ENERGY CENTER OF WISCONSIN

ECW Report Number 233-1

# Energy Savings from Daylighting

## *A Controlled Experiment*

May 2005

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## Project Funding

The Association of State Energy Research and Technology Transfer Institutions, Inc. (ASERTTI), the National Association of State Energy Officials (NASEO), the Department of Energy (DOE) and the Energy Smart Schools Program funded and conducted a joint project that encompassed applied research, field testing and technology integration.  As part of the overall project there were eight distinct tasks outlined, each with their own set of goals, activities and deliverables.  This report encompasses data on *Task 4: Advanced Daylighting Research, subtask 3 HVAC Performance Research.*

## Project Manager

Scott Pigg

## Acknowledgements

Energy Resource Station staff member Curt Klassen provided much assistance throughout the design, implementation and subsequent data analysis for the project.

Former Energy Center project manager Craig Schepp was instrumental in scoping the project, specifying the configuration of the rooms and test parameters, and getting the project up and running.

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## REPORT SUMMARY

Demonstrating actual lighting and HVAC impacts from high performance glazing and daylighting strategies is difficult. While many schools and other buildings have been designed and built using these strategies, attempts to compare energy consumption with standard designs must confront the myriad ways in which any particular building differs from other buildings. This leaves open the question of the extent to which measured differences in energy consumption are due to deliberate design strategies versus other uncontrolled factors such as operation schedules and building orientation.

This report presents the results of an experiment conducted at the Energy Resource Station near Des Moines, Iowa in two sets of identical rooms with independent lighting and HVAC delivery systems. One set of four rooms ("test rooms") was configured with high-performance glazing with reduced visible transmittance as well as direct/indirect electric lighting with photosensor dimming. The other set of four rooms ("control rooms") was configured with standard clear-glass glazing and ceiling-mounted fluorescent fixtures with no dimming. The rooms were identical in all other respects.

Both configurations were operated as typical variable air-volume (VAV) systems with a central chilled-water coil and terminal hydronic re-heat coils. This set-up allowed for a direct comparison of lighting and HVAC energy consumption through the analysis of the more than 600 parameters that are recorded at one-minute intervals in this highly instrumented facility.

The experiment was conducted in three rounds during the summer, fall and winter of 2003, comprising a total of 70 days of operation. Within each round, three slightly different configurations of the test rooms were evaluated: (1) base case, as noted above; (2) reduced fenestration, simulated by partially covering windows in the high-performance rooms with exterior panels; and, (3) addition of an interior light shelf to improve the distribution of natural light to the interior of the rooms.

### RESULTS

The lighting and HVAC operating cost savings for the high-performance rooms are considerable, and represent a savings of more than 20 percent on operating costs of about $1.13 per square foot. The table below shows how these costs and savings break out:

| | Standard configuration (control rooms) annual operating costs | Savings for High-Performance configuration (test rooms) | |
|---|---|---|---|
| | (cents/ft$^2$) | (cents/ft$^2$) | Percent |
| Lighting energy | 22 | 7 | 32% |
| Cooling energy | 19 | 5 | 25% |
| Heating energy | 6 | -0.1 | -1% |
| Fan energy | 13 | 0.3 | 3% |
| Demand charges | 53 | 12 | 24% |
| Total | 113 | 24 | 22% |

### Lighting Energy

The dimmable fixtures in the high-performance rooms operated at reduced output much of the time. These fixtures used about half the electricity as the fixtures in the standard rooms on sunny days. The overall savings is somewhat less due to occasional overcast conditions, shorter days during the winter, and the fact that daylighting was not possible for the interior room, which represents a quarter of the floor space.

### Cooling Energy and Chiller Sizing

On a weather normalized basis, the high performance rooms require 25 percent less cooling than the standard rooms. These savings derive from three differences between the two configurations: (1) reduced need to remove heat from dimmed electric lighting; (2) reduced heat gain through the high-performance windows; and, (3) reduced cooling required to condition ventilation air, which increases as the other cooling loads increase. Under hot conditions, the last two factors dominate, as electric lighting represents only about 10 percent of the building's cooling load.

In terms of chiller sizing, analysis of hourly cooling loads on the two systems show that the high-performance configuration results in 26 percent lower cooling load at the Des Moines summer design temperature of 93$^o$F.

### Heating Energy

Reduced solar gain and electric lighting loads should translate into higher winter heating costs for the high performance configuration, and the data do reflect this effect at temperatures below about 40$^o$F. However, the data also reveal that the high performance rooms require less re-heat energy at higher temperatures. This is presumably due to less time in which cooling is needed in only one or two rooms to deal with high solar and electric lighting loads. On balance, these two effects effectively cancel out, and the impact on heating energy is negligible.

### Fan Energy

The high performance rooms required somewhat less fan energy during hot weather, due to reduced need for VAV-system airflow to meet the cooling load. At other times the two systems used about the same amount of fan energy.

### Demand Charges

Analysis of 15-minute combined lighting and HVAC system demand shows a substantial reduction in monthly (and rolling annual) demand charges, representing more than half of the total operating cost savings. These savings are predicated on the assumption that a school or office using the high performance configuration tested here would have a 25 percent smaller chiller, with a comparable reduction in chiller power draw when it is operating. Demand savings without chiller downsizing would be much smaller, since it only takes one 15-minute period of chiller operation in a given month to set the peak demand charge for the month (as well as the rolling demand charge for the year). These results reinforce the need to couple high performance glazing and lighting specifications with chiller sizing. Demand charge savings are mostly due to reduced cooling requirements in warmer months; while electric lighting saves energy during the winter, the shorter days mean that the dimming system still operates at full power during part of the day.

### Effect of Different High Performance Configurations

The data did not show large differences in lighting or HVAC energy use across the three high-performance configurations tested. The configuration with reduced fenestration area had somewhat higher lighting energy use due to decreased daylight availability, and the configuration with the light shelf was between this level and the standard configuration. None of the configurations showed statistically significant differences in HVAC energy, though this is at least partly a consequence of less statistical precision when analyzing across varying weather conditions rather than being able to directly compare energy use across configurations under identical conditions.

Overall, the results of this experiment show that that there is significant potential for reduced lighting and HVAC operating costs—as well as upfront capital costs for chillers—through careful attention to glazing characteristics and lighting configuration.

## EXPERIMENTAL SETUP

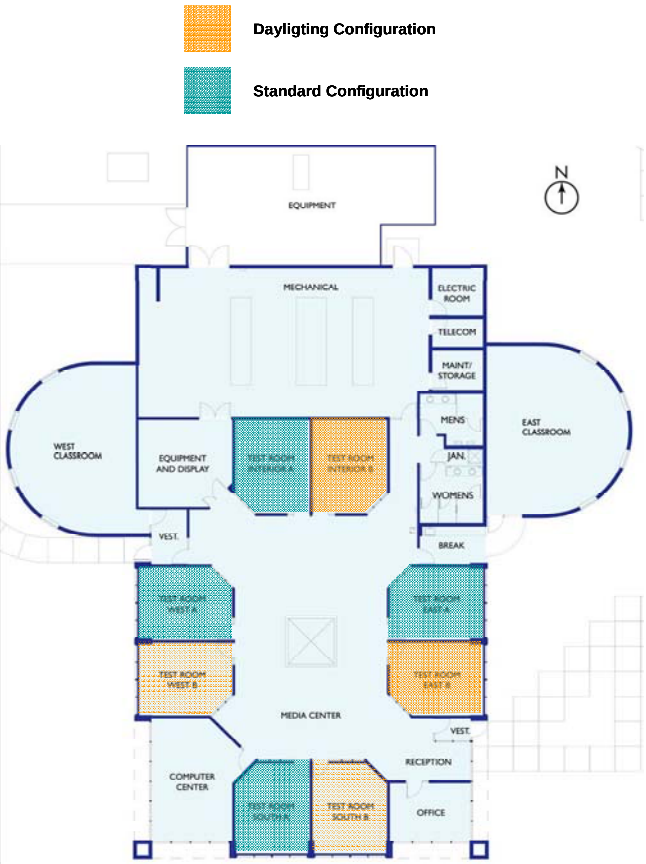
### TEST CONFIGURATION

FIGURE 1. ENERGY RESOURCE STATION.

The experiment was conducted at the Energy Resource Station, in Ankeny, Iowa just north of Des Moines (Figure 1). Associated with the Iowa Energy Center, this highly instrumented facility is specifically designed for multiple, full-scale tests and demonstrations involving commercial building lighting and HVAC systems.

The facility contains eight test rooms, each measuring 267 square feet in size. The rooms are paired into “A” and “B” sets, which are served by separate but identical HVAC systems. There are pairs of rooms on the east, south, and west faces of the building, as well as a pair of interior rooms.

FIGURE 2. FLOOR PLAN WITH EXPERIMENTAL ROOM ASSIGNMENTS.



For the project, the “A” rooms were assigned to represent a typical standard configuration for electric lighting and window specification—referred to here as the control rooms. The “B” rooms represented a daylighting design that included dimming controls on the electric lighting (except in the interior room), and windows with reduced visible transmittance to reduce glare (Figure 2). These rooms are referred to in this report as the test rooms. This approach created a case/control experimental design, with the control rooms serving as the baseline against which energy consumption in the test rooms could be compared.

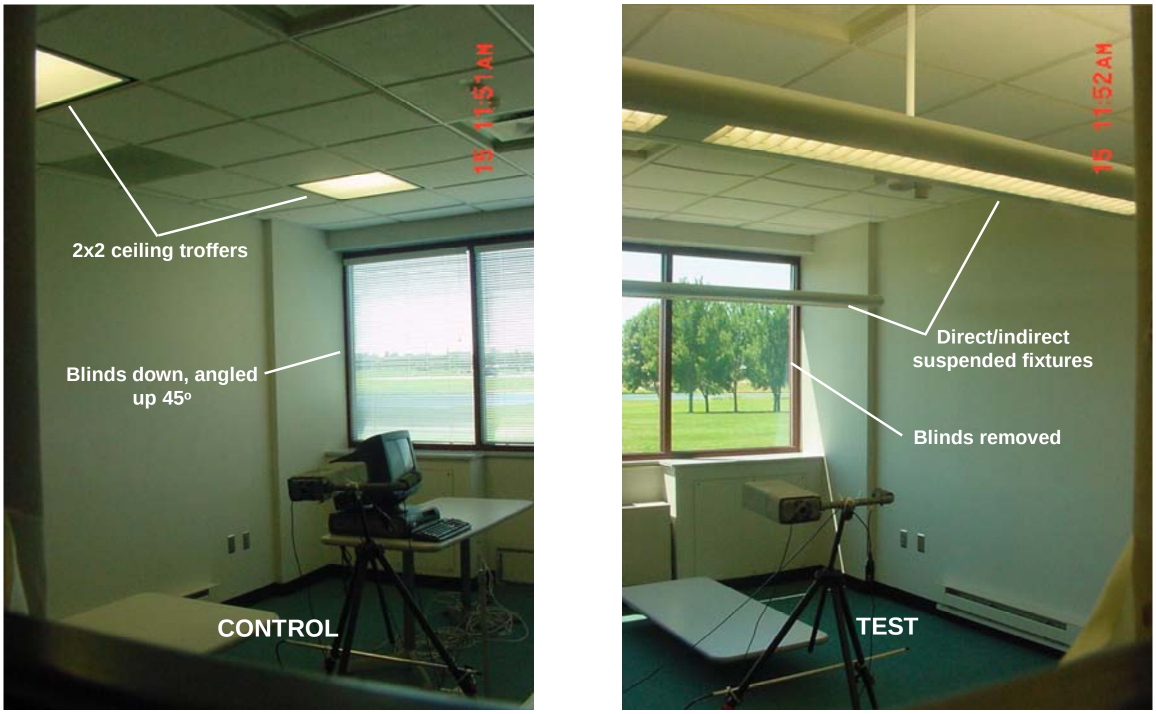
In addition to dimming controls, the test rooms also used direct/indirect suspended luminaries in place of the ceiling-mounted troffers used in the standard rooms. Blinds were used in the control rooms, but were removed in the test rooms. In other respects, the two sets of rooms were configured and operated identically. Electric lighting was turned on from 7 a.m. to 6 p.m. in both sets of rooms to simulate a typical classroom or office environment.

