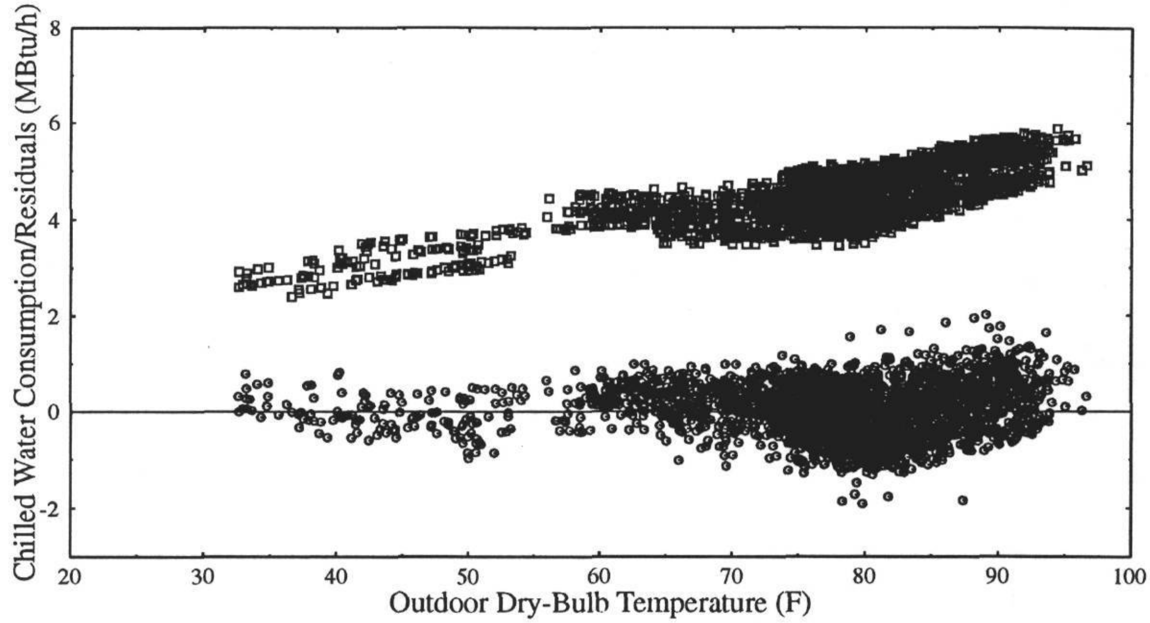

Figure 21 – Simulated Chilled Water Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (Calibrated).

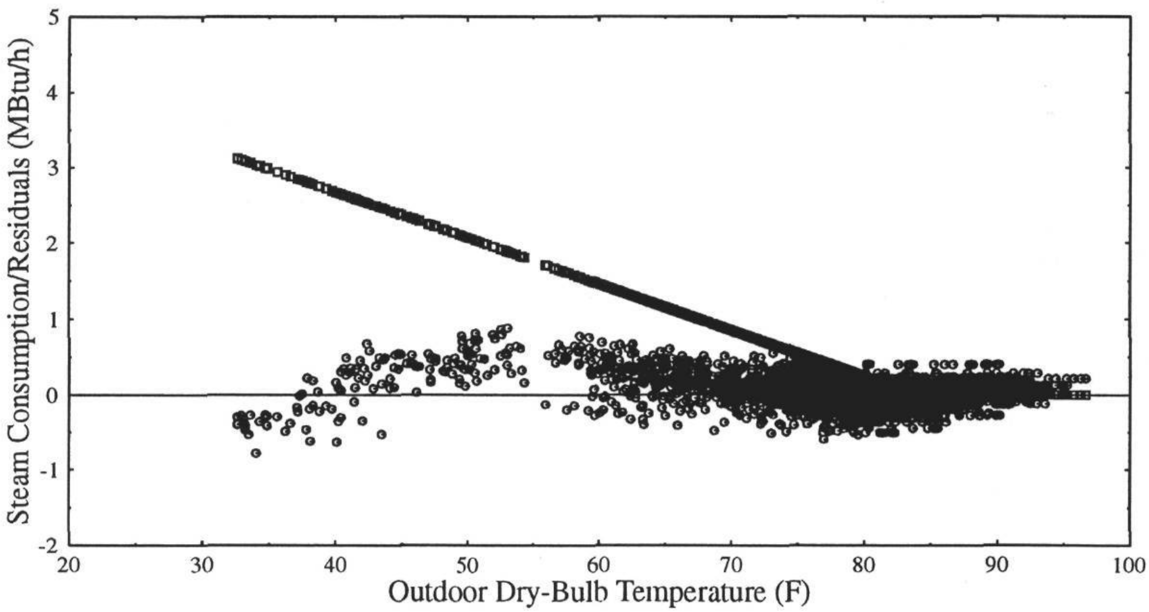

Figure 22 – Simulated Steam Consumption and Residuals as a Function of Outdoor Dry-Bulb Temperature (Calibrated).

