Field Investigation of Duct System Performance in California Light Commercial Buildings

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Synopsis

This paper discusses field measurements of duct system performance in fifteen systems located in eight northern California buildings.

Abstract

Light commercial buildings, one- and two-story with package roof-top HVAC units, make up approximately 50% of the non-residential building stock in the U.S. Despite this fact little is known about the performance of these package roof-top units and their associated ductwork. These simple systems use similar duct materials and construction techniques as residential systems (which are known to be quite leaky). This paper discusses a study to characterize the buildings, quantify the duct leakage, and analyze the performance of the ductwork in these types of buildings.

The study tested fifteen systems in eight different buildings located in northern California. All of these buildings had the ducts located in the cavity between the drop ceiling and the roof deck. In 50% of these buildings, this cavity was functionally outside both the building's air and thermal barriers. The effective leakage area of the ducts in this study was approximately 2.6 times that in residential buildings.

This paper looks at the thermal analysis of the ducts, from the viewpoint of efficiency and thermal comfort. This includes the length of a cycle, and whether the fan is always on or if it cycles with the cooling equipment. 66% of the systems had frequent on cycles of less than 10 minutes, resulting in non-steady-state operation.

1. Introduction

Light commercial buildings, primarily one- and two-story buildings with individual HVAC package roof-top units serving floor areas less than 10,000 ft^2, make up a significant portion (50%) of non-residential building stock in the U.S. and California. Commercial retail strip-malls are among the largest percentage of light commercial buildings. This stock also consists of offices, restaurants and professional buildings.

First-cost dominates construction practices in these buildings. This potentially leads to short-cuts in construction practices and/or using lower grade materials. Often resulting in buildings which appear visually distressed five to ten years after they are built; moisture damage due to leaky roofs, and uncontrolled infiltration being the most common visual indicators of problems. The buildings use package roof-top units for HVAC, and as with the buildings, if not to a greater degree, first-cost dominates, with the same potential problems of poor construction practice and/or lower grade materials.

Slowly the industry, and research community, is acknowledging that duct-work in residential HVAC systems leak, sometimes by a very large amount. Roof-top units in commercial buildings use the same duct-work and installation techniques as residential systems (combinations of sheet-metal, duct-board, and flex-duct). Considering construction standards and practices, it would be a surprise if ducts in small commercial systems did not leak. The industry acknowledges that the ducts “may” leak, but since, in commercial buildings, the ducts are largely inside the building, there has been little interest in their performance, and in quantifying the extent of and the impact of duct leakage. While the ductwork may be physically inside the building, inside the ceiling cavity, this cavity is often outside the building’s thermal and air barrier, thus ducts in many light-commercial buildings are subject to the same loss mechanisms as residential ducts located in attics.

1.1 Other Work

Researchers have recently documented the leakage characteristics of residential ducts. This study uses data obtained at LBNL for various studies (Jump, et al., 1996). Other than anecdotal evidence, the only significant work in the area of small commercial systems is from the Florida Solar Energy Center (FSEC). FSEC looked at the entire building envelope in a study titled “Uncontrolled Air Flow in Non-Residential Buildings” (Cummings, et. al., 1996). Their primary concern was with uncontrolled flow across the building envelope, and they did envelope leakage studies in 70 light-commercial buildings. Since ducts often dominate building leakage, they also performed duct leakage measurements in 43 of these buildings.

1.2 Goals

The goals for this current study fell in three basic areas: characterization of the building and HVAC systems, measurement of duct leakage area, and measurement system and register flows. Characterization involved identifying HVAC unit sizes, occupied areas, and the location of the thermal and air barriers. The goals of duct leakage information were measurement of fan and register flows, along with direct leakage area measurements.

The goals for this current study fell in three basic areas: building and HVAC system characterization, duct leakage, and duct thermal losses. Characterization involved identifying unit sizes, occupied areas, the location of the thermal and air barriers, and the system-fan and register flows. Duct leakage came from direct leakage area measurements. Single-day temperature monitoring yielded information on thermal losses.

2. Methods

Building selection consisted of buildings with package roof-top cooling systems, whose owners/occupants were willing to cooperate with the study. All of the buildings in this study were occupied, which meant working around the schedules of the occupants. This required the tests to be as non-obtrusive as possible, and consisted of three distinct parts: walk-through characterization, leakage and flow measurements, and thermal measurements.

2.1 Buildings

There were eight buildings involved in the current study, three of which were LBNL office spaces. The remainder were: a Stockton area office building, an office space located in a Sacramento area industrial park, a shoe repair store located in a Sacramento area strip-mall, a health food store in Marin county, and a Marin county gymnastics facility. In total, we tested a total of fifteen HVAC systems in these eight buildings.

2.2 Walk-Through Information

A simple walk-through with the occupants yielded most of the characterization information. Major items of importance were the name plate information on the HVAC equipment, duct material and location, building thermal barrier, and building air barrier. Other items such as occupancy schedules, internal loads, etc. were obtained by filling out a questionnaire with the building occupants. Appendix A contains the questionnaire and the protocol for this study.

2.3 Flow Measurements

Fan-flow measurements were measured with the tracer-gas method outlined by Delp, et al. (Delp, et al. 1996). Due to the restrictions of working in an occupied building, register-air flows were measured using a flow hood only.

2.4 Leakage Measurements

This study measured effective leakage areas using a modified duct pressurization method, as shown in Appendix B. The method uses a single set-up to measure the combined leakage area of both the supply and return duct systems. By using the HVAC unit as a flow meter, it is possible to determine the breakdown of return and supply leakage. Leakage area calculations are proportional to the flow into the system. Therefore, uncertainties in leakage area are proportional to the uncertainties in measuring the flow into the system. Since the uncertainties in measuring the flow are within 5%, the same range applies to the leakage are values.

2.5 Thermal Measurements

This study used small, battery-operated self-contained temperature loggers for all the thermal measurements. These loggers have a resolution and accuracy of approximately 0.2^{O}C, and store 1,800 data points. A collection interval of 12 seconds allows six hours of data storage. The loggers have a delayed start feature, allowing them to be left in place to start simultaneously at a pre-determined date. We collected the following temperatures: outside air, ceiling cavity, room, supply plenum, and at least one supply register.

3. Results

Results are presented in four primary sections: building and HVAC characteristics, duct leakage area, thermal issues, and occupant interactions.

3.1 Building and HVAC Characterization

Figure 1 shows the floor area versus the unit size, for both the LBNL and the FSEC data sets. Since the FSEC data set is larger, whenever the appropriate data is available, we use it for comparison. The important point here is the floor area served by each unit. This figure shows that the California (LBNL) buildings are similar to those in Florida (FSEC). Light-commercial buildings frequently have a greater load density (ton/ft^2) than single-family residential homes, due to internal loads such as equipment, lights, and people. Unfortunately with most light-commercial buildings accurate load information is not available during design, and contractors/engineers resort to a rule-of-thumb approach to equipment selection, often resulting in oversized equipment. It is worth noting the values in the figures are installed capacities, and do not necessarily correspond to actual space loads.

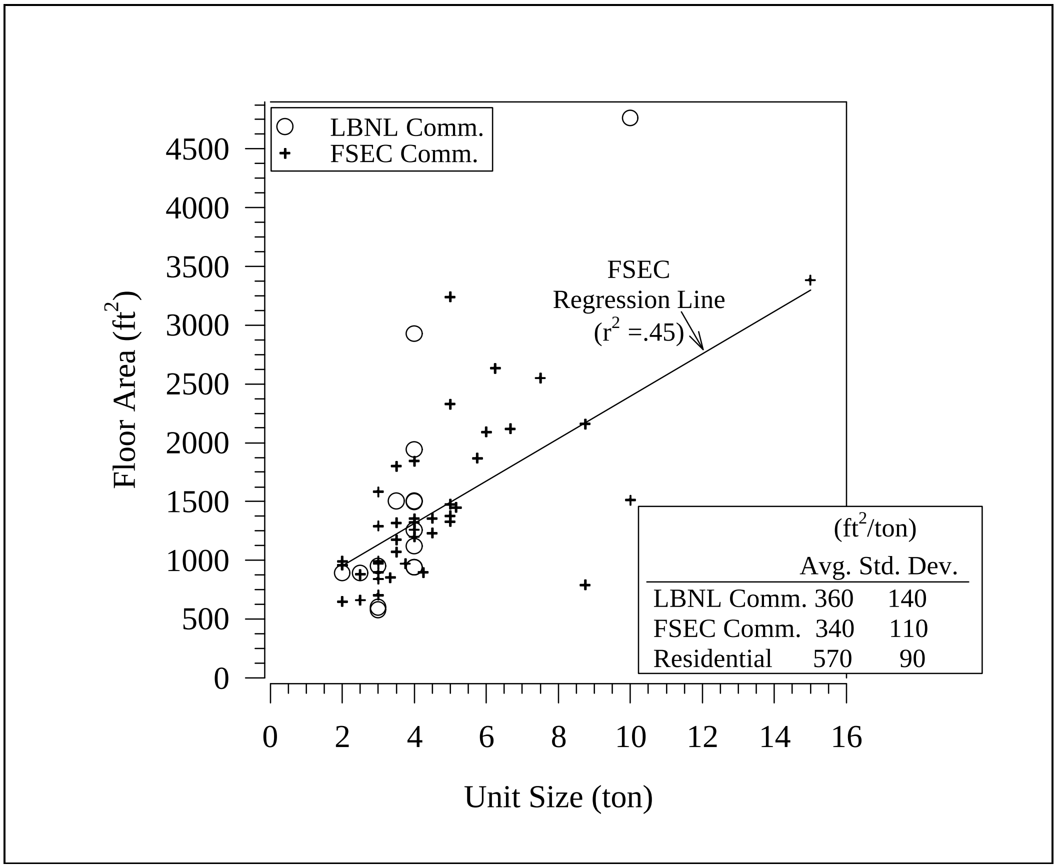

Figure 1. Floor area -vs- unit size: using the current LBNL and FSEC (Cummings, et al., 1996) commercial data along with residential (Jump, et al., 1996) summary information. The FSEC unit size is derived from the total installed capacity in the building divided by the number of units.

The fifteen systems had an average unit size of 3.9 tons, this compares with the FSEC data of 4.5 tons, and the residential of 2.9 tons.