Table 1, along with Figure 3 and Figure 4 highlight the key differences between the two sets of rooms in terms of lighting and fenestration. Appendix A provides additional details about the placement of the lighting in the rooms.

TABLE 1. ELECTRIC LIGHTING AND WINDOW CONFIGURATIONS

| | | **DAYLIGHTING CONFIGURATION (TEST ROOMS)** | **STANDARD CONFIGURATION (CONTROL ROOMS)** |
|---|---|---|---|
| **ELECTRIC LIGHTING** | | | |
| | Fixtures | Two rows of suspended direct/indirect fixtures. | Four 2x2 lay-in troffers. |
| | Lamps | Two 12’ T-8 lamps per fixture | Three T-8, U-tube lamps per fixture |
| | Controls | Photosensor-controlled continuous dimming from 40% to 100% of full output | None |
| **WINDOWS** | | | |
| Transmittance | *visible* | 23% | 73% |
| | *solar energy* | 14% | 52% |
| | *ultraviolet* | 5% | 36% |
| | Shading coefficient | 0.26 | 0.76 |
| | Solar Heat Gain Coefficient (SHGC) | 0.22 | 0.66 |
| U-value | *winter night time* | 0.31 | 0.33 |
| | *summer daytime* | 0.33 | 0.35 |
| | Blinds | Removed | Down, angled up 45° |

FIGURE 3, DAYLIGHTING AND STANDARD CONFIGURATIONS (SOUTH ROOMS).



The two sets of rooms were operated identically in terms of space heating and cooling. Each set of rooms is served by a separate HVAC system. The Energy Resource Station has the flexibility to operate in a number of HVAC configurations. For the purposes of this project, the system was configured as a typical variable-air-volume (VAV) system with hydronic reheat.

Cooling is provided by a single 10-ton (nominal) air-cooled chiller, which supplies chilled glycol solution to separate central coils for each air handler loop. The system was configured to economize cooling energy by introducing outdoor air when outdoor conditions were amenable.

Heating is provided by a single condensing boiler that provides hot water to reheat coils in the VAV boxes for each room.

Each set of rooms has separate air distribution systems that comprise supply and return fans, economizer dampers to control the introduction of outdoor air, and dampers for each room. The VAV system works by modulating airflow to the rooms based on the demand for heating or cooling from the thermostats in each room, with a minimum flow of 200 cfm to each room. The heating and cooling setpoint schedule is shown in Table 2.

For the test, ventilation air was based on 15 cfm per person with an assumed occupancy of six people per room. The outdoor air damper was set at a fixed position to achieve this ventilation rate at 1,800 cfm total supply air.

In addition, 1 kW of electric resistance heat was introduced into each room between 8:00 a.m. and 6 p.m. to simulate internal heat gains from equipment and people.

FIGURE 4. VIEW OF WINDOWS DURING INSTALLATION (EAST ROOM).

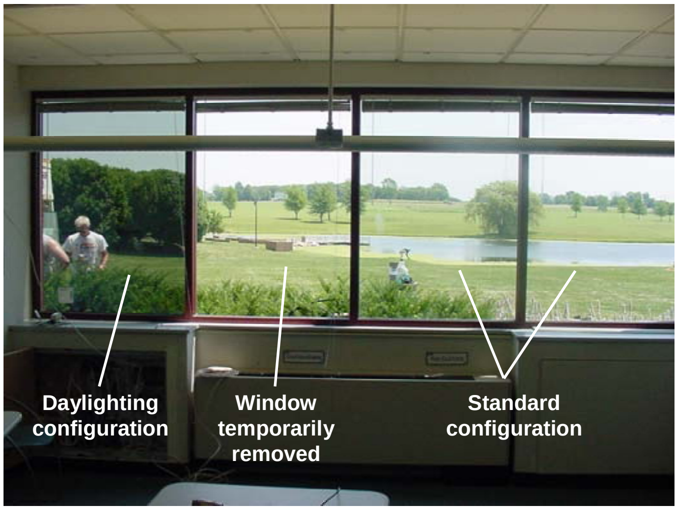


FIGURE 5. HVAC LAYOUT.

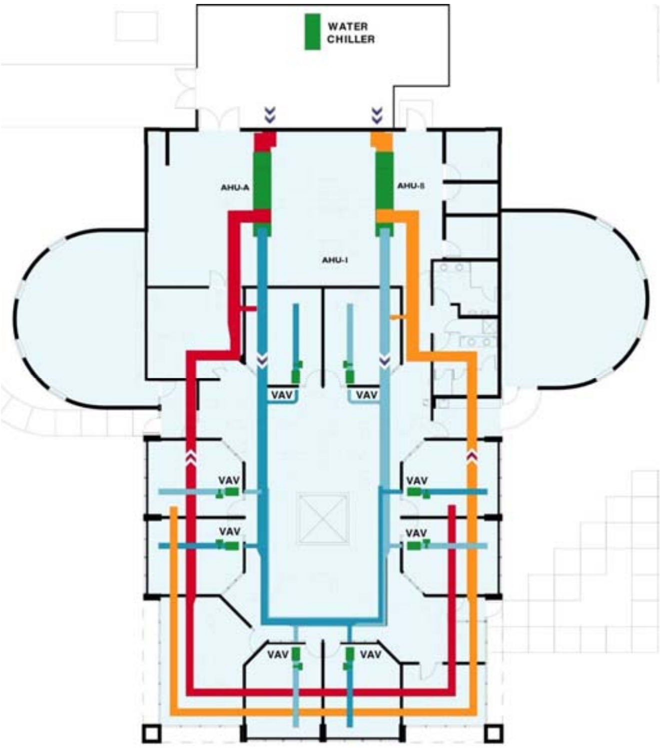


TABLE 2. HEATING AND COOLING SCHEDULE.

| | Occupied Period (07:00 — 18:00) | Unoccupied Period (18:00 — 07:00) |
|---|---|---|
| Heating Setpoint ($^oF$) | 72 | 60 |
| Cooling Setpoint ($^oF$) | 75 | 80 |

## TEST ROUNDS

Three rounds of testing to capture seasonal variation were conducted for the project:

- Summer (July 11 to August 7)
- Fall (September 23 to October 26)
- Winter (December 9 to January 8)

Within each round, three slightly different configurations of the test rooms were employed for roughly one-week each:

- Base case
- Reduced fenestration
- Interior light shelf.

The base case was as described above. The daylighting rooms were simply operated with blinds up and photosensor dimming.

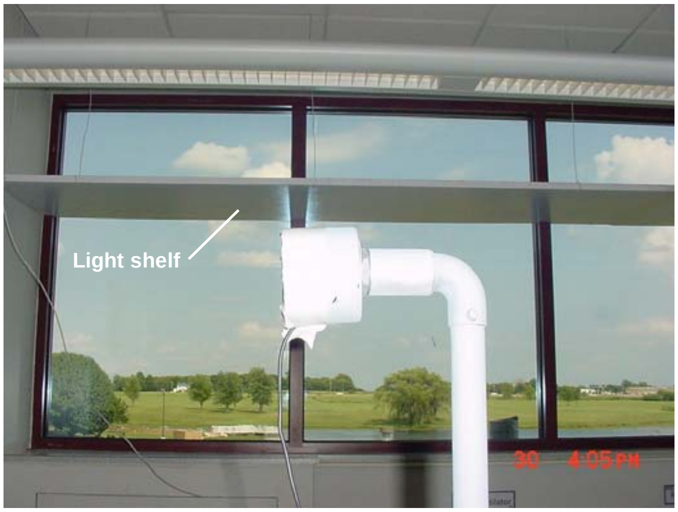
For the second configuration, exterior panels were used to effectively reduce the window area in the daylighting rooms by about one third (Figure 6). These panels had an insulating value approximately equivalent to the wall sections they were meant to simulate.

In the third configuration, a temporary interior light shelf was created to help daylight reflect more deeply into the rooms (Figure 7).

The strategy behind these configurations was to allow for additional comparisons across the daylighting configuration variants. However, while the overall experimental design allowed for a direct comparison between the test and control rooms at any point in time under identical weather conditions,

FIGURE 6. REDUCED FENESTRATION CONFIGURATION.

FIGURE 7. LIGHT SHELF CONFIGURATION.



comparisons across the daylighting variants were subject to differences in weather conditions, since they were implemented sequentially within each test round.

## MONITORING

As noted above, the Energy Resource Station is highly instrumented, since it is designed for experiments such as this one. Data are routinely collected and stored for approximately 600 parameters at the facility, many of which were not relevant for this project. Key monitoring points for this project are as follows:

- Lighting energy — average lighting power draw by room.
- Interior light levels — several vertical and horizontal illuminance levels captured for each room (see Appendix A).
- Space cooling load — calculated at the overall "A" and "B" loop level based on an energy balance across the chilled-water coils using flow rate and temperature difference across the coil.
- Space heating load — calculated for each room based on an energy balance across the hydronic re-heat coils for each room.
- Air flow — measured for loop supply and return (from which outdoor ventilation air can be deduced), as well as for VAV supply to each room.
- HVAC energy — power draw for all fans and pumps; also power draw for single air-cooled chiller serving both sets of rooms.
- Indoor conditions — temperature and relative humidity recorded for each room.
- Outdoor conditions — temperature and relative humidity; also exterior light levels (by compass direction) and solar beam radiation.

All of these parameters were captured and recorded as one-minute averages.

## DATA RECOVERY

Table 3 shows the test periods in 2003 by round and test condition. Two of the periods were slightly less than a week in length, but several others approached two weeks

TABLE 3. DATA COLLECTION PERIODS

| | **DAYLIGHTING CONFIGURATION** | | |
|---|---|---|---|
| **ROUND** | Base | Reduced Fenestration | Light Shelf |
| Summer | 7/11 – 7/18 (8 days) | 7/24 – 7/29 (6 days) | 7/31 – 8/7 (8 days) |
| Fall | 9/23 – 9/29 (7 days) | 10/1 – 10/13 (13 days) | 10/16 – 10/26 (11 days) |
| Winter | 12/9 – 12/21 (13 days) | 12/24 – 12/29 (6 days) | 1/1 – 1/8 (8 days) |

One additional notable issue arose with respect to calculated cooling loads. During the first part of the summer test round, a valve was inadvertently left open. This created chilled-water circulation flows such that the recorded flow through the chilled-water coils did not accurately reflect the actual flow. Energy Resource Center staff were able to work out correction factors to translate the recorded flow into actual flow for this period (see Appendix B).

## RESULTS

Results are presented below in terms of lighting, cooling and heating energy end-uses. The final sub-section summarizes overall differences in estimated operating costs.

### LIGHTING ENERGY

As expected, lighting energy was substantially lower for the test rooms with direct/indirect fixtures and dimming controls. Table 4 shows the average operating wattage for each room, by test round and test condition.[1] Electric lighting in the control rooms averaged 353 watts of power consumption during operating hours, with less than one percent variation across rooms and test period. Overall, the test rooms averaged 208 watts of power draw during operating hours, indicating an average savings of 41 percent relative to the control rooms. These savings are diluted to 32 percent when the interior rooms—which were identical for the test and control setups, and account for a quarter of the floor space—are included.

These lighting energy savings varied by season, room orientation, and test condition. As Figure 8 demonstrates, the length of the day and the angle of the sun are predictable determinants in the ability for the daylighting system to reduce electric lighting use. Savings are thus highest in the summer when the days are longer, and somewhat higher for the south room, which receives more sunlight.