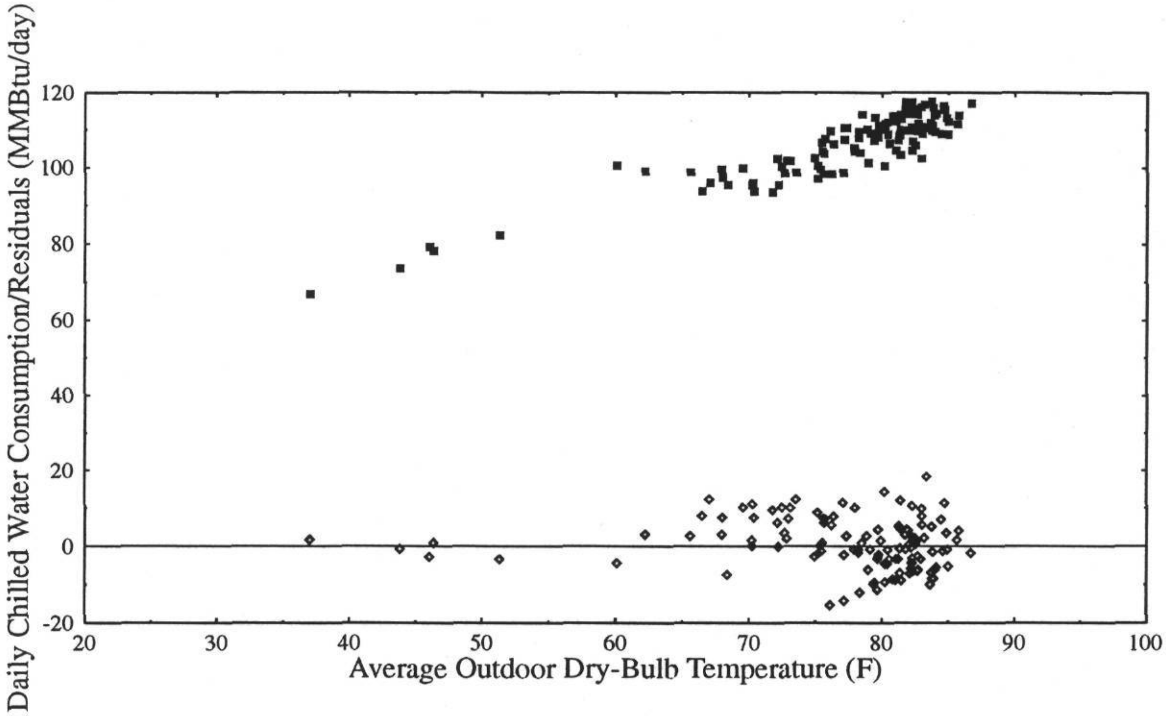

Figure 23 – Simulated Daily Chilled Water Consumption and Residuals as a Function of Average Daily Outdoor Dry-Bulb Temperature (Calibrated).