The average floor area served by each unit was 1,500 ft^2 for the current study, 1,400 ft^2 for the FSEC buildings, and 1,800 ft^2 for the residential buildings. Since the area served by each unit is similar among all three data sets, the light commercial units are 30 to 50% larger than those found in single family residential houses.

Figure 2 shows the total number of registers (supply and return) versus the unit size. Since FSEC data on the number of registers was not available, the comparison is only with the residential data set. In both cases there is a widespread range in the number of registers for any given unit size, e.g., the number of registers found on a four-ton commercial unit ranged from 4 to 16. In general, the commercial units have fewer registers per ton than the residential units, because commercial spaces are usually open plan, with a few large rooms and fewer, but larger, registers.

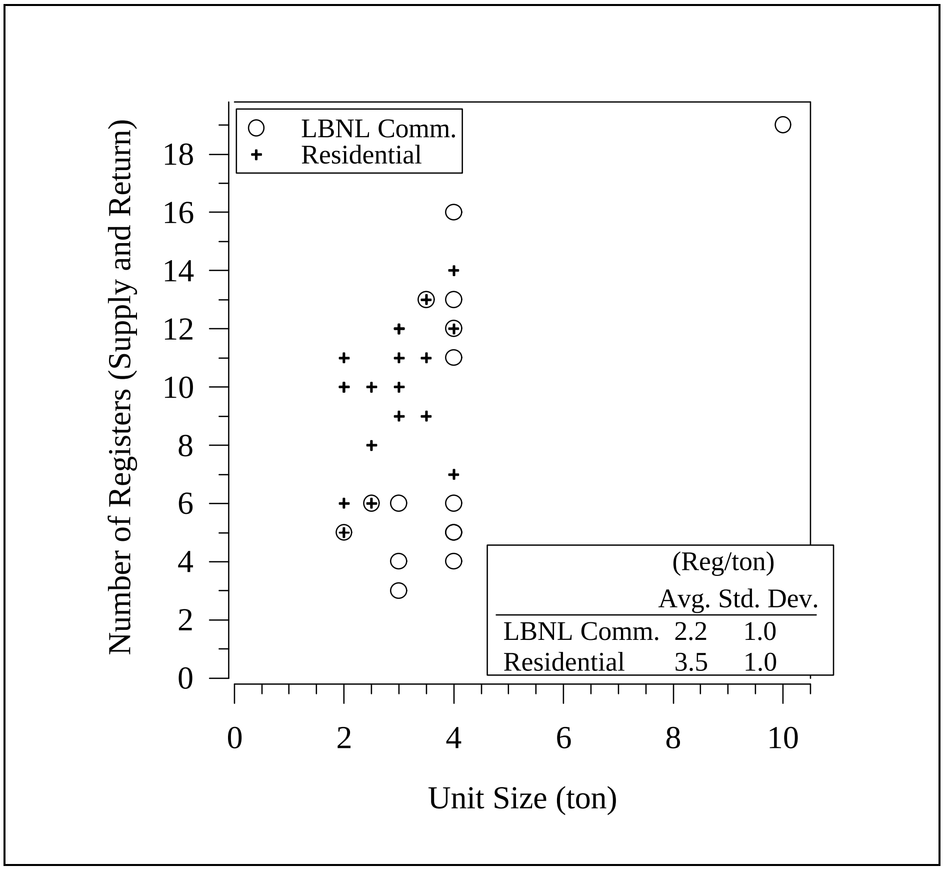

Figure 2. Number of registers -vs- unit size: using the current LBNL commercial and residential (Jump, et al., 1996) data.

In order to understand the dynamics of duct losses, details of the building need to be addressed. Figure 3 summarizes many of the characterization details pertaining to the buildings. Each of the buildings had a drop ceiling with the ducts run in the ceiling cavity. Because of this, two critical building details are the location of the thermal and the air barrier. 25% of the buildings had insulation placed both at the roof deck and on top of the ceiling tiles. The remainder of the buildings were divided between roof-only and ceiling-only insulation. 40% of the buildings had a directly vented ceiling cavity. In these buildings, the lay-in acoustical ceiling tiles formed the major air barrier. In 50% of the buildings the primary thermal barrier was at the ceiling tiles, which implies that the ducts are entirely outside the conditioned space. In 25% of the buildings the ceiling cavity acted like a buffer zone, with the temperature floating between the room and outside temperatures. With these buildings, the thermal barrier is in-between the roof and ceiling. In the remainder of the buildings the thermal barrier was at the roof, however even in these buildings, the ceiling cavity temperature was slightly higher than the room.

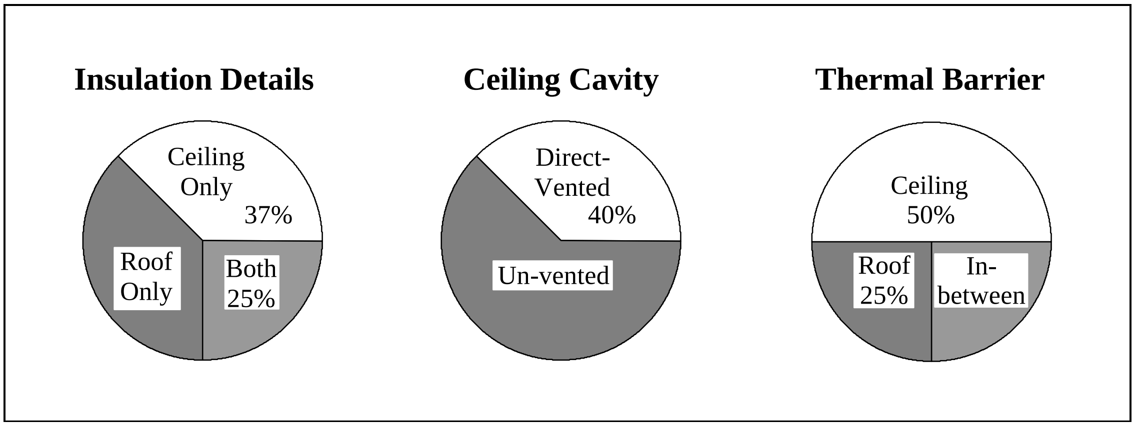

Figure 3. Building thermal and air barrier characterization for the current LBNL commercial buildings.

Figure 4 summarizes HVAC unit characterization details. Duct material fell into two basic types: all metal trunk-and-branch, and flex-octopus. 60% of the systems had all metal ducts, while the remainder had some form of flex-octopus. There are two types of basic ductwork configurations found with the typical light-commercial package roof-top unit: bottom discharge, and side discharge. Bottom-discharge eliminates ductwork exposed outside since it penetrates the roof directly under the unit. The typical side-discharge installation includes 90^0 elbows directly off the unit, ideally cutting down on the amount of duct exposed on the roof. Economics and local practice govern which method is used. Bottom discharge units require the use of a special curb to support the unit, while side discharge units typically use a field-fabricated platform for the unit. 33% of the HVAC units had bottom-discharge ductwork, while the remainder used a side-discharge arrangement.

Air side economizers minimize cooling energy use when it is cooler outside than inside. 47% of the units had some sort of economizer; however, they were not checked for functionality. Only one unit had functioning minimum outside air (an intentional opening in the return duct directly to outside). All of the others either had no outside air provisions, or had the dampers permanently shut.

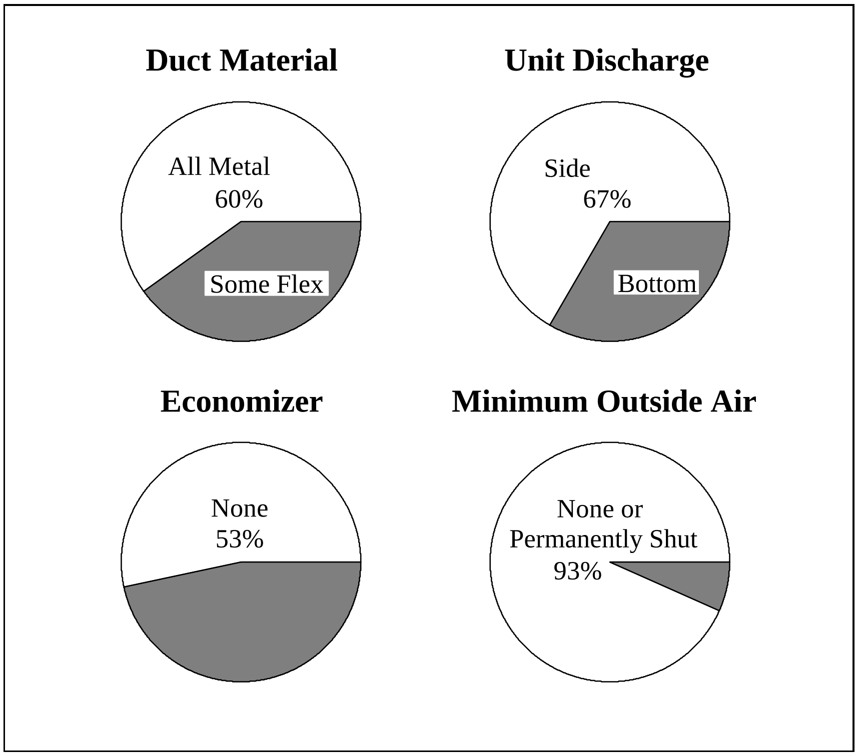

Figure 4. Characterization of HVAC unit details for the current LBNL commercial buildings.

Figure 5 shows system fan flow versus unit size. Fan flow is often used as an indicator of installation quality. The general rule-of-thumb is a flow of 400 cfm/ton. This corresponds to the rated flow, required to obtain the published efficiency ratings, for the HVAC system. When the flow drops below approximately 250 cfm/ton, the coil operates with a temperature conducive to frost formation. Even in dry climates, flow rates this low impair system efficiency. Most of the units had adequate flow, which makes sense with units with few registers, hence low pressure drop systems. Two systems had flows close to the 250 cfm/ton range.