TABLE 4. MEAN OPERATING WATTAGE, BY ROOM, TEST ROUND, AND TEST CONDITION.

| **(Mean operating wattage)** | | **Summer** | | | **Fall** | | | **Winter** | | |
|---|---|---|---|---|---|---|---|---|---|---|
| | | Base | Red. fen. | Light shelf | Base | Red. fen. | Light shelf | Base | Red. fen. | Light shelf |
| **East** | Control room | 353 | 355 | 354 | 352 | 353 | 353 | 354 | 354 | 354 |
| | Test room | 147 | 191 | 159 | 184 | 213 | 220 | 251 | 290 | 267 |
| | Difference | 205 | 163 | 195 | 168 | 141 | 133 | 103 | 63 | 87 |
| | % difference | 58% | 46% | 55% | 48% | 40% | 38% | 29% | 18% | 25% |
| **South** | Control room | 351 | 352 | 351 | 351 | 353 | 353 | 353 | 353 | 354 |
| | Test room | 174 | 206 | 174 | 157 | 167 | 175 | 206 | 221 | 238 |
| | Difference | 178 | 146 | 178 | 195 | 185 | 177 | 147 | 132 | 116 |
| | % difference | 51% | 41% | 51% | 55% | 53% | 50% | 42% | 37% | 33% |
| **West** | Control room | 348 | 349 | 349 | 354 | 355 | 355 | 355 | 355 | 355 |
| | Test room | 179 | 214 | 186 | 203 | 227 | 232 | 231 | 270 | 243 |
| | Difference | 169 | 135 | 163 | 151 | 128 | 123 | 125 | 85 | 112 |
| | % difference | 49% | 39% | 47% | 43% | 36% | 35% | 35% | 24% | 31% |

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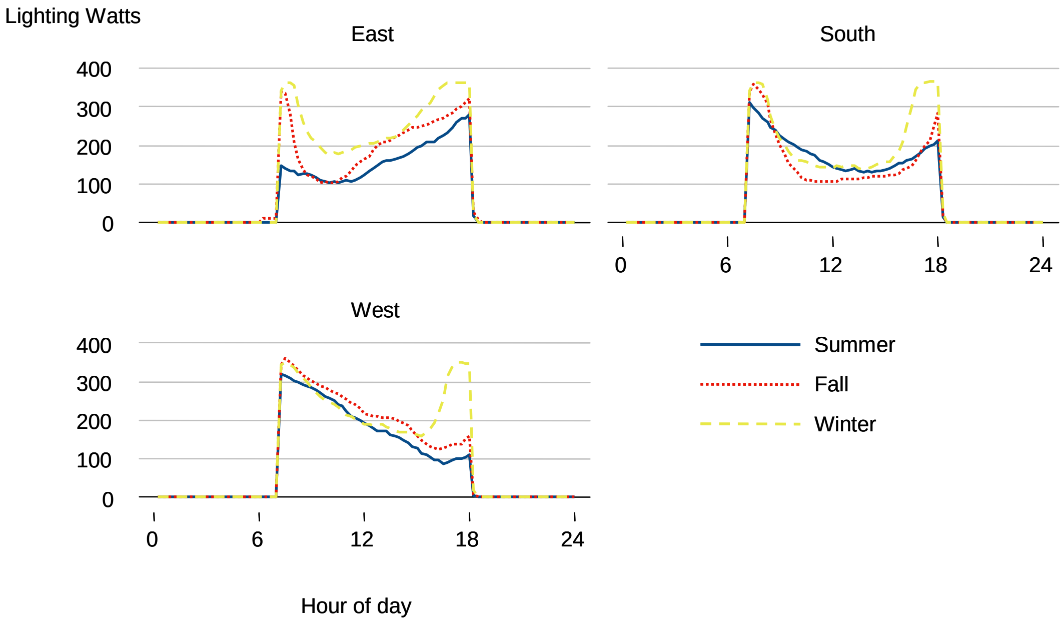
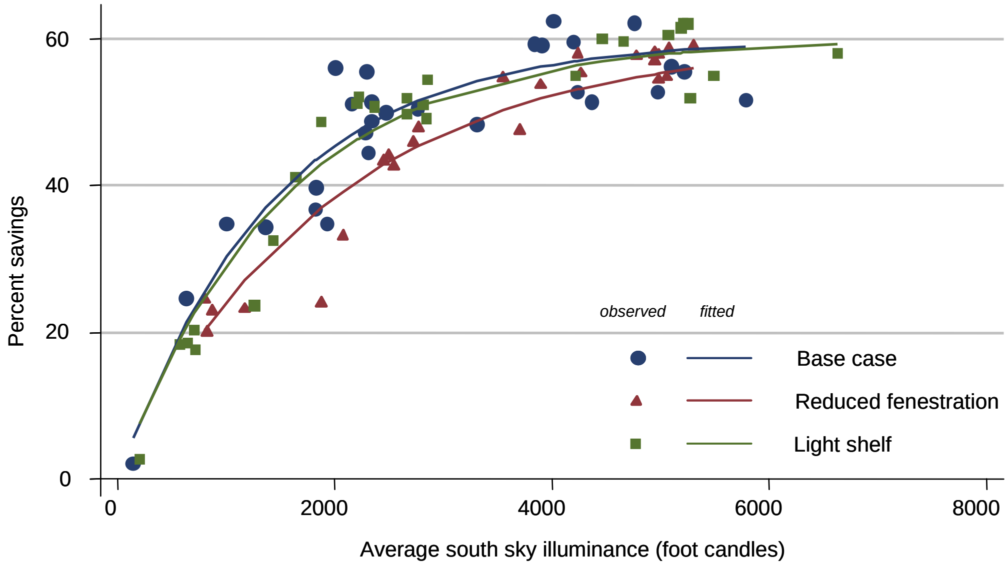
[1] The operating wattage for the two interior rooms is omitted from the table since these rooms were not a part of the experimental setup (though they do contribute to the overall HVAC load). These rooms averaged about 530 watts of operating lighting power, with a difference of less than one percent between the test room and the control room.

FIGURE 8. AVERAGE TEST-ROOM LIGHTING WATTAGE PROFILE, BY ROOM AND SEASON.



Differences in lighting energy use across the three test conditions (base case, reduced fenestration and light shelf) were minor. As Figure 9 shows, the reduced-fenestration test case exhibited somewhat lower savings for given sky conditions than the other two test conditions. Because differences in lighting energy across the test conditions were small, we did not distinguish among these in subsequent analyses.

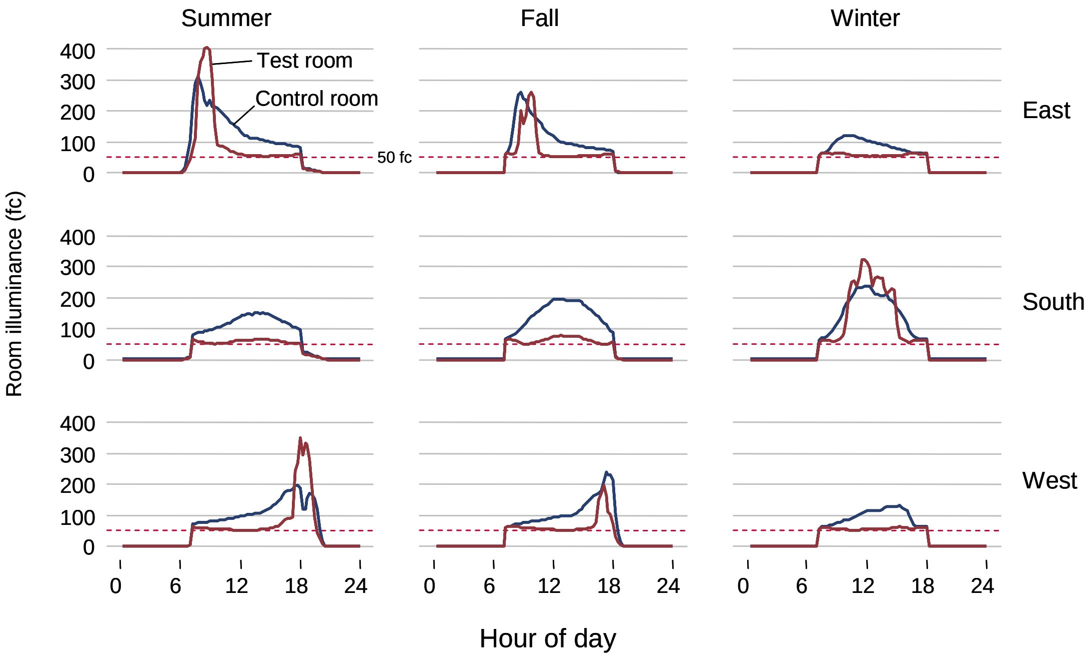
FIGURE 9. DAILY LIGHTING ENERGY SAVINGS VERSUS EXTERIOR LIGHT LEVEL, SOUTH ROOM



## LIGHT LEVELS

Center-of-room, vertical illuminance profiles are shown in Figure 10.[2] Light levels in the control rooms were generally higher than the target 50 foot-candles because of daylight penetration into the rooms, which were configured for 50 foot-candles of uncontrolled electric lighting. Light levels in the test rooms were at about 50 foot-candles except during periods of direct solar beam penetration through the windows.

FIGURE 10. AVERAGE CENTER-OF-ROOM VERTICAL ILLUMINANCE PROFILE, BY TEST ROUND AND ROOM



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[2] These measurements were made at 30 inches above floor level for the control rooms and 9 inches above floor level for the test rooms. The difference was designed to account for differences in the height of the ceiling lighting fixtures in the former rooms and the suspended direct/indirect fixtures in the latter.

## HVAC ENERGY

### Cooling

Because the most accurate measure of cooling energy use was at the system level, we analyzed the load on the central chiller coil for each set of rooms.

As Figure 11 shows, cooling energy was used for temperatures down to about 45$^{o}$F, with a reasonably linear relationship between cooling load and outdoor temperature.

To assess cooling energy savings, we directly modeled the observed *difference* in daily average cooling load between the two sets of rooms. To do so, we used a two-slope function that allowed for a separate savings relationship when the systems were in the temperature range for economizer operation. Figure 12 shows the relationship that best fits the daily data. As the figure shows, cooling energy was substantially lower for the test rooms in hot weather: the observed difference in cooling load between the test rooms and the control rooms averaged about 45 percent on hot days that reached above 90$^{o}$F, but was only about 10 percent for days where the temperature reached into the 70s.

