Contract Report

Evaluating the Impacts of Uncontrolled Air Flow and HVAC performance Problems on Florida's Commercial and Institutional Buildings

Final Report

FSEC-CR-1210-00
October 31, 2000

Contract Number
FEO: 97-SE-20-06-15-05-271

Submitted to:
Edward A Cobham, Jr.
Florida Energy Office
2555 Shumard Oak Blvd.
Tallahassee, FL 32399

Submitted by:
James B. Cummings
Don B. Shirey
Charles Withers
Richard Raustad
Neil Moyer

Florida Solar Energy Center
1679 Clearlake Road
Cocoa, Florida 32922

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Submitted by:
James B. Cummings
Don B. Shirey, III
Chuck Withers
Richard Raustad
Neil Moyer

Florida Solar Energy Center
1679 Clearlake Road
Cocoa, FL 32922-5703

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ABSTRACT

Twenty commercial and institutional buildings located in central Florida were tested for uncontrolled air flow and cooling system problems. Tested buildings were selected based on three criteria; 1) buildings with large exhaust fans, 2) buildings with leaky ceilings, and 3) education buildings. They ranged in size from a 650 square foot portable classroom to a 50,000 square foot elementary school, with an average floor area of 6518 square feet.

Diagnostic inspection and testing was performed in each building to identify problems related to uncontrolled air flow or HVAC system or control problems. The tests were performed to characterize or identify building airtightness, HVAC flow rates, building ventilation, air flow and pressure imbalances, cooling system performance, and HVAC control strategy. Short-term monitoring (typically for a week or two) of indoor parameters such as temperature, relative humidity, ceiling space temperature, carbon dioxide levels, and indoor pressure was implemented in seven buildings. After testing and inspection (and in some buildings short-term monitoring), a report was written for each building summarizing test results and presenting recommendations for correcting any deficiencies. These reports are found in Appendix B.

Following testing, a variety of retrofits were performed on six buildings. Improvements in cooling energy use and indoor temperature/humidity control resulting from the repairs were monitored. One or more retrofits were implemented in each building. Depending upon the match between cooling load and cooling capacity, some retrofits reduced cooling energy use, others reduced only indoor temperatures (toward greater comfort), and others reduced both cooling energy use and indoor temperature. In one of the six buildings, monitoring was begun and the first retrofit (cooling system tune-up) implemented when the facility changed ownership. As a result of major renovations performed by the new owner which greatly changed the thermal characteristics of both the building and the HVAC systems, this monitoring effort was abandoned because there would be no opportunity to identify the change in energy use resulting only from the retrofits implemented in this project.

A variety of retrofits were implemented, including application of a foam insulation material to both tighten the building and improve the thermal barrier (four buildings), change the color of roof and exterior walls to white (one building), correct unbalanced return air problems (one building), reduce the number of hours of cooling system operation (one building), repair duct leaks (one building), addition of make-up air (one building), service (tune up) the cooling systems to bring refrigerant charge and air flows into design specification (two buildings), and install a dedicated outdoor air pre-conditioning system for improved humidity control (one building).

EXECUTIVE SUMMARY

Twenty commercial and institutional buildings located in central Florida were tested for uncontrolled air flow and cooling system problems. Following testing, a variety of repairs were performed on six buildings. Improvements resulting from the repairs were monitored. Following is a summary of the findings from this work.

Of the twenty buildings tested, ten were education buildings, eight were restaurants, one a government-owned community center, and one a charity office/thrift shop. These buildings were selected based on three criteria: 1) buildings with large exhaust fans, 2) buildings with leaky ceilings, and 3) education facilities. These selection criteria were established at project initiation and were incorporated in the project work plan.

Table 1
Twenty buildings tested for uncontrolled air flow and HVAC failures.

BUILDING DESCRIPTION	LARGE EXHAUST	LEAKY CEILING	EDUCATION
1. Apollo elementary school	x		x
2. Chicken cafeteria restaurant	x		
3. Community center / clubhouse	x	x	
4. Floral Avenue elementary school		x	x
5. Eagle Lake elementary school		x	x
6. Sea food chain restaurant	x		
7. River front restaurant	x		
8. Steak house chain restaurant	x		
9-12. Kennedy middle school portables		x	x
13-15. McNair middle school portables		x	x
16. Bar and grill	x	x	
17. Convenience store / fast food	x	x	
18. Ocean front restaurant	x	x	
19. Greek restaurant	x	x	
20. Charity office / thrift shop		x	

Ten of the twenty buildings had large exhaust fans, including all eight restaurants plus the community center which had a restaurant and large dining room, and a 50,000 square foot elementary school which had a kitchen.

In thirteen buildings, leaky ceilings were identified as an important issue. Sixteen buildings actually have leaky ceilings, but in three cases the leakiness of the ceiling was not an issue either because the ceiling space was not vented or there was little or no uncontrolled air flow interacting with the ceiling space. Seventeen of the twenty had suspended T-bar ceilings over part or all of the building, two buildings (portables) had large vinyl ceiling panels, and one was entirely gypsum board.

Interestingly, the one with gypsum board (bar and grill) was included among those with a leaky ceiling because the soffit area above bar was very leaky.

Ten of the twenty buildings fall into the category of education buildings. Seven of these are portables, located at two middle schools. The average floor area of the portables is 718 square feet.

Overview of Buildings with Large Exhaust Fans

In the ten buildings with large exhaust fans, the large exhaust fans are associated exclusively with food preparation. The presence of large exhaust fans in buildings tends to cause depressurization of the building unless the mechanically induced air flows into the building are greater than the exhaust air flow. The average exhaust fan flow rate in the ten buildings was 5976 cfm. On average, combined make-up air and outdoor ventilation air was 4228 cfm, representing 71% of the exhaust flow rate. Six of the ten buildings had mechanical make-up air, six of the ten had outdoor ventilation air, and four of the ten had both make-up air and outdoor ventilation air.

In all cases, these buildings operated at negative pressure (with respect to outdoors) with the kitchen exhaust (and make-up air/outdoor ventilation air, if available) operating, which in most cases was the normal mode of building operation. The degree of space depressurization depends upon the net exhaust flow rate (exhaust air flow less incoming air flow) and the airtightness of the building.

- The greatest level of depressurization occurred in the two restaurants with the tightest building envelope as indicated by ACH50 (air changes per hour with the building at a pressure of 50 pascals). The river front restaurant with a very tight envelope (3.8 ACH50) operated at -45 pascals and the chicken restaurant with a moderately tight envelope (6.7 ACH50) was at -22.8 pascals. Pressure in the remainder of these ten buildings fell in the range of -0.2 pascals to -3.8 pascals.

- The two buildings with the greatest envelope leakage (31.8 ACH50 and 34.4 ACH50) experienced building pressure of -0.2 pascals and -1.3 pascals. Average ACH50 for all 10 buildings with large exhaust fans was 15.8 ACH50 and average building pressure was -8.0 pascals.

Space depressurization is of concern for several reasons. It can do the following:

- draw radon gas into the building
- create elevated ventilation rates which can cause elevated humidity levels during hot and humid weather
- draw high dew-point temperature outdoor air into the exterior wall cavities of the building. This humid air, when in contact with the cool gypsum wall board, creates a high humidity condition at the surface of the wall board which may cause growth of mold and mildew, especially when vinyl wall paper is on the wall surfaces
- draw hot and humid air into the building through a leaky suspended T-bar ceiling, which in many cases causes elevated heat, humidity, and comfort problems
- draw sewer gases into a building when drain traps in floor drains, sinks, etc. dry out or when toilets are not properly sealed to the sewer line

- draw air down combustion appliance vent pipes and blow out pilot lights
- slow the flow of air up the vent pipe of combustion appliances and therefore cause spilling of some fraction of combustion by-products into the space
- reverse the flow in the vent pipe (referred to as backdrafting) so that 100% of combustion by-products enter the space
- cause a sharp increase in the production of carbon monoxide in the combustion appliance. This is caused when reverse flow in the vent pipe (caused by space depressurization) slows flow up the flue (the flue is the vent inside the appliance) and deprives combustion of the necessary oxygen.
- cause flame roll out, a condition where the combustion of the fuel occurs outside the water heater combustion chamber, and flames lick up around the water heater jacket. This is caused when reverse flow in the vent pipe actually reverses flow in the flue and therefore pushes combusting fuel out of the combustion chamber.

Specific problems that were observed in these restaurant buildings included flame roll-out in the chicken restaurant on two occasions (as reported by the store owner/manager). Water heater backdrafting was reported at the river front restaurant (building at -45 pascals). Staff at the river front restaurant had responded to this problem by keeping a kitchen exterior door open during exhaust fan operation, and in this circumstance building pressure was only -2 pascals.

Drawing of attic air into the occupied space was causing serious problems in four restaurants; the bar and grill, the convenience store, the ocean front restaurant, and the Greek restaurant. The bar and grill had no make-up air for the approximately 1000 cfm of kitchen exhaust air. The convenience store had only 155 cfm of outdoor ventilation air for the 656 cfm of kitchen exhaust. The ocean front restaurant had no make-up air or outdoor ventilation air for the 2349 cfm of exhaust (except for passive make-up air vents in the kitchen). The Greek restaurant had 167 cfm of make-up air for the 871 cfm of exhaust air.

At the bar and grill, air was coming from the very hot residential style vented attic space through openings immediately above the bar, and this was causing comfort problems especially at the bar and also throughout the restaurant. The air-conditioning systems (17.5 tons for 4300 square feet) at the convenience store ran almost continuously through the day and staff complained of the space being too warm. Comfort problems and high humidity had been recurring problems at the ocean front restaurant. Additional air-conditioning systems had been installed in response.

In the Greek restaurant, the excessive amount of air drawn in from the unconditioned ceiling space and exterior soffit area (like an attic above the sidewalk) was causing indoor temperatures to routinely run at 80F to 85F in the dining area. The owner estimated that he had lost 50% of his clientele due to the heat and humidity. His business was suffering substantially and was on the brink of going out of business. At the time that we first began testing his building, he was installing an additional 3.5-ton air-conditioning unit to serve the dining area, bringing total building cooling capacity to 8.5 tons. In effect, the owner was using the "brute force" approach to solving the uncontrolled air flow problems in his building. Instead of correctly diagnosing the problem (i.e., unbalanced exhaust air because of undersized make-up air), he was throwing money and energy at the problem.

As part of this project, two retrofits were implemented at the bar and grill. First, a make-up air system was installed equal to 82% of the exhaust flow rate. Building pressure went from -2.0 pascals to -0.2 pascals. Total cooling energy use declined by 13.5% as a result of the make-up air. Second, a spray foam insulation product was applied to the bottom of the roof deck (R-11 thermal resistance rating) and the attic vents were sealed off by the foam insulation. Many of the insulation batts which had been attached to the bottom of the trusses had been pulled out of location because of various contractors performing work in the attic space. After the second retrofit, the attic space was now inside the air and thermal boundaries of the building. As a result, the attic space changed from being very hot and humid to warm and dry, and building ACH50 declined from 17.1 to 13.5 (the building became 21% tighter). Total cooling energy use declined by another 11.6% as a result of the foam insulation application which made the building more airtight and the insulation system more complete. No retrofits were performed at the Greek restaurant.

Overview of Buildings with Leaky Ceilings

Suspended T-bar ceilings are leaky. Even when installed with care and to manufacturer's specifications, they are inherently leaky because there are cracks on all four sides of each ceiling tile. In some cases, these ceilings are much more leaky for several reasons. Sometimes sections of ceiling tile are broken out, or are penetrated by wires or conduit. In some buildings, ceiling tiles are found to be sagging substantially because of elevated humidity, and this causes the interface between the ceiling tile and the T-bar grid to become more leaky. In other cases, there are speakers, sprinklers, recessed lights, and other features which cause increased leakage. Testing performed in three buildings has found that standard suspended T-bar ceilings have leakage of about 5 CFM50/square foot. This can also be thought of as 0.625 square inches of cumulative hole per square foot of ceiling surface area. In one of the test buildings, the community center, the ceiling of the auditorium section of the building had leakage of about 25 CFM50/square foot, five times greater than standard suspended ceilings. Adding to the leakiness were speakers and sprinkler heads, but the major factor causing the extreme ceiling leakiness was recessed ceiling coves with integrated light shelves. Light shelves commonly leak where the light shelf extends horizontally out from the ceiling cove. Where the metal studs of the light shelf penetrate the side wall, it creates a gap approximately 3 inches high by the length of the light shelf.

Even without the added leakage factors, suspended T-bar ceilings can only be thought of as a visual barrier with large holes in it. The Florida Energy Code recognizes the leakiness of suspended T-bar ceilings and greatly restricts their use. The following section is from the 1997 version of the Florida Energy code.

406.1.ABCD.1.4. Building Cavities.

406.1.ABCD.1.4.1 *Where vented dropped ceiling cavities occur over conditioned spaces, the ceiling shall be considered to be both the upper thermal envelope and pressure envelope of the building and shall contain a continuous air barrier between the conditioned space and the vented unconditioned space that is also sealed to the air barrier of the walls.*
IMPORTANT NOTE: See the definition of air barrier in section 202.

Where unvented dropped ceiling cavities occur over conditioned spaces that do not have an air barrier between the conditioned and unconditioned space (such as T-bar ceilings), they shall be *completely sealed from the exterior environment (at the roof plane) and adjacent spaces by a continuous air barrier that is also sealed to the air barrier of the walls. In that case, the roof assembly shall constitute both the upper thermal envelope and pressure envelope of the building.*

Unconditioned spaces above separate tenancies shall contain dividing partitions between the tenancies to form a continuous air barrier that is sealed at the ceiling and roof to prevent air flow between them.

406.1.ABCD.1.4.2 *Building cavities designed to be air distribution system components shall be sealed according to the criteria for air ducts, plenums, etc. in section 410.1.ABCD.3.6.*

Also from the 1997 Florida Energy Code is the definition of *Air Barrier*.

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AIR BARRIER---

Relating to air distribution systems, a material object(s) which impedes or restricts the free movement of air under specified conditions. For fibrous glass duct, the air barrier is its foil cladding; for flexible non-metal duct, the air barrier is the non-porous core; and for sheet metal duct and air handling units, the air barrier is the metal in contact with the air stream. For mechanical closets, the air barrier may be a uniform panelized material such as gypsum wall board which meets ASTM C36, or it may be a membrane which alone acts as an air barrier which is attached to a panel, such as the foil cladding of fibrous glass duct board.

Relating to the building envelope, air barriers comprise the planes of primary resistance to air flow between the interior spaces of a building and the outdoors and the planes of primary air flow resistance between adjacent air zones of a building, including planes between adjacent conditioned and unconditioned air spaces of a building. To be classed as an air barrier, a building plane must be substantially leak free; that is, it shall have an air leakage rate not greater than 0.5 cfm/ft^2 when subjected to an air pressure gradient of 25 pascals. In general, air barriers are made of durable, non-porous materials and are sealed to adjoining wall, ceiling or floor surfaces with a suitable long-life mastic. House wraps and taped and sealed drywall may constitute an air barrier but dropped acoustical tile ceilings (T-bar ceilings) may not. Batt insulation facings and asphalt-impregnated fiberboard and felt paper are not considered air barriers.

End of code language.

The Florida Energy Code requires the ceiling assembly to be 0.5 cfm/square foot or tighter when tested at 25 pascals (pressure across the assembly). This is approximately equivalent to 0.8 CFM50 per square foot. **Therefore, suspended T-bar ceilings are at best six times more leaky than the standard, and therefore cannot be used as the primary air barrier between the occupied space and a vented ceiling space.**

As noted, the leakiness of the suspended T-bar ceiling often contributes to the overall leakiness of the building. On average, the ACH50 of the twenty buildings tested during this project is 19.1, indicating that these buildings are very leaky. However, a clear difference is seen between those with leaky ceilings combined with vented ceiling spaces and the remainder. Eleven of the twenty buildings were identified as having both a vented ceiling space and a suspended ceiling. ACH50 averaged 26.0 for these buildings, much more leaky than the average commercial building (Table 2)

Table 2
Size and tightness of buildings with suspended T-bar ceilings and vented ceiling space.

building descriptor	floor area	CFM50	CFM50/ft^2	ACH50
community center	23934	155047	6.48	31.8
classroom	5000	16644	3.33	25.0
classroom	5000	18509	3.70	27.8
portable	706	2080	2.95	23.2
portable	756	1945	2.57	19.9
portable	756	4217	5.58	43.2
portable	720	1871	2.60	20.8
portable	720	2680	3.72	27.9
convenience store	4320	10776	2.49	15.4
ocean front restaurant	5977	16738	2.80	16.3
Greek restaurant	1500	7736	5.16	34.4
average	4490	21659	3.76	26.0

A study of 69 small commercial buildings found an average ACH50 of 16.7 (Cummings et al., 1996). By comparison, the average ACH50 for single family homes is about 12.7 (Cummings et al., 1991). The remaining nine buildings, having either an *unvented* ceiling space or a vented ceiling space with a non-suspended ceiling (gypsum board or other panels), have an average 10.7 ACH50 (Table 3).

Table 3
Size and tightness of buildings which have an unvented ceiling space
or do not have a suspended T-bar ceiling.

building descriptor	floor area (ft^2)	CFM50	CFM50/ft^2	ACH50
elementary school	50139	63200	1.26	8.6
chicken restaurant	3161	3165	1.00	6.7
seafood restaurant	8204	13242	1.61	9.7
river front restaurant	5850	3722	1.28	3.8
steakhouse restaurant	6115	12021	1.97	14.1
portable	650	588	0.90	6.8
portable	720	1623	2.25	16.9
bar and grill	2400	6599	2.75	17.4
charity office/thrift shop	3741	6841	1.83	12.5
average	8998	12333	1.65	10.7

The 26.0 ACH50 for the eleven buildings with vented ceiling space can also be expressed as 3.76 CFM50 per square foot of building foot-print area (floor area is generally the same as ceiling area

since all eleven are single story buildings). CFM50 per square foot ranges from 2.49 to 6.48 (see Table 2). Some readers will ask; since the suspended T-bar ceilings in these eleven buildings by themselves have leakage of at least 5.0 CFM50 per square foot, why is the overall building airtightness lower than 5 CFM50 per square foot in eight of the eleven buildings. The reason is that the roof assembly (including the exterior walls above the ceiling level) also represents resistance to air flow -- in series with the ceiling leakiness. This resistance can vary from a little resistance to a lot. In cases where the ceiling space is very well vented, then the full leakiness of the ceiling will be represented in the test number. From an air flow point of view, it will be as if the roof was not there. The community center/clubhouse is an example of a building with extensive attic venting, and CFM50/ft^2 is the highest of all tested buildings (6.48 CFM50/ft^2).

If the ceiling space (which may be an attic space) has only a modest amount of venting (to outdoors), then the roof assembly may be a greater resister to air flow than the ceiling. Recalling that a standard suspended T-bar ceiling has an equivalent hole of 0.65 square inches per square foot, then a 4000 square foot ceiling has an equivalent hole of 18 square feet. If the net free area of the ceiling space vents is less than 18 square feet, then the roof will be the primary air barrier. If the net free area of the ceiling space vents is greater than 18 square feet, then the ceiling will be the primary air barrier.

The 10.7 ACH50 for the nine buildings with unvented ceiling space or panel ceilings can be expressed as 1.65 CFM50 per square foot of building foot-print area (eight of nine are single story buildings; river front restaurant is two stories). CFM50 per square foot ranges from 0.90 to 2.75 (see Table 3).

Testing for Primary Air Barrier. The primary air barrier can be identified during a blower door test. When the occupied space is depressurized to -50 pascals, pressure drop across the ceiling is measured. If the pressure drop across the ceiling is less than 25 pascals, then the roof assembly is the primary barrier. If the pressure drop across the ceiling is greater than 25 pascals, then the ceiling is the primary barrier. In cases where the ceiling space is not vented, it is common to find a pressure drop across the suspended T-bar ceiling of less than 1 pascal, indicating that the roof represents essentially all resistance to air flow. In cases where the ceiling space is very well vented, it is common to find a pressure drop of 40 pascals across the suspended ceiling. In cases where the pressure drop is 25 pascals across the ceiling and across the roof assembly, then the equivalent ceiling leak hole is equal in area to the vent openings of the ceiling space. In cases where the ceiling is gypsum board or other panel construction, pressure drop across the ceiling is generally greater than 45 pascals, especially if the ceiling space is vented.

Good practice building tightness. Good building construction practice calls for building airtightness between 4 and 8 ACH50. Buildings tighter than 4 ACH50 are more likely to experience extreme pressure differentials which may be dangerous. Buildings more leaky than 8 ACH50 are likely to experience excessive infiltration during strong winds or large indoors-to-outdoor temperature differentials, and it will be more difficult to maintain the desired building pressure.