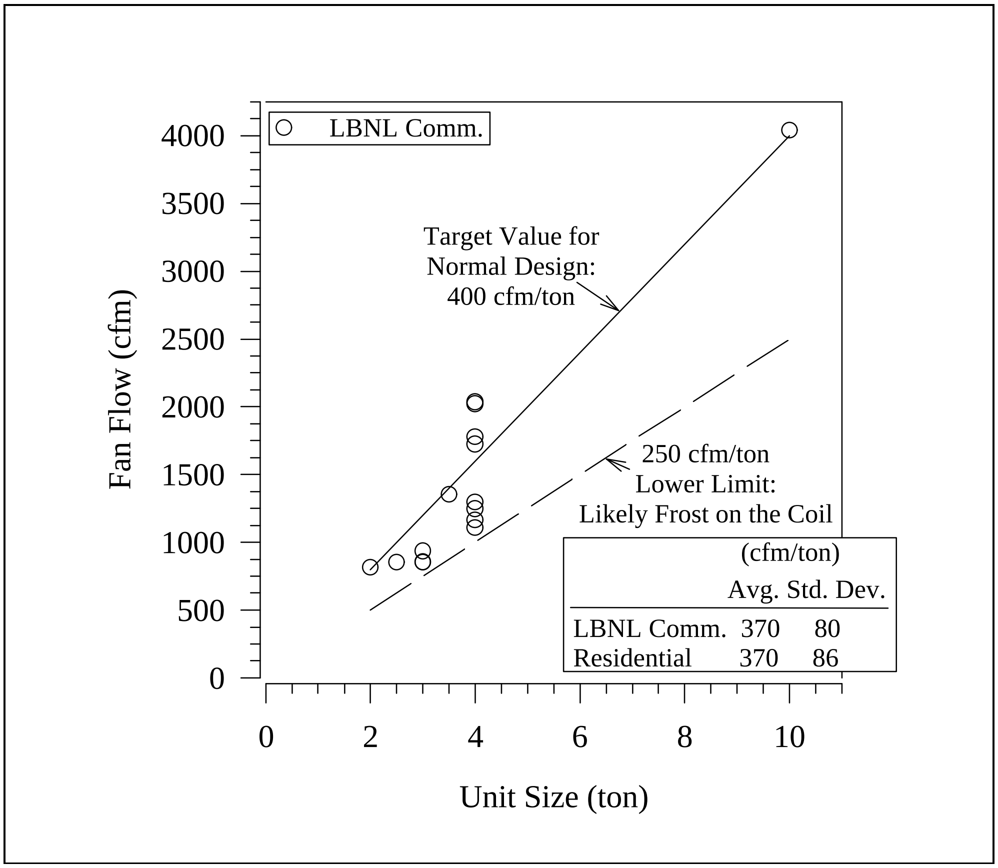

Figure 5. Fan flow -vs- unit size for the current LBNL commercial buildings along with residential (Jump, et al., 1996) summary information.

3.2 Leakage Area of the Duct Systems

The main emphasis of the current study was to measure the leakage area of the ducts. There are several ways to compare the systems to each other, and to other data sets. The goal of comparison is to find a way to normalize the data, making direct comparison of different systems possible.

Figure 6 shows the combined leakage area (ELA_{25}) versus the unit size for both commercial data sets. The data have a large spread in values. A linear regression on the FSEC data only had a r^2 of 0.29. The LBNL and the FSEC leakage values fall in the same general range for any given unit size. Normalizing leakage area with the unit size (cm^2/ton) does not yield a constant due to the large spread in values. However, the residential and FSEC data sets had similar average values for leakage are per ton (cm^2/ton), while the LBNL small commercial buildings had a 40% higher average value, due to outliers.

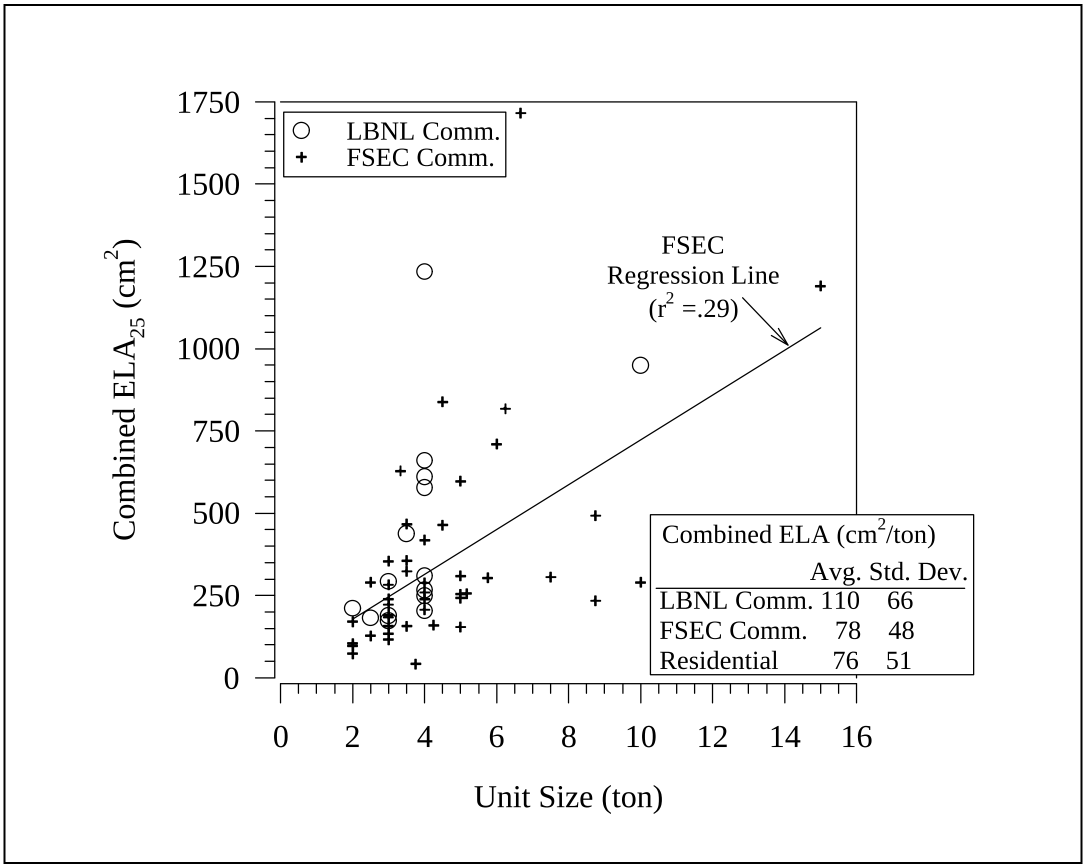

Figure 6. Combined leakage area (ELA_{25}) -vs- unit size using the current LBNL and FSEC (Cummings, et al., 1996) commercial data along with residential (Jump, et al., 1996) summary information. Combined leakage area includes both supply and return leakage. The FSEC unit size is derived from the total installed capacity in the building divided by the number of units.

Figure 7 shows the combined leakage area (ELA_{25}) versus the floor area for both commercial data sets. Again, the data show a large spread in values. A linear regression on the FSEC data only had a r^2 of 0.26. The LBNL data grouping is similar to, and slightly higher than, the FSEC data. It is common to present building envelope leakage results by normalizing leakage area with floor area (cm^2/m^2). The average cm^2/m^2 in the LBNL data set was over 2.5 times that of the residential data, while the FSEC data was just over 2 times the residential. These data suggest that light-commercial duct systems leak air at a rate much greater than residential systems, for any given floor area.

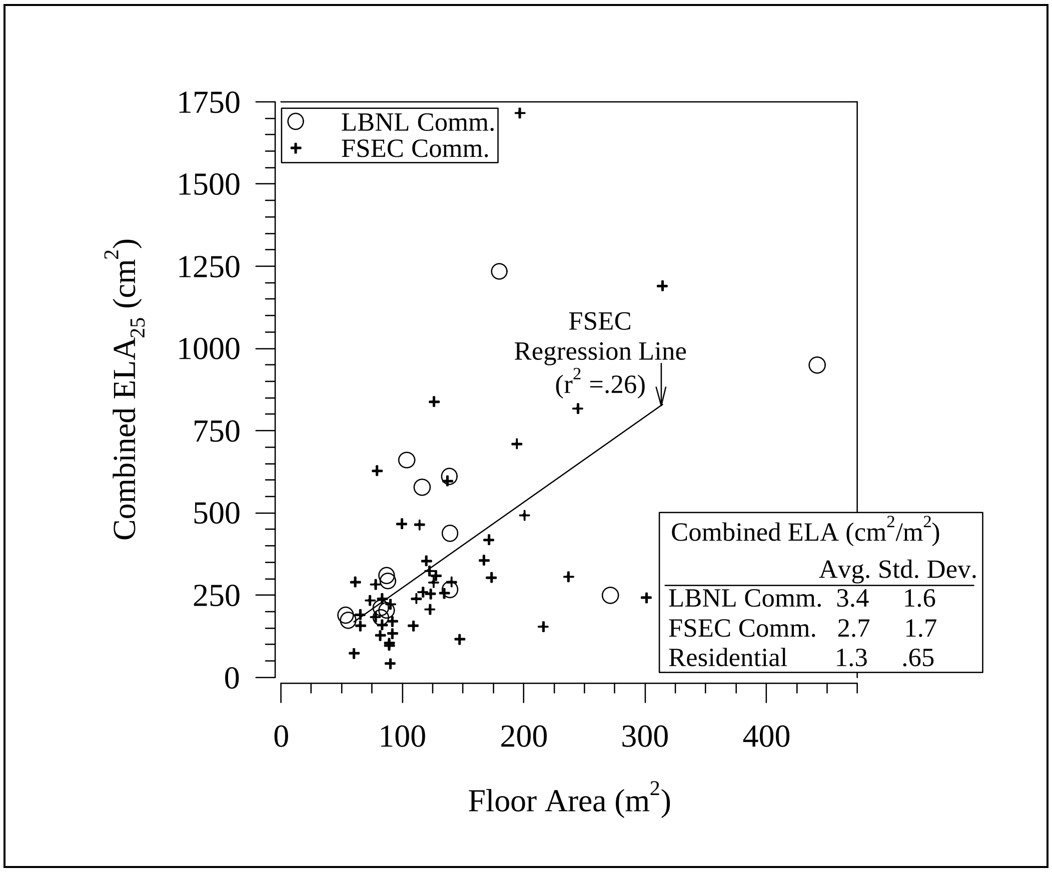

Figure 7. Combined leakage area (ELA_{25}) -vs- floor area using the current LBNL and FSEC (Cummings, et al., 1996) commercial data along with residential (Jump, et al., 1996) summary information. Combined leakage area includes both supply and return leakage.