FIGURE 11. DAILY AVERAGE COOLING LOAD VERSUS OUTDOOR TEMPERATURE, TEST AND CONTROL ROOMS

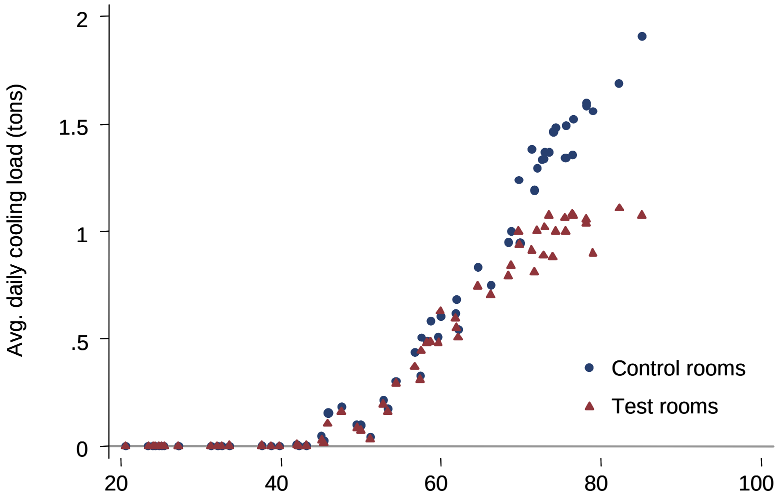
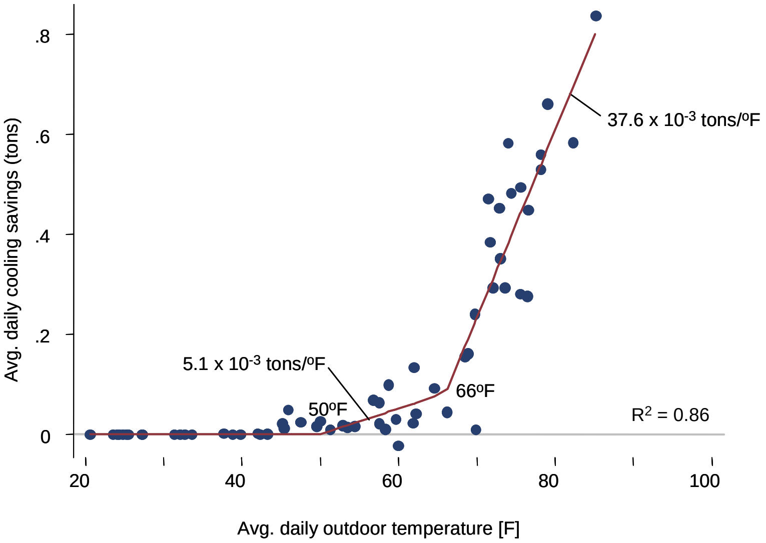


FIGURE 12. AVG. DAILY COOLING LOAD SAVINGS VERSUS OUTDOOR TEMPERATURE.



These savings derive from three differences between the two configurations: (1) reduced need to remove heat from dimmed electric lighting; (2) reduced heat gain through the high-performance windows; and, (3) reduced cooling required to condition ventilation air, which increases as the other cooling loads increase. Under hot conditions, the last two factors dominate, as electric lighting represents only about 10 percent of the building's cooling load on 90$^{o}$F+ days. Indeed, the data suggest that about half of the cooling energy savings arise from reduced need to condition outdoor air introduced by the system.

The differential impact of direct solar gain through the windows is exemplified in the fact that on a number of warm summer evenings, solar gain through the window of the west control room pushed the room temperature above the unoccupied-mode setpoint of 80°F, which triggered additional cooling operation for the control rooms. While temperature in the west test room also rose during these periods, it never exceeded the thermostat setpoint. Figure 13 demonstrates this effect for July 25, a sunny day in which the outdoor temperature peaked at about 90°F.

FIGURE 13. ROOM TEMPERATURES AND COOLING LOAD ON JULY 25.

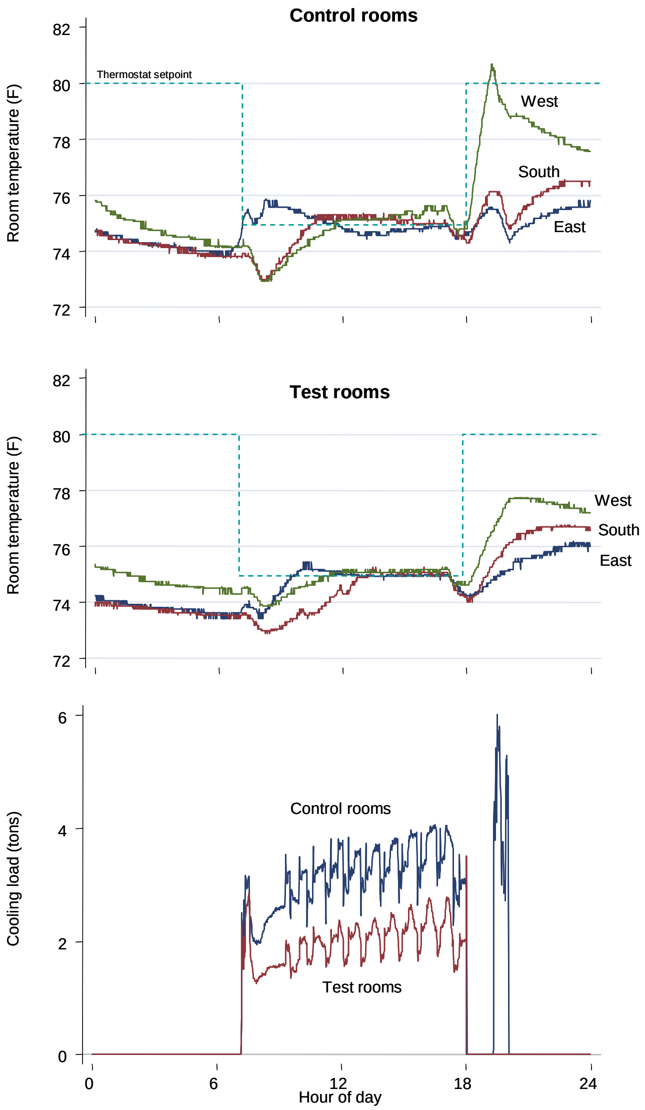
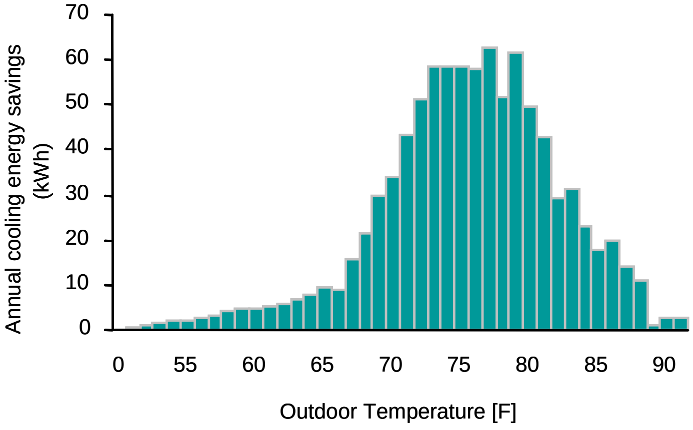


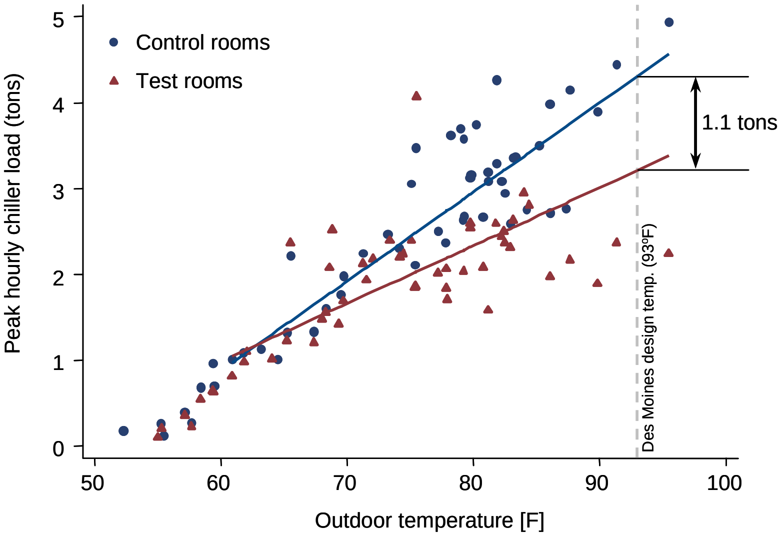
FIGURE 14. ESTIMATED ANNUAL COOLING ENERGY SAVINGS BY OUTDOOR TEMPERATURE.



We combined the relationship shown in Figure 12 with long-term temperature data for Des Moines Iowa to estimate annual cooling energy savings. This analysis also took into account the fact that the efficiency of the chiller tends to go down as outdoor temperature goes up.[3] The results indicate that the test rooms will average about 25 percent less cooling energy than the control rooms, for a savings of about 0.83 kWh/ft$^2$.[4] Most of these savings occur on days when the temperature averages between 70 and 80$^{o}$F (Figure 14).

## Chiller Sizing

FIGURE 15. PEAK HOURLY CHILLER LOAD VERSUS OUTDOOR TEMPERATURE.



Reduced cooling loads from daylighting strategies such as the one tested in this project means that a smaller chiller can be installed in daylit buildings. To analyze the implications of the cooling load savings for our test, we examined differences in peak hourly chiller loads and outdoor temperatures for the test- and control-room systems. For the majority of days, this peak load occurred between 2 and 5 pm in the afternoon when outdoor temperature tended to be highest.

At the Des Moines 0.4% design temperature of 93$^{o}$F, the data suggest that the test rooms have about a 26 percent lower peak cooling load than the control rooms (Figure 15). In terms of square feet per ton of cooling load, this represents a 34 percent increase, from about 260 to 340 square feet per ton.

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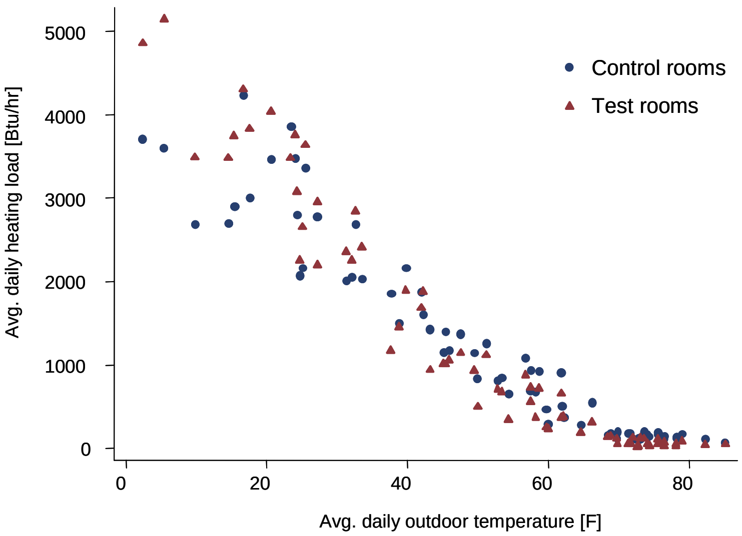
[3] The exact relationship (derived from hourly chiller electricity consumption and cooling output data) was: chiller EER = 18.62 – 0.126*outdoor temperature.

[4] Note that the savings per square foot includes the interior rooms, which were operated identically. Expressed per square foot affected by the daylighting, the savings were 1.1 kWh/ft$^2$.

### Heating Energy

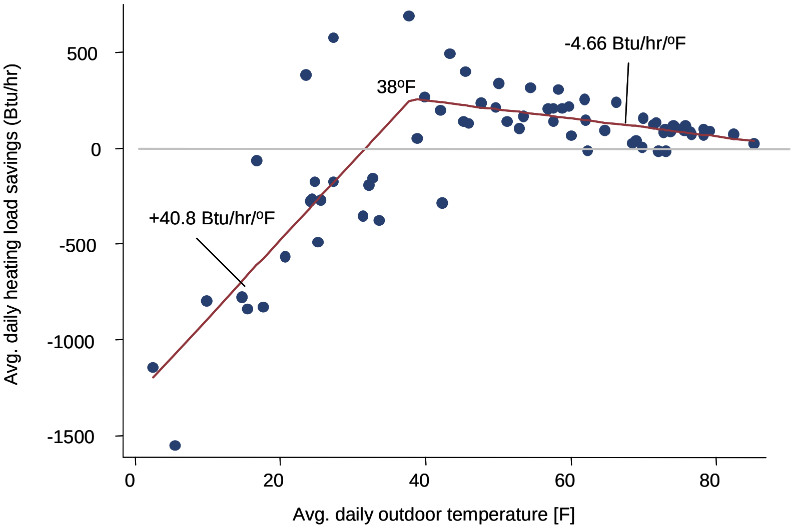
We analyzed heating energy use in a fashion similar to cooling energy; that is, by modeling total daily heating energy consumption versus outdoor temperature. Figure 16 shows daily average heating load versus outdoor temperature for the test and control room systems. The data indicate that heating energy is used on days when the outdoor temperature averages about 68°F or less. On warmer days, heating is mostly needed for re-heat in the VAV system when some rooms are calling for cooling but others are not; on days when the heating load on the building exceeds the building balance-point temperature, heating is needed to maintain the heating setpoint temperature.