Consider a hypothetical example. A 4000 square foot restaurant has airtightness of 3.0 ACH50 and contains an atmospherically vented gas water heater. Exhaust air is 4000 cfm, make-up air is 2500 cfm, and outdoor ventilation air is 1500 cfm. With these air flows the building pressure is neutral with respect to outdoors. However, consider what happens when the belt driving the make-up air fan breaks. With make-up air stopped, building pressure goes to approximately -60 pascals (-0.24 inWC) and the gas water heater begins to experience flame roll-out. If the building were more leaky, then the degree of negative pressure would be diminished. Note that it is also recommended to locate atmospherically vented combustion appliances in a zone that is well isolated from indoors and well connected to outdoors. An exterior combustion closet with a louvered exterior door should achieve this goal.

Consider another hypothetical example. A 5000 square foot office space has an airtightness of 11.0 ACH50. Bathroom exhaust fans draw 200 cfm from the building and outdoor ventilation air is delivered at 500 cfm into the building. Net air flow is 300 cfm into the building. This will generate +0.4 pascals positive pressure which is not sufficient positive pressure to assure that air flow will go from indoors to outdoors most of the time. Wind and stack effects can readily overcome this pressure. On the other hand, if the building airtightness was 6 ACH50, then building pressure would be +0.8 pascals, a pressure more likely to achieve the desired level of control over the direction of air flow. Furthermore, during periods that the space conditioning equipment is turned off (nights and weekends), a tighter building will prevent rapid entry of humid outdoor air into the building during hot and humid weather.

Notice that the tightest of the twenty buildings (the river front restaurant with 3.8 ACH50; see Table 3) also has the greatest level of space depressurization. The building has exhaust fans (primarily kitchen) drawing 5318 cfm from the building. Make-up air, delivered directly in front of the exhaust hood, provides 1738 cfm. The unbalanced exhaust air of 3580 cfm causes the building to operate at -45 pascals with respect to outdoors. An atmospherically vented gas water heater is located within the building, which if exposed to this pressure would experience backdrafting and flame roll-out. In order to avoid this excessive space depressurization, an exterior kitchen door is propped open during hours of exhaust fan operation, and as a result the building operates at -2.2 pascals, a pressure which is not sufficient to cause water heater operation problems.

The leakiness of suspended T-bar ceilings dominates overall building leakage when the ceiling space is vented. As indicated, the eleven buildings with suspended ceilings and vented ceiling space had an ACH50 of 26.0, substantially greater than the average commercial building and much greater than the recommended airtightness. This 26 ACH50 also converts to 3.8 CFM50 per square foot of floor area. By contrast, the other nine buildings (with no ceiling space venting or with tight ceiling construction) are 60% tighter, with 1.6 CFM50 per square foot.

Suspended T-bar ceilings are on the order of six times more leaky than the 0.5 CFM25/ft^2 requirement of an air barrier (equal to about 0.8 CFM50/ft^2) as defined by the 1997 Florida Energy Code. Therefore, these suspended ceilings cannot be counted on to provide significant resistance to air flow. As a consequence, in the presence of air flow, there is great potential for suspended ceilings to cause unwanted increases in cooling and heating loads, ventilation rates, indoor humidity levels, and space conditioning energy use. Whether this potential is realized depends for the most part upon the thermal and humidity conditions in the ceiling space (the zone between the ceiling and the roof deck).

The thermal and humidity conditions of the ceiling space, in turn, depend upon the location of the air boundary and the thermal boundary of the building. If the ceiling space is not vented, then the primary air boundary of the building is normally the roof deck. If it is vented, then the ceiling is generally the primary air boundary. However, if the ceiling space is vented and the ceiling is suspended T-bar construction, then the building in effect has no air boundary at its top. The thermal boundary (the insulation) is sometime located on top of the ceiling tiles, sometimes attached to the bottom of the truss members, sometimes is secured to the bottom of the roof deck, and sometimes is integrated into the roofing system.

There are three primary ceiling space configurations based on the location of the air boundary and the thermal boundary (indicated in parentheses are ceiling space conditions during hot and humid weather; *for discussion purposes, define cool indoors as 76F, warm ceiling space as 85F, hot outdoors as 90F, and very hot ceiling space (or attic) as 105F or greater*):

- thermal boundary at ceiling and air boundary at ceiling (very hot and humid)
- thermal boundary at ceiling and air boundary at roof (very hot and dry)
- thermal boundary at roof and air boundary at roof (warm and dry)

In some cases the ceiling space is used as a return plenum. There are two factors which can make this successful. 1) The roof and exterior walls above the ceiling must be insulated and of course unvented. 2) The return plenum should operate at neutral or positive pressure with respect to outdoors. The reader may wonder how a return plenum can operate at positive pressure? The answer is this; the building should operate at positive pressure and the pressure drop across the ceiling (from occupied space to plenum) should be small. For example, the occupied space could operate at +2.0 pascals with respect to outdoors. The return plenum could operate at -0.5 pascals with respect to the occupied space. Consequently, the return plenum will be at +1.5 pascals with respect to outdoors. When the ceiling space is a return plenum, that space will be cool (perhaps 80F) and dry.

Most commercial buildings use suspended T-bar ceilings, and therefore there is great potential for air flow between the occupied space and the ceiling space. Three forms of mechanically-induced uncontrolled air flow create driving forces that move air across the suspended ceilings – duct leakage, unbalanced return air, and unbalanced exhaust air. The impacts of uncontrolled air flow vary according to ceiling space configuration.

- If the ceiling space is hot and humid, then uncontrolled air flow will tend to increase cooling load, the building ventilation rate, and indoor humidity. Air leaking to or from ductwork in the ceiling space will tend to be lost. Unbalanced return air will push room air into the ceiling space in some areas and draw hot and humid air from the ceiling space in other areas. Unbalanced exhaust air depressurizes the occupied space. This depressurization causes air to be drawn into the occupied space. Most of this incoming air comes from the hot and humid ceiling space because the ceiling is leaky and the ceiling space is also leaky [vented] to outdoors.

- If the ceiling space is hot and dry, then uncontrolled air flow will increase cooling loads, will not significantly impact ventilation rates, and may reduce indoor humidity (because the increased load will increase air conditioning system duty factor and improve its dehumidification performance). Duct leakage and unbalanced return air will transport air from the ceiling space to the occupied space, increasing the sensible cooling load but not significantly increasing ventilation or the latent cooling load. Humidity will likely decrease. Unbalanced exhaust depressurizes the occupied space. Unbalanced exhaust air will draw air from outdoors and the ceiling space. The load impact, however, will be much less than with the hot and humid ceiling space configuration for two reasons: 1) air is drawn into the occupied space not primarily from the ceiling space but from wherever the building envelope leaks are largest and 2) the air that comes into the occupied space from the ceiling space has much less latent load.

- If the ceiling space is warm and dry, then uncontrolled air flow will have little impact on cooling loads, ventilation rates, or humidity. Air leakage into or from ductwork in the ceiling space will, for the most part, be regained and not lost. This is because the duct leakage, whether return or supply leakage, occurs inside of the air and thermal boundaries of the building. Unbalanced return air will push room air into the ceiling space in some areas and draw air from the ceiling space in other areas, but because the ceiling space is not vented and its air is warm and dry, there will be little impact upon loads, ventilation, or humidity. Unbalanced exhaust air will draw air from outdoors and the ceiling space, but the load impact will be less than with the hot and humid ceiling space configuration. In other words, placing the building air boundary and thermal boundary at the roof creates a "forgiving" building – that is, it is a building design that can eliminate most of the negative impacts of most uncontrolled air flow problems, especially with regard to duct leakage and unbalanced return air.

In summary, best practice design is to locate the insulation system at the roof deck and not ventilate the ceiling space. With this design, the leakiness of the suspended ceiling creates no problems, the building will generally be quite airtight, and the impacts of uncontrolled air flows are greatly muted.

Overview of Education Buildings

Ten of the twenty buildings fall into the education category. One is a large 50,000 square foot elementary school, two are identical site-built 5,000 square foot classroom buildings, and seven are "portables", manufactured classroom buildings with average floor area of 718 square feet.

The 34-year old elementary school is moderately airtight with 8.6 ACH50. It has suspended T-bar ceilings throughout. However, the ceiling space is not vented and insulation is at the roof deck, therefore the leakiness of the suspended ceiling has little or no bearing on building energy use, ventilation rate, or humidity. Bathroom exhaust fans run throughout the occupied hours. The kitchen exhaust fan operates from early morning to early afternoon. Together these fans draw 10,184 cfm of air from the building (kitchen = 8100 cfm; bathrooms = 2084 cfm). Outdoor ventilation air moves 4236 cfm into the building. No make-up air is provided for the kitchen or bathroom exhaust fans. When all exhausts are operating, building pressure is -1.6 pascals with respect to outdoors. When the kitchen exhaust is off and the bath exhausts are on, the building operates at +1.1 pascals.

The kitchen operates at -43 pascals with kitchen exhaust operating. When the door from the kitchen to the dining room is opened, kitchen pressure is -6.7 pascals. Testing of the chilled water cooling system found that many of the 34 year old fan/coil units were providing 46F chilled water entered the air handlers; however, because of fouling of the coils (on the outside and probably on the inside), some of the air handlers were delivering only 25% to 28% of their rated capacity. Nevertheless, required cooling was being provided to all zones except the kitchen, where only token cooling was provided.

The school district of Polk County in Florida has developed a "quad" permanent classroom building design. They can be constructed with four, eight, or twelve classroom per building. We tested two four-classroom units. They are slab on grade, frame construction, metal roof over vented attic, and suspended T-bar ceilings two feet below the metal truss system. Insulation batts are located at the bottom of the truss members, supported on radiant barrier material stretched tight between the trusses. In both buildings, about 25% to 30% of the insulation batts were missing or were suspended over electrical conduit and other obstructions, causing a substantial reduction in effective R_value. Because of the vented attic and the suspended ceiling, these two building were very leaky, with ACH50 of 25.0 and 27.8 for each. CFM50 per square foot was 3.3 and 3.7.

There was essentially no ductwork (there are two ducts to a central zone), with each classroom served by PTHP (packaged terminal heat pump) unit. Retrofits were performed to make these buildings more airtight. Spray foam insulation was applied to the radiant barrier material on which the insulation batts were lying to create an air barrier where the thermal barrier is located. Gaps around the PTHP units were also sealed. In addition, insulation batts were added and adjusted to create a continuous thermal barrier. On average, these buildings became 37% more airtight. CFM50 was reduced to 11,091 in both buildings (both having the same post-retrofit airtightness is a coincidence). This can also be expressed as 17.9 ACH50, for both buildings. By way of comparison, identical floor plan buildings were also constructed using aerated concrete panels for both walls and roof, and no venting of the ceiling space. These had an average airtightness of 7.3 ACH50. Therefore, it appears that substantial ceiling air leakage remains. Monitoring found that cooling energy use was reduced by 12.4% one school (white roof) and 15.4% in the other (blue roof).

Seven "portable" classrooms were tested at two middle schools. These units were either three, four, or nine years old. Five of the seven had suspended T-bar ceilings located below a vented attic space. Two had large vinyl ceiling panels attached to the bottom of the trusses. ACH50 averaged 22.7 for all seven. Those with suspended ceilings averaged 27.0 ACH50, while those with panel ceilings averaged 11.8 ACH50. All had package air-conditioning units hung on an exterior wall. Only one of the seven had a duct system. The remainder had only a through-the-wall return grill and a through-the-wall supply diffuser which throws air across the entire room. They each have outdoor air openings in the package unit, and outdoor air ranged widely, from 40 cfm to 210 cfm, averaging 79 cfm. Return air leakage (from outdoors) was also measured (using tracer gas); it averaged 52 cfm. Combined outdoor air and return leak air equals 131 cfm. With average occupancy of 28 persons, the design ventilation should be 420 cfm, or somewhat less if an occupancy use factor is employed. Therefore, none of these classrooms is receiving the necessary ventilation.

Ventilation was also evaluated by means of tracer gas decay. With the air handler blower operating continuously, ventilation varied from 1.27 to 2.51 air changes per hour (ach), with an average of 1.77 ach. Given an average occupied space volume of 5601 cubic feet, this indicates a ventilation rate of 165 cfm. Allowing that there may be some passive ventilation (induced by wind, temperature, or door opening and closing effects; recall that these buildings have very leaky envelopes), this 165 cfm of ventilation is in reasonable agreement with the measured mechanically induced ventilation of 131 cfm.

Description and Impacts of Retrofits

Six buildings were selected for retrofits. We report results on five buildings because problems occurred in one of the buildings, a convenience store with a small food preparation operation. At this store retrofits were implement to improve the performance of two HVAC systems. However, the store was sold to another owner within weeks of this first retrofit (others had been planned). While this new owner authorized us to continue our experiments and monitoring, they unexpectedly made major changes to the building, including moving walls, replacing the suspended ceiling and insulation, replacing west windows, and changing the metal roof color from light gray to dark green. Given these drastic changes, we had at this point no pre-retrofit data, and therefore we discontinued our monitoring and retrofit efforts at this building.

Community Center/Clubhouse

Three retrofits were implemented. First, the schedule of operation of the HVAC systems was modified. The cooling system consists of 100 ton chiller and five air handlers, four of which have outdoor ventilation air. The building operated at 70% relative humidity much of the time, had moisture dripping from the ductwork onto the ceiling tiles, total energy use was high, and mold growth was observed behind vinyl wallpaper in several locations. In most zones, the air conditioners operated 24 hours per day with primarily constant fan operation (some VAV). Cooling output of the air handlers is modulated almost entirely by means of three-way valves which modulate the flow rate of chilled water through the coils (thus changing the coil temperature). During part load operation, the cooling coil is quite warm and little or no dehumidification occurs. Furthermore, humid outdoor air is drawn in continuously throughout the day due to continuous air handler operation.

The first retrofit – application of spray foam insulation to the bottom of the roof deck and sealing off all attic vents – produced cooling energy savings of 5.3%. Indoor relative humidity remained approximately the same at about 70%. Continuous entry of largely untreated outdoor air is the primary cause of the high humidity.

The second retrofit consisted of rescheduling of cooling system operation. Air handler (and chiller) operation time was reduced from approximately 23 hours per day to about 13 hours per day (except air handler 1 was cut back to three hours operation per day). Savings occurred in three ways; 1) total cooling load is substantially reduced because the outdoor air brings in large amounts of latent and sensible load and the air handlers generate considerable heat from their operation, 2) air handlers consume substantial amount of electricity, and 3) the chilled water pumps consume substantial amount of electricity. Cutting back air handler operation by an average 12 hours per day and chiller operation and chilled water pumping by 10 hours per day reduced total HVAC energy use by 23.3%.

The third retrofit consisted of installing a dedicated outdoor air conditioning system. This unit, using chilled water and a heat pipe, takes 4300 cfm of outdoor air, conditions it, and pumps it into the building. 430 cfm is ducted directly into the auditorium. The remaining 3870 cfm is directed into the return air of the air handlers which serve the office, hallway, and dining room zones of the building. As a result of this retrofit, total cooling energy use increased by ~10% during summer months and indoor humidity levels declined from approximately 70% to 60%.

Two Polk County Classroom Buildings

Retrofits were performed to airtighten the ceiling plane of the building and restore the thermal barrier to standard. Spray foam insulation was applied to seal radiant barrier material which was suspended tightly across the bottom of the metal truss system. Insulation batts were added and rearranged to provide continuous attic insulation. These retrofits reduced cooling energy use by 12.4% and 15.4%, respectively.

Bar and Grill

Two retrofits were implemented; 1) make-up air was installed and 2) spray foam insulation was added to make the building tighter and improve the thermal boundary. Installation of a make-up air system providing 82% of the exhaust air flow produced cooling energy savings of 11.5%. Application of the spray foam insulation yielded cooling energy savings of 13.6%.

Charity Thrift Shop/Office

Three retrofits were implemented. This metal building had cream/tan color exterior walls and a galvanized metal roof. A white roof coating was applied. The exterior walls were painted white. No energy savings were observed. However, indoor temperature (from 3 PM to 5 PM) decreased from 84.7F to 80.7F. Indoor relative humidity increased slightly from 48.1% to 50.5%. The reason that no energy savings were observed was that the air-conditioning systems ran almost continuously from 7:30 AM to 5 PM even after the retrofits. This building has the smallest cooling capacity (per floor area) of any building in the project, with 2.0 tons per 1000 square feet.

Unbalanced return air problems were corrected. Before repair, the return of one of the two AC systems was located in a storage room that was closed much of the time. With the door closed, the room was depressurized. This in turn caused air to be pulled through the suspended T-bar ceiling into the room from the unvented ceiling space that often reached 120F. The return air was modified so that the majority of the return air was taken from the hall, and the return air in the storage room was decreased to match the supply air flow to that room. This correction of unbalanced return air produced cooling energy savings of 15%.

Duct repairs were performed as well. Difficulties were experienced in making repairs because of the fragility of the flex duct (the inner liner tended to fall apart when disassembled). As a result, only 30% of the leakage was repaired. As a result, one would expect only a modest amount of energy savings. However, and to our considerable surprise, there was actually an <u>increase</u> in cooling energy savings after duct repair of 8.6%. There are factors, however, which may account for the increased energy use after repair: slightly higher roof deck and ceiling space temperatures, significantly higher outdoor dewpoint temperatures, and potential adjustments to thermostat setpoints by the occupants.

PROJECT OBJECTIVES

The project objectives were to investigate ways to substantially reduce HVAC system energy consumption, improve occupant comfort, reduce building degradation, and improve indoor air quality by repairing uncontrolled air flow (UAF) and HVAC system performance/control problems in commercial and institutional buildings. Specifically these objectives involved:

- gaining knowledge of failure modes in buildings
- evaluating diagnostic methodologies which can uncover these failures
- measuring energy savings which result from repair of UAF and HVAC problems
- developing recommendations for improved management of air flows and HVAC systems in non-residential buildings
- developing advanced diagnostic and research capabilities that will facilitate the obtaining and disseminating of information regarding building functioning.

Meeting these objectivee involved *field testing, retrofits, and monitoring*:

- performing field testing of 15 small-to-medium sized commercial and institutional buildings (for purposes of discussion, in this report "commercial" will refer to both commercial and institutional buildings).
- performing short-term monitoring of such things as temperature, relative humidity, carbon dioxide level, and pressure differential in 7 buildings.
- preparing summary reports that also contain retrofit recommendations for many of the buildings in which testing and short-term monitoring was performed.
- performing retrofits on 6 buildings and monitoring changes in such things as energy consumption, temperature, relative humidity, carbon dioxide levels, and building pressure differentials.

Meeting these objectives also involved the *design and construction of a building science laboratory and training facility* at the Florida Solar Energy Center campus. This facility has been designed and constructed (70% complete by contractor in November 1999 and 90% complete at time of this report) with wide flexibility – that is, the ability to be transformed into many building types. *Just a few examples:* the building envelope can be made airtight or leaky, well insulated or poorly insulated, it can be configured as an office building or a restaurant or a retail store, its ceiling space can be vented to outdoors or not, the ceiling plane can be leaky or airtight, the ceiling insulation can lie on the ceiling tiles or be placed at the roof level, the ducts can be tight or leaky, and the return air can be central return, plenum return, or hard-ducted to each room. Multiple air-conditioning, heating, and ventilation systems can also be tested in combination with the various building configurations.

In terms of field testing, three commercial/institutional building groups were targeted: 1) buildings with large exhaust fans, 2) buildings with leaky ceilings, and 3) education buildings. A total of 20 buildings were tested, averaging 6518 ft^2 of floor area.

Short-term monitoring was done on 7 of the 20 buildings. Retrofits and long-term monitoring were done on 6 of the 20 buildings.

Project Focus: Uncontrolled Air Flow and HVAC Problems

The focus of the testing was on uncontrolled air flows and HVAC system performance. Uncontrolled air flow (UAF) falls into four categories; 1) duct leakage, 2) restricted (or unbalanced) return air, 3) unbalanced exhaust air, and 4) leaky building envelopes (mostly related to the leakiness of suspended t-bar ceilings). Regarding UAF, we tested for building and air distribution system airtightness, building envelope air boundaries, air moving system flow rates, infiltration/ventilation rates, and building pressure differentials. Regarding HVAC, we examined cooling system maintenance status (dirty coils, filters, refrigerant charge), air flow rates, efficiency, operation schedule, system controls, and dehumidification performance.