Figure 8 shows the combined leakage area (ELA_{25}) versus the total number of number of registers for the LBNL and residential data sets (the number of registers was not available from FSEC). For the same number of registers the commercial buildings consistently have higher leakage areas. The average cm^2/register among the LBNL data is 2.3 times that of the residential buildings. This makes sense, since the likely leakage site is at any connection, and as commercial buildings use larger ducts than residential, the larger connection sites have a greater potential for leakage. A linear regression on the LBNL data had a r^2 of 0.80. This suggests that leakage area normalized by the number of registers can be used as a metric for identifying problematic systems.

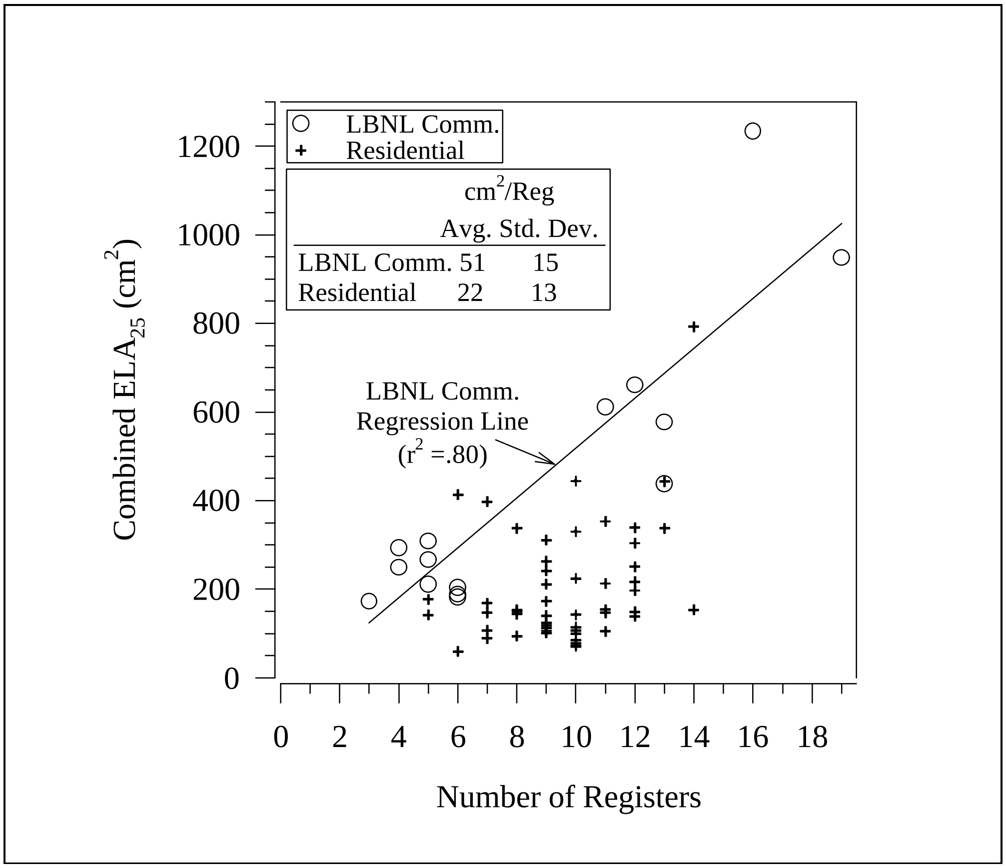

Figure 8. Combined leakage area (ELA_{25}) -vs- number of registers using the current LBNL commercial and residential (Jump, et al., 1996) data. Combined leakage area includes both supply and return leakage.

3.3 Thermal Issues

Figure 9 shows typical temperature profiles for an office conference room over a six-hour period, and it illustrates several points concerning thermal issues. The change in temperature between the supply air plenum, down-stream of the coils, and the register affects system efficiency. It also impacts thermal comfort, the longer the duct the greater the temperature rise, which leads to uneven temperature distribution by the system. The figure illustrates this, as register 1 is closer to the plenum than register 2. The length of cycle duration also effects the temperature rise. Energy from the air stream cools the ducts until they reach a steady-state temperature. The figure shows that air temperature in this particular system never reached steady state, the longer the cycle the lower the temperature rise from the plenum. Finally, this figure shows that the fan does not cycle on and off with the cooling equipment. The cooling equipment shuts off at the bottom of each spike on the plot. The plenum and both registers continue to rise to approximately the same temperature before the beginning of the next cycle. This recovers the energy used to cool the ducts, but it also starts heating the room, as the unit is on the roof in the sun. With the fan running, and the cooling equipment off, the unit acts as a heat exchanger heating the room below. For the 6 hour period represented in this figure, the hot air delivered during the off-cycle accounted for 33% of the total space cooling load.

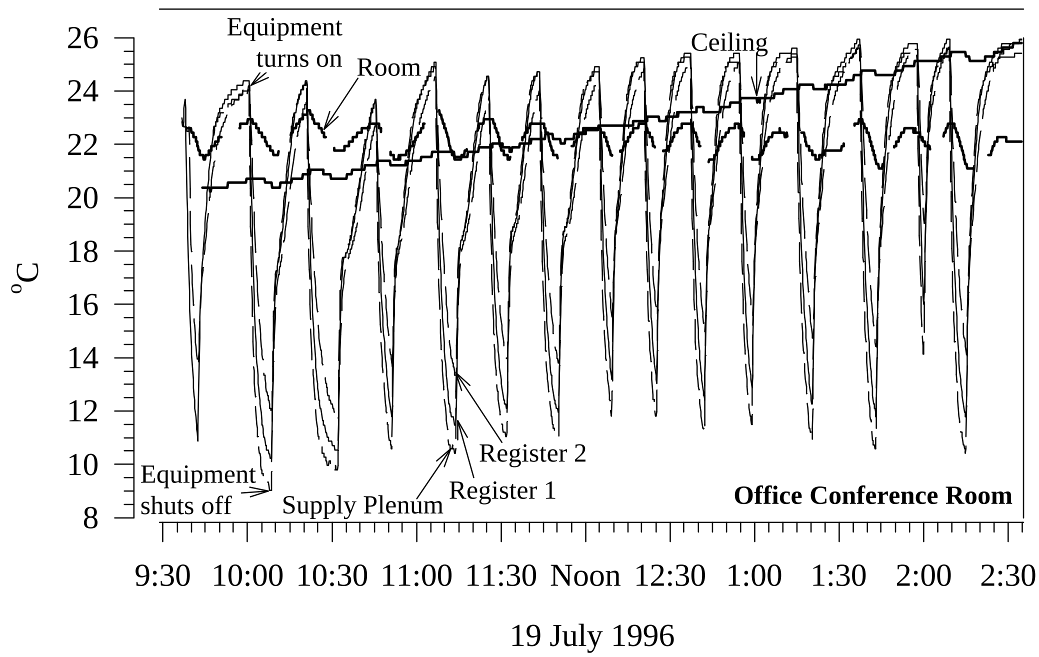

Figure 9. Typical temperature data for 6 hours of operation

3.3.1 Temperature Rise From Supply Plenum to Register

As stated above, the temperature rise from the supply plenum to a supply register affects both system efficiency and thermal comfort. In order for the temperature to rise after leaving the plenum, there needs to be a potential difference between the temperature in the plenum and the temperature of the ambient surroundings (i.e., it has to be hotter outside the duct than inside the duct). Figure 10 shows the temperature rise from the supply plenum to the register plotted versus this temperature difference, between ambient and plenum. With the ducts located, in all cases, in the ceiling cavity, the ceiling cavity serves as the ambient temperature. All points represent conditions in the mid-afternoon (~ 2 to 4 p.m.), and as close to a steady-state operation as possible. Systems that never reach steady-state are labeled as transient, and the temperatures are from the end of the longest on-time cycle available. The temperature rise ranged from 0.5^{O}C to almost 6^{O}C. Both extremes, high and low, occurred near the greatest potential difference; however, the temperature difference between ambient and the plenum alone does not correlate well with the temperature rise.

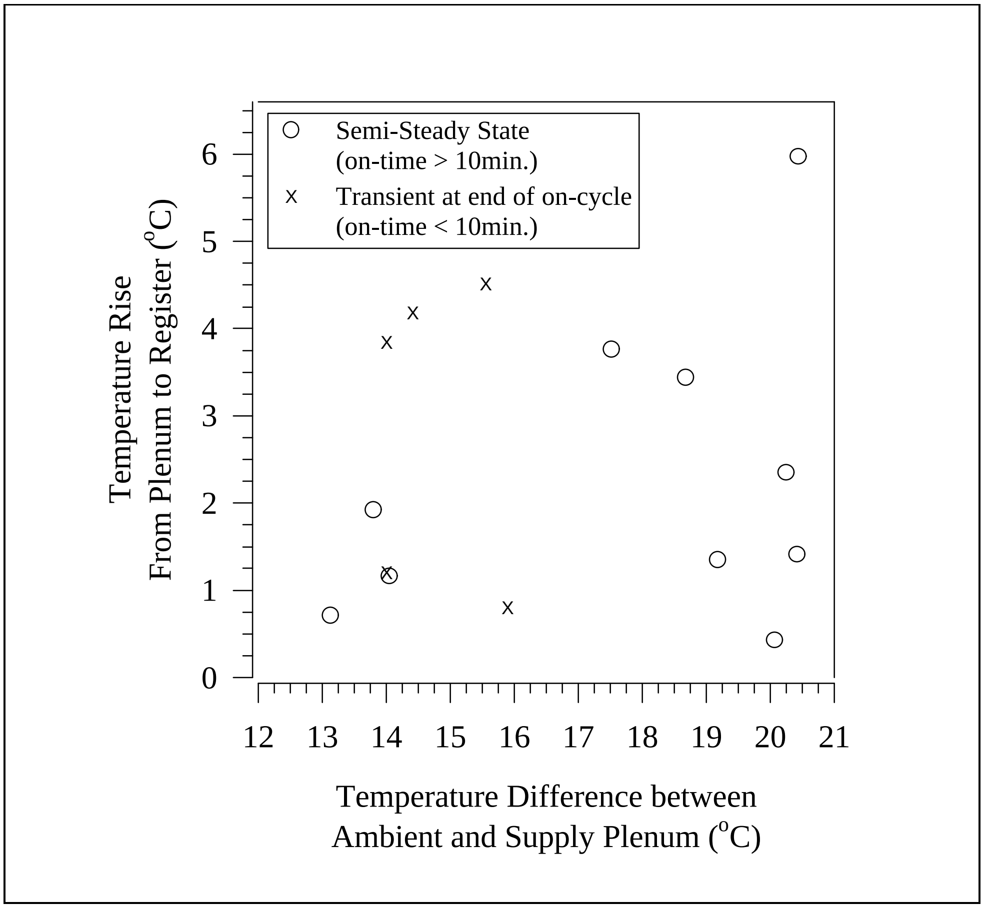

Figure 10. Temperature rise from supply plenum to register -vs- temperature difference between ambient and supply plenum using the current LBNL commercial data. All data taken in mid-afternoon, at the end of an on-cycle.