FIGURE 16. AVERAGE DAILY HEATING LOAD VERSUS OUTDOOR TEMPERATURE, TEST AND CONTROL ROOMS.



As with the analysis of cooling energy, we directly analyzed the difference in heating energy requirements between the test rooms and the control rooms as a function of outdoor temperature (Figure 17). As one would expect, the reduced internal loads in the test rooms mean that additional heating is required when the building is in heating mode. However, the data also reveal that less heating energy is needed for re-heat purposes at warmer temperatures. This appears to be due to reduced cooling loads in the exterior rooms requiring less re-heat energy for the interior room to prevent overcooling.

FIGURE 17. AVERAGE DAILY HEATING SAVINGS VERSUS OUTDOOR TEMPERATURE.



When combined with long-term temperature data for Des Moines, the data suggest that there is only a small net difference between the test and control rooms in terms of heating energy needs. Extra heating requirements for the test rooms under the relatively infrequent cold conditions are nearly offset by reduced heating needs for cooling-mode reheat (Figure 18). The end result is an estimated 1 percent difference between the two sets of rooms in heating energy requirements.

## Fan Energy

Each set of rooms is served by a supply and return air handler. Fan energy tends to be highest under cold and hot conditions when heating and cooling loads are more extreme, and lowest at moderate temperatures (Figure 19).

The data suggest that the test rooms have somewhat lower fan energy when the system is in cooling mode and the outdoor temperature averages more than 66°F (Figure 20); at other temperatures there is no statistically significant difference in fan energy between the two sets of rooms. Extrapolated to long-run Des Moines temperatures, the analysis indicates about 3 percent lower fan energy consumption for the test rooms, or about 0.06 kWh per square foot per year.

FIGURE 18. ESTIMATED ANNUAL HEATING ENERGY SAVINGS BY OUTDOOR TEMPERATURE.

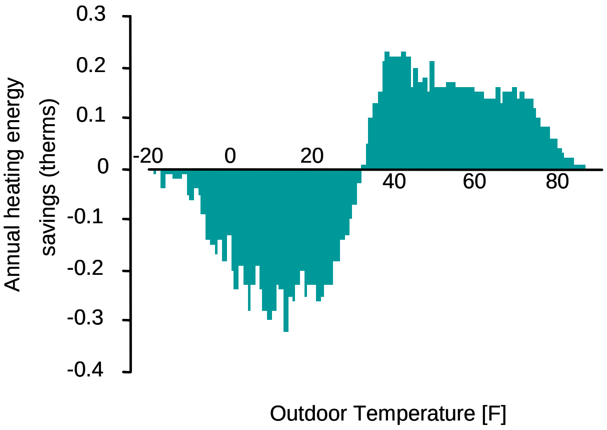


FIGURE 19. AVERAGE DAILY FAN POWER, TEST AND CONTROL ROOMS.

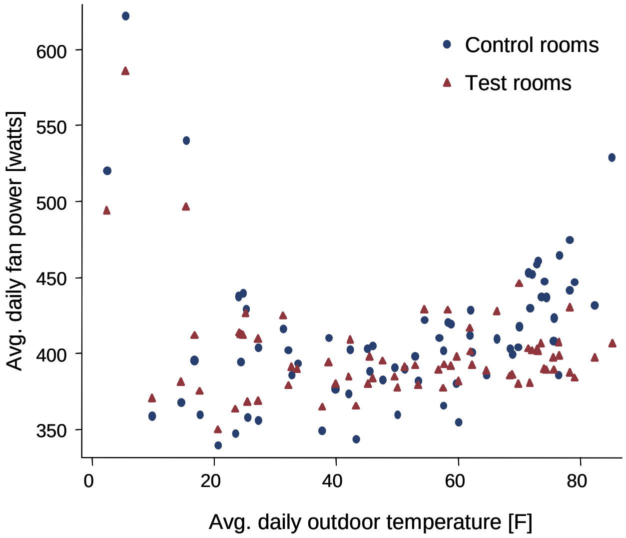
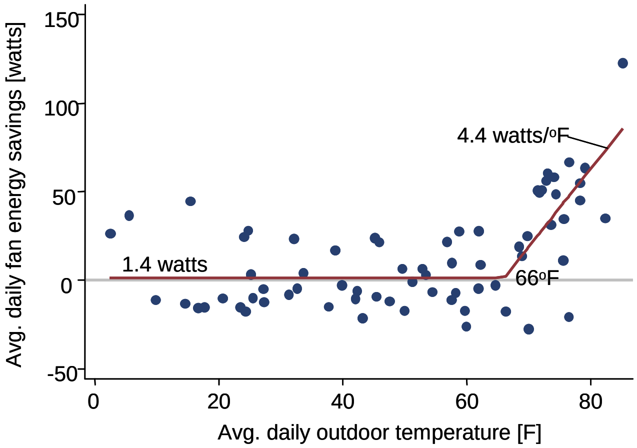


FIGURE 20. FAN ENERGY SAVINGS BY OUTDOOR TEMPERATURE.



## DEMAND CHARGE SAVINGS

Most commercial facilities incur utility charges not only for the electrical energy used in the building but also for the maximum power draw, or demand. These charges are typically divided into a monthly demand charge for the maximum 15-minute average power draw during the billing period and a rolling annual (or ratchet) charge for the largest power draw during the preceding year. Chiller electrical load and lighting are often key contributors to peak demand charges. We analyzed the difference in monthly peak and annual rolling demand between the two sets of rooms as follows:

1. We assumed a 25 percent smaller chiller for the test rooms compared to the control rooms.
2. We collapsed the 1-minute data to 15-minute averages for chiller load, HVAC fan and lighting electrical demand.
3. We analyzed the chiller load data for each 15-minute period. If cooling load was present but was half or less than the full capacity of the assumed chiller size, we assigned one-half the full chiller power draw to the 15-minute period; if the cooling load was more than half the full output capacity, we assigned the full chiller power draw to the period. This step was intended to reflect the operation of the two-stage chiller at the site. Chiller power draw was also adjusted for the empirical variation in efficiency with outdoor temperature, as noted previously.
4. We then combined the lighting, chiller power and air handler 15-minute demand values, and found the maximum total electrical demand for each day.
5. Finally, we analyzed differences in the maximum daily lighting and HVAC demand between the test and control rooms, and extrapolated these to typical monthly weather for Des Moines.

While it is possible that peak demand could be determined by end uses other than lighting and HVAC in a given building, typically it is these end-uses that drive demand charges. Differences between the test and control rooms in the setup here thus are probably a reasonable indicator of demand charge savings.

Figure 21 shows the resulting estimates of daily peak demand as a function of temperature, and Figure 22 shows the difference between the test and control rooms for the same. Chiller operation dominates the difference between the two sets of rooms, and there is little difference in peak demand on days with no chiller operation. While it would seem intuitive that there should be substantial demand savings from the electric lighting alone, in fact the daylighting system calls for close to full output at some point during the short days of the winter months.

To calculate annual demand savings, we applied the relationship in Figure 22 to long-term weather data for Des Moines. Specifically, we calculated the average maximum daily temperature for each month of the year, and calculated the peak demand and peak demand savings for this temperature (Table 5). The results indicate about 22 percent savings on monthly demand charges, and 29 percent savings on the rolling annual charges which typically involve the high cooling demand on hot summer days. Note that these savings are predicated on the assumption that a smaller chiller is installed for the daylighting configuration.

FIGURE 21. DAILY PEAK LIGHTING + HVAC DEMAND VERSUS OUTDOOR TEMPERATURE.

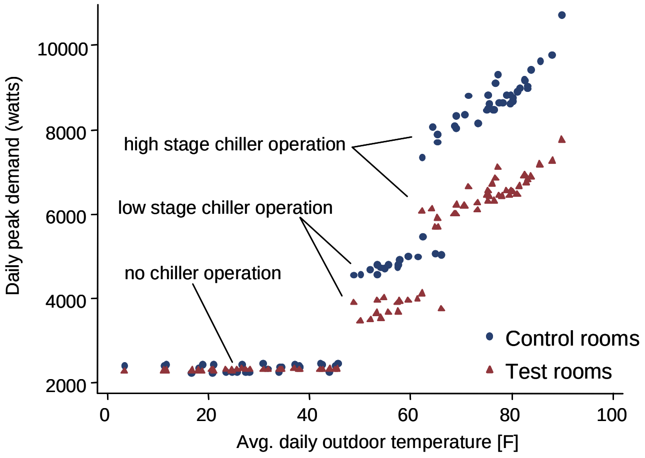


FIGURE 22. DAILY PEAK DEMAND SAVINGS VERSUS OUTDOOR TEMPERATURE.

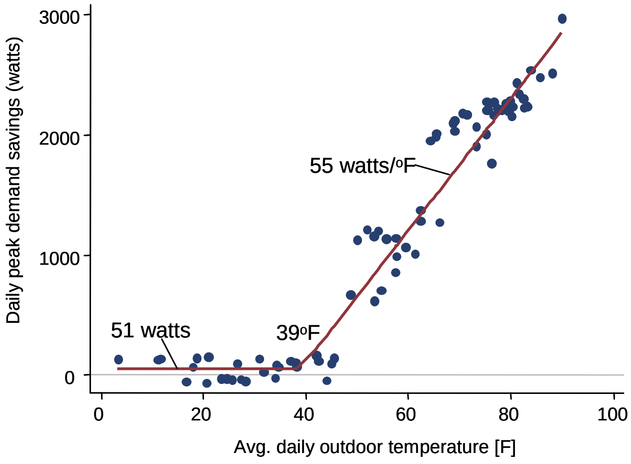


TABLE 5. MONTHLY AND ANNUAL ESTIMATED PEAK DEMAND AND SAVINGS.

| Month | Typical max. daily temp.[a] | Control room demand | Savings | |
|---|---|---|---|---|
| | (F) | (kW) | (kW) | % |
| Jan | 39.3 | 2.4 | 0.1 | 3% |
| Feb | 44.4 | 4.4 | 0.3 | 8% |
| Mar | 58.1 | 6.9 | 1.1 | 16% |
| Apr | 68.2 | 7.5 | 1.7 | 22% |
| May | 74.8 | 8.0 | 2.0 | 25% |
| Jun | 81.6 | 8.6 | 2.4 | 28% |
| Jul | 85.2 | 8.9 | 2.6 | 29% |
| Aug | 82.9 | 8.7 | 2.5 | 28% |
| Sep | 78.7 | 8.3 | 2.2 | 27% |
| Oct | 69.8 | 7.6 | 1.7 | 23% |
| Nov | 55.8 | 4.6 | 1.0 | 21% |
| Dec | 43.5 | 4.3 | 0.3 | 7% |
| Annual | Mean | 6.7 | 1.5 | 22% |
| | Max | 8.9 | 2.6 | 29% |

[a] Represents the average maximum of daily average temperatures.

## OVERALL OPERATING COST SAVINGS

Using the savings estimates described in the previous sections, we applied typical Midwestern utility rates for commercial buildings (Table 6) to calculate the total operating cost savings between the test and control rooms.

The results are shown in Table 7 in terms of operating costs and savings per square foot. Overall the calculations suggest about 22 percent savings for the test rooms, or about 24 cents per square foot savings on annual lighting and HVAC operating costs of a bit more than a dollar per square foot. Demand charge savings—stemming mainly from reduced cooling loads—make up nearly half of this amount. Energy savings for lighting and cooling make up most of the rest of the savings.