Previously, UAF had been studied in isolation in a group of 70 buildings. Retrofits had been performed in 20. However, HVAC problems had been largely ignored, and consequently some excellent retrofit opportunities had been missed. The basic concept of the current project was to 1) develop diagnostic approaches to commercial buildings and 2) develop holistic solutions to buildings, taking into account both UAF and HVAC systems. Combining these two issues together is important since they are closely interrelated, and problems caused by these represent such a large portion of serious failures in buildings. The principal investigators of this project have observed that a vast majority of Florida buildings experience serious and sometimes even catastrophic failures related to UAF and HVAC systems performance problems. These failures often create real human and pocketbook problems related to indoor air quality, indoor comfort, humidity control, moisture damage to building materials, or high utility costs, and may in some cases cause catastrophic building failures.

Lab Building

A building science laboratory and training facility was designed, constructed, and commissioned as part of this project (see Appendix D for more details). The funds for construction of the "bricks and mortar" portion of the building came from outside this project. However, the cost of developing the initial conceptual design; developing the construction design, specifications, and documents; commissioning the facility; and installation of specialized research features came from the project budget. These specialized features include:

- data loggers for the data acquisition system
- conduit and instrumentation trays for running sensor wires through slabs and the ceiling space
- a wide range of sensors for temperature, humidity, moisture levels, air flow rates, pressure differentials, and carbon dioxide
- walkway grid systems (4 foot on centers) and walkway platforms running across the grid in the ceiling space
- separate supply duct systems for two heating and cooling systems
- multiple return air systems for each heating/cooling system
- specialized dampers for creating repeatable adjustments to supply and return air flows
- motorized dampers for vent openings in the ceiling space and the occupied space, allowing remote adjustment of the building airtightness.
- special features of the building structure which allow specialized research, including 1) the 18 foot height of the building and extraordinarily deep metal trusses which allow 6-foot vertical

clearance in the ceiling space, and 2) forty-two 8 x 16 inch vent openings in the exterior walls of the building above and below the ceiling.
- oversized electrical service to accommodate a wide range of HVAC systems, diagnostic test equipment, and internal load generation.

Two specialized HVAC systems were donated by industry:

- equipment for two independent heating and cooling (split DX) systems of 3.5 tons and 5.0 tons were donated by Carrier Corporation. These two heat pump systems provide the flexibility to operate cooling or heating systems appropriate for various types of commercial buildings, and with enough capacity to meet a variety of loads associated with a variety of space utilizations, internal loads, and uncontrolled air flow scenarios.
- an exhaust hood was donated by Greenheck Corporation. This 4.5 foot by 10 foot hood is a special design with unique capabilities. The make-up air can be directed in three different directions – short circuit discharge, curtain discharge, and face discharge. This will give us the capability to assess exhaust hood performance and make-up air capture effectiveness with any and all discharge configurations.

Advisory Committee

Another project task was to establish a program advisory infrastructure that would provide guidance to the project team and assist in expeditiously transferring the findings of this study to the non-residential marketplace, such as through utility demand-side management programs, improved HVAC equipment design, modified engineering design methods, better HVAC equipment/controls selection criteria, revised inspection practices and codes, and training for those involved in designing, constructing, and maintaining buildings. A list of committee members is found in Appendix A.

The project scope of work consisted of 10 tasks, as laid out in the project workplan. These are discussed briefly in the following section while detailed results are presented in Appendices B (test results), C (retrofit results), and D (lab building).

PROJECT TASKS

Task 1: Advisory committee

An advisory committee was formed in 1997 with total membership of 23 persons, representing groups that have a stake in the outcome of the research. A list of the Advisory Committee members is contained in Appendix A (affiliation shown as of 1998). The first and only meeting of the Advisory Committee took place on December 8, 1997 at the Florida Solar Energy Center. Nineteen members of the committee were in attendance. FSEC staff presented:

- findings from the preceding uncontrolled air flow research project
- questions that remained unanswered from that earlier work
- how the present project addresses those questions
- what the present project proposes to accomplish.

Presentations were made to the committee regarding activities which had occurred during the first six months of the project. Plans for design and construction of a building science research building were also presented.

It was proposed to the committee that the second and probably final meeting would be scheduled for late summer or fall of 1998. However, due to delays in completing the field investigation portions of the project, and the greater length of time required to get the planning of the lab building completed, the second meeting was canceled and never rescheduled. Progress reports were sent to committee members four times per year. Members of the committee were also invited to attend the dedication ceremony for the new lab building on February 17, 2000.

Task 2: ***Perform one-time field tests on at least 15 buildings to characterize the UAF and HVAC failure modes which exist in those buildings.***

Three groups of buildings were targeted in this project: 1) buildings with large exhaust fans, 2) buildings with leaky ceilings, and 3) education buildings. These groups were chosen because we had not had a chance to investigate thoroughly the issues related to unbalanced exhaust air, leaky ceilings, and schools in a preceding project in which 70 commercial buildings were tested for UAF (repairs were performed in 20 buildings in that study). In the current project, testing was done on a total of 20 buildings (listed in Table 4). Note that for the purposes of counting buildings that were tested, four portable classrooms at Kennedy middle school were counted as one education building and three portable classrooms at McNair middle school were counted as one education building. Therefore, for purposes of meeting project scope of work requirements, 15 buildings were tested.

Typical testing included:

- a blower door test to determine the airtightness of the building envelope
- a building air boundary test to determine the primary resistor to air flow across the building envelope
- pressure mapping to identify air pressure in various zones or cavities of the building with the HVAC systems operating in standard mode
- measurement of various air flows including return, supply, outdoor ventilation air, exhaust air, and make-up air flows
- tracer gas decay test to determine air exchange between indoors and outdoors with the HVAC systems in standard operation
- a return leak fraction test to determine the fraction of air leaking into the return ducts from outdoors or other unconditioned spaces
- visual inspection to identify the location of leak sites in the building envelope or ductwork and to determine if backdrafting or other problems are occurring with vented combustion devices
- measuring the cooling performance of air-conditioning equipment
- measuring the energy demand characteristics of air-conditioning equipment
- inspecting coils on air-conditioning equipment to determine whether service work is needed
- at some building, a duct system airtightness test was performed.

Description of the testing and test results are found in Appendix B.

Table 4
Diagnostic testing was performed in 20 buildings.

BUILDING DESCRIPTION	LARGE EXHAUST	LEAKY CEILING	EDUCATION
1. Apollo elementary school	x		x
2. Chicken cafeteria restaurant	x		
3. Community center / clubhouse	x	x	
4. Floral Avenue elementary school		x	x
5. Eagle Lake elementary school		x	x
6. Sea food chain restaurant	x		
7. River front restaurant	x		
8. Steak house chain restaurant	x		
9-12. Kennedy middle school portables		x	x
13-15. McNair middle school portables		x	x
16. Bar and grill	x	x	
17. Convenience store / fast food	x	x	
18. Ocean front restaurant	x	x	
19. Greek restaurant	x	x	
20. Charity office / thrift shop		x	

Tables 5 and 6 identify the tested buildings by category and building use.

Table 5
Summary of buildings by categories. The total number of buildings listed in this table is 33, because some buildings fall into two categories.

building category	number of buildings
large exhaust fans	10
leaky ceilings	13
education	10

Table 6
Use of buildings tested in this project.

building use	number of buildings	avg. floor area (ft^2)
restaurants	7	4744
schools	10	6517
convenience store	1	4320
community center	1	23934
charity thrift shop/office	1	3741
total/average	20	6518

Task 3: ***Perform short-term monitoring of 7 or more buildings to characterize UAF and HVAC failure modes, and user interactions in the time domain.***

Short-term monitoring was performed in 7 buildings during this project, as shown in Table 7. The goal of the short-term monitoring was to observe in the time domain the patterns of building occupancy, HVAC operation, and building environment response. While the one-time testing gave us a "snap-shot" of the building, the short term monitoring gave us a "video tape" showing the building over a period of time, including occupied and unoccupied periods. Along with the one-time testing, the short-term monitoring provides data upon which to determine if the building is operating in an efficient and healthy manner, to to develop a diagnosis, and to make recommendations for remediation.

Table 7
Buildings in which short-term monitoring was performed.

building type	floor area
elementary school	50139
Greek restaurant	1500
convenience store	4320
community center	23934
chicken cafeteria restaurant	3161
steak restaurant	6115
seafood restaurant	8204

Tasks 4 and 5 (discussion combined since these tasks deal with the same buildings)

Task 4: ***Select at least 6 of the 15 tested buildings for retrofits. These retrofits will be targeted toward resolving UAF and HVAC performance/controls problems.***

Task 5: ***Monitor changes to the building and HVAC systems which result from retrofits, including changes in energy use, indoor temperature, indoor relative humidity, pressure differentials, and ventilation rates.***

Six buildings were selected for retrofits and long-term (approximately one year) monitoring (See Table 8).

Table 8
Retrofits and long-term monitoring were done on six buildings.

building description	retrofit description
convenience store	1) improve HVAC systems
bar and grill	1) add make-up air, 2) improve air/thermal barrier
charity thrift store/office	1) change exterior to white, 2) balance return air, 3) duct repair
classroom building	1) improve air/thermal barrier
classroom building	1) improve air/thermal barrier
community center	1) improve air/thermal barrier, 2) reduced operation time, 3) outdoor air dehumidification

One of the monitored buildings – the convenience store – did not have a successful retrofit. The circumstances unfolded as follows. We set up a data acquisition system to monitor temperatures, humidity, AC system energy use, building air pressure, solar radiation, and outdoor temperature and humidity. After several months of monitoring (pre-retrofit data), we arranged to have AC system retrofits done. These included bringing the AC system air flows (including outdoor air) and refrigerant charge as close as possible to design specs and modifying the thermostat of one of the two AC units so that the fan would not run continuously (this fan could literally not be turned off by the thermostat). Within two weeks of completing these retrofits, the store was sold (without warning and to our surprise) to another owner (February 26, 1999). We spoke with the new ownership and received permission to continue our monitoring, but we did not know the extent of the changes which were coming. The roof, which had been light gray color, was painted a dark green. Some of the exterior walls were painted. The suspended ceiling and the ceiling insulation were removed and replaced by new product. The west facing (tinted) windows were replaced. Additionally, all condenser units for both AC and refrigeration were replaced. Given the change in ownership and all the retrofits, there was no opportunity to collect data reflecting changes resulting from the retrofits alone. We therefore abandoned monitoring at this building.

Descriptions of the retrofits and the resulting changes (energy savings, etc.) are contained in Appendix C for five of the six buildings.

Task 6: ***Data collected by reviewing construction documents, visual inspection of the building and HVAC systems, one-time testing, short-term testing/monitoring, and long-term monitoring will be analyzed.***

Data collected from one-time testing, short-term testing/monitoring, and long-term monitoring has been analyzed. Throughout the long-term monitoring portion of the project, the data collected from the preceding day were graphed and reviewed on a daily basis to observe proper system operation, identify AC system or monitoring system malfunctions, etc. In some cases (clubhouse/community center and charity office/thrift shop), daily electrical energy usage versus outdoor dry-bulb temperature was calculated and plotted on a regular basis to observe this relationship as the baseline data was being collected. This analysis was used to determine when one phase of the monitoring/retrofits was complete (when we had sufficient data) so we could move on to the next phase.

Task 7: ***A final report will be written summarizing the project findings.***

This report is in fulfillment of Task 7.

Task 8: ***Develop plans for a laboratory facility which can be used for uncontrolled air flow and HVAC performance experiments.***

During the fourth quarter of 1998, the first stage of the design process (conceptual drawings) was completed by the architect. Additionally, the architect provided an estimate of probable costs to construct the lab building ($200,000). In the first quarter of 1999, final construction documents (drawings and specifications) were developed by the same architect. The process of developing and finalizing the final design was time consuming, to a large extent because we wanted to accommodate a great deal of flexibility into the facility, both for research and for training. There were a number of meetings with the architect, reviews of interim versions of the drawings and specifications, and numerous internal meetings and discussions regarding design details.

A descriptive, promotional brochure was finalized during the first quarter of 1999. A copy of the brochure was included in the fourth quarter of 1999 report. Additionally, a two-page brochure was developed. The purpose of these brochures was first to secure funding for the bricks and mortar construction, and then later to search for funding for research to be done in this building. The eight-page and the two-page brochures are contained in Appendix D, along with a description of the facility.

Funding was obtained to construct the lab buildings. A bid package was assembled and made available to contractors on March 25, 1999. A pre-bid meeting was scheduled for April 8, 1999 and bids were opened on April 22, 1999. Construction began the last week of May 1999. The construction to be done by the contractor was completed the second week of November 1999.

When the contractor was done with his work, only an estimated 70% of the work was complete. Additional tasks had be done to complete the facility:

- design of two heating and cooling systems ***
- installation of two heat pumps **
- installation of two independent supply ductwork systems *
- installation of fully ducted return air ductwork (return to each room) *
- installation of walkway platforms throughout the ceiling space ***
- installation of manual vent covers ***
- installation of motorized vent covers *
- installation of insulation batts on the suspended ceiling
- installation of a data acquisition system, including dataloggers and instrumentation *
- installation of window blinds
- run water line to building and hook-up ***
- run sewer line to building and hook-up ***
- coating of the roof with silver roof paint
- installation of kitchen exhaust fan, hood, and make-up air *

As of the date of publication for this final report, some of the above listed tasks were still incomplete. A single (*) indicates items 50% to 75% complete, two (**) indicate items which are 75% to 99% complete, and (***) indicates items which are complete.

Task 9: Installation of a data acquisition system and advanced energy/control features for the lab facility.

Conduit was specified in the construction documents to accommodate the running of instrumentation wiring throughout the building, and electrical boxes have been located in the northeast room to house the datalogger system.

At the beginning of the construction phase, instrumentation was installed in the slab and soil beneath the slab. Thermocouples and pressure sampling tubes were located at five locations under the building foot print (four corners and the center). Before installation, the thermocouples were calibrated in water baths of 32°F, 77.3°F, and 107°F. Four thermocouples were located at each of the five locations, in the slab and at depths of 1 inch, 12 inches, and 36 inches below the slab. The ones at the corners were located 24" from the interior surface of both adjacent block walls. Pressure tubing, 1/2" under the slab and 36" under the slab will allow us to sample soil pressure at various locations and depths beneath the building. Soil moisture probes were installed at two locations 12" below the top of the soil; 1) at the center of the building footprint and 2) in the northeast corner of the building (42" from the north wall and 48" from the east wall).

This instrumentation was installed after the foundation footers were poured, three rows of block were laid, and the soil under the slab had been filled and compacted. We dug 4" deep trenches to carry the wire and tubing and dug 6" diameter holes with a post hole digger to position the thermocouples and tubing. We then filled the trenches and holes. In order to facilitate the entry of this mass of thermocouples and tubes (20 thermocouples, 20 tubes, and 2 instrument wires) from the soil into conduit in the west wall of the northeast room, PVC "street elbows" (elbows with a broad curve) were attached to the bottom of the vertical conduit runs. See Appendix D for additional information on the lab building and instrumentation.

In addition to the instrumentation under the building footprint, we realize the need for temperature probes in the soil outside of the building footprint. These thermocouples will be installed at a later date when the need for these soil temperatures and the best locations for these soil temperatures is well known. In order to facilitate running these thermocouple wires into the building, two additional conduits have been installed under the slab to the exterior on the north side and the south side of the building.

<u>Task 10</u>: Oversight and commissioning of construction of a laboratory facility.

Construction began in May 1999. Throughout the construction process, we observed the construction closely. This lead to catching a number of problems before there was considerable cost involved in correction.

For example, we had specified 1" extruded polystyrene wall insulation on the inside surface of the block walls. The contractor brought 3/4" polystyrene to the job site. We brought this to the subcontractor's attention. He then brought in 1" foil faced polyisocyanurate to the job site and before we found out what he was doing, he had covered 40% of the walls with the wrong product. We specifically did not want the polyisocyanurate product because it changes in R_value over time, and because this is a research facility, we wanted building thermal properties to remain as constant as possible. The insulation contractor pulled that all down, obtained the correct product, and then put up the correct product. However, he failed to seal the seams between adjacent sheets of insulation board (per specifications). After we pointed this out and made clear our expectations, he placed foil tape over all seams.

Perhaps the biggest crisis was faced when the steel trusses came in, and they were much smaller (in the vertical dimension) than called for in the plans. The construction documents showed the trusses being 6 feet minimum height, but they actually came in 4.5 feet minimum height. As a consequence, the walkway grid system was going to be 1.5 feet higher than planned, and this would mean reduced maneuverability in the ceiling space. After considerable discussion, a heavy duty "hanger system" was designed to allow the walkway grid to be suspended at the original planned elevation. This ended up costing the project $5000 because the hanger system was considered a contract modification.

About one month after taking over the building, we discovered a water leak which occurred twice, each time after a lengthy rain. Water came out from the bottom of the north exterior wall and puddled on the linoleum floor. The contractor spent about two weeks testing (spraying water on various surfaces of the building) and sealing joints, seams, etc. to try to figure out how water was getting into the north exterior wall. In the end, his conclusion was that the roof membrane was leaking in an area where water was ponding and was finding its way into the wall. A roof membrane patch has been applied to the suspected leak area. The contractor said that this was the most difficult water leak he has ever had to solve. Since the repair, no water intrusion has been noted.

Appendix D contains a detailed description of the Building Science Laboratory and Training Facility and its capabilities.

Appendix A

Advisory Committee Roster

Eric Althouse
Florida Department of Education

Larry S. Nelson
Florida Power and Light

Van Baxter
Oak Ridge National Laboratory

Morton Blatt
Electric Power Research Institute

Roy Crawford
Honeywell Inc.

Jack Davis
Florida Power Corporation

John McMillan (FACCA)
Florida Air Conditioning Contractors
Association

Randy Harris
City Gas Co.

Eli Howard
Sheet Metal and Air conditioning
Contractors National Association

Esher Kweller
US Department of Energy

Neil Leslie
Gas Research Institute

Mark Modera
Lawrence Berkeley Laboratory

David Odom
CH2M Hill

Mike Philo
Florida Power and Light

Bill Seaton
ASHRAE

Terry Sharp
Oak Ridge National Laboratory

Mike Schell
Engelhard Corp.

Phil Simmons
Orange Co. Building Dept.

Dwaye Sloan
Underwriters Laboratory

Ann Stanton
Energy Code Program
State of Florida Dept. of Community Affairs

Dilip Vyavaharkar
Carrier Corporation

Philip Wemhoff
Wemhoff and Associates
Jacksonville, Florida

Wayne Dunn
Sunbelt Engineering

Appendix B

Diagnostic Test Results for 20 Buildings

One of the primary objectives of this research project was to inspect and test commercial and institutional buildings, especially buildings with energy use, comfort, humidity, or indoor air quality problems, from a diagnostic approach. In other words, the inspection and testing was not just for the sake of gathering information but to actually determine how the building envelope, air moving systems, and HVAC systems were operating and determine what corrections, if any, would restore the building to proper working order. Tests were done in a total of 20 buildings selected because they had either large exhaust fans, leaky ceilings or they were educational facilities (Table 1).

Table 1
Twenty buildings tested for UAF and HVAC failures

Building Description	Large Exhaust	Leaky Ceiling	Education
1. Apollo elementary school	x		x
2. Chicken cafeteria restaurant	x		
3. Community center / clubhouse		x	
4. Floral Avenue elementary school		x	x
5. Eagle Lake Elementary School		x	x
6. Sea food chain restaurant	x		
7. River front restaurant	x		
8. Steak house chain restaurant	x		
9-12. Kennedy Middle School portables		x	x
13-15. McNair Middle School portables		x	x
16. Bar and grill	x	x	
17. Convenience store / fast food	x	x	
18. Ocean front restaurant	x	x	
19. Greek restaurant	x	x	
20. Charity office / thrift shop		x	

The following reports that summarize the test findings and retrofit recommendations were sent to each building owner/occupant.

1. APOLLO ELEMENTARY SCHOOL

Testing was performed to characterize the airtightness, air flow, pressure differentials, and HVAC system performance of the main building at Apollo Elementary School. The building was constructed in 1966. Recommendations were developed for possible retrofits.

Building Description

This is a one-story building with 50,139 ft^2 of conditioned floor area, including kitchen. An unconditioned and vented mechanical room is located adjacent to the kitchen. Acoustical tile ceilings are at 8'4" throughout the building except for the cafetorium (room doubles as cafeteria and auditorium) with 10'6" ceiling and the library with approximately 9'6" ceiling. Insulation is built into the roof system which is located on a metal corrugated roof deck. The roof acts as both primary air and thermal barrier, so the ceiling space can be considered to be largely within the conditioned space. Therefore, air leakage across the ceiling or into or out of ducts has relatively little consequence in this building. Interior layout consists of 31 classrooms, an office suite, a cafetorium, a library, a kitchen, 4 large bathrooms, 2 medium size bathrooms, 12 small bathrooms located in classrooms, 2 janitor closets, and 2 storage rooms. Cooling is provided by a central chiller which serves the entire building and several additional classroom buildings.