The length of the duct impacts the temperature rise in addition to the ambient conditions. Figure 11 plots the temperature rise from the plenum versus the length of duct. Again the data shows a high degree of scatter. The longest duct has one of the lowest temperature rises, illustrating that the length does not properly correlate with the temperature rise.

There are several factors not taken into account by either of these two previous attempts at correlating the temperature rise. Among these factors are, the effective temperature of the ambient, taking into account radiation effects, differing amounts of insulation on the ducts, different duct sizes, and finally, different air velocities within the duct. Unfortunately, not enough data was taken to easily compare the temperature rise from one system with another.

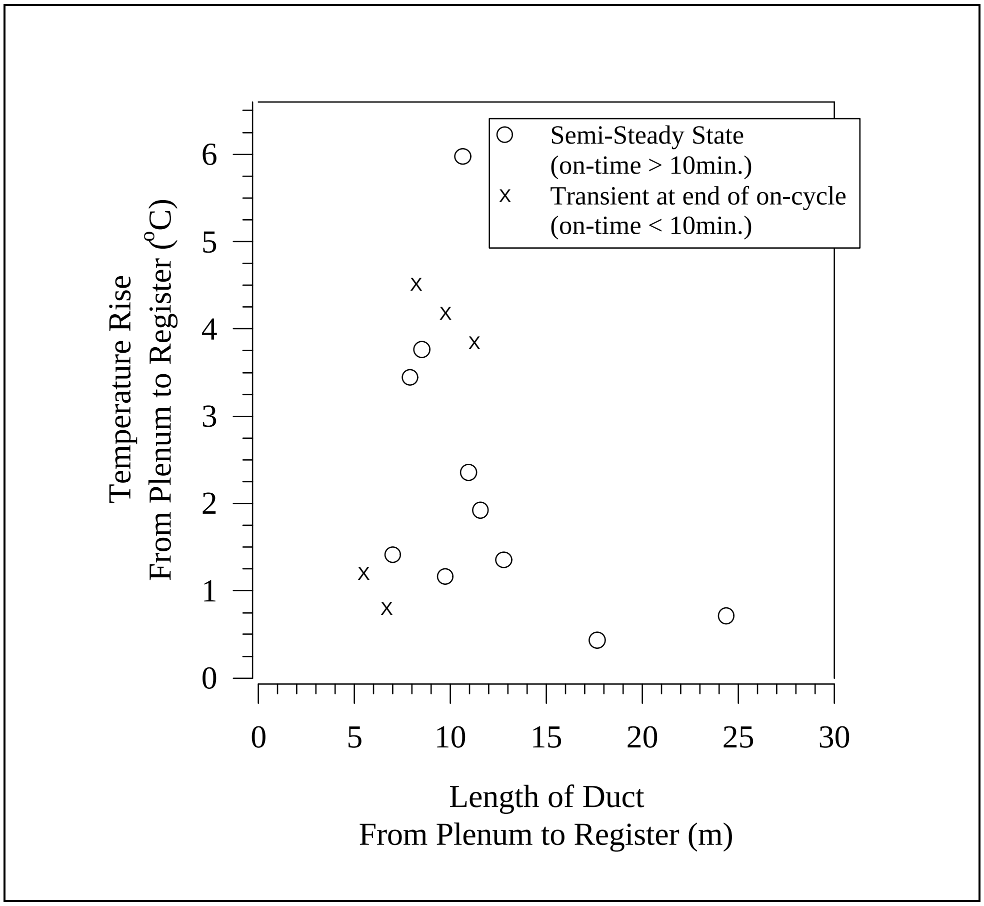

Figure 11. Temperature rise from supply plenum to register -vs- the length of the duct the from the supply plenum to register using the current LBNL commercial data. All data taken in mid-afternoon, at the end of an on-cycle.

3.3.2 Cycle Duration

Figure 9 showed that on-cycle time impacted the plenum to register temperature rise. The system shown in this figure had no period of steady operation with an average on-time of slightly over 5min. Figure 12 shows a typical temperature rise versus time. This figure is for a different system. It shows that the ducts effectively reached a steady-state condition in approximately ten minutes. This particular register was far from the plenum (24 m). Registers closer to the plenum reach steady-state operation sooner, leading to periods of uneven temperature distribution.

66% of total number of systems tested operated a significant portion of the time with the on-time cycles less than 10 minutes. 33% of total number of systems had no single on-cycle longer than 10 minutes. This results in non-uniform (from register to register) and constantly changing temperatures (at all registers) in the distribution system, and the space, leading to thermal comfort issues.

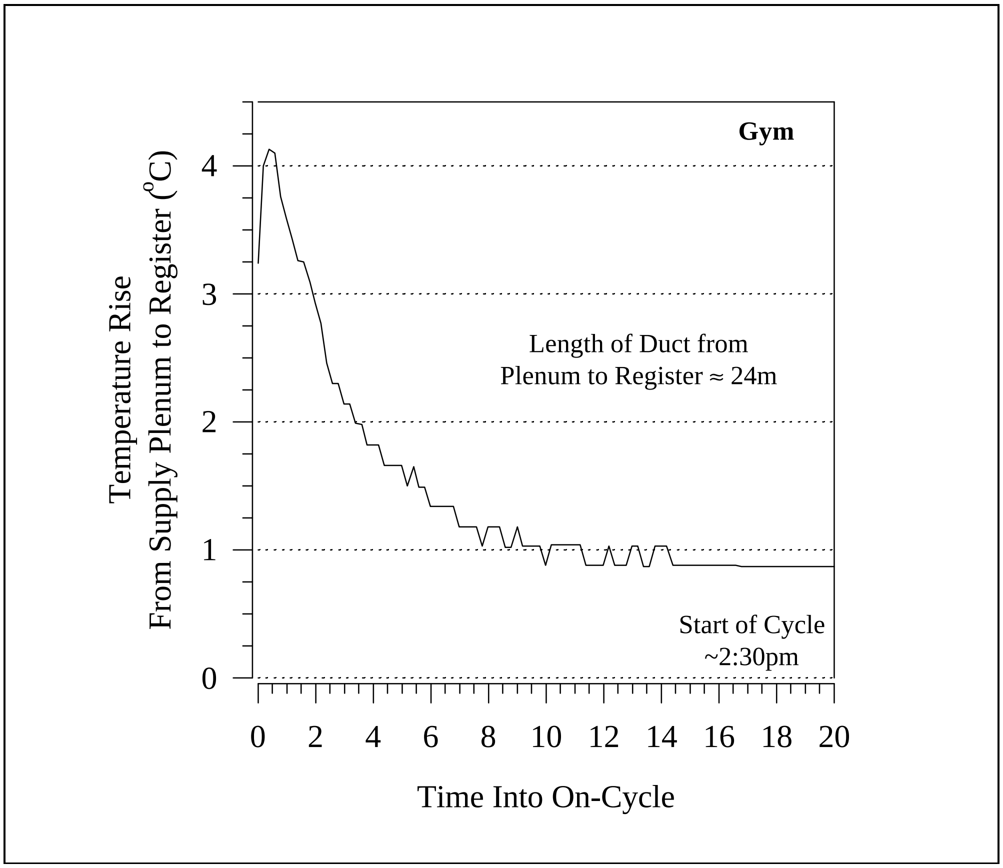

Figure 12. Typical temperature rise from supply plenum to register -vs- time after start of on-cycle. Plot is for a single cycle for a single system.

3.3.3 Fan Operation

Figure 13 shows two hours of temperature data for two different systems. With one of the systems, the store, the fan cycles with the compressor. While the other system, the gym, regardless of whether or not the compressor is on, the fan stays on all of the time. Keeping the fan on serves two functions (1) maintains air movement and (2) (assuming provisions exist) provides outside air. During the off-cycle, in the store, the plenum temperature rises dramatically, while the register temperature approaches the room temperature. At the beginning of the on-cycle the plenum temperature is more than 15^{O}C higher than the register temperature. This is because the fan is off, and the unit and plenum are on the roof, the effective temperature, including solar-radiation, warms the air in the plenum and unit considerably. In the gym, the fan stays on during the equipment off-cycle, and both the plenum and register temperatures warm approximately the same before the beginning of the next on-cycle, approaching and slightly exceeding the room temperature.