TABLE 6. UTILITY RATES USED IN THE ANALYSIS.

| | | | |
|---|---|---|---|
| Electricity | on-peak energy | 6 | cents/kWh |
| | off-peak energy | 3 | cents/kWh |
| | monthly peak demand | 6 | $/kw |
| | annual peak demand ratchet | 1 | $/kw |
| Gas | | 75 | cents/therm |

TABLE 7. ANNUAL OPERATING COSTS AND SAVINGS PER SQUARE FOOT.

| | Control room annual operating costs (cents/ft$^2$) | Test room savings (cents/ft$^2$) | Percent |
|---|---|---|---|
| Lighting energy | 21.6 | 6.8 | 32% |
| Cooling energy | 19.2 | 4.8 | 25% |
| Heating energy | 6.1 | -0.1 | -1% |
| Fan energy | 13.0 | 0.3 | 3% |
| Demand charges | 53.3 | 12.5 | 24% |
| Total | 113.2 | 24.4 | 22% |

# DISCUSSION

The data from this experiment demonstrate clear and substantial reductions in lighting and HVAC energy consumption due to the lighting and window specifications. About two-thirds of the operating cost savings are due to reduced cooling loads in the building, and much of the cooling load reduction appears to be attributable to the high-performance windows, which have a solar heat gain coefficient that is one-third that of the windows in the control rooms.

The high performance windows—and to a lesser extent, the lighting controls—affect cooling both directly by reducing the amount of heat introduced into the conditioned space, and indirectly as reduced cooling loads translate first into reduced VAV system airflow and then into reduced outdoor air drawn into the system. It is somewhat surprising that about half of the cooling load savings in the test rooms are attributable to reduced need to condition ventilation air. This suggests that strategies to actively control the amount of outdoor air drawn into the system have the potential to further reduce cooling energy consumption. Alternatively, this angle also suggests that savings from the daylighting strategies tested here may be less in buildings that use active control of the outdoor-air damper to mitigate the introduction of excessive outdoor air as the need for cooling rises.

It is also important to note that half of the overall operating cost savings arise from the estimated reduction in monthly demand charges. These estimates are based on the important assumption that a 25 percent smaller chiller would be installed in buildings that use the daylighting strategy tested here. Without such downsizing, these demand charge savings would mostly be eliminated, since a larger chiller draws more power when it operates, even if it operates less frequently. In addition, the analysis here considers only HVAC and lighting loads. In buildings, actual occupants and plug-load equipment, and electrical demand for other end-uses no doubt affects the overall electrical demand profile, particularly during colder months. This could affect the savings estimates produced here.

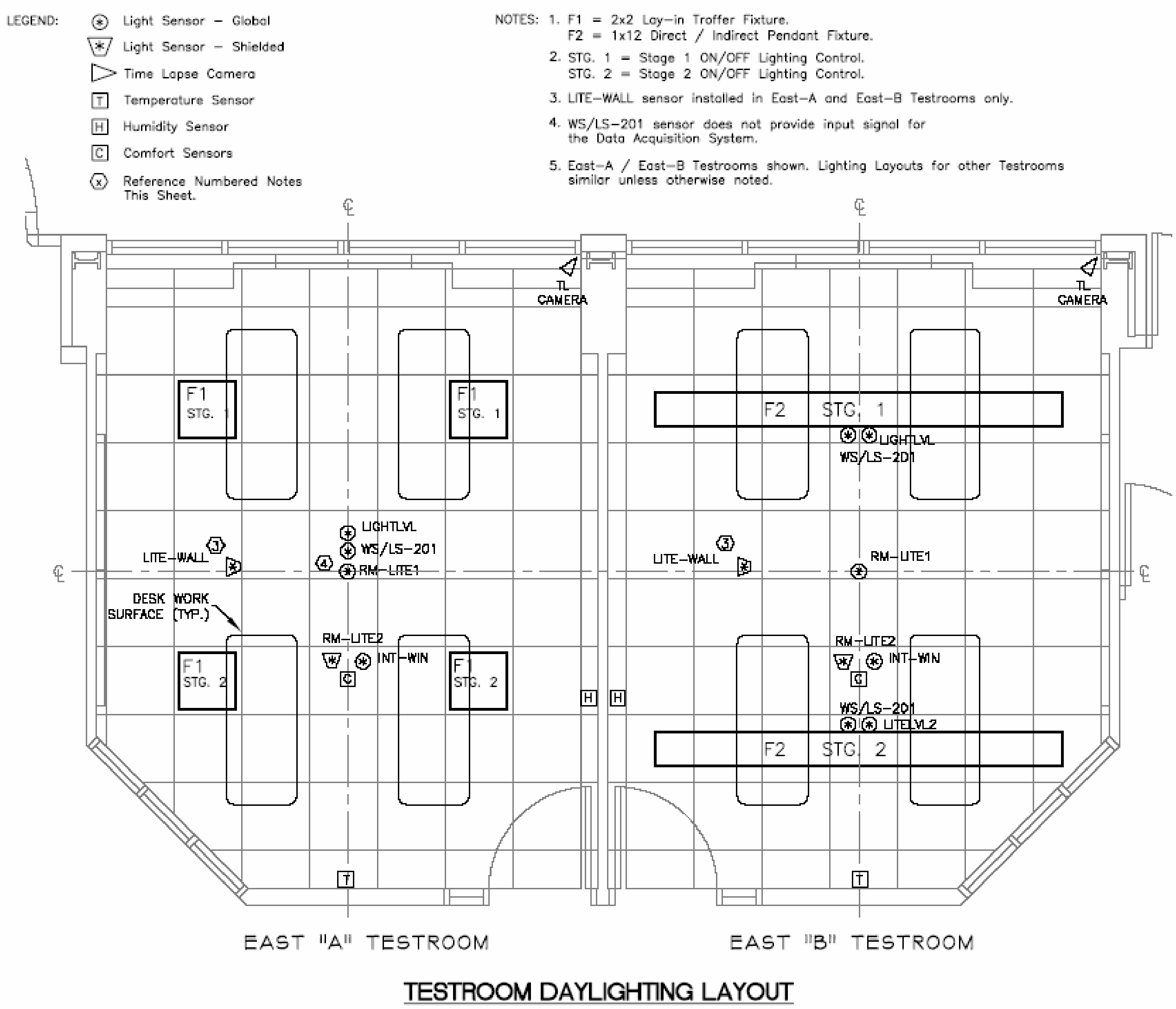
Note also that the results here reflect a building geometry in which 75 percent of the square footage is in perimeter areas amenable to side daylighting, and 25 percent is in building core areas without side daylighting capability. Savings would undoubtedly be different for buildings where these ratios differ.

In terms of lighting energy, the observed lighting energy savings, while substantial, could be increased further by employing controls that shut the lights off entirely when daylight levels are sufficient. This also raises the question of the use of blinds with reduced visible-transmission glass and daylighting controls. The experiment for this project effectively assumed no use of blinds, and therefore maximum daylight harvesting; in reality, use of blinds might mitigate the lighting savings.

The results here suggest several avenues for additional research. These include:

- More field research on the use of blinds in buildings with reduced visible-transmission glass and daylighting controls.
- Assessing how lighting and space-cooling affect demand charges in typical buildings.
- More field research on the actual operation of outdoor air dampers in commercial buildings.

## APPENDIX A: ROOM LAYOUTS



NOTES:
1. F1 = 2x2 Lay-in Troffer Fixture.
   F2 = 1x12 Direct / Indirect Pendant Fixture.
2. STG. 1 = Stage 1 ON/OFF Lighting Control.
   STG. 2 = Stage 2 ON/OFF Lighting Control.
3. LITE-WALL sensor installed in East-A and East-B Testrooms only.
4. WS/LS-201 sensor does not provide input signal for the Data Acquisition System.
5. East-A / East-B Testrooms shown. Lighting Layouts for other Testrooms similar unless otherwise noted.

TESTROOM DAYLIGHTING LAYOUT

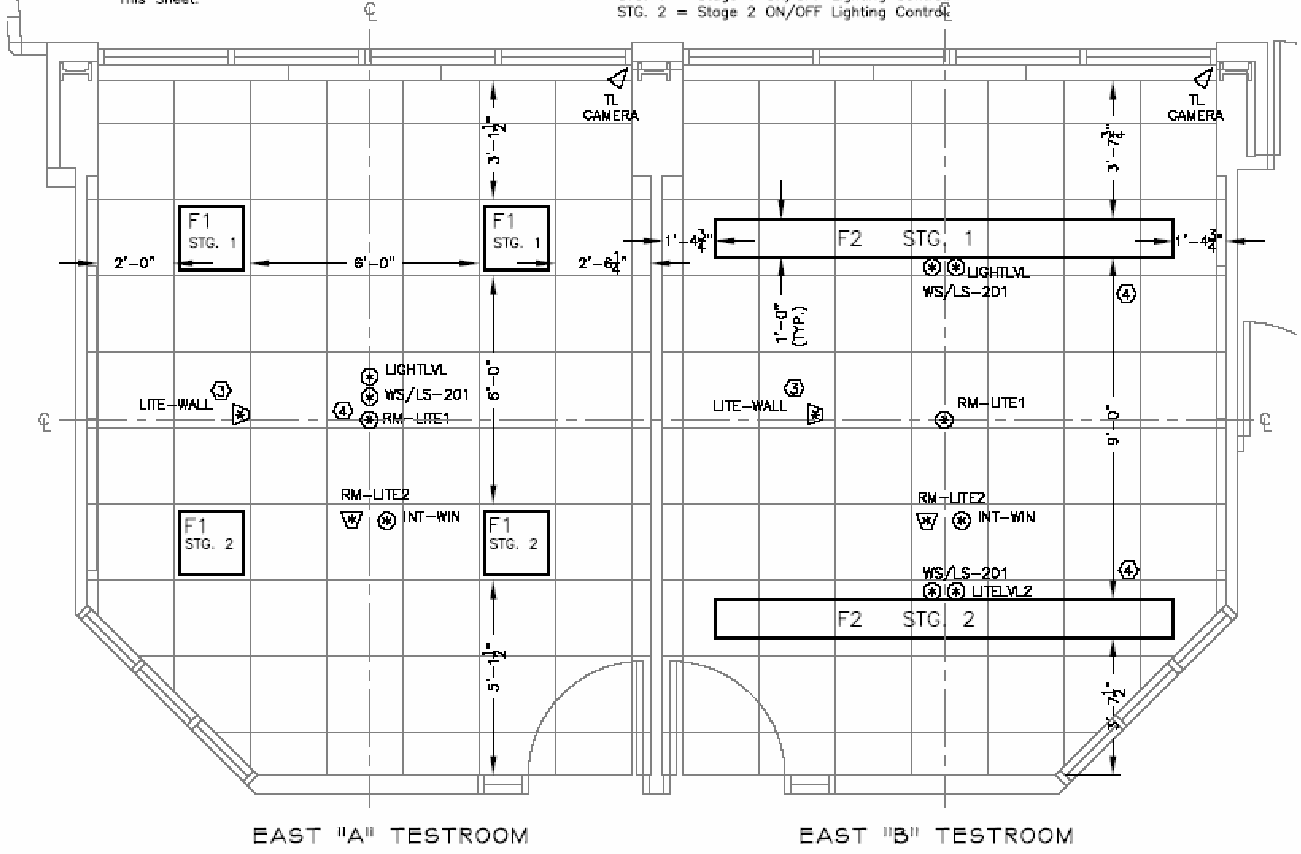
**Testroom Daylighting Layout**
**Daylighting Performance**
**NASEO / ECW / LBL / IEC**

**ENERGY RESOURCE STATION**
Data Date: 07/08/03 By: DLP
Drawing Date: 08/05/03 By: DLP
Revision Date: By:

SHEET NO.
**ECW**
**DL-0**

LEGEND:
- ⊛ Light Sensor – Global
- ⊛ Light Sensor – Shielded
- ▷ Time Lapse Camera
- [T] Temperature Sensor
- [H] Humidity Sensor
- [C] Comfort Sensors
- ⓧ Reference Numbered Notes This Sheet.