Test Results

Building air flows

Cooling is provided by chilled water. Three types of air handling systems serve the building; air handlers, fan coil units, and unit ventilators. Two air handlers serve the cafetorium, two air handlers serve the library, and three air handlers serve the office suite. Air flow rates were measured at all return and supply registers served by the air handlers. Supply and return air flow rates for the unit ventilators and fan coil units have not been measured, except for four unit ventilators. Note that on the day of our testing, October 17, 1997, the air flow rate from the unit ventilator in Room 17 was virtually zero; therefore this unit requires service.

A number of fan coil units serve the hallways and kitchen (2 in the kitchen). Each of the 31 classrooms has a unit ventilator. Outdoor air ducts enter the building from the roof. In the case of the classrooms, two rooms are served from each outdoor air (OA) duct. Based on measurements, the unit ventilators and air handlers draw a total of 4236 cfm of OA into the building. In the classrooms alone, OA equaled 3201 cfm or 5.6 cfm per student (desk) with “as found” fan settings (we counted a total of desks including teachers). With the classroom unit ventilators set to “high” fan speed, OA for the classrooms increased to 3943 cfm, or 6.9 cfm per student, and OA for the entire building increased to 4978 cfm. For several unit ventilators, changing the fan setting had little or no impact on fan speed and consequently little impact on OA flow. Note that measured OA in the classrooms varies considerably, ranging from 3 cfm per desk to about 8 cfm per desk, as found. With the unit ventilators set to “high” fan speed, OA varies from 3.3 cfm per desk to about 9 cfm per desk. These measurements were taken <u>after</u> the outside air damper modifications were completed (see below).

Originally, the control strategy caused OA flow to the unit ventilators to vary inversely with the cooling or heating load on the space (i.e., as the cooling or heating load increased, the OA flow decreased, and vice versa). In a hot and humid climate, a severe humidity control problem results from this control approach. During part-load cooling conditions, the cooling coil becomes warmer while the amount of entering humid air increases, and the warmer coil cannot remove this moisture. Brevard County school personnel recently remedied this problem by disconnecting the OA damper mechanism in each ventilator and setting it to a fixed position, thereby providing a constant volume of OA to each room regardless of the level of cooling or heating required. The OA flow quantities were reportedly set at approximately 5 cfm per person, based on statements made by school district personnel. The original construction blueprints for the facility did not delineate the OA requirement per person through the unit ventilators, but the OA settings for each ventilator were either 20% (16 units) or 15% (15 units) of the supply air flow. Assuming 30 students and 1 teacher per classroom, this equates to 7.1 cfm/person. Based on the current number of desks (574 including teachers), design OA (per original plans) for the building equates to 11.9 cfm/person.

One of the primary complaints in the school building is high temperatures in the kitchen. The kitchen does not have cooling except for two fan coil units, one near the serving line and one in the kitchen staff lunch room. The kitchen is primarily cooled by operation of the exhaust air (EA) system which depressurizes the kitchen and draws conditioned air from the school building when the doors from school into the kitchen are open. The original design intent was to have an unconditioned kitchen. The large exhaust fan was designed to draw air from the kitchen and make-up air would enter from outdoors through large louvered vent openings located at the base of the exterior kitchen walls, thereby providing some means of relief from the heat and moisture generated by the ovens and other cooking appliances. There is no mechanical make-up air. The original design intent has apparently been abandoned (louvered vents have been closed) in lieu of the current strategy to improve comfort for the kitchen staff. This strategy, however, causes the building to operate under negative pressure with respect to outdoors when the kitchen EA is operating.

The kitchen EA draws 8100 cfm from the kitchen when the doors between the kitchen and the school are open and 6500 cfm from the kitchen when the doors between the kitchen and building are closed (the difference is due to change in kitchen pressure). (Note that the construction documents specify that the kitchen EA should be 12,150 cfm; therefore, flow is only 67% of the rated flow with kitchen door open. This shortfall may be partly because of leaks in the fan housing (rust has caused large openings) and dirt on the blower wheel.) We also measured air flow from the cafetorium to the kitchen (through the door to the right of the stage) with only one door between the kitchen and building open, and that air flow was approximately 7000 cfm.

Exhaust fans are also located in the bathrooms. In four bathrooms near the kitchen, total measured flow was 1764 cfm versus 1740 cfm design. In two bathrooms near the office suite, the total measured EA flow was only 80 cfm (these flow rates were not specified in the design). There are 12 classroom bathrooms with estimated average 20 cfm EA in each, for a total of 240 cfm (these flow rates were not specified). For the entire building, bathroom EA total was measured at 2084 cfm.

In summary, the building air flow balance, as originally found, was 10,184 cfm of EA and 4236 cfm of OA, causing building pressure to be negative. With the kitchen EA off, the building pressure was positive because OA was 2152 cfm greater than EA.

Building pressure differentials

Building pressure and zone pressure differentials were measured. With the OA and bathroom EA operating but the kitchen EA turned off, the whole building pressure was +1.1 pascals with respect to (wrt) outdoors. This is consistent with the fact that OA exceeded EA flows by 2152 cfm (4236 cfm - 2084 cfm = 2152 cfm). When the kitchen EA was also turned on, building pressure went to -1.6 pascals. Again, this is consistent with the fact that EA now exceeds intake air flows by 5948 cfm (10184 cfm - 4236 cfm = 5948 cfm). In Florida's hot and humid climate, buildings should operate with slight positive pressure wrt outdoors, to avoid a number of problems, but especially moisture accumulation in wall cavities and other building cavities. The exhaust fans also depressurize individual rooms (see Table 2). The kitchen operated at -43 pascals wrt outdoors with the kitchen EA on and the doors between the kitchen and the rest of the school closed. (It operates at -6.7 pascals wrt outdoors when the two doors from the kitchen to the building are open.) Three of the four bathrooms near the kitchen were depressurized to -4.7, -11.5, and -8.5 pascals wrt the hallway because EA flow exceeded supply air. The fourth bathroom operated at +3.5 pascals because the supply air flow was greater than the EA flow. Ideally, these bathrooms should operate at a slight negative pressure wrt the hallways but at positive pressure wrt outdoors. Therefore, ideally the building would operate at say +2 pascals wrt outdoors and the bathrooms would operate at say -1 pascals wrt to adjacent portions of the building.

Table 2
Rated and actual air flows of exhaust fans and resulting zone pressures in Apollo Elementary School.

	boys-sw	girls-sw	boys-nw	girls-nw	boys-office	girls office	class-bath.	kitchen	total
rated cfm	435	435	435	435	30[1]	30[1]	360[1]	12150	14310
actual cfm	504	569	426	265	45	35	240	8100[3]	10184
dP (pa)[2]	-4.7[3]	-11.5[3]	-8.5[3]	+3.5[3]	na	na	-0.6	-43	
supply cfm	255	299	189	313	112	405	0	0	

[1] estimated flow based on specified exhaust fan
[2] pressures are with respect to outdoors
[3] note that pressures in these bathrooms reflect air flow balance between EA and the supply air from designated cooling systems that draw air from hallway, cool it, and deliver it into the bathrooms.
[4] note that kitchen EA was 6500 cfm with kitchen doors to cafetorium closed. Flow was reduced to about 5000 cfm when new grease extraction type filters were installed (doors open) and 4600 cfm with doors closed.

When visiting the school in October, we found that the existing screen-type filters had been replaced by grease extraction filters in the kitchen exhaust hood. Since these new filters cause considerably greater resistance to air flow and therefore reduce air flow, we found that pressure in the closed kitchen was only -22 pascals, down from -43 pascals with the old filters. When one of the 10 filters was removed, kitchen pressure went to -38 pascals. When two filters were removed, kitchen pressure was -47 pascals. Based on an airtightness curve for the kitchen (from a blower door test that we performed on the kitchen), we estimate that kitchen EA flow with the new filters was 4600 cfm with kitchen doors closed and 5000 cfm with doors from the kitchen to the building open. Therefore, these filters reduce kitchen EA by about 2000 cfm with the kitchen doors closed and 3000 cfm with the kitchen doors open.

Pressure mapping throughout the building found that pressures in the closed classrooms were generally slightly positive (mostly +0.1 to +1.5 pascals wrt the hallways). The closed rooms in the office suite generally were at +2 to +4 pascals wrt the main part of the office suite. The office suite was at +0.1 pascal wrt the remainder of the school. The library was at +0.9 pascals wrt the hallway. These small pressure differentials are of little concern. They primarily indicate that OA is being brought into those rooms causing them to be at positive pressure. The positive pressure in these rooms pushes air into the ceiling space. However, since the air pushed into the ceiling space remains within the building (since the ceiling space is inside the air and thermal boundary of the building), we estimate that there is little loss associated with air being pushed up into the ceiling space.

Building and duct system airtightness

A blower door test was performed to determine the airtightness of the building excluding the kitchen (doors between kitchen and building remained closed during the test). During the test, air handlers were left on, OA ducts remained open, and 6 bathroom exhaust fan were left running. Two blower doors were installed in an exterior hallway doorway (adjacent to the library and boy's restroom). With one blower door operating at full capacity, the building was depressurized 1.6 pascals. With two blower doors operating at full capacity, the building was depressurized 3.4 pascals. This indicates a building airtightness of 63,200 CFM50, or 8.6 ACH50, an acceptable building airtightness and substantially tighter than average for small commercial buildings (average airtightness of 69 small commercial buildings was found to be 16.7 ACH50; Cummings et al., April 1996).

A separate test was done to characterize the airtightness of the kitchen. With virtually all of the vent windows closed and doors to the remainder of the building closed, CFM50 was 7234 and ACH50 was 35.1 (EA discharge was also masked off). While this indicates a fairly leaky space, it is not surprising since there are so many louvered vents that leak considerably even when closed. The primary purpose of this blower door test was to obtain an airtightness curve that could be used to determine the air flow rate of the kitchen exhaust fan. From this curve we were able to fairly accurately estimate, for example, the EA flow rate with the grease filters (section 2.2).

No tests were done to measure airtightness of the duct systems because the ducts are located inside the primary air and thermal boundaries of the building and because performing duct airtightness tests is very time consuming and requires shutting down the air handlers. Because the ducts are located inside the primary air and thermal boundaries of the building, the energy penalties and changes in building infiltration (ventilation) associated with duct leakage are expected to be relatively small.

Cooling system performance

Air conditioning is provided to this building and the rest of the school complex by a chilled water system. Water leaves the chiller at 44F and enters the cooling units at about 46F, based on existing thermometers mounted in the chilled water lines. Tests were done to characterize the air flow, temperature drop, and dew-point temperature drop in several of the cooling systems. For these tests, thermostats were set to provide full cooling prior to the measurements. Testing found that in general, air flows are lower than specified and the enthalpy drops (based on temperature and dew-point drops) were much below rated values (Table 3). Measured cooling output in the office suite was only 25% of specified, and in the library was only 28% of specified. This is partly the result of air handler air flows that are 30% to 40% lower than specified in the office suite and library and partly because the temperature and dew-point temperature drops are less than expected. According to the construction documents, supply air should be 58.5F and 92% RH when entering air (return air + OA) is 80F and 50% RH, yielding an enthalpy drop of 7.0 Btu/lb (dry air). Measured enthalpy drops were averaging about 2.5 Btu/lb or less.

Table 3a

Air conditioner performance for the office suite at Apollo Elementary school, including air flows (cfm), temperatures (degrees Fahrenheit), and enthalpy (Btu/lb dry air)

(Nomenclature: AH = air handler, Tr = return temperature, Ts = supply temperature, RHr is return air relative humidity, RHs is supply air relative humidity, dh is delta enthalpy or the enthalpy change from return to supply (Btu/lb dry air), and kB/h is kBtu/hour cooling output.)

system	rated AH flow	rated OA	OA flow	return flow	supply flow	Tr	RHr	Ts	RHs	dh B/lb	actual kB/h	rated kB/h
1	1725	250	na	770	1322	73.5	58	63.4	87	1.9	11.2	49.2
2	1400	140	na	1050	1228	73.5	58	63.4	87	1.9	10.4	40.0
3	2055	300	na	830	1745	73.5	58	63.4	87	1.9	14.7	55.5
total	**5180**	**690**	**478**	**2650**	**4295**						**36.3**	**145**

Table 3b

Air conditioner performance for the library at Apollo Elementary school, including air flows (cfm), temperatures (degrees Fahrenheit), and enthalpy (Btu/lb dry air)

system	rated AH flow	rated OA	OA flow hw (RLF) [1]	return flow	supply flow	Tr	RH	Ts	RH	dh	actual kB/h	rated kB/h
6	2800	240	111 (191)	1242	1708	67.8	72	59.6	87	3.0	22.8	67.8
7	2800	240	16 (80)	1215	1541	68.2	66	58.3	94	2.3	15.8	67.8
total	**5600**	**480**	**127 (271)**	**2459**	**3249**						**38.6**	**136**

[1] "hw" = air flow measured by hot wire anemometer and RLF is air flow as measured by return leak fraction.

Recommendations

Following are recommendations submitted to the school district. The objectives of these recommendations are to achieve comfort in all zones of the building (with focus on the kitchen), achieve acceptable ventilation (at least meeting the original design), control humidity levels with the objective that they remain below 60% most of the time, and to achieve the preceding objectives with a minimum of energy use.

HVAC systems

a) Install covers (hinged doors) over the filter access for the air handlers since a significant fraction of the return air is being drawn from the ceiling space. Without these access covers in place, ceiling air has been bypassing the filters and a large amount of dirt has built up on the surface of the cooling coils. The dirt insulates the cooling coils and causes poor cooling/heating performance, and the dirt also slows air flow across the coil. Cleaning these coils and ***immediately*** installing filter access covers should provide the following benefits: improved cooling/heating performance, improved filtration of the air being circulated to the occupied space, and increased outdoor ventilation air (measured outdoor ventilation air for the air handlers was well below design, and simply installing these access covers should increase the OA flow somewhat).

b) **After** the filter access doors have been installed and the cooling coils have been thoroughly cleaned, measure the OA (ventilation air) provided by the air handlers and adjust the flow rates to match the design specifications. Also, re-measure the temperature and humidity drops across the cooling coil for comparison with design values. If significant discrepancies exist (after adjusting for any differences between design and actual entering air conditions), confirm the proper air flow rate across the cooling coil and proper chilled water flow through the coil. In spite of measured cooling output being far below specified levels (Table 3 of this report), we have received no reports of cooling problems on hot summer days, except that the cafetorium becomes hot toward the latter portion of the lunch serving period.

Kitchen

a) Closely scrutinize the EA requirements for the kitchen hood. The EA flow rate should be as small as possible in order to save energy while it should be sufficient to capture and remove heat and contaminants emanating from the cooking appliances. The original design called for 12,150 cfm of exhaust. When we first arrived on site, the exhaust flow was measured at around 8,100 cfm as typically operated with the doors between the kitchen and the cafetorium open. Subsequent installation of grease extraction filters has further decreased the exhaust flow to around 5,000 cfm. Surprisingly, feedback from school personnel indicates that kitchen workers are more comfortable since the grease filters were installed, indicating that the hood may be effectively capturing most of the cooking by-products at this lower EA flow rate.

   The 1997 Standard Mechanical Code currently mandates 150 cfm/ft^2 for a canopy hood which is open on all sides (as this one is). This would set the current code exhaust requirement for this hood at nearly 14,500 cfm. However, the Code allows these exhaust requirements to be reduced if the hood is designed and certified for proper performance by a licensed architect or professional engineer, or if an approved prefabricated hood is installed which is tested and certified by the manufacturer. Since the existing hood was not prefabricated, tested, and certified by a hood manufacturer, the exhaust requirement may be reduced if a licensed architect or professional engineer certifies that the equipment is properly capturing and exhausting cooking fumes. As a point of reference, typical minimum exhaust flow rates for listed (prefabricated, tested and certified) hoods for light and medium duty cooking equipment range from 150 to 300 cfm per linear foot of hood (per side in the case of the double canopy hood under consideration). Since this hood has 11 linear feet on each of the two sides, this equates to between 3300 cfm and 6600 cfm of total flow, which is <u>much</u> lower than the Code minimum for a non-listed hood and points out the advantage of using a listed hood.

   The minimum exhaust rates noted above are somewhat misleading, because it is really the types of cooking equipment and the food products being prepared that dictate the exhaust requirements. Given a specific list of cooking equipment and food products being prepared, most major hood manufacturers can provide guidance regarding the EA requirements for your specific application.

   In the case of the exhaust hood for Apollo elementary, the current exhaust rate may well be sufficient. We recommend contacting a hood manufacturer and/or kitchen hood expert to verify the proper exhaust rate for this hood. In addition, adding "eyebrow" extensions to the existing hood **<u>or</u>** moving the cooking equipment further underneath the existing hood will increase the capture efficiency of the hood (see recommendation b. below).

Another item to keep in mind is air velocity in the exhaust duct. A minimum velocity is usually required, primarily to minimize grease buildup on the duct work and reduce the risk of fire. It is our understanding that the types of food cooked at this school do not produce grease-laden vapors. This is also consistent with our observation that there is no evidence of grease build-up on louvers at the fan discharge. Check with an exhaust hood manufacturer and/or kitchen hood expert to make sure that exhaust duct velocities meet any minimum velocity requirements which may apply.

b) Move the kitchen cooking appliances (ovens, stoves, etc.) more fully under the exhaust hood. According to hood manufacturers, the 1997 Standard Mechanical Code, and the 1995 ASHRAE Applications Handbook, the exhaust hood should overhang the equipment they serve by a minimum of 6", but most manufacturer's suggest 12" overhangs to ensure proper capture and containment of heat, moisture and other contaminants generated by the cooking process. ***This overhang should be measured from the <u>inside</u> edge of the grease trough located at the perimeter of the hood.*** If there is insufficient room under the existing hood to meet the overhang requirements, "eyebrow" extensions could be added to the existing hood to broaden the capture area of the hood. Eyebrow extensions would only be required on the sides of the hood where the overhang distances noted above are not met. Consult with a hood manufacturer or exhaust hood expert for proper installation of these eyebrow extensions. Proper capture and containment of heat and moisture from the cooking appliances would help to maintain comfort for the kitchen staff.

It is also worth noting that in this particular application, cooking fumes that are not captured immediately by the hood are soon re-captured by the hood, since there is a strong flow of air toward the hood from all portions of the room. This recapture is quite effective in the Apollo kitchen because the kitchen is well isolated from the remainder of the building (by walls) and because there is no air distribution system in that space which can pick up the escaped cooking fumes and distribute (and condition) them.

c) The kitchen exhaust air fan housing on the roof is rusting, such that there are **<u>significant</u>** openings which allow considerable air leakage into the fan housing from outdoors. Also, the blower wheel needs cleaning. Once the hood exhaust requirements have been established (item a. above), consider fixing the housing and cleaning the blower wheel. Also, replace the new grease filters with the original screen-type filters since the grease filters are unnecessary (no grease-laden vapors produced by the cooking at this school) and they only create an additional pressure drop for the fan. All of these actions should increase exhaust flow, so the fan speed may have to be reduced to match the newly-established exhaust requirements (energy savings!). Since the existing motor/fan assembly is nearing the end of its useful life, it may be more cost effective to replace the entire unit with a new, more efficient unit. This new unit may be significantly smaller than the existing unit, depending on the final determination of EA requirement.

d) Add an air handler or air handlers (but no cooling coil) and ductwork to move air from the hallways into the kitchen. The total air flow for this air handler (or air handlers) should be roughly 80% of the new hood exhaust requirement (see item 3.2a above). In effect, this will pull air out of the building and dump it into the kitchen in a more controlled manner than is currently being used (currently drawing air from the cafetorium through open doorways). The purpose of this is twofold. First, it is "make-up air" to match a large fraction of the EA drawn from the room. Second, it is a cooling system for the kitchen staff, providing spot cooling to specific locations in the kitchen. We suggest providing diffusers which can provide directed streams of air and which can be adjusted to match the working locations of the staff. This arrangement should provide improved cooling and a greater sense of control over their environment.