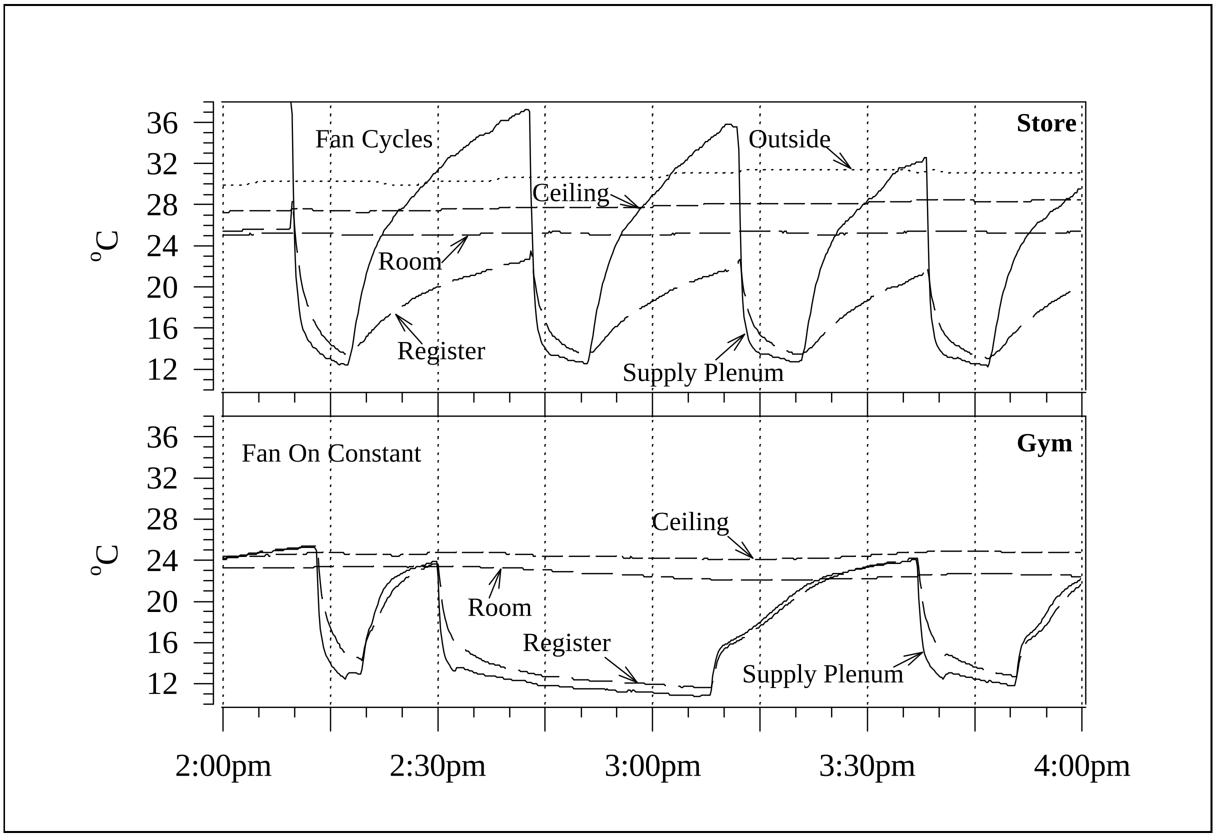

Figure 13. Two systems: in the store the fan cycles with cooling equipment, and in the gym the fan stays on all the time.

3.3.4 Conduction Losses

The basic definition of the conduction effectiveness concerns the fraction of the capacity lost. In the simplest terms, neglecting any leakage, the term is simply (1)

$$e_s(t) = \frac{(T_{reg}(t) - T_{room}(t))}{(T_{plenum}(t) - T_{room}(t))} \tag{1}$$

Where:

$e_s(t)$: Conduction effectiveness at time *t*

$T_{reg}(t)$: Register temperature at time *t*

$T_{room}(t)$: Room temperature at time *t*

$T_{plenum}(t)$: Supply plenum temperature at time *t*

Equation (1) yields an instantaneous value for effectiveness at any given time *t*. It includes both conduction and impacts of thermal cycling. By summing each of the quantities in the above equation one arrives at a cumulative effectiveness for any given time *t'* (2)

$$\bar{e}_s(t') = \frac{\int_0^{t'} \left(T_{reg}(t) - T_{room}(t)\right)dt}{\int_0^{t'} \left(T_{plenum}(t) - T_{room}(t)\right)dt} \tag{2}$$

Where:

$\bar{e}_s(t')$: Cumulative conduction effectiveness up to time *t'*

The cumulative effectiveness in equation (2) gives us a running total of the ratio of the energy delivered at the register to the potential at the plenum. The uses of the above formulations have subtle differences between the fan cycles with unit and fan-on constant cases.

3.3.4.1 Fan Cycles with Unit

Figure 14 shows a single cycle for a case where the fan cycles with the cooling unit (fan on/off case). The building is the same store shown in Figure 13. This cycle was the first of the day, and both the plenum and register temperatures started off high. The instantaneous and cumulative effectiveness started low since the ducts were warm and had to be cooled. An important point here is the fact that the cumulative lagged behind the instantaneous effectiveness; at the end of the on-cycle, the instantaneous was over 90% while the cumulative was around 80%. The difference between the two effectiveness values is a consequence of the energy stored in the ducts as a fraction of the energy supplied to the plenum. Nothing was known about the flow out of the registers after the fan shut off, as a result, what happens to this stored energy is not clear from this information. The overall analysis for the fan-off cases only includes on-cycle information.

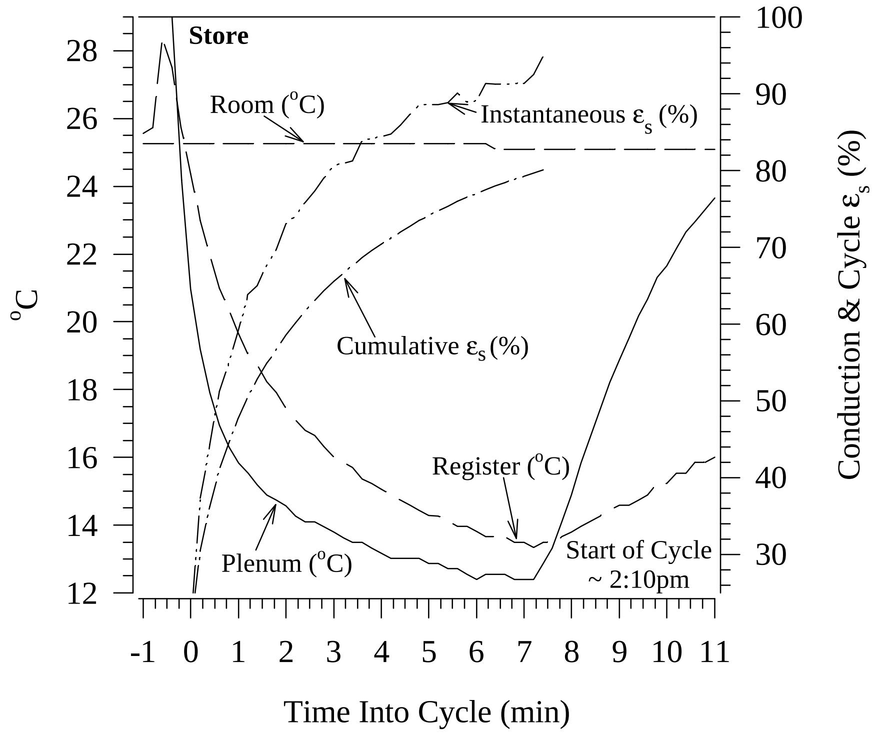

Figure 14. Analysis of an individual cycle with a fan on/off case.

Figure 15 shows the cumulative effectiveness versus time for all four of the cycles captured in the store. With each consecutive cycle the ducts start off a little cooler. This accounts for the fact that the effectiveness starts at a higher value for each consecutive cycle. The final three cycles end near the same value (~85%); however, due to the short on-times they have not reached a steady-state.

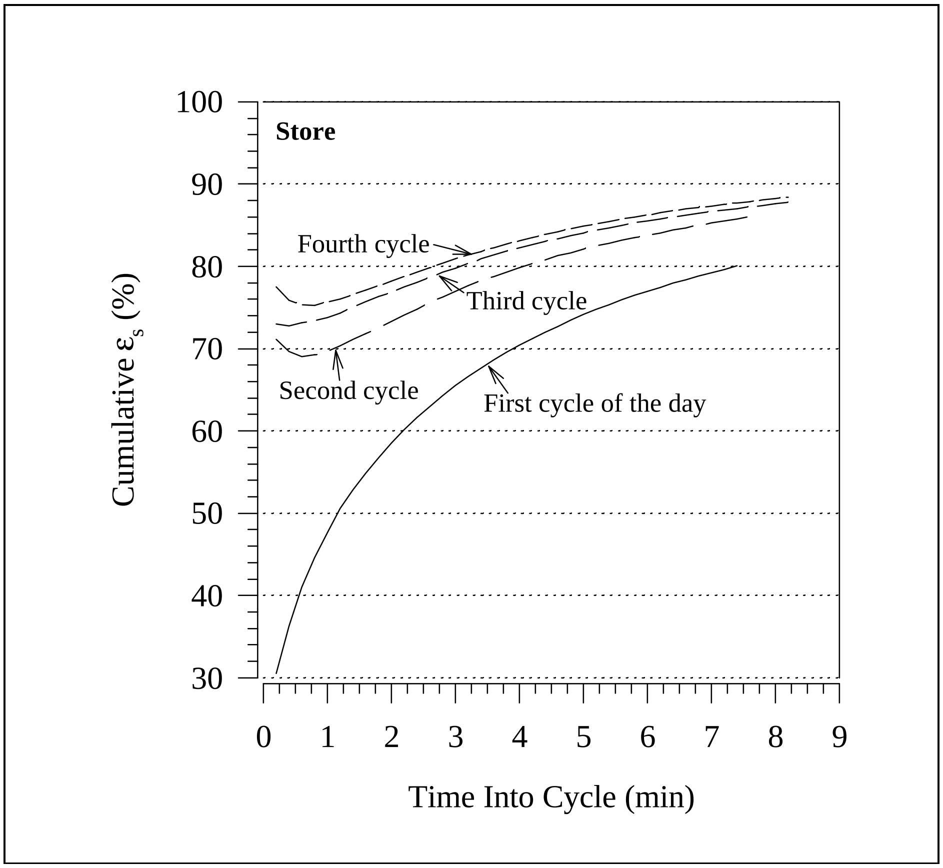

Figure 15. Cumulative supply effectiveness (ε_s) -vs- time for four different cycles in a fan on/off case.

3.3.4.2 Fan Stays On

Figure 16 shows three hours of temperature information for a fan-always-on case. This is the same gym in Figure 13, only with different time and temperature scales. The length of the off-cycle is important since after a period of time, during the off-cycle, the system starts heating the space instead of cooling it. Once the system starts heating the space, the concept of instantaneous effectiveness is meaningless, and we need to redefine cumulative effectiveness (2).

If we define the point in time just when the system starts heating the room ($T_{plenum} = T_{room}$) as t^*, cumulative effectiveness becomes after time t^*:

$$\bar{e}_s(t')\Big|_{t'>t^*} = \frac{\int_0^{t'} \left(T_{reg}(t) - T_{room}(t)\right)dt}{\int_0^{t^*} \left(T_{plenum}(t) - T_{room}(t)\right)dt} \tag{3}$$

From the time the compressor shuts off to t^* the system recovers the energy stored in the ducts and coil.