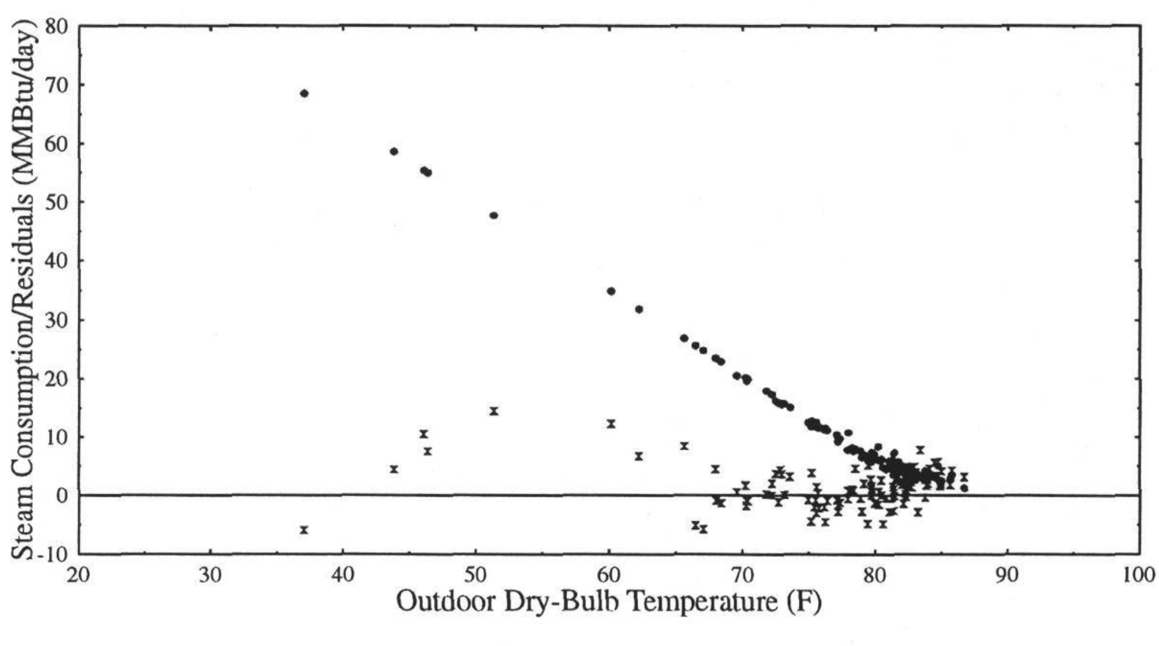

Figure 24 – Simulated Daily Steam Consumption and Residuals as a Function of Average Daily Outdoor Dry-Bulb Temperature (Calibrated).

the loads on the PCL 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 PCL building. However, several of the system parameters for the DDCV and SDCVR 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 retrofit of the DDCV system to a VAV system did not change any temperature controls in the PCL building. Therefore, the same cold and hot deck temperatures used in the simulation of the post-retrofit system were used for the pre-retrofit system. The ventilation rate used in the simulation of the post-retrofit energy use was also used in the simulation of the pre-retrofit energy use.

The field measured horsepower for the four DDCV systems was 275 hp, and for the eight SDCVR systems it was 310 hp. Assuming a total pressure of about 5" of water in the ducts and total efficiency of 0.65, the total air flow rate is assumed to be 225,000 cfm for the DDCV system and 255,000 cfm for the SDCVR system. The total flow rate is 480,000 cfm, which is little over 1 cfm per square feet of conditioned area.

The inputs to the DDCV model were the 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 external and the intermediate zones are conditioned by the DDCV systems and the core is conditioned by the SDCVR systems.

Figure 25 shows the pre-retrofit simulated chilled water consumption for the PCL as a function of the outdoor dry-bulb temperature. The bottom scatter plot is for the external and intermediate zones (DDCV), the middle scatter plot is for the core zone (SDCVR) and the top is the total. Both the DDCV and SDCVR system consumptions show only a slight variation with the outdoor air temperature. The variation is mainly due to fresh air intake.

The measured post retrofit consumption for the same period is shown in Figure 11. The consumption varies from 6 MMBtu/h to about 2 MMBtu/h. The consumption drops below 60 F outdoor dry-bulb temperature because of the partial use of the economizer cycle. The peak consumption in the pre-retrofit mode was about 8.5 MMBtu/h compared to 6 MMBtu/h in the post-retrofit mode. The reduction is due to two reasons: (i) the use of variable air volume system and (ii) turning off the reheat valves in the single duct systems.

Figure 26 shows the pre-retrofit simulated steam consumption for the PCL as a function of the outdoor dry-bulb temperature. The bottom scatter plot is for the external and intermediate zones (DDCV), the middle scatter plot is for the core zone (SDCVR) and the top is the total. The change in the consumption pattern at 80 F outdoor for the DDCV and the total is due to the change in hot deck reset temperature. The total consumption varied from a high of 6 MMBtu/h to about 3 MMBtu/h due to the outdoor air intake and envelope losses. The measured post retrofit consumption for the same period is shown in Figure 17. The consumption varied from 4 MMBtu/h to about 0. Again the reduction is due to two reasons: (i) the use of variable air volume system and (ii) turning off the reheat valves in the single duct systems.

CONCLUSIONS

The energy use of the PCL 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. Since the PCL was supposed to employ a vent cycle (temperature based economizer cycle) below outdoor dry-bulb temperatures of 55 F, the model was calibrated in two steps: (A) for outdoor dry-bulb temperatures above 55 F and (B) for outdoor dry-bulb temperatures below 55 F. The parameters that were adjusted in the calibration process (A) include: (i) zone set point temperature (78 F to 76 F), (ii) minimum supply fan speed (30% to 40%)