A note of caution: don't use conventional supply air diffusers close to the hood. The supply air flow, if improperly located and directed, may disrupt the air flows around the exhaust hood and thereby decrease capture efficiency. Using conventional supply air diffusers, which tend to spread supply air over a large area, may interfere with proper EA flows around the hood. For diffusers near the hood, we suggest using designs that direct air toward the floor, such as perforated plate supply diffusers with the 4-way directional grilles or circular plates removed, or other types of diffusers which allow air to be directed down toward the floor but not toward the hood. We also suggest that these diffusers be located no closer than 3 feet from the outer edge of the exhaust hood. For diffusers located near the hood, the supply air flow rates should be fairly slow; to optimize exhaust hood capture, velocity should be about 40 feet per minute at work level. The air velocity can be higher at locations further away from the hood (e.g., at the food serving line), and in fact high velocity jets of air may provide greater comfort for staff. With respect to air velocity near the hood, there is a trade-offs between discharge air velocity, comfort, and hood capture efficiency. Higher air velocities will yield greater comfort for the kitchen staff in a hot kitchen, yet these higher velocities also have the potential to disrupt exhaust hood capture effectiveness. In this particular application, it may be reasonable to err on the side of higher velocities because the cooking by-products which may escape initial capture will be completely recaptured within a short time. On balance, more comfort may be produced by higher velocity supply air than is compromised by reduced hood capture efficiency (this assumes, of course, that the supply air is not directed toward the hood, rather downward toward the floor).

The fan for this air handler (or air handlers) should be interlocked with the exhaust hood fan control, so that it (they) operates whenever the kitchen EA operates. It should also have an override time control which would allow turning on the air handler fan when the EA is off, for up to say 2 hours at a time, to provide adequate cooling to the kitchen workers after the cooking appliances and EA have been turned off for the day.

Note that this make-up air being delivered to the kitchen is being drawn from the surplus of OA that is being provided to the building. It is part of the overall building air flow balance. If the kitchen EA is 8100 and the make-up air is 6500, and the OA flow rates are increased to the original design specifications, then the building will still operate at positive pressure even with the kitchen EA operating. See later section called "Overall Building Pressure" for discussion of building air flow balance.

Bathrooms

a) All bathrooms should operate at a slightly negative pressure (say, -1 pascal or greater) wrt the surrounding occupied spaces to ensure that odors are properly exhausted from the building. As shown in Table 1, several bathrooms are depressurized beyond 1 pascal. Greater depressurization of the bathrooms is both good and bad. It is good because it will better contain odors within the bathroom space. It is bad because bathroom walls that are building exterior walls may experience moisture accumulation in the walls. However, since these bathrooms have been at significant negative pressure for many years and do not have apparent moisture problems, the level of depressurization does not appear to be critical. In addition, one bathroom operates at positive pressure, promoting the transport of bathroom odors into the adjacent areas. Two other bathrooms near the office suite operate at near neutral pressure wrt the hallway. To obtain proper depressurization, the supply air provided by the AC system should be slightly less than the EA leaving the bathroom. Pressure measurements should be taken <u>after</u> the air flows are adjusted to confirm proper depressurization of the bathroom.

b) In the toilet rooms located at the west end of the building, the design documents call for 435 cfm of exhaust per room, with a fan coil unit providing 400 cfm of conditioned supply air from the adjacent hallway. The exhaust specification equates to roughly 2.5 cfm/ft^2, versus the current 1997 Standard Mechanical Code which requires only 2 cfm/ft^2. Consider reducing the exhaust requirement per bathroom from 435 cfm to approx. 350 cfm (confirm actual bathroom floor area, then multiply by 2 cfm/ft^2 to obtain final exhaust flow). Then the supply air through the fan coil units could be reduced to about 315 cfm, and this amount of air should still provide adequate cooling to the bathroom. You will see in Table 1 that fan coil supply air flows for these toilet rooms (boys-sw, girls-sw, boys-nw, girls-nw in Table 1) currently range from 265 cfm to 569 cfm. Note that as an energy saving measure, it may be possible to stop conditioning the supply air into those bathrooms and still maintain acceptable temperature control in those zones. This could be achieved by adjusting the thermostats to those bathrooms so that heating and cooling is disabled, or shutting off the chilled water to the coils.

Outdoor air

a) We suggest that OA to the building air handlers (3 for the office, 2 for the library and 2 for the cafetorium) should be reset to the original specifications (construction blueprints) following installation of filter access panels and thorough cleaning of the cooling coils.

b) Currently the amount of OA to each classroom is uneven, ranging from about 3 to 8 cfm per desk (5.7 cfm/desk average). Therefore, consider making OA more uniform throughout the classrooms by adjusting the OA dampers on the unit ventilators. ***Be aware that the OA duct entering half of the ventilators includes flow from a return air duct from the common work room (one work area for 4 classrooms). The air flow rate through this return duct must be accounted for when measuring the "OA flow" to these unit ventilators (i.e., the outside air duct entering these unit ventilators is actually a mixture of OA and return air from the work room). Consult the construction blueprints for clarification.***

c) Consider increasing the OA flow rates to the classrooms. When we measured OA, the flow rates to the classrooms were 5.6 cfm/desk on average. When we set all unit ventilators to "HIGH" fan speed setting, the outdoor ventilation increases to about 6.9 cfm per desk. (Reportedly, teachers change fan speeds to adjust room comfort. Of the 31 unit ventilators, 4 were off, 6 were set to low, 6 were set to medium, and 17 were set to high.) The original design specified a minimum of about 7 cfm/person, which matches well with the current settings <u>***IF***</u> the teacher leaves the fan speed at the "HIGH" setting. Our measurements, however, found that on average, the outside air flow as measured was 22% less than the "HIGH" setting (5.6 cfm/desk versus 6.9 cfm/desk).

In order to maintain the original design intent of 7 cfm/person (minimum), you may want to increase the OA to 9 cfm/person with all fans on "HIGH" to achieve roughly 7 cfm/person as normally operated (this assumes that this same diversification factor will exist in the future).

d) Another idea to consider is increasing ventilation to the current State requirement of 15 cfm/student (per Chapter 6A-2, Florida Administrative Code). This OA rate is significantly higher than the original design rate, and the existing unit ventilators may not be able to provide proper cooling and dehumidification of this large quantity of OA. Consult with the unit ventilator manufacturer to determine if proper cooling/ dehumidification capacity will be available from the existing ventilators before increasing the OA to this high level of 15 cfm/person.

e) Consider installing controllers to vary the OA flow rate on schedule, particularly for the 7 air handlers. OA can be shut off when not needed. It is our understanding that the HVAC system is operated from roughly 6 AM to 9 PM, 5 days per week based on a time clock. When the HVAC system operates, the OA is continuously introduced to the building regardless of whether school is in session. Providing OA to the building during the 6 AM to 8 AM morning hours dilutes any contaminant build-up over nights and weekends, plus it provides make-up air for the kitchen exhaust hood which typically operates first thing in the morning for breakfast and lunch preparation. The OA must, of course, be provided when school is in session for the students and staff. While OA can be varied during occupied hours based on occupancy levels (using carbon dioxide sensors/controllers to open and close OA dampers), it is unlikely that this strategy would be cost effective because of the distributed way the OA is introduced to the building (roughly 20 OA intake ducts).

We assume that operation of the HVAC system after 3:30 PM is for the cleaning crew and/or the periodic nighttime events (various large group meetings, etc.). A large amount of energy is required to condition the outdoor ventilation air during these periods of little or no occupancy. There may be an opportunity to save a significant amount of energy by reducing the outdoor ventilation air from 3:30 PM to 9 PM.

Consider installing dampers in the outdoor ventilation air ductwork for the air handlers (total of 5 ducts: 1 duct for the office air handlers, 2 for the cafetorium and 2 for the library). From 6 AM to 3:30 PM, the dampers can be fully open to provide the necessary OA. From 3:30 PM to 9 PM, the dampers can be mostly closed (perhaps 20% to 30% of maximum), leaving enough air for the cleaning crew and accounting for the use of cleaning supplies and the extra ventilation required to dilute these contaminants (consult industrial hygienist for proper ventilation for cleaning crew). The dampers would remain 20% to 30% open during nights and weekends, but the air handlers will be turned off at these times. The motorized OA dampers can be controlled by a single time clock located in the office. An override switch can be provided for the periodic night time event (large group meetings, etc.). Since the kitchen exhaust hood will not be operating from 3:30 PM to 9 PM, the building should remain pressurized even with the reduced OA.

Overall building pressure

a) In Florida's hot and humid climate, it is recommended to keep buildings at slightly positive pressure wrt outdoors to minimize wall cavity moisture accumulation problems. Slight positive pressure means that relatively cool and dry air from the conditioned space is flowing outward through any cracks and crevices in the building envelope, instead of hot and humid air being sucked into the building if the space is depressurized. This pressurization should exist in all conditioned spaces that are adjacent to unconditioned spaces or outdoors. It is not essential that the building be at positive pressure at all times. It is important, however, that the building operate at positive pressure a substantial portion of the time.

b) Positive pressure in the building is produced when a positive air flow balance exists for the building as a whole and when interior building zones are fairly well connected. A positive air flow balance means that the total amount of OA coming into the building exceeds the total of all exhaust fan flow rates. Consider some examples. If OA flow rates are reset to original specifications – 1700 cfm for cafetorium, 480 for library, 690 for administration, and 6812 for 31 classrooms – then the total OA will be 9682. In this case, the air flow balance will be positive if the kitchen EA remains at the current 5000 cfm, since total building EA (kitchen plus bathrooms) will be 7160 cfm. If the kitchen EA increases to 8100 cfm (this with screen filters), then the building will be at a slight negative air flow balance, since total building EA will be 10,260 cfm. Therefore, during the hours when the kitchen EA operates, the building will operate at a very slight negative pressure (estimated to be -0.1 pascals). This level of space depressurization during only a portion of the day will not cause problems, since the building will be at a strong positive pressure when the kitchen EA is off. If the classroom OA flow rates are set to 15 cfm/person (574 desks), then total OA would be 11,480 cfm and the building would operate at positive pressure at all times even with 8100 cfm kitchen EA.

Actual building pressure will be dictated by the exhaust and OA decisions described previously in an earlier section. The goal is to operate the building with slight positive pressure most hours when the HVAC system operates. The preceding paragraph describes several possible scenarios, but they are only examples.

Library

a) Seal up both the passive vent and exhaust fan in the ceiling space above the library. The exhaust fan is apparently inoperable, and even if it were operable, there is no reason to exhaust air out of that ceiling space (the insulation is located at the roof deck). According to the construction documents, this fan is rated at 3540 cfm. If it were to operate, the building would operate at negative pressure. Currently, the exhaust fan and the passive vents simply act as holes in the building, allowing air to leak into the building when the HVAC systems are turned off and diminish the ability of the building to operate at significant positive pressure when the HVAC systems are operating.

[This concludes the report of testing and recommendations at Apollo Elementary School.]

2. CHICKEN CAFETERIA RESTAURANT

Initial testing of this restaurant occurred in 1994 as part of another project. Additional testing was done in 1998 as part of this project.

Test report 1994.

1) Based on blower door tests, CFM50 was 6995 and ACH50 was 14.8. The building is therefore moderately leaky. It would be reasonable to expect ACH50 of 6 in a building like this.
2) There are 35 tons of air conditioning in four air conditioning units. Each has outdoor air.
3) Four kitchen exhaust fans and two bathroom exhaust fans are discharging 10,616 cfm (cubic feet per minute) of air from the building.
4) Air entering the building by mechanical means totals 8410 cfm, by means of two "make-up air fans" and "outdoor air" on each of the four air handlers.
   a) Two make-up air fans, that run simultaneously with the kitchen exhaust fans, draw 6157 cfm of air into the building.
   b) Four air handlers draw outdoor air into the building when they operate. When all four air handlers are operating simultaneously, they draw 2253 cfm of outdoor air into the building.
5) When all the exhaust fans and all the air handlers are on, the building experiences 2196 cfm net air leaving the building (10,616 cfm - 8,410 cfm = 2196 cfm).
6) This net air flow of 2196 cfm out of the building depressurizes the building to about -8 pascals (0.032 inWC).
7) The water heater drafts properly when the building is at -8 pascals. This means that the strength of the water heater draft is sufficient to overcome the negative pressure of -8 pascals.
8) The air handlers do not stay on all the time. In fact they cycle according to the cooling load. When the thermostat setting is satisfied, the air handler turns off, and the outdoor air ceases. When all four air handlers are off, the building depressurization is -17 pascals. At this pressure, the water heater will not draft. Rather, air is drawn down the vent pipe from outdoors and the combustion gases emerging from the water heater flue are discharged into the water heater closet, and then into the room. We measured the water heater exhaust gases and found that carbon monoxide levels in the flue were low (10 parts per million).
9) The owner reported that on two occasions he had observed flame roll-out from the water heater. We did not observe flame roll-out during our testing. Even with the outdoor air for the four air handlers blocked off, and the building pressure at -23 pascals, the water heater did not produce flame roll-out. We surmise that flame roll-out will occur under the following conditions. All four air handlers are off and the water heater has been off, and then the water heater turns on. At the moment when the water heater fires, a strong flow of air is coming down the vent pipe and also down the flue (inside the appliance), and this air pushes the gas down and out of the fire chamber. Thus, an observer would see fire burning up around the sides of the water heater.
10) As noted above, the restaurant building is relatively leaky. We measured one nearby fast-food restaurant, for example, that is about five times more airtight. The importance of this is that if your building were more airtight, the pressure imbalances would be much greater. If the building was five times more airtight, the level of space depressurization would be more than five times greater. In fact, another restaurant from the same chain located about 10 miles away is much tighter and operates at -43 pascals to -63 pascals, depending upon which air conditioning fans that are operating.

11) The mold and mildew problems, as well as the wall paper separating from the wallboard, is also a consequence of negative pressure in the building. This is how it occurs. The negative pressure caused by the exhaust fans draws humid outdoor air into the building and into interstitial cavities in the walls. The humid air in the walls comes into contact with the cool surface of the back side of the gypsum board. Because the gypsum board temperature is cool and the dewpoint temperature of the air in the wall cavity is high, the surface relative humidity is at or near 100%. As a result, moisture accumulates in the gypsum board material. If the interior surface of the gypsum board (facing into the room) were paint or vapor permeable wall paper, then the gypsum board could dry to indoors (assuming the air conditioners maintained reasonably low humidity conditions). However, in your building, the wall paper is vinyl, which restricts the flow of moisture vapor. Consequently, the gypsum board cannot dry to inside. As the moisture content of the gypsum board increases, mold and mildew begin to grow, and the wall paper begins to fall off the walls.

The following solutions were suggested:

1) The building should operate at positive pressure. This means the amount of air brought into the building should be greater than the amount leaving the building through the exhaust fans. This can be done by decreasing the flow rate of the exhaust fans or increasing the amount of air brought into the building. Consult with a qualified engineering firm to determine the best means to achieve the desired air flow balance.

2) Do not plan to use the "outdoor air" of the air handlers to balance the building air flow. This is because the air handlers do not run all the time, and the building will be depressurized when the air handlers are off. If you should think that the air handlers could be operated continuously, I would not suggest against that. Cycling the air handlers with the compressor operation provides the best dehumidification performance of the air conditioners. If the air handlers are operated continuously, then they will remove much less humidity and the interior relative humidity will increase.

3) <u>Close off the water heater closet from the rest of the building</u> and provide code required combustion/dilution air to the closet. This will, in effect, put the water heater outdoors and largely protect it from the pressures which may occur within the building. Consider, for example, what would happen to building pressure if one of the two make-up air fans stopped, perhaps due to motor burn-out. The building could be considerably depressurized and flame roll-out could be even a greater concern.

Test Report 1998

This restaurant is a 3161 square foot slab on grade with block walls, suspended t-bar ceiling at 9 ft, insulation batts located on ceiling tiles, and metal roof deck. Four air conditioners of 7.5, 7.5, 10.0, and 10.0 tons capacity are located on the roof with ductwork in the ceiling space. Each AC unit has outdoor air.

Test results

Testing was done on May 11, 1998 and May 19, 1998. On the first date, one of the AC units would not turn on. Transformer replacement got the AC unit to work on the second test day.

Building airtightness was determined by means of a blower door test. The building airtightness was found to be 3165 CFM50 or 6.7 ACH50 which makes this building moderately tight. Airtightness was measured at 14.8 ACH50 in 1994 – reasons for the greater airtightness in 1998 have not been identified. When the building was at -50 pascals, pressure in the ceiling space was found to be -48 pascals wrt outdoors, thus indicating that the roof deck and not the ceiling is the primary air boundary. Note that the thermal boundary, however, is at the ceiling.

Building pressure was measured. Normal operating pressure was found to be -22.8 pascals with respect to (wrt) outdoors. This indicates that the exhaust fans are greater than the sum of make-up air, outdoor air, and return leak air. In 1994, building pressure was normally about -8 pascals. The tighter building envelope causes the space depressurization to increase.

Make-up air (MA) and outdoor air (OA) were measured using the building as a capture tent. Blower doors were installed in an exterior doorway to blow air into or out of the building to substitute for various fans that were turned off (and/or sealed).

> In the first instance, the OA intakes were sealed (using duct mask) and the blower doors had to blow 1656 cfm into the building to return the building to a pressure of -22.8 pascals. Therefore, we conclude that OA flow is 1656 cfm.
>
> In the second instance, MA was turned off and sealed while OA was still sealed. The blower doors had to blow 7893 cfm into the building to return the building to a pressure of -22.8 pascals. Therefore, we conclude that MA flow is 7893 - 1656 = 6237 cfm

Exhaust air (EA) was determined by two methods. The first uses the air flow measurements of OA and MA, and the calculated Net EA flow based on the building airtightness curve. The second method uses a dedicated capture tent with blower door.

1) From the building airtightness curve, $Q = C\,(dP)^n$, we can calculate Net EA flow (which is also equal to the flow rate of air into the building because of the -22.8 pascal building pressure). Based on the building airtightness curve, Net EA flow is $Q = 312.67\,(22.8)^{.59} = 1978$ cfm. EA flow rate can be calculated by the following equation, where RL and SL are the return leak and supply air flows. RL was measured by means of the tracer gas return leak fraction test. SL, however, is not know. Based on the fact that one quarter of the roof top air handlers (package units) are on the supply side of the system, we estimate that SL is equal to 300 cfm.

EA	=	OA + MA+ Net EA + RL - SL
	=	1656 cfm + 6237 cfm + 1978 cfm + 868 cfm - 300 cfm
	=	10439 cfm

In summary,

OA + RL	=	2524 cfm
MA	=	6237 cfm
Net EA	=	1978 cfm
EA	=	10439 cfm

Measurements from 1994 found similar air flow rates:

OA + RL = 2253 cfm
MA = 6157 cfm
Net EA = 1986 cfm
EA = 10396 cfm

2) EA was also measured by means of a capture tent with blower door positioned over the exhaust "mushrooms". Total EA flow was determined to be 9534 cfm.

Air distribution system air flow rates were measured. Return and supplies were measured with air flow hoods. Return leakage and outdoor air were measured by means of tracer gas "return leak fraction" test.

Table 4

	AH#1 (7.5 ton)	AH#2 (7.5 ton)	AH#3 (10 ton)	AH#4 (10 ton)	TOTAL
Return	2365	2333	2334	2205	9237
*Return leakage	43	170	197	458	868
Outdoor air	352	298	530	471	1651
Return total	2760	2801	3061	3134	11756
Supply	2915	2839	2973	3301	12028
**Supply x 1.05	3061	2981	3122	3466	12630

* Return leakage flow was determined with the OA vents masked off.
** This adjusts supply air flow for an estimated 5% supply leakage.

Based on these measurements, it appears that total air handler flow rate is in the range of 12,000 cfm, or about 340 cfm per ton, which is generally within an acceptable range.

Building ventilation was measured by means of tracer gas decay. With EA, MA, OA, and AHs all operating, the building ventilation rate was 3.0 ach. Based on a volume of 28,449 cubic feet, this equates to a ventilation flow of 1424 cfm across the building envelope.

Air conditioner performance. Air conditioner cooling output is substantially below rated capacity.

AC unit #1 had a measured cooling output of 46,000 Btu/hr, or 51% of its rated capacity.
AC unit #2 had a measured cooling output of 79,800 Btu/hr, or 89% of its rated capacity.
AC unit #3 had a measured cooling output of 81,900 Btu/hr, or 68% of its rated capacity.
AC unit #4 had a measured cooling output of 86,900 Btu/hr, or 72% of its rated capacity.

In combination, these four AC units are providing cooling of only 294,600 Btu/hr, or 70% of the rated cooling output of 420,000 Btu/hr.

Servicing of these four AC units, and especially unit #1 would likely bring their performance back to or close to standard and would reduce cooling costs.

Retrofit recommendations:

The following retrofits were recommended to improve system and building performance.

1) Modifications to EA system. Move broiler cooking to one end of the exhaust hood and close down one exhaust fan when not needed. Remove filters in portion of hood that is not required and replace with plugs (plates). If greater amount of broiler cooking is required, then that EA fan can be turned only when needed. Turning off one EA fan will reduce total exhaust by about 2600 cfm, thus bringing total kitchen EA flow to about 7800 cfm.