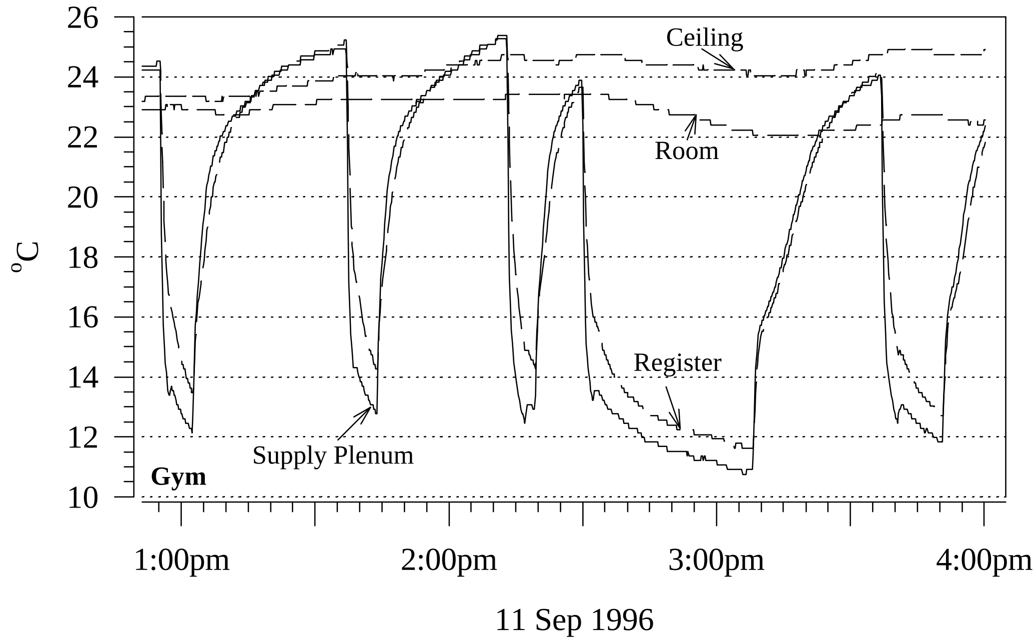

Figure 16. Three hour temperature data for a fan-always-on case.

Figure 17 shows a single cycle with a short compressor on-time for the fan-always-on gym case. The start of the cycle was around 1:37pm, the compressor shut off approximately 6 minutes later, and *t** was reached about 15 minutes after the start of the cycle. When the compressor shut, off the system had not reached “steady-state” as shown by the rapidly increasing instantaneous effectiveness. Once the compressor shut off, the cumulative effectiveness increases representing a regain of the energy stored in the system. This peaks upon reaching *t**, after which the effectiveness decreases, because the system is now heating the space. The cumulative effectiveness was 73% when the compressor shut off, peaked at 89%, and then fell off to 58%.

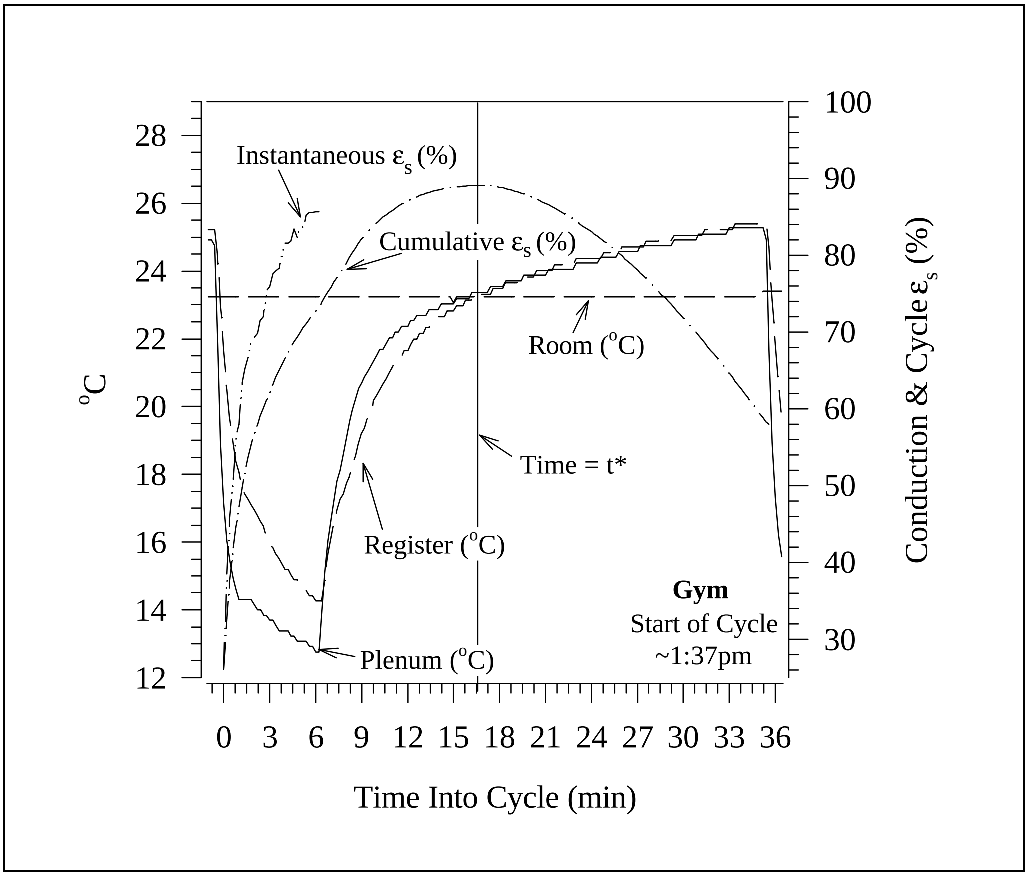

Figure 17. Analysis of an individual short on-time cycle with a fan-always-on case.

Figure 18 shows a single long on-time cycle for the fan-always-on gym case. The start of the cycle was around 2:30pm, this time the compressor shut off approximately 37 minutes later, and *t** was reached about 54 minutes after the start of the cycle. The system reached a steady-state value after approximately 10 minutes of operation, as indicated by the instantaneous effectiveness leveling out at approximately 94%. The cumulative effectiveness when the compressor shut off was approximately 89%, and it peaked at 92%, and the fell down to slightly under 90% by the end of the cycle. The energy recovered in the long cycle is a much smaller percentage of the energy supplied over the longer on-time cycle.

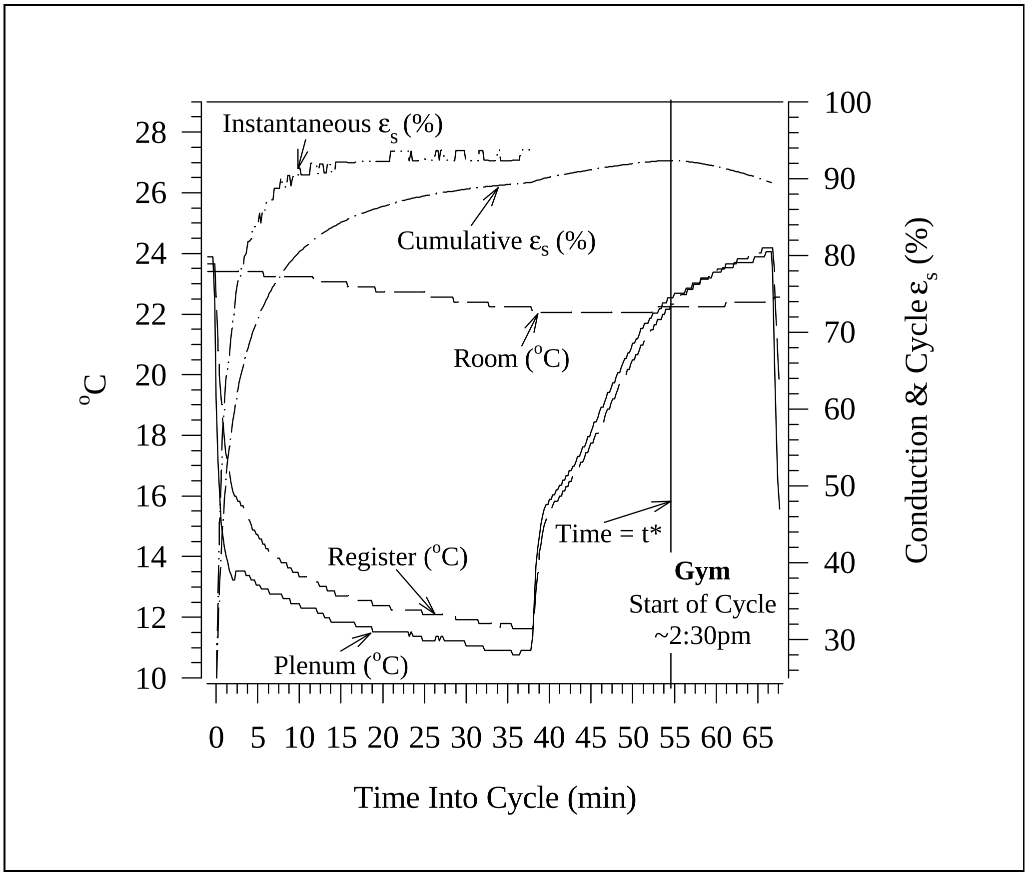

Figure 18. Analysis of an individual long on-time cycle with a fan-always-on case.

Figure 19 shows the cumulative effectiveness versus time for three cycles of varying on-time duration in the gym. The circles represent when the compressor shut off, after which the system entered into an energy recovery mode (i.e., recovering the energy stored in the equipment coils, and ducts). In each of the cases, the effectiveness peaked when the system started heating the room. With each case this peak was close to the steady-state instantaneous effectiveness of 94%. The longer the off-cycle, the lower the effectiveness was for the individual cycle. This is because of the increased heating time due to long off-cycles with the fan on constantly.