NOTES:
1. Dimensions shown are to the face of wall, window or edge of light fixture. AFF is the abbreviation for Above Finished Floor.
2. East–A / East–B Testrooms shown. Lighting Layouts for other Testrooms similar unless otherwise noted.
3. LITE–WALL sensor installed in East–A and East–B Testrooms only.
4. F1 = 2x2 Lay–in Troffer Fixture.
   F2 = 1x12 Direct / Indirect Pendant Fixture.
5. ℄ is Testroom centerline.
6. STG. 1 = Stage 1 ON/OFF Lighting Control.
   STG. 2 = Stage 2 ON/OFF Lighting Control.

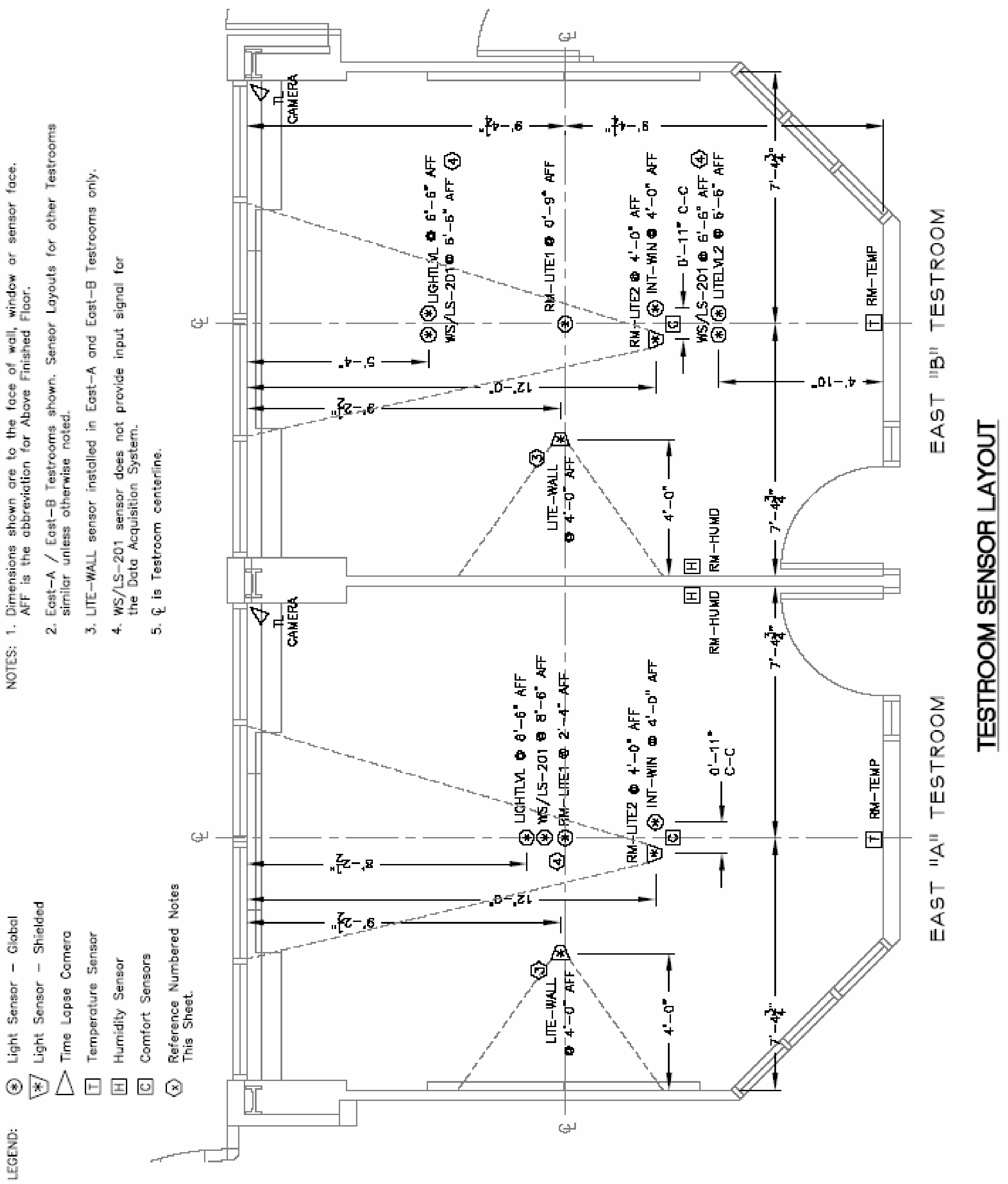


**TESTROOM LIGHTING LAYOUT**

**Testroom Lighting Layout**
**Daylighting Performance**
**NASEO / ECW / LBL / IEC**

**ENERGY RESOURCE STATION**
Data Date: 07/08/03 By: DLP
Drawing Date: 08/05/03 By: DLP
Revision Date: By:

SHEET NO.
**ECW**
**DL-1**



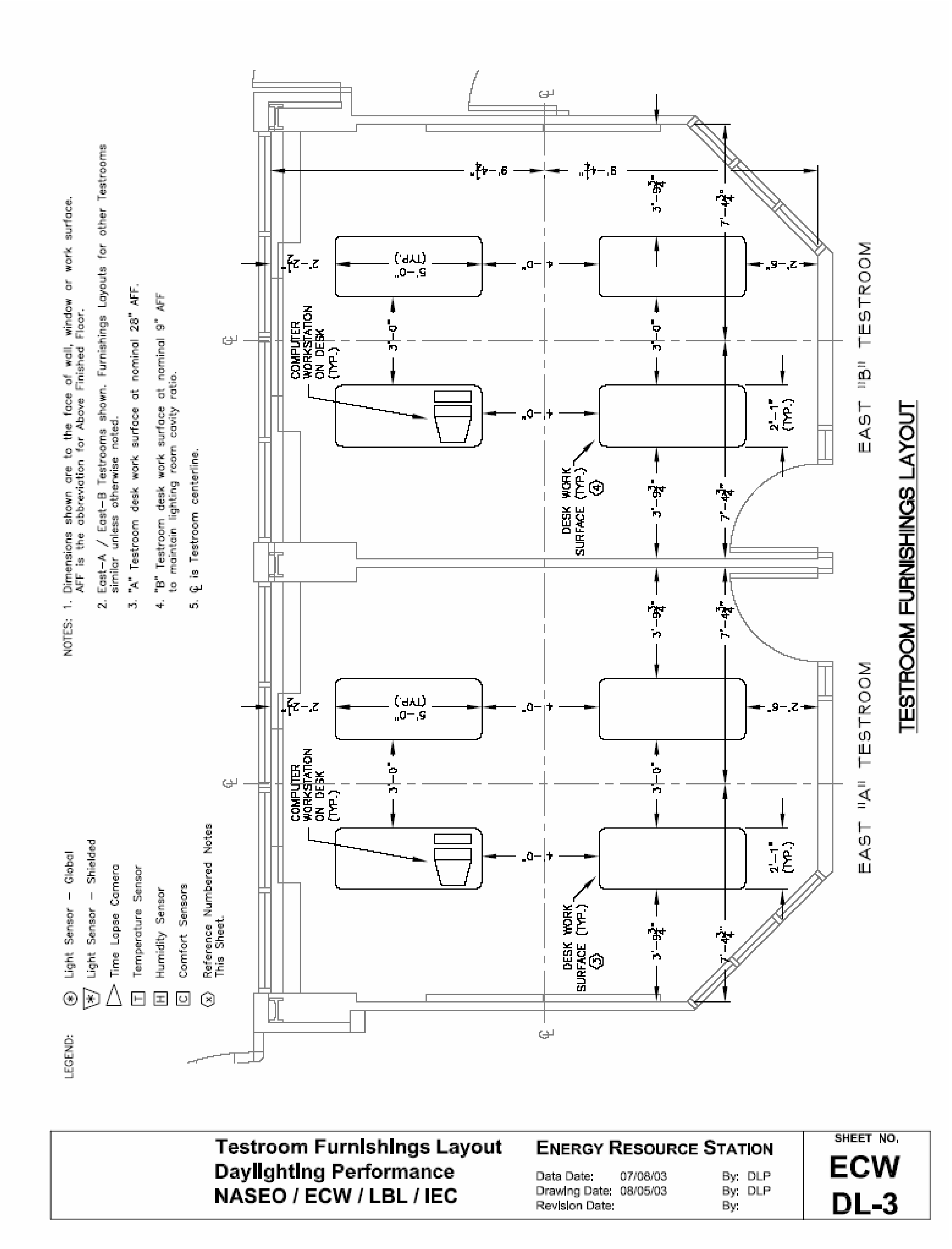
LEGEND:
- ⊛ Light Sensor – Global
- Light Sensor – Shielded
- △ Time Lapse Camera
- T Temperature Sensor
- H Humidity Sensor
- C Comfort Sensors
- Reference Numbered Notes This Sheet.

NOTES:
1. Dimensions shown are to the face of wall, window or sensor face. AFF is the abbreviation for Above Finished Floor.
2. East-A / East-B Testrooms shown. Sensor Layouts for other Testrooms similar unless otherwise noted.
3. LITE-WALL sensor installed in East-A and East-B Testrooms only.
4. WS/LS-201 sensor does not provide input signal for the Data Acquisition System.
5. ℄ is Testroom centerline.

**Testroom Sensor Layout**
**Daylighting Performance**
**NASEO / ECW / LBL / IEC**

**ENERGY RESOURCE STATION**
Data Date: 07/08/03 By: DLP
Drawing Date: 08/05/03 By: DLP
Revision Date: By:

SHEET NO.
**ECW**
**DL-2**



NOTES:
1. Dimensions shown are to the face of wall, window or work surface. AFF is the abbreviation for Above Finished Floor.
2. East-A / East-B Testrooms shown. Furnishings Layouts for other Testrooms similar unless otherwise noted.
3. "A" Testroom desk work surface at nominal 28" AFF.
4. "B" Testroom desk work surface at nominal 9" AFF to maintain lighting room cavity ratio.
5. ℄ is Testroom centerline.

LEGEND:
- ⊛ Light Sensor – Global
- ▽ Light Sensor – Shielded
- △ Time Lapse Camera
- T Temperature Sensor
- H Humidity Sensor
- C Comfort Sensors
- ⓧ Reference Numbered Notes This Sheet.