2) With EA at about 7800, current MA flow of 6237 will be correctly sized (about 80% of EA flow). OA flow can be reduced from 1656 cfm to about 1200 cfm. After these changes, Net EA will decrease from current +1978 cfm to -505 cfm (intake air flow will be 505 cfm greater than EA flow) and the building will operate at positive pressure.

3) If EA flow is not decreased, then increase MA from 6237 cfm to 8000 cfm (73% of EA flow). Increase OA from 1656 to 2000 cfm. These two changes will decrease Net EA from current +1978 cfm to -129 cfm (intake air flow will be 129 cfm greater than EA flow) and the building will operate at slight positive pressure when all four air conditioners are operating.

4) Move several cooking appliances farther back under the exhaust hood in the kitchen. If this is not feasible, then consider installing an "eyebrow" hood extension (out into the room) to capture a greater portion of the cooking by-products that currently escape the hood especially when the cooling appliance doors are opened.

5) Move return air grills that are located in close proximity to the exhaust hoods. Because they are close to the hoods, fumes that escape from the hood are quickly drawn into the air distribution system and add to the cooling load. By locating the returns farther away, the fumes are more likely to be drawn into the hood before getting into the cooling system.

6) Move several supply registers that are close to the exhaust hoods farther away. Also, remove the "circle plates" located in these diffusers so that air flow will be directed downward rather than outward toward the hoods. Both of these modifications will cause less disruption of the hood capture.

7) Service the AC units so that they produce at least 80% of their rated cooling output.

Proposal:

The Florida Solar Energy Center (FSEC) proposes to assist Boston Market in the implementation of and cost of these retrofit measures. FSEC will provide consultation and additional testing at no cost. We will assist by meeting with contractors to arrange bids and oversee retrofit work. We will pay the cost of bringing in an exhaust hood expert for consultation, if needed. We will also pay for up to $5000 in retrofit costs.

We would ask Boston Market for the following: 1) to allow FSEC to monitor energy use, AC performance, and indoor temperatures and humidity for a period of 12 months; 2) to allow contractors to visit the store to provide estimates and make retrofits; 3) to pay for the balance of the retrofits chosen for implementation (to be jointly agreed upon); and 4) arrange for (or allow FSEC to arrange for) those retrofits to occur at approximately half way through the monitoring period.

Short-term Monitoring Results at Chicken Cafeteria Restaurant

Short-term monitoring was performed on this chicken cafeteria restaurant during the fourth quarter of 1997. A short-term monitoring plan was developed before installation. Monitoring ran for 4 weeks. The following parameters were monitored: four roaster temperatures (exterior temperature for operation status), four exhaust fan discharge temperatures, outdoor temp and RH, outer and inner roof surface temperatures, ceiling space temp and RH, dining room temp and RH, temp and RH at supply and return in 2 dining room AC units, temp and RH in front and back kitchen areas, DHW flue temp as status indicator, building pressure, CO in water heater closet, CO_2 in dining area, and run time for all four AC units. We found the building operating at -25 to -32 pascals under normal operation depending upon how many of the four AC units were operating. The AC units cycle on and off in accordance with cooling and heating requirements determined by four separate thermostats. Therefore, outdoor air cycles on and off, and building pressure fluctuates up and down in response in the range of -25 to -32 pascals.

Testing had been performed on this building several years previously. The building was operating at -3 to -17 pascals pressure depending upon the number of roof-top AC units operating. Recommendations were made at that time regarding how to bring the building to positive pressure. These recommendations are contained in a published paper (Cummings, Withers, and Shirey, 1997). These recommendations had not been followed. In fact, we were surprised that in fact the building depressurization had increased.

Two major consequences of depressurization had been observed in the previous testing; 1) mold and mildew growth behind the vinyl wallpaper of the newly constructed store and 2) backdrafting and flame roll-out from the gas water heater located in a closet. Originally, combustion/dilution air to the closet was provided by large grills in the closet door and above the closet door. We had recommended that these vents be closed up and combustion/dilution air ducts be provided to outdoors through the roof. They did install two vertical ducts for combustion/dilution air but kept the door vents open. Therefore, the closet was exposed to essentially the same level of depressurization as the main body of the restaurant. To stop flame roll-out, they had installed butterfly dampers at the top of the flue, and these would open only after the combustion started. Nevertheless, the water heater continued to experience backdrafting of all its combustion gases whenever it operated. The butterfly dampers have apparently stopped the flame roll-out incidents.

3. COMMUNITY CENTER / CLUBHOUSE

This community center and golf country club facility was five years old at the time of testing and had problems with high humidity, mold/mildew growth on ceiling tiles and behind vinyl wallpaper in the auditorium portion of the building, comfort problems in some zones, and high utility bills. Extensive testing was performed on HVAC system performance and for uncontrolled air flow. The 24,000 square foot facility is served by a 100 ton chiller and five air handlers. The chillers, as found, were scheduled to run 24 hours per day 365 days per year. The air handlers operated continuously

23 hours per day, and cooling output was controlled primarily by means of three-way valves that modulate the flow rate of chilled water through the coils. Therefore, under part load conditions, the cooling coils are warm and therefore dehumidify poorly.

Two of the systems also modulate cooling output by means of partial variable air volume operation (inlet guide vanes; the VAV control in the air handlers was not operating, so in effect these were constant volume units). This control system causes warm coil temperatures, poor dehumidification, and high indoor RH (typically 65% to 80% in the summer and fall). Electric energy bills are about $5000 per month. Pump power (for the chilled water circulation) and fan power (for the air handlers) together are consuming about $1300 per month, or about 25% of total building energy and perhaps 40% of total HVAC energy use. Heat generated by the motors of the air handlers alone accounts for approximately 7 tons of cooling load. The ductwork is located in the attic space above the ceiling and insulation (which is "floating", that is, attached to the bottom of the truss system which is located from two to six feet above the suspended ceiling). This attic space is below red asphalt shingles on a 5/12 sloped roof over plywood decking (like a residential attic space) and is quite hot during the summer.

Uncontrolled air flow findings

In terms of uncontrolled air flow, several important issues were identified. Duct leakage was measured on two of the five systems.

Table 5

Duct leakage measurements ($CFM25_{total}$) on two of the building's five AC systems

System	return CFM25	supply CFM25	combined CFM25
27-ton	212	408	620
13-ton	132	229	361

Duct leakage was found to be 620 $CFM25_{total}$ for the 27-ton system and 361 $CFM25_{total}$ for the 13-ton system. Since the ductwork actually operates at an average pressure of about 50 pascals, actual operating duct leakage may be about 50% greater than these test numbers, or a total leakage for these two systems of 1500 cfm, or 3700 cfm projected for the entire building (five AC systems). This would represent about 12% of total air handler air flow. Note that the duct leakage measurements are to "everywhere", not just to outdoors. However, since the ductwork is located almost exclusively in the attic space, the largest majority of the leakage is with respect to unconditioned space.

Four of the five AC systems have outdoor air (OA). During the duct tightness test of the 27-ton system, the OA ductwork was included in the test as well. This duct, which is about 30 feet long, has leakage of 121 $CFM25_{total}$. With an estimated OA duct operating pressure of -50 pascals, the actual leakage would be about 180 cfm. Since this duct is largely inside the building, the leakage into the OA duct reduces outdoor air and therefore building ventilation.

Leaky Ceilings Cause a Leaky Building

The most dominant uncontrolled air flow feature of the building is the leaky ceiling. This is of special interest because the ceiling was found to be extremely leaky (much more so than the typical suspended t-bar ceiling), the attic space above the ceiling is very well ventilated to outdoors, and "leaky ceilings" is one of the three categories of buildings which we want to study in this project. To examine the leakiness of the ceiling, we performed several tests. First, we performed a blower door test on the building and found an ACH50 of 31.8; this is a very leaky building, especially when considering that the building volume is especially large (it is the denominator of the ACH50 calculation) because of an average ceiling height of 12 feet.

Building leakiness is associated with two primary factors; 1) the suspended, t-bar ceiling is quite leaky and 2) the attic space above the ceiling is very well vented to outdoors. The ceiling and the roof can be thought of as two series resisters – that is, resistors to air flow. If air is going to flow out of the building through the ceiling, it also has to flow through the roof system. As the building was found, both the ceiling and the roof deck were very leaky. The ceiling is leaky because it is a suspended, t-bar ceiling with many cracks, etc. (also there are architectural features – such as light shelves – in the auditorium that make it very leaky). The roof is very leaky (to air but not water) because of a great amount of attic venting (one of the most ventilated attics we have witnessed). If either the ceiling or the roof plane were fairly airtight, then the building would be fairly tight, as seen by the blower door.

To explore the potential for airtightening the building, the attic vents were temporarily sealed with masking materials. This effort was time consuming because there are many and large attic vent openings. There are five types of attic vents, and each of these vent types were sealed incrementally. We carefully measured the dimensions of the attic vents, and estimated the net free area of each vent type. We also disaggregate the vent area between the two major zones of the building, 1) the auditorium and 2) the balance of the building (a concrete block fire wall separates and isolates the two zones of the building). Gross area and net free area for each vent type and zone are reported in Tables 6 and 7. The sequence of testing and the cumulative net free area sealed is presented for each vent type in Table 8.

After sealing 17 *gable vents* with combined 58 square feet of net free vent area, ACH50 dropped from 31.8 to 23.9. When all *ridge vents* with combined 12 square feet of net free vent area were also sealed, ACH50 dropped to 19.9. When the *facia vents* (located only on the auditorium) were also sealed, ACH50 decreased to 16.7. When *vents in the drive through attic* were also sealed, ACH50 decreased to 14.7 (there is an attic above the drive through area which is very well connected to the main attic space). When an estimated 86% of the *soffit vents* were also sealed (the remaining 104 lineal feet were not sealed because of access and time constraints), ACH50 declined to 11.5. (Figure 1 shows the sequential sealing of the attic vents and the resulting building airtightness.)

It should be noted that our sealing of the vents was not complete in all cases, because of the difficulty of getting 3" masking tape (widest we could find at the time) to span some of the facia vent openings and adhere to the eave vents. Our best estimate is that only 80% of the facia and soffit vents and about 75% of the total soffit vents were sealed. Based on these tests, we predicted that airtightening of the attic system could achieve ACH50 of 9 or less, and using a spray foam application could achieve an even tighter building. At that point, this building would then be tighter than the average new commercial building.

Building and Ceiling Space Pressures With Respect to Outdoors With Five Blower Doors (~28,300 cfm) at a Clubhouse/Community Center

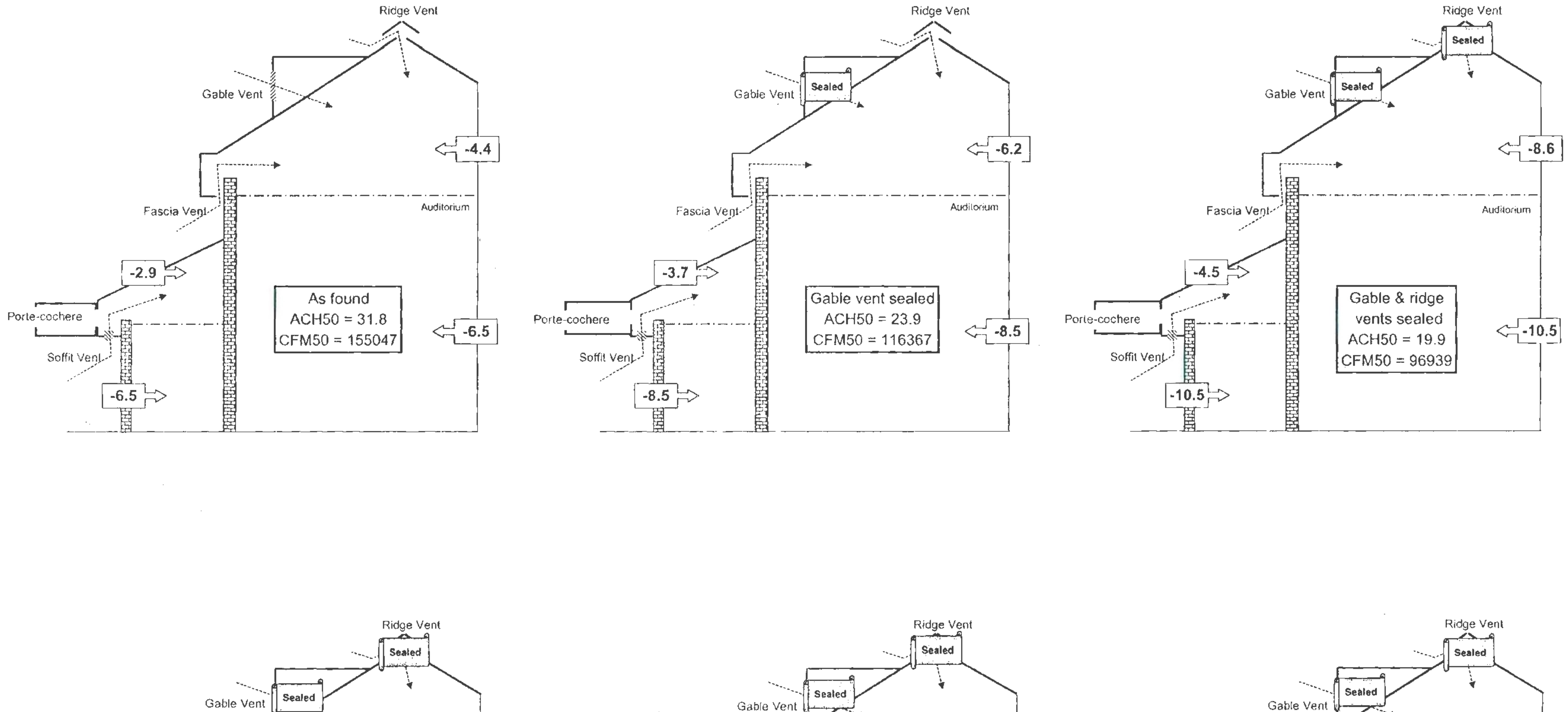


Table 6
Gross area (ft^2) of various types of vents in the attic of Country Club.

	gable	ridge	facia	drive-up	soffit	total
auditorium gross area	48	8	49	0	0	105
non-auditorium gross area	67	18	0	8	127	220
total gross area (ft^2)	**115**	**26**	**49**	**8**	**127**	**325**

Table 7
Net free area (NFA; ft^2) of various types of vents in the attic of a country club.

	gable	ridge	facia	drive-up	soffit	total
auditorium NFA area	24	4	44	0	0	72
non-auditorium NFA area	34	8	0	4	57	103
total NFA area (ft2)	**58**	**12**	**44**	**4**	**57**	**175**

Table 8
Cumulative net free area (NFA; ft^2) of various types of attic vents as the vents are incrementally sealed.
(Listed in the order that they were sealed during the building airtightening experiment.
Note: "hallway" = non-auditorium portion of building.)

	NFA auditorium (ft^2/% of aud.)	NFA hallway (ft^2/% of hallway)	NFA bldg (ft^2/% of total)
gable	24/34	34/33	58/33
ridge	28/39	42/41	70/40
facia	72/100	42/41	114/65
drive-up	72/100	46/45	118/67
soffit	72/100	91/89	163/94

As part of the blower door test, we determined the location of the air boundaries of the building. In order to do this, we depressurize the occupied space to -50 pascals (or as close to that pressure as possible) and measured pressure in the ceiling space above the auditorium, restaurant, and hallway while the blower doors were operating. From this we determine the pressure drop across the ceiling and across the roof deck. *In the auditorium*, 32% of the pressure drop was across the ceiling and 68% was across the roof deck; thus we were able to determine that the roof deck is the primary air boundary. These pressure drop numbers indicate that the accumulation of all ceiling holes is approximately twice as great as the vent openings in the roof. Since we knew the approximate leakiness of the roof deck because we had measured the vent openings, we could predict the approximate cumulative hole size in the auditorium ceiling. This lead us to look closely at the auditorium ceiling in order to find the big holes. *In the remainder of the building,* on the other hand, the ceiling is the primary air boundary since only 42% of the pressure drop is across the roof while 58% is across the ceiling. This means that the ceiling hole size is slightly less than the accumulative hole size in the roof vents.

Icynene Retrofit Tightened the Building

Icynene foam insulation was applied in the attic space and the attic vents were sealed in the summer of 1998. As a result of this, building airtightness decreased to 7.3 ACH50 (33,564 CFM50; C=3593.92 n = 0.57 r = .9960). This is very much in line with the airtightness that we had projected based on temporarily sealing the attic vents.

We recommend that building airtightness should, in general, fall in the range of 4 to 8 ACH50. Tighter than 4 ACH50 can lead to large pressure differentials from unbalance air flows which may cause a number of problems, including moisture accumulation in walls and other wall cavities, gas intrusion from soil (radon) and sewer lines, and combustion safety problems. Leakier than 8 ACH50, it becomes difficult to control unwanted weather-induced infiltration and is more difficult to achieve a positive pressure in the building (this positive pressure is important in preventing wall moisture accumulation, radon intrusion, and combustion safety problems). The testing of incremental tightening of the attic vents was valuable because it indicated that the building could be airtightened to a significant extent by sealing only the intentional attic vents. This testing was also valuable because we were able to identify the location of the primary air boundary of the building under various sealed-vent configurations.

Ceiling Airtightness

A reasonably accurate approximation of ceiling airtightness (cumulative ceiling hole size) can be made by examining these pressure relationships and assuming that we know the vent area of the attic. In order to make this approximation, we carefully measured roof vent openings and estimated net free area (net free area takes into account blockage by vent louvers, screens, hole perforations, and other obstructions). These are presented in Tables 6 and 7. On average, net free area is about one half of gross vent area, based on our observations.

The total NFA for the attic vents that service the auditorium is 72 square feet. Since the pressure drop across the roof deck is about two times that across the ceiling, the ceiling leakage (hole size) should therefore be about 2 times that of the roof deck, or about 144 square feet. In order to extend the analysis further, the relationship between CFM50 and hole size should be explained. In general terms, CFM50 can be converted to hole size (square inches) by dividing by 8. Conversely, hole size can be converted to CFM50 by multiplying by 8. Therefore, 144 square feet (20,736 square inches) of estimated ceiling leak area can be converted to 166,000 CFM50. Since there are 6500 square feet of floor area (and ceiling area) in the auditorium, this equates to 24.8 CFM50 per square foot of ceiling area.

The total NFA for the attic vents that service the non-auditorium portion of the building is 103 ft^2. Since the pressure drop across the roof deck is 0.72 times that of the roof deck, the ceiling leakage (hole size) should therefore be 0.72 times that of the roof deck, or 74 ft^2. Therefore, 74 ft^2 of estimated ceiling leak area can be converted to 85,206 CFM50. Since there are 17,434 ft^2 of floor area (and ceiling area) in the non-auditorium portion of the build, this equates to 4.9 CFM50/ft^2 of ceiling area. Two refereed papers have been published based on the ceiling airtightness data from this building and from a couple other buildings (Cummings and Withers, May 2000 and Withers and Cummings, May 2000). The results from several buildings in which we have measured ceiling airtightness have also found that standard suspended T-bar ceilings have leakiness of about 5 CFM50 per square foot.

Testing was done to evaluate the airtightness of typical suspended t-bar ceiling. To do this, an airtightness test was performed on the storage room adjacent to the auditorium. This room has linoleum over a concrete slab floor, block and gypsum board walls, one double-door to the auditorium, and a suspended ceiling above which is a highly vented attic space. Since the floor and walls of this room are nearly airtight, practically all of the air leakage in the room can be attributed to the suspended t-bar ceiling. With cracks along the door sealed, the room was taken to -50 pascals with respect to the ceiling space above the room. Airtightness for the room was found to be 1820 CFM50, or 5.2 CFM50/ft^2 of ceiling area (we estimate that 95%+ of the room leakage is in the ceiling). Therefore, the ceiling airtightness is about 5 CFM50/ft^2, which is in close agreement with the estimated airtightness of the ceiling for the non-auditorium portion of the building and with ceiling airtightness tests done on several other buildings, with each being very nearly 5 CFM50/ft^2. *Therefore, the ceiling in the auditorium is indicated to be about five times as leaky as normal suspended t-bar ceilings, whereas the ceiling of the remainder of the building is fairly standard for a suspended ceiling.*

One spin-off from this research project is a proposal which we have written to US Department of Energy to investigate the impacts of leaky suspended T-bar ceilings on building airtightness, ventilation rates, energy use, and indoor relative humidity, and develop improved suspended T-bar ceilings. This proposal, titled *Investigation of Current and Improved Technologies Related to the Airtightness of Suspended T-bar Ceilings* was submitted to US DOE on April 3, 2000. We received notice in September of 2000 that DOE would not be funding this research.