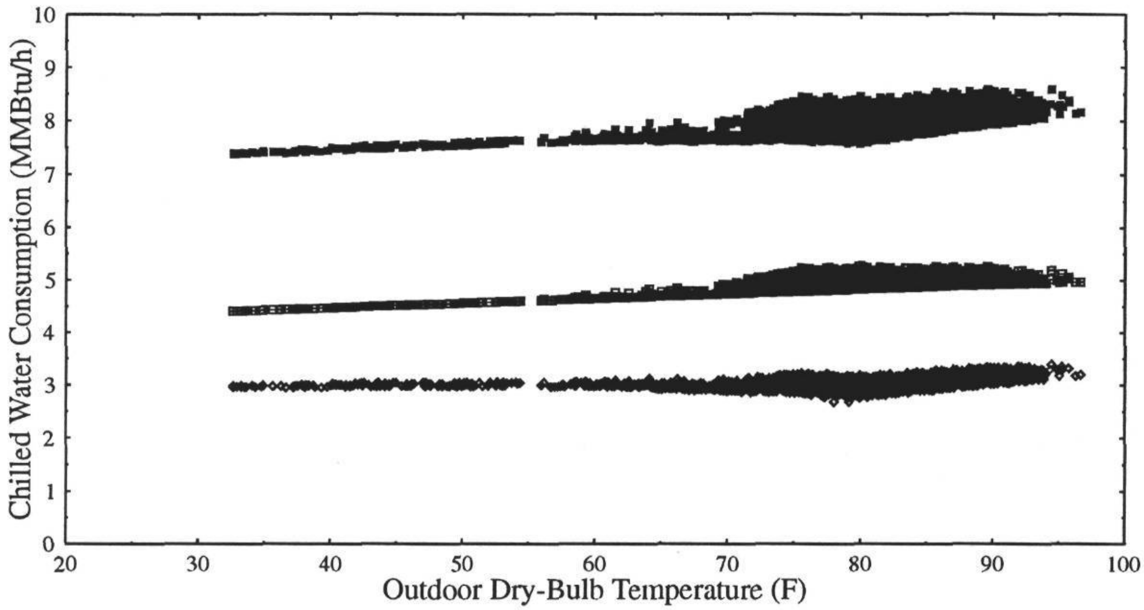

Figure 25 – Chilled Water Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature With Pre-Retrofit HVAC System (July through October 1991).

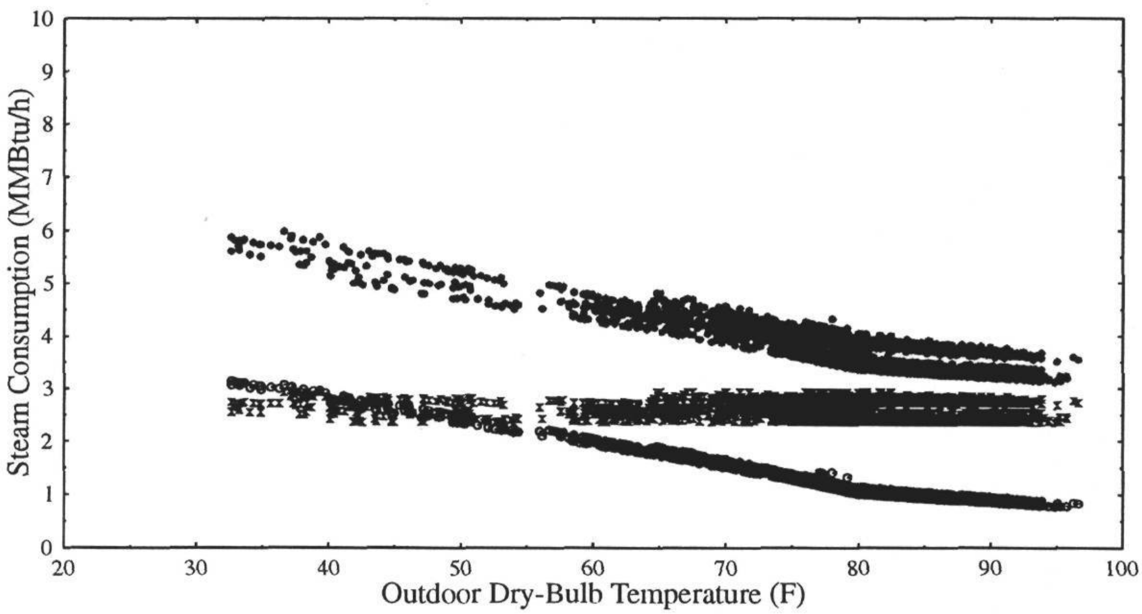

Figure 26 – Steam Consumption for PCL as a Function of Outdoor Dry-Bulb Temperature With Pre-Retrofit HVAC System (July through October 1991).

and (iii) ventilation rate (10,000 cfm to 15,000 cfm). The calibration process (B) involved increasing the ventilation rate from 15,000 cfm to 45,000 cfm below 55 F outdoor dry-bulb temperature.

Although the hourly residuals (simulated - measured) varied from 0 - 20%, the simulated daily consumption compared well with the measured daily consumption (0-10%). One of the possible reasons for the high variation in the hourly residuals was the use of CLF value of one. One method of estimating the actual CLFs is by comparing the typical hourly profile with the simulated and the measured. The typical hourly simulated profile could not be compared with the typical hourly measured profile because the measured profile was discrete (in steps of 200 kBtu/h).

The calibrated loads were used to simulate the energy use of the PCL building with the pre-retrofit HVAC system (DDCV).

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