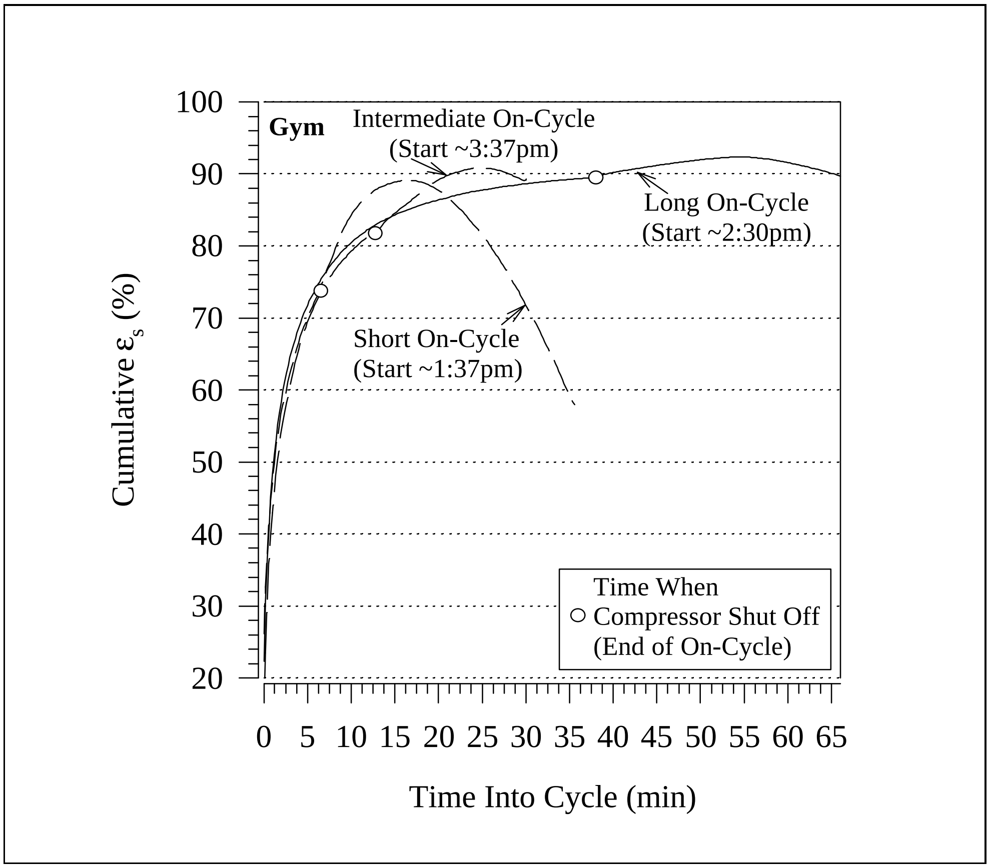

Figure 19. Cumulative ε_s -vs- time for three cycles of different on-time duration in a fan-always-on case

3.4 Occupant Interactions

The fan was on constantly in the gym analyzed in section 3.3.4.2 because the occupants were not aware of how to operate the system. The thermostat had a switch labeled FAN/ON, and they thought for the unit to work it must be in the ON position. However, it is common to leave the fan on constantly in buildings where the system provides some outside air to the space for indoor air quality issues.

Figure 20 shows temperature data for two apparently identical systems, serving opposite sides of a hallway, east and west, in an LBNL office setting. One short-cycles, the west side, and the other, the east side, does not. Each side of the hallway had similar internal loads, roughly the same numbers of desks, computers, and people. Investigation revealed that the occupants in the east zone were controlling the load by opening the large sliding windows, while the windows were shut in the other zone. The load served by the east system was three times that served by the west system.

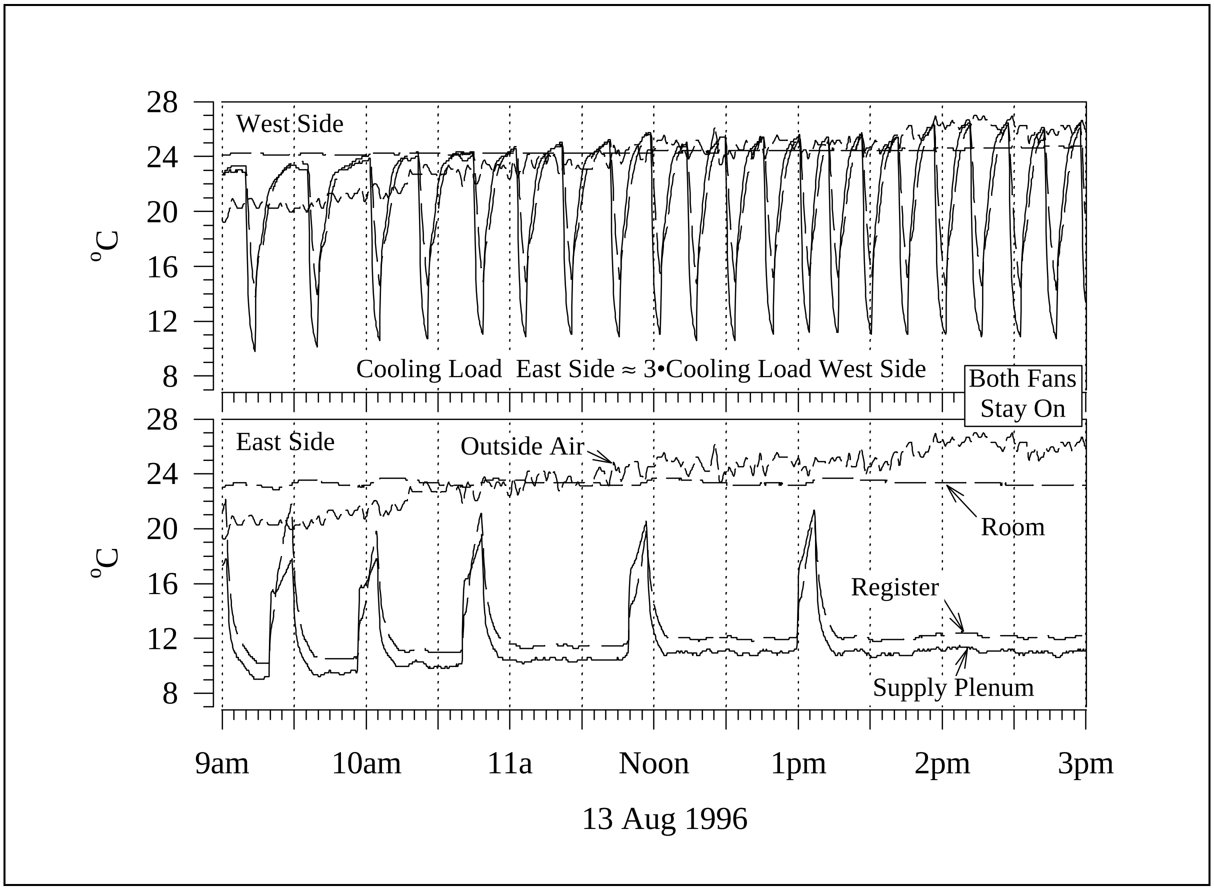

Figure 20. Two identical systems serving “mirror-image” offices located across the hall from each other.

Figure 21 shows temperature data for a shoe repair shop and the occupant solution to “ventilation” control. A shoe repair shop uses a lot of volatile chemicals. The occupants said that they open a window to control odors. Looking at the two days of operation, one day the system ran at very nearly steady-state, while the other day this did not happen until after 3pm. The occupants are probably controlling the load, as in the office in Figure 20, rather than strictly odor control.

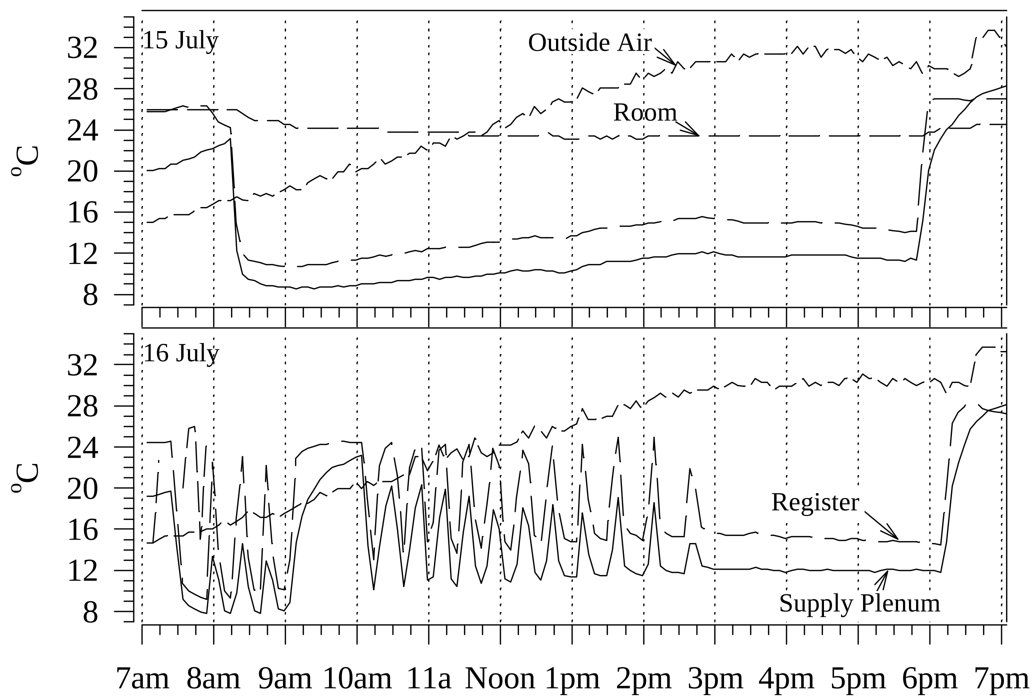

Figure 21. Operator control for "ventilation" purposes

4. Summary and Conclusions

In these buildings visited, repeated observations suggested problems a result of first-cost economics during construction: torn and missing external duct wrap, poor workmanship around duct take-offs and fittings, disconnected ducts, and improperly installed duct mastic. Where there was ceiling tile insulation, installation was, at best, very uneven. Visual indicators alone are not good at identifying poor systems. While systems that appeared poor usually had high ELA_{25}'s, the systems with the highest ELA_{25} looked, upon initial inspection, like good systems.

When the fan cycles with the cooling equipment, they both shut off at the same time. From the viewpoint of recovering energy stored in the ducts; thermal analysis suggests there is an optimum time to shut off the system fan following the cooling equipment. This is not always possible since the fans are left on in many of these systems for ventilation requirements.

This study did not attempt to quantify the amount of outside air entering each building. However, observations made during the characterization phase of this project suggests the buildings visited in this study will have very low quantities of outside air.

For the set of light-commercial buildings visited in this study, the following can be said:

- ◊ Light Commercial ductwork leaks: at a rate over twice that of residential systems
- ◊ Duct systems are outside the direct conditioned space
- ◊ The primary air barrier is often located at the drop ceiling: this is effectively **NO** barrier
- ◊ With the fan on and the compressor off, the system eventually acts as a heater to the space

A relatively small data set forms the basis for these conclusions; additional data is needed to better characterize this large stock of buildings. Thermal measurements require additional, and more complete, data in order to fully understand the differences between the fan on/off and fan-always-on cases, as well as energy recovery mechanisms in the fan on/off case. We have plans to continue with the same characterization and leakage measurement work by testing additional systems, while getting more complete thermal data.

Acknowledgments

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