Visual inspections were performed to discover why the auditorium ceiling is exceptionally leaky, *and the leaks were identified.* They are associated principally with the indirect light shelf system around the perimeter of the room and around the chandeliers in the auditorium. Fluorescent light fixtures are located on shelves that extend 30 inches into the room. The shelf support consists of 3.25 inch metal framing running horizontally from the wall. Where these 3.25 inch members emerge from the wall, a 3.25 inch high gap exists in the gypsum board, and the total length of this gap is 520 lineal feet. In addition, where the shelves meet other architectural features (building cavities), there are twenty-two 12" x 7.5" openings into these building cavities throughout the auditorium. During inspection, we could feel cool breezes blowing through these openings and onto the inspectors (inspections were done on a cool winter day). In total, these identified leak sites account for 144 square feet of leak area. In addition to these leaks, there are of course the leaks associated with normal t-bar ceiling construction (cracks on four sides of each ceiling tile, which is normally about 5 CFM50/ft^2), and there are 32 recessed lights, 32 sprinkler heads, and 16 speakers.

Looking at Figure 1, one can see that as the roof vents are sealed, the roof becomes more and more the dominant air boundary *for the auditorium* portion of the building. Pressure drop across the roof deck increases steadily from 68% to 93% of the total pressure drop from indoors to outdoors as vents are sealed. In the remainder of the building, masking off roof vents actually causes the ceiling to become the more dominant air boundary until the soffit vents are sealed. Pressure drop across the roof deck first declines from 45% to 39% of the total pressure drop from indoors to outdoors, and then increases to 59% when most of the roof vents are sealed.

HVAC System Performance and Controls

The HVAC systems were examined and tested to determine system performance and verify proper operation control. A number of deficiencies were identified. The following list (in italics) was submitted to city staff for repair. A majority of the indicated problems had been fixed within the quarter following the submission of this list to the city.

Country Club HVAC System Maintenance List

Notes:

* Items that need to be fixed as soon as possible
^ Fix these items if they can be completed by the time the other repairs are completed.

1) AHU-1 Auditorium:

*Electric Heater	-	Replace blown fuse and verify proper operation of electric heater (40 kW, currently controlled as 2 stages with 20 kW each).
*Outside air intake	-	Bug screen at inlet to building (3 inlet grilles on east wall near back stage entrance) are extremely dirty. Clean or replace as required. Reset and verify outdoor air flow rate according to construction drawings (2070 cfm). Outside air duct is not air tight, and probably leaks where the duct connects to the inlet grilles. Seal ductwork/intake grille interface when cleaning bug screen.
^Inlet guide vanes	-	Supply air fan inlet guide vane actuator inoperable, repair/replace as required. Clean and lubricate guide vanes. Verify operation of pressure sensor in supply air duct which controls inlet guide vane actuator.
*System air flows	-	Supply air flows: measured supply flow rates are <u>higher</u> than design in back stage dressing areas and <u>well below</u> design in the north and south auditorium partitioned areas. Noticed that flex duct was used in north/south partitioned areas instead of hard duct with dampers as per the construction drawings, possibly causing some extra resistance to flow. Supply air flows nearly match design in open seating area near stage. Return air flows: <u>total</u> return flow nearly matches design, but excess return flow in main seating area near stage (dual returns on either side of stage stairs at 130% of design) and in back stage dressing rooms. Returns in north and south auditorium partitioned areas are well below design (77% and 35% of design flow, respectively). Reset supply and return air flows to design values as required (see attached air flow diagram).
Air handler cabinet	-	Insulation missing inside of AHU-1 at top of unit near fan, repair as needed.

2) AHU-2 Kitchen:

General	-	Overall, this system is in pretty good shape. Total supply and return air flows nearly match design, although a few individual supply/return flows are higher or lower than design (see attached air flow diagram). Filters and cooling coil are generally clean.

*Hood exhaust	-	Have kitchen hood exhaust and make-up air flow rates checked and reset according to the construction documents if necessary.
Fan access door	-	Excessive leakage observed at fan access door; repair/replace seal as required.
Dishwash exhaust	-	Exhaust fan over dishwasher only operates at 65 cfm vs. 300 cfm design. Investigate and correct cause of low flow. Fan controlled by potentiometer dial located on the wall.

3) AHU-3 Dining Room:

^Inlet guide vanes	-	Clean and lubricate supply air fan inlet guide vanes. Verify operation of pressure sensor in supply air duct.
*System air flows	-	Supply air flows: measured supply air flows in dining room are consistently low, averaging only 57% of design flow. Return flows are also low, averaging 81% of design. Reset air flows per construction documents (see attached air flow diagram). Verify outdoor ventilation air flow rate matches construction documents (after supply/return flows have been reset).
*Return air grilles	-	Clean dirty return grille intakes (2 locations in ceiling). Excessive dirty build-up is probably contributing to low return air flows.
Airhandler cabinet	-	Excessive leakage through top of air handler cabinet, repair as required. Repair insulation on inside of cabinet near supply air fan. Repair tape on return duct insulation.

4) AHU-4 Office/Lobby:

^Inlet guide vanes	-	Supply air fan inlet guide vane actuator inoperable, repair/replace as required. Clean and lubricate guide vanes. Verify operation of pressure sensor in supply air duct (located above ceiling at north end of hallway connecting mezzanine to front lobby).
*System air flows	-	Low supply air flow in lobby, hallway and manager's office. Test and balance as required (see attached air flow diagram). Supply air flows to concession area (adjacent to auditorium) do not need to be as high as design. Original design values included provisions for a small kitchen with exhaust hood. If this kitchen will not be installed, then the supply flows to this concession area can stay at their current values (no adjustment required). Reset outdoor ventilation air flow rate to design value of 1900 cfm.
Hall Bathrooms	-	Bathroom exhausts within 85-89% of design flow since fan belt was replaced by facility staff.

5) AHU-5 Pro Shop:

*Electric Heater	-	Replace high temperature cutout on heater circuit and verify proper operation of electric heater (25 kW, currently controlled in 2 stages).
*Cooling coil	-	Dirty coil, clean as required.
*B_room exhaust	-	Current exhaust from bathrooms adjacent to pro shop is less than 50% of design flow. Increase exhaust air flows to match design (490 cfm each).

*System air flows - Return air flows match design values fairly well, but the supply air flow rates are low for nearly all diffusers in the pro shop and office. Shortfall in supply air flow seems to be the results of no outdoor air intake. Reset the outdoor ventilation air intake to the design value of 1190 cfm. (Our measurements indicate less than 40 cfm of flow at the intake grille. Visual inspection indicated installed intake duct was <u>much</u> smaller than the size shown on the construction drawings). Then recheck supply air flows versus design values. One complication is that the original design was to take return air from the pro shop and transfer it to the bathrooms. Apparently there were comfort complaints in the bathroom with the original design, so the transfer grilles were converted to supply air diffusers. Thus, the supply air to the pro shop and office will be decreased by the amount of supply air now being supplied directly to the bathrooms and adjacent hallway entrance. (See attached air flow diagram).

Air handler cabinet - Loose insulation near fan inlet, repair as required.

Retrofit Recommendations

Once the maintenance items listed above are complete, we recommend the following retrofits.

- Consider sealing the south exterior block wall of the auditorium from the cart storage side with sealants to restrict both air and moisture flow into the block wall. It is our understanding that there are spray-on materials that may achieve this purpose.

- Consider eliminating the attic ventilation. This would involve sealing the eave vents, ridge vents, and gable/cupola vents (latter from the inside, so the building aesthetics would not change). This would increase the temperature in the attic, but would greatly reduce the humidity levels. This would eliminate condensation on the ducts and lower relative humidity in the occupied spaces. While the sensible cooling load may go up somewhat because the attic will be hotter, the latent cooling load will go down because the humidity in the attic space will be lower.

  One option for reducing the temperature in the attic would be a thermostatically controlled attic exhaust fan (turn on at say 95F), installed at a high point in the building, perhaps at the top of the cupola. An exhaust fan of say 1000 cfm would exhaust hot air from the attic space and reduce the building cooling load. It would also have the effect of depressurizing the attic space relative to the occupied space, thus ensuring that air flow would be from the occupied space to the attic most of the time, and not vice versa. As long as the total building intake air-to-exhaust air differential is positive (according to design, intake should be more than 3000 cfm greater than exhaust), then this arrangement should maintain positive space pressure (with respect to outdoors) and prevent air flow from the attic to the occupied space. (Note that recommendation number 7 emphasizes the importance of having a Test and Balance firm verify the building air flow balance.)

  Note that the tighter the building becomes, the easier it will be to control pressures and the direction of air flows in the building.

- Also consider installation of a spray-on insulation material on the bottom of the roof deck. This would greatly reduce heat influx into the attic space and thereby considerably reduce the building cooling load. Three products come to mind -- wet-spray cellulose, urethane foam, and Icynene. Each of these materials can be sprayed onto and stick to the bottom side of the plywood decking. This would move the thermal barrier to the newly established air barrier, and create a more effective total building envelope. Spray of 1 inch (R4) of Icynene, for example, onto the bottom of the entire roof deck system might cost in the range of $10,000 to $15,000, but it might save $5000 per year in heating and cooling costs. It would also reduce the amount of time and cost involved in maintaining the current insulation system. (Note that the batts sometimes fall down on their own and in other instances are moved aside by contractors accessing the space.) Since the insulation at the roof deck would stop most of the heat flux into the building, the importance of maintaining the integrity of the batt insulation system would be reduced.

- In the future, if wallpaper is changed, consider replacing vinyl wallpaper with types that allow moisture transport. This will allow wall assemblies to dry to indoors.

- Insulation batts have fallen in a number of places in the attic. It would be good to reattach or replace any batts that have fallen or are missing. When the insulation is improved and the building made more airtight, the 100 tons of air conditioning should be more than enough to meet loads. If the roof deck is also insulated, building design cooling requirements will decrease much more, to perhaps 60 tons or less. In this case, it might be useful to have a mechanical HVAC engineer examine the cooling system to see if air handler air flow reductions might be in order; lower flow rates would allow the coils to operate at lower temperatures and provide improved dehumidification.

- Alternatively, you may want to consider reducing the amount of outdoor. Currently, the design calls for total outdoor air (for all air handlers combined) of 6940 cfm, or enough for more than 300 persons. (We did not measure outdoor air flow rates.) You may want to consider whether that amount of outdoor air can be decreased, especially in the non-auditorium portions of the building. A mechanical engineer could make that determination.

  Also consider installation of ventilation demand control. Since you generally have much fewer than 300 persons, you could save considerable energy and reduce indoor humidity levels by automatically modulating ventilation based on occupancy. Demand control would consist of carbon dioxide sensors controlling dampers on the outdoor air ducts, thus regulating the amount of ventilation air brought into the building based on the carbon dioxide levels. Note that carbon dioxide is given off by people, so the more people in the building the higher the carbon dioxide level. Carbon dioxide is not a significant air pollutant in itself, but indicates the ventilation-to-occupancy ratio. Outdoors carbon dioxide is about 350 ppm (parts per million). An indoor level of 1000 ppm indicates about 20 cfm per person of ventilation (in occupancy/ventilation equilibrium). So a controller set to 800 ppm of carbon dioxide would achieve 20 cfm or more per person when the building is full (with a margin for error), and greatly reduce ventilation and energy use during most of the time when the building has low occupancy.

- Once the building has been made more airtight by sealing up much of the attic ventilation, it would be good to hire a Test and Balance firm to bring all air flows up to design (either original or modified). According to the current mechanical plans, when all the exhaust air, make-up air, and outdoor air are fully operational, the building should be operating at positive air flow (+3300 cfm) and positive pressure. (Outdoor air plus make-up air = 9880 cfm. Exhaust air = 6590 cfm.) (It would be good to select a Test and Balance firm based on asking them questions about how they determine the air flow balance and pressure balance in the building and specifically how they account for any duct leakage which may exist in exhaust ducts, make-up air ducts, and outdoor air ducts.) The amount of positive pressure will depend upon how tight the building becomes after the attic tightening has occurred. The positive pressure will have the benefit of pushing relatively dry indoor air outward through walls, other building cavities, and attic space to outdoors. This should help to eliminate moisture accumulation on ducts and in wall cavities.

Postlude

The retrofits that actually occurred at this building:

- Re-scheduling of operation times of the AC systems
- Sealing of all attic vents and application of about 3 inches of spray foam insulation to the under side of the roof deck – thus the air barrier and thermal barrier of the building was moved to the roof deck
- Addition of a dedicated outdoor air conditioning system to reduce indoor humidity levels

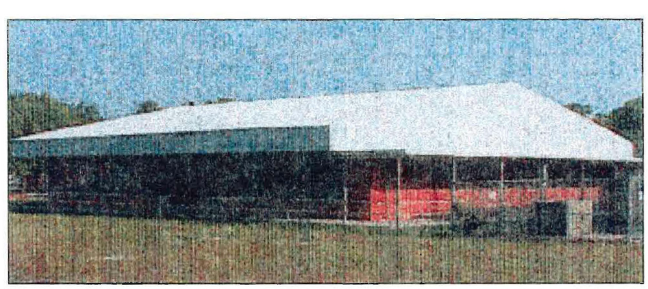
The energy savings and other impacts of these retrofits are discussed in detail in Appendix C.

4. & 5. ELEMENTARY SCHOOL CLASSROOM BUILDINGS

Polk County (Florida) school district has recently constructed classroom buildings that are designed to be easy to construct, energy efficient, and easy to maintain. Two of these units were selected for testing and monitoring in this project. The objective of the monitoring was to determine if improving the air boundary and thermal boundary of the building at the ceiling plane would yield reduction in cooling energy use.

Building description

Floral Ave Elementary School (Bartow, FL) and Eagle Lake Elementary School (Eagle Lake, FL) are of similar construction, the only significant difference is in the color of the roof. Floral Ave has a white metal roof (Figure 2). Eagle Lake has a blue metal roof. The buildings are constructed with metal framed wall studs and a metal roof truss system. The exterior wall finish is brick. The standing seam metal roof deck has



2. Floral Avenue Elementary

approximately 1 inch of vinyl coated fiberglass insulation directly beneath. The ceiling is a typical suspended T-bar assembly located about 2 feet below the metal truss rafter system. A layer of foil radiant barrier material is attached to the bottom of the trusses and R-19 fiberglass batt insulation lies on top of this foil.

Both buildings are slab-on-grade with the surrounding grade sloped slightly away from the building. The entrances and windows of the four classrooms are protected from both sun and rain by an overhang that extends out approximately 8 feet. Doors and windows are located only on two opposing sides of the buildings. The floor covering consists of tile laid directly on the concrete slab.

These two classroom buildings are essentially identical with about 5000 ft^2 of floor area, and are located about 20 miles apart. They consist of four classrooms, four bathrooms, and a central teacher planning area (Figure 3). This modular design gives the school board the flexibility to link the "quads" to together in a row to form as many classrooms as needed.

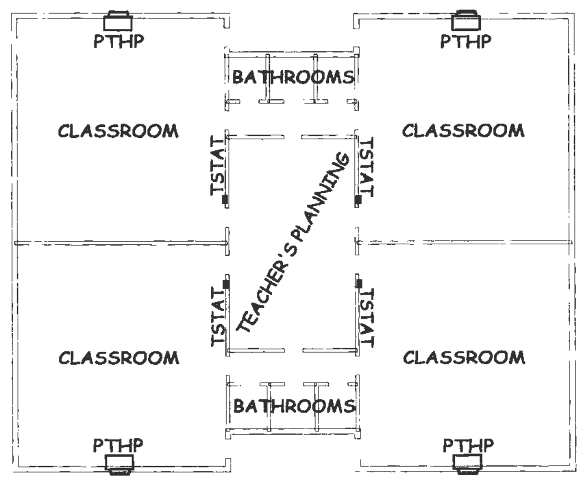


Figure 3. Typical layout of a "quad" containing 4 classrooms, teacher planning area, and bathrooms. Each classroom has its own heating, cooling and ventilation unit.

The heating, cooling, and ventilation needs of the building are met by four PTHP (Package Terminal Heat Pump) units – one in each classroom. The units are manufactured by the Crispaire Corporation of Cordele, GA., the product line is the Marvair and the model is the Scholar II. Secondary supply ducts are run from the supply plenums of two of the four PTHP units to provide air to the centralized teacher planning area. Separate thermostat control is provided for each of the PTHP units. Thermostats are located on the wall near the doorway that leads to the central prep room. There are four-hour crank timers connected in line with the thermostats to limit PTHP runtime. Ventilation air is brought into the classroom via the PTHP unit. The outside air is filtered and delivered to a space between the evaporator coil and the return air filter. Manufacturer specifications indicate that up to 400 cfm of outside air may be introduced to the space when the control damper is in the full open position (we did not measure this directly, but tracer gas decay tests found an average of 200 cfm of ventilation per classroom). There is also a filtered pathway for exhausting air through the PTHP. There are also exhaust fans in each of the bathrooms and janitor's closet. These fans are interlocked with the light switch and have a preset timed delay before they turn off.

Building Airtightness Test Results

A multi-point blower door test was completed on each of the buildings. Testing was done with exterior doors and windows closed and PTHP units in the off mode (PTHP outdoor air and exhaust vents were not sealed). Interior doors connecting the classrooms to the centralized teacher planning area were open during the test.

Airtightness tests found 16,430 CFM50 (cubic feet per minute at 50 pascals) at Floral Avenue and 17,756 CFM50 at Eagle Lake. This airtightness can also be expressed as 21.9 ACH50 (air changes per hour at 50 pascals) and 23.7 ACH50, respectively. Given that we recommend that building tightness fall in the range of 4 to 8 ACH50, these buildings are very leaky.

Building leak paths

A CFM50 of 17,000 indicates an equivalent leak opening of approximately 15 square feet, or a square hole nearly 4 feet on a side. The fact that the buildings were very leaky prompted a search for such large leakage. Large leak paths were identified in two general locations.

- Ceiling plane - consisting of a T-bar assembly, an air space of approximately 18 inches, a metal grid covered with a radiant barrier material, and an R-19 fiberglass batted insulation. The perimeter vertical (wall) section between the T-bar ceiling and the radiant barrier covered grid did not have a good air barrier in many areas of the building, especially above the bathrooms (Figures 4-7).

Figure 4. Missing air barrier & insulation that had been moved to gain access to attic

Figure 5. Incomplete air barrier at connection of wall assembly to ceiling

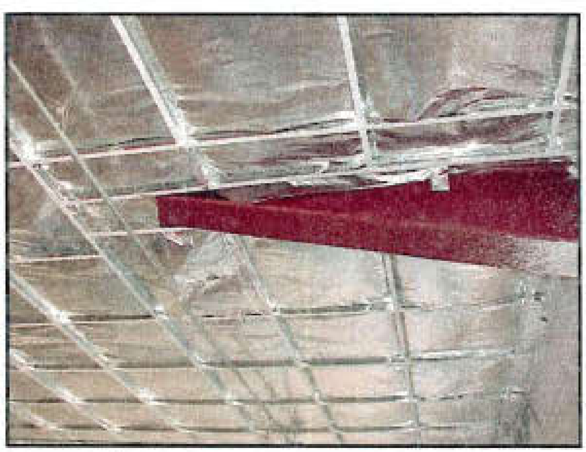


Figure 6. Metal grid system covered with radiant barrier acting as air barrier. Numerous gaps exist, especially at penetrations of the grid assembly.

Figure 7. Incomplete air barrier over t-bar ceiling.

- PTHP assembly - the unit rolls on wheels and is slid into an opening (about the size of the PTHP unit) in the exterior wall of each classroom (Figures 8 and9). There is large leakage between the unit and the wall opening. In general, the gap is 1 inch to 2 inches wide depending on the rough opening dimensions of the wall and closeness of the unit to the wall. There was no seal between the wall and unit.

Figure 8. The interior decorative trim has been removed to show the space between the unit and wall. This space is approximately 1 inch.

Figure 9. Exterior view of PTHP unit with grille assembly removed. Approximately 1 inch of clearance exists between the unit and the wall.

Ceiling insulation deficiencies

Three major problems were identified with insulation above the ceiling. First, 20% to 25% of the insulation batts were missing. Second, many batts were not lying flat, but rather were elevated above the radiant barrier material (which was supporting the batts) by electrical conduit or truss elements. Third, about 25% of the batts at Floral Avenue were R-11 instead of R-19.

Air barrier and thermal barrier retrofit

After testing was complete, it was determined that a retrofit would be undertaken to make the ceiling plane substantially more airtight and correct deficiencies in the ceiling insulation system. Since a major focus of this project was to address leaky ceilings, considerable thought and investigation went into selecting the correct retrofit for this application. No easy and simple solutions readily identified. The suspended T-bar ceiling could not be made more airtight, at least not with technologies we were aware of or with methods that would leave the ceiling as a modular suspended T-bar ceiling as we know it. Therefore, we focused on the plane formed by the radiant barrier material which was attached to the bottom of the trusses and was located two feet above the suspended T-bar ceiling.