TESTROOM FURNISHINGS LAYOUT

**Testroom Furnishings Layout**
**Daylighting Performance**
**NASEO / ECW / LBL / IEC**

ENERGY RESOURCE STATION
Data Date: 07/08/03 By: DLP
Drawing Date: 08/05/03 By: DLP
Revision Date: By:

SHEET NO.
**ECW**
**DL-3**

## LIGHTING SENSOR SCHEDULE

| SYMBOL | POINT NAME | DESCRIPTION | MANUFACTURER & PART NO. | MOUNTING HEIGHT | ORIENTATION | FIELD OF VIEW | VIEWING |
|---|---|---|---|---|---|---|---|
| ⊛ | INT-WIN | LIGHTING SENSOR, PHOTOMETRIC, GLOBAL | MFR: LI-COR, MODEL LI-210SZ WITH EME MODEL UTA VOLTAGE OUTPUT AMPLIFIER | 4'-0" A.F.F. | HORIZONTAL | 180 DEG. GLOBAL | VIEWING ALL EXT. WINDOWS |
| ⊛ | LIGHTLVL | CELESTIAL AMBIENT LIGHTING SENSOR, INDOOR RANGE. | MFR: PLC. MODEL MK7-B-CCF-0/10. | 6'-6" A.F.F. | VERTICAL | 60 DEG. CONICAL | TOWARD FLOOR |
| ⊛ | LITELVL2 | CELESTIAL AMBIENT LIGHTING SENSOR, INDOOR RANGE. | MFR: PLC. MODEL MK7-B-CCF-0/10 | 8'-6" A.F.F. | VERTICAL | 80 DEG. CONICAL | TOWARD FLOOR |
| ⏢ | LITE-WALL | LIGHTING SENSOR, PHOTOMETRIC, BLINDERS INSTALLED. | MFR: LI-COR. MODEL: LI-210SZ. | 4'-0" A.F.F. | HORIZONTAL | SHIELDED | TOWARD INT. WALL |
| ⊛ | RM-LITE1 | LIGHTING SENSOR. PHOTOMETRIC. | MFR: LI-COR. MODEL: LI-210SZ. | TESTROOMS:<br>"A": 2'-4" A.F.F<br>"B": 0'-9" A.F.F. | VERTICAL | 180 DEG. CONICAL | TOWARD CEILING |
| ⏢ | RM-LITE2 | LIGHTING SENSOR. PHOTOMETRIC, BLINDERS INSTALLED. | MFR: LI-COR. MODEL: LI-210SZ | 4'-0" A.F.F. | HORIZONTAL | SHIELDED | TOWARD 2 CENTER EXT. WINDOWS |
| ▷ | | TIME LAPSE CAMERA | MFR: LOGITECH. MODEL: QUICKCAM PRO 4000 | 3'-0" A.F.F. | HORIZONTAL | CONICAL, ADJUSTABLE | TOWARD INT. WALL |
| ⊛ | | WS / LS-201 LIGHT LEVEL SENSOR / CONTROLLER. | MFR: THE WATT STOPPER, INC. MODEL: LS-201 | 6'-6" A.F.F. | VERTICAL | 60 DEG. CONICAL. | TOWARD FLOOR |

## LIGHTING FIXTURE SCHEDULE

| FIXT. NO. | DESCRIPTION | FIXTURE SIZE | MANUFACTURER & MODEL NO. | MOUNTING | LAMPS | BALLASTS | REMARKS |
|---|---|---|---|---|---|---|---|
| F1 | FLUORESCENT LAY-IN CEILING TROFFER | 2'x2' | MFR: H. E. WILLIAMS, INC. CAT. NO.: EPG-D263-FWKA125 | RECESSED 8'-6" A.F.F. | (3) U-TUBE LAMPS PER FIXTURE, SYLVANIA-OSRAM T8 LAMPS FB031/841. COLOR TEMP. 4100 K | 277 V, USED WITH (1) ADVANCE MARK VII VZT-3S32 ELEC. DIMM. BALLAST PER FIXTURE. | 0.125" ACRYLIC LENS. |
| F2 | FLUORESCENT SUSPENDED INDIRECT LINEAR FIXTURE | 1'x12' | MFR: LEDALITE ARCH. PRODUCTS MODEL: ACHIEVA 8616T02WN1221EW | SUSPENDED 8'-6" A.F.F. | (6) LAMPS PER FIXTURE. SYLVANIA-OSRAM T8 LAMPS FO32/841/ECO. COLOR TEMP. 4100 K | 277 V, USED WITH (2) ADVANCE MARK VII VZT-3S32 ELEC. DIMM. BALLASTS PER FIXTURE. | WHITE LOUVER WITH 80% LIGHT UP, 20% DOWN OPTICS. |

**Testroom Schedules**
**Daylighting Performance**
**NASEO / ECW / LBL / IEC**

**ENERGY RESOURCE STATION**
Data Date: 07/08/03 By: DLP
Drawing Date: 08/05/03 By: DLP
Revision Date: By:

SHEET NO.
**ECW**
**DL-4**

## APPENDIX B: EVALUATION OF CHILLED WATER FLOWRATES

# ENERGY RESOURCE STATION

## CHILLED WATER FLOWRATE EVALUATION

### ECW Daylighting Tests, Summer 2003

September 21, 2004

1. Objective

The purpose of this document is to evaluate questionable chilled water flowrate values that had been collected during the summer months of 2003 at the ERS. The Energy Center Wisconsin Daylighting (ECW DL) Tests had been underway during this time period and their energy balance analysis had found potential errors in air handler chilled water flow measurements during the course of their test.

This document will investigate the data set provided for this test, evaluate the accuracy of such data, and analyze logged operations activities which may have affected this test.

2. Values Evaluated

Two points associated with branch chilled water flowrates, i.e., AHU-A \ CHWP-GPM and AHU-B \ CHWP-GPM, were both found to be inaccurate over a specific time period. The reasons for the inaccuracies are noted below.

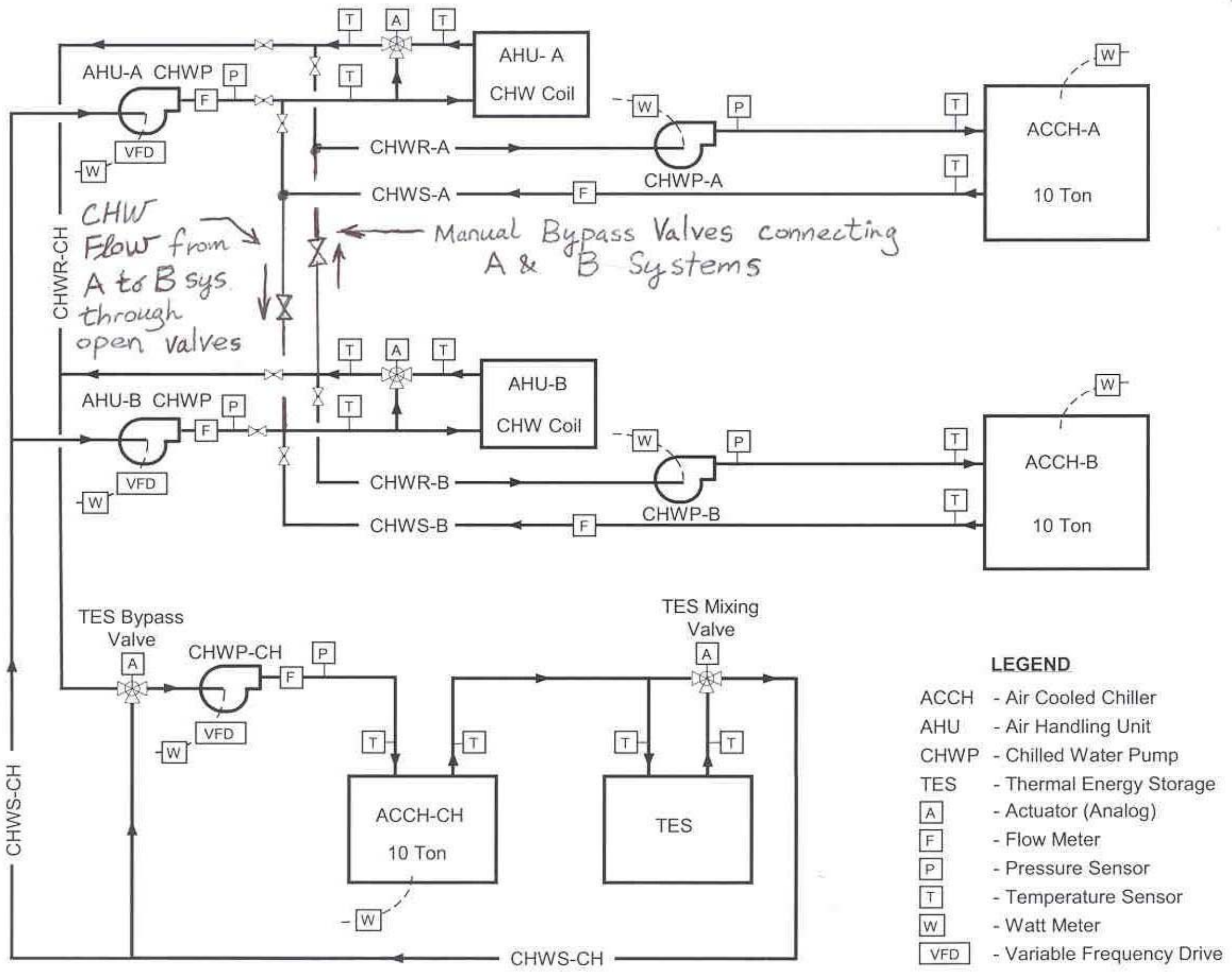
3. Reasons for Inaccurate Readings

Two factors contribute the inaccuracy of these flow measurements. The primary reason for the inaccuracies was that a set of two-way manual bypass valves had been left in an open position for part of this test, allowing a portion of the A system flow to be diverted to the B system branch. See attached image of chilled water system showing these manual bypass valves.

These bypass valves needed to be closed for this test in order to ensure that the A and B system flow readings being reported would actually represent the flows being received at each chilled water coil circuit. With the bypass valves open, the individual A and B system measured chilled water flowrates would not truly represent their respective air-handler chilled water coil flowrates.

The second factor contributing to flowrate inaccuracies were bias errors associated with our flow measurement instrumentation. The chilled water system contained biological contaminants and corrosion products previous to 07/30/03, potentially causing problems in flow measurement. The chilled water system, including mag-meter flowheads, had been cleaned and flushed on 07/30, improving our ability to accurately measure liquid flowrates from before.

# ENERGY RESOURCE STATION – CHILLED WATER SYSTEM



**LEGEND**

| | |
|---|---|
| ACCH | - Air Cooled Chiller |
| AHU | - Air Handling Unit |
| CHWP | - Chilled Water Pump |
| TES | - Thermal Energy Storage |
| A | - Actuator (Analog) |
| F | - Flow Meter |
| P | - Pressure Sensor |
| T | - Temperature Sensor |
| W | - Watt Meter |
| VFD | - Variable Frequency Drive |

**NOTES**

1. Test Systems AHU A and AHU-B provided with chilled water from either dedicated chillers ACCH-A and ACCH-B *or* by common chiller ACCH-CH.
2. ACCH-CH / Thermal Energy Storage system configuration shown as Chiller Priority. Valve manifold allows Chiller Priority or TES Priority configuration.
3. Design Chilled Water Supply and Return Temperature: 44 °F CHWS / 54 °F CHWR. Heat Transfer Fluid: nominal 21% Propylene Glycol solution.

## 4. Calculations of Actual Flowrates

Review of test data surrounding 07/28/03 reveals some important information. Because the primary and secondary, if applicable, loop pumps were operated at fixed speeds, their flowrates should be reasonably constant during each air handler operating mode.

Since the A and B systems are the only two chilled water systems available, the sum of A and B system chilled water flowrates should be a constant. No other flow path is available, unless leakage is present.

The average flowrates for the A and B systems, along with their average summation have been calculated for several days surrounding 07/28. Flow reading changes attributed to bypass valve closures have been evaluated as simple biases to previous flow readings. Bias errors were present in the data posted before 19:00 07/28 because, during that time, a small amount of flow had been diverted in a bypass circuit, as stated previously.

## 5. Post-processing of Chilled Water Flowrates

Two relevant water flow conditions occurred during this test. The first condition existed during the air handler Occupied mode of operation, (from time 07:00 – 18:00), and the second condition existed when the air handlers were placed in the Setback mode, (06:00 – 07:00 and 18:00 – 23:59).

The chiller had been commanded to an ice making mode of operation during part of this test, (from time 0:00 – 06:00). During the ice make mode, the air handlers were segregated from the ice storage operations and the A and B system chilled water flowrates were zero. No corrections are necessary for chilled water flowrate data that was sampled during the ice make operation.

Both of the Occupied and Setback conditions require the application of chilled water flowrate corrections in their dataset, as shown below. Table 1 gives the offsets that need to be applied to chilled water flowrate data for various times during their respective modes of air handler operation. Posted data previous to 07/31 should be corrected, as shown in Table 1.

Data posted for date 07/30 will probably be unsuitable for research use because maintenance activities were performed at that time, disrupting steady-state flow conditions. Chilled water flow readings taken during or after 07/31/03 should be accurate.