The air leakage was occurring at seams where sections of radiant barrier material overlapped, where sections of radiant barrier material had been pulled aside to gain access to the attic space, and in the exterior wall sections located above the suspended T-bar ceilings. The leakage of these exterior wall sections occurred because much of it had no gypsum board (though the kraft paper backed batts were in place). Therefore, the ceiling space was well connected to the eave soffit area which is also open to the attic. After considering options, it was decided that the retrofit would consist of the following:

- Application of a spray foam insulation material (with R-3.7 per inch) to the insulation batts in the exterior wall sections located above the suspended T-bar ceiling. This would make the exterior walls between the ceiling and the radiant barrier material fairly airtight.

- Re-install sections of radiant barrier material that had been pulled open

- Application of the spray foam insulation to the seams of the radiant barrier material

- Rearranging insulation batts already in the attic space so they fit close together and were not elevated above the radiant barrier material, replacing R-11 batts with R-19 batts as needed at Floral Avenue, and then installing additional insulation batts to provide complete coverage.

Long-term monitoring to determine indoor conditions and cooling energy.

Detailed monitoring of these two classroom buildings was performed for a period of approximately one year (April 1999- May 2000) to compare the energy consumption, equipment operation, and interior conditions. Following are significant events that occurred during this monitoring:

- In May 1999 the interior temperature of the two buildings was set at 75F and the 4 hour crank timers were disconnected, thus allowing the cooling equipment to operate as needed throughout the entire day. The adjustment damper of the ventilation air in the PTHP units were closed to minimize outside air entry.

- Retrofits were performed on both buildings in July. 24-hour thermostat control continued through mid-August 1999.

- In mid-August, the crank timers were reconnected and control of HVAC equipment was returned to the teachers and maintenance personal. Normal school operations resumed through May 2000).

A more detailed discussion of the retrofit implementation, long-term monitoring effort, and retrofit impacts is found in Appendix C.

Building testing

Building airtightness and building ventilation rates were measured by means of blower door tests and tracer gas decay tests.

Building airtightness testing

While only two classroom buildings were tested as part of this project, two other replacement classroom buildings were tested in another project. These other two buildings have a similar floor plan and essentially the same volume. However, their construction is substantially different. Instead of metal frame wall construction, their walls are autoclave aerated concrete panels (AAC). The roof consists of 8 inch thick AAC panels (lain across steel trusses) with a metal roof covering. There is no attic insulation other than the AAC panels, which have a nominal R-11.5 thermal resistance, but yield a higher comparative thermal resistance during the cooling season when the mass of the panels is taken into account. There is no venting of the space between the roof deck and the suspended T-bar ceiling. Test results from all four buildings are presented here.

A building airtightness test was performed on each of the four buildings, with a Minneapolis Model 3 Blower Door installed in one of the exterior doorways. A multi-point blower door test was completed on each of the classroom buildings. The testing was completed with the exterior doors and windows closed and PTHP units in the off mode. All of the interior doors that connect the classrooms to the centralized teacher's planning area were open. Test results are shown in Table 9 and Figure 10.

The metal framed buildings averaged 17,577 cfm at 50 Pascals (CFM50 = 17,577) or about 3.3 CFM50 per square foot of conditioned floor area. Typical airtightness in commercial type buildings is in the range of 1 to 3 cfm per square foot of floor area at a pressure of 50 Pascals. While the frame construction classrooms The AAC buildings had an average CFM50 of 4874 or 1.0 CFM50 / square foot, or 72% more airtight than the metal framed buildings.

Table 1
Airtightness Test Results for Four Classroom Buildings

School	CFM50	ACH50	CFM50 / Ft^2	C	n	Comments
Floral Ave	16644	19.9	3.3	1641.9	0.59	As found[1]
	13692	16.4	2.7	1409.8	0.58	PTHP grills sealed[2]
Difference	2952	3.5	0.6	17.7%	reduction	
Eagle Lake	18509	22.1	3.7	1649.5	0.62	As found
	16771	20.1	3.3	1513.8	0.61	PTHP grills sealed
Difference	1738	2	0.4	9.4%	**reduction**	
Spook Hill	4916	4.2	1.0	470.2	0.60	As found
	2272	2.0	0.5	212.3	0.61	PTHP grills sealed
Difference	2644	2.3	0.5	53.8%	reduction	
Elbert	4832	4.2	1.0	480.5	0.59	As found
	2433	2.1	0.5	232.7	0.60	PTHP grills sealed
Difference	2398	2.1	0.5	49.6%	reduction	

1- The exterior doors were closed, all interior doors open, all exhaust fans off and the HVAC units were off.
2- The exterior grill of the PTHP units were sealed off temporarily with an air impermeable cover. This eliminates any air passing through or around the unit from the outside.

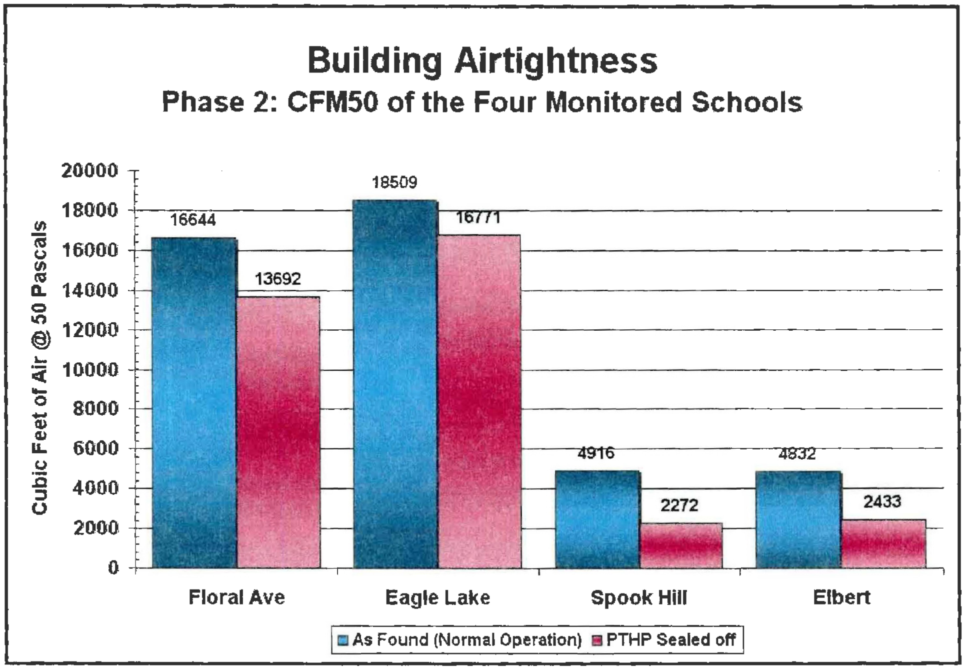


Figure 10. Building airtightness results. Sealing the PTHP resulted an average reduction of 2433 CFM50.

The first set of tests included leakage through the outdoor air intake and exhaust openings in the PTHP units as well as the gap between the equipment and the wall assembly. Testing was repeated with the PTHP units isolated from the building by temporarily sealing all of the exterior grills. This eliminates both the outdoor air intake and exhaust openings in the PTHP units as well as the gap between the equipment and the wall assembly. The airflow was reduced to 15,232 (CFM50=15232) for the metal framed buildings and to 2353 (CFM50=2353) for the AAC buildings. From this we conclude that leakage around the PTHP units averaged 2433 CFM50 for these four outdoor air intake and exhaust openings in the PTHP units as well as the gap between the equipment and the wall assembly buildings, or about 14% of the leakage of the metal framed buildings and 50% of the leakage in the concrete panel buildings. CFM50 can be divided by 13 to yield equivalent hole size in square inches. Thus, the equivalent hole size of this leakage around and through the PTHP units is 316 square inches per building or 79 square inches per PTHP unit.

Infiltration/ventilation rates

Air change rates were measured using a Miran 101 gas analyzer and the ASTM E 741, "Standard Test Method for Determining Air Leakage Rate by Tracer Dilution". Low concentrations of a tracer gas Nitrous Oxide (N_2O) were injected into the four classrooms and teacher planning area. After a short mixing period, the concentrations of the gas are recorded at regular time intervals. The rate of decay of the tracer gas concentration gives the air change rate (Equation 1).

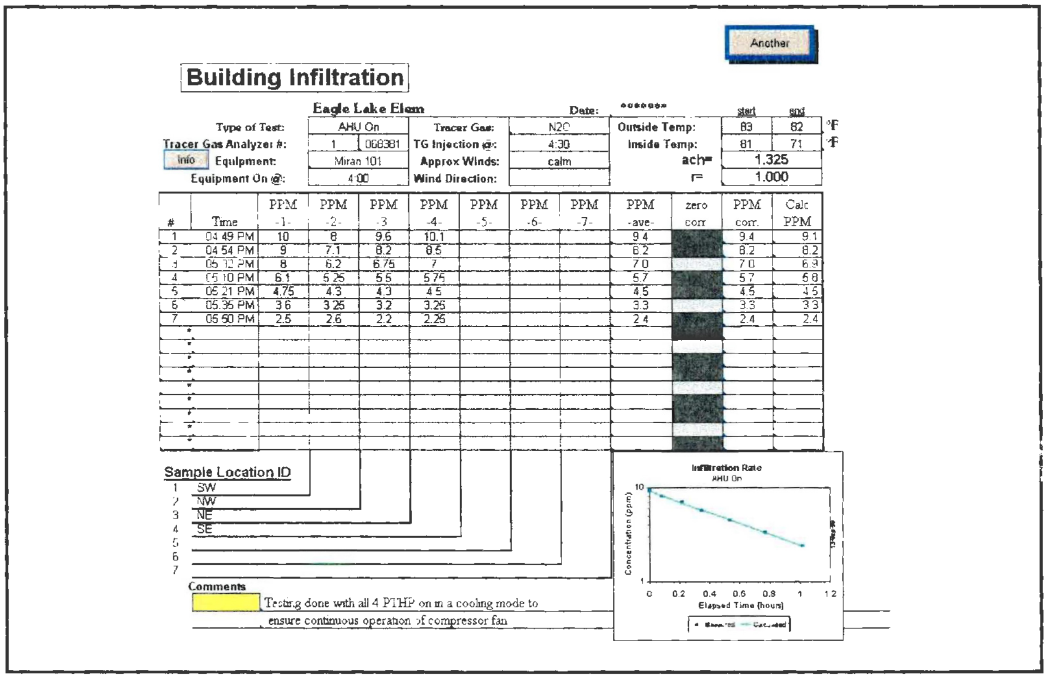
$$I = \ln (C_0 / C_f) / t \qquad \text{Equation 1}$$

where:

I = air change rate (1/h or ach)
C_o = initial concentration
C_f = final concentration
t = time (h)

The tracer gas was measured at 5-10 minute intervals for approximately one hour at the center of each classroom. The results, which are only a snapshot in time, show that all four buildings have approximately the same infiltration rate during a normal classroom operating condition (i.e., the air handler units on), about 1 air change per hour (average = 1.08 ach; see Figure 11 and Table 10). This equates to just over 200 cfm per classroom.

The school board uses a time averaging technique to determine the amount of outside air required. It is based on the average occupancy of a classroom for the entire school day. The assumption for the elementary school is that the room will be occupied 77.7% of the time. If the classroom holds 25 students, then the total ventilation needed would be 291 cfm per classroom (15 cfm/student x 25 students x 77.7% occupancy loading = 291 cfm). The ventilation rates of the these four buildings (during the test period) were on average 22% less than the target.


Another

Building Infiltration

Eagle Lake Elem Date: *******

Type of Test: AHU On | Tracer Gas: N2O | Outside Temp: start 83, end 82 °F
Tracer Gas Analyzer #: 1 068381 | TG Injection @: 4:30 | Inside Temp: start 81, end 71 °F
Info Equipment: Miran 101 | Approx Winds: calm | ach= 1.325
Equipment On @: 4:00 | Wind Direction: | r= 1.000

#	Time	PPM -1-	PPM -2-	PPM -3	PPM -4-	PPM -5-	PPM -6-	PPM -7-	PPM -ave-	zero corr	PPM corr.	Calc PPM
1	04 49 PM	10	8	9.6	10.1				9.4		9.4	9.1
2	04 54 PM	9	7.1	8.2	8.5				8.2		8.2	8.2
3	05 ?2 PM	8	6.2	6.75	7				7.0		7.0	6.9
4	05 10 PM	6.1	5.25	5.5	5.75				5.7		5.7	5.8
5	05 21 PM	4.75	4.3	4.3	4.5				4.5		4.5	4.5
6	05 35 PM	3.6	3.25	3.2	3.25				3.3		3.3	3.3
7	05 50 PM	2.5	2.6	2.2	2.25				2.4		2.4	2.4

Sample Location ID
1 SW
2 NW
3 NE
4 SE
5
6
7

Comments
Testing done with all 4 PTHP on in a cooling mode to ensure continuous operation of compressor fan

Figure 10. Example of tracer gas test form used. Eagle Lake shown with a ventilaltion rate of 1.325 ach.

Table 2
Building infiltration rates with HVAC operational

School	ach	cfm[1]	cfm/classroom[2]	difference from 291 cfm	percentage difference
Floral Ave	1.00	839	210	81	27.9%
Eagle Lake	1.33	1114	278	13	4.3%
Spook Hill	1.04	874	219	72	24.9%
Elbert	0.98	819	205	86	29.6%
average	**1.08**	**912**	**228**	**63**	**21.7%**

Notes:
1- Hourly ventilation rate during the test period with outside dampers as found.
2- Total cfm in previous column divided equally among the 4 classrooms. The ventilation air damper was left in the "as found" position, RA and OA filters were clean, and the PTHP were ON.

It is interesting to note that ventilation rates are not significantly different in the leaky frame construction buildings compared to the tight AAC buildings. This indicates that ventilation is dominated by the mechanical systems, and that the excessive leakiness of these frame buildings is not a major factor in their ventilation/infiltration rates.

6. SEA FOOD CHAIN RESTAURANT

PART 1 (written in April 1999 based on testing March 15, 1999)

Background

This restaurant (Figure 12) was opened in July 1998. There were reports of high humidity problems during the first few months of operation. Significant pressure difference was noted between the kitchen and dining area, and transfer grills were installed above two doorways to the kitchen to relieve this pressure. Humidity has reportedly been under control since then.

Figure 12

This 8200 square foot building's HVAC needs are served by five roof top unit (RTU) air conditioners with gas heat and economizers, four kitchen exhaust fans, a dishwasher exhaust fan, a restroom exhaust fan, and one make-up air unit.

Building airtightness testing

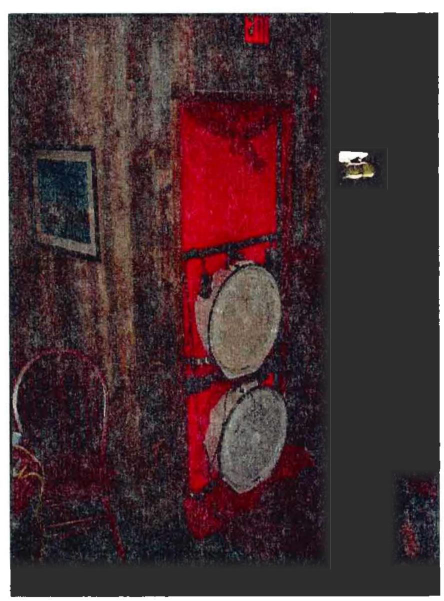


Figure 2. Blower door

A blower door test was performed to characterize the airtightness of the building (Figure 13). In preparation, the kitchen and bathroom exhaust fans were turned off and masked off. The kitchen make-up air was turned off and masked off. The dishwasher exhaust fan was not disabled or masked off, since in our opinion this fan would often be off and therefore would represent a "hole" in the building most of the time (this assumption may be in error). Using two blower doors, a multi-point blower door test was performed. The building was depressurized to a number of pressures ranging from -11.0 pascals to -37 pascals. Test results found a building airtightness of 13,407 CFM50 (C=1302.66, n = 0.59). CFM50 is the air flow rate through all the cracks, penetrations, and holes in the building envelope when the building is depressurized to -50 pascals. Though we did not achieve -50 pascals, a computer program projects air flow at -50 pascals with considerable accuracy based on the multiple pressures points, so we have great confidence in the projected value.

Given the floor area of 8204 square feet and estimated volume of 82,040 cubic feet, the building airtightness can also be expressed as 9.7 ACH50 (this means that the building air volume is exchanged 9.7 times every hour when the building is depressurized to -50 pascals). ACH50 is the best measure of tightness for comparison to other buildings. The average new small commercial building has an airtightness of about 14 ACH50, so this restaurant is tighter than average. A blower door test summary report is attached to the end of this report. However, this building is still more leaky than our recommendation of 4 to 8 ACH50.

Review of test and balance report

A Test and Balance report performed by Melink Corporation reports that air flows were well balanced after adjustments were made to exhaust, make-up, and distribution air flows in July 1998. According to the design documents, the sum of incoming air flows (OA + MA) exceeds exhaust air by 675 cfm. After adjustments to exhaust and make-up air flows, Melink found a positive incoming air flow balance of 288 cfm, and therefore the building would presumably be operating at positive pressure. Instead, building pressure was indicated to be negative (-0.01 inWC or -2.5 pascals) upon completion of the TAB work.

The TAB report found total kitchen exhaust hood flow of 13,302 cfm which was then reduced to 11,062 cfm (16.8% reduction). The make-up air air flow was initially 8748 cfm, and was reduced to 7436 cfm (15% reduction). While the exhaust fan flows were reduced by 2240 cfm and the make-up air was reduced by 1312 cfm (per TAB report), building pressure went from +5 pascals to -2.5 pascals (pressures with respect to outdoors). This change in pressure is certainly unexpected, given the greater reduction in exhaust air flow. Note that the net reduction in exhaust air of 928 cfm (2240 cfm - 1312 cfm = 928 cfm; based on exhaust air and make-up air alone) would increase building pressure from +5 pascals to +7.5 pascals with respect to outdoors, all other variables being constant based on the building airtightness curve (blower door test). The fact that building pressure went from +5 to -2.5 pascals could indicate that the outdoor air flow rates were changed (reduced) by modifications carried out by the TAB firm or it could be that the economizers were operating when the first measurement was performed and were not operating when the final pressure was measured. Variability in wind speed could also be another factor.

Air flow measurements

In our testing, building air flows that relate to building air flow balance were measured. To do this, we first measured the normal operating pressure (NOP) of the building. This was done at about 8:30 AM, and we found that the building pressure was +4.5 pascals compared to -3.0 pascals with all systems turned off. (Because the wind was strong, we had to take a number of running average measurements to even out the fluctuations.) Using the blower door airtightness curve, we were able to calculate the net air flow into the building necessary to increase the building pressure from -3.0 pascals to +4.5 pascals. This amount was 5694 cfm. Therefore, the incoming air (combination of make-up air and outdoor air) was greater than that removed by the exhaust fans by 5694 cfm (this projected from the blower door test). This means that net exhaust air (EAnet) was -5694 (a negative number for net exhaust means that incoming air flows are greater than exhaust air flows).

We characterized the make-up air flow by the substitution method. In this method the blower doors substitute for the make-up air fans. We turned on the exhaust fans and opened up the outdoor air intakes while leaving the make-up air turned off and masked. With the exhaust fans on and the outdoor air open, we then used the blower doors to move air into the building to achieve the same +4.5 pascals NOP. We found that it required 9980 cfm of air to achieve that NOP; thus the make-up air flow was = 9980 cfm. This is considerably greater than the 7436 cfm that was measured by the TAB firm.

We repeated this test method with the make-up air (MA) on and the outdoor air (OA) closed (masked off but air handler fans still running) in order to find the OA flow rate; OA was found to be 8840 cfm. Given that the net exhaust flow rate was -5694, we can then calculate that the approximate exhaust air (EA) flow rate is:

EA = MA + OA + EAnet = 9980 cfm + 8840 cfm - 5694 cfm = 13,126 cfm

This estimated value for exhaust air of 13,126 cfm is fairly close to the July 1998 Test and Balance measurement of 12,846 cfm and the design value of 12,450 cfm. Since the building was operating at +4.5 pascals during our test, this positive pressure would tend to slightly *increase* the exhaust fan flow rate because the positive building pressure would assist the air flow out through the exhaust ducts, and would tend to slightly *decrease* the make-up air flow rate because the positive building pressure would resist the air flow into the building.

Inspection of the air conditioning systems found that by 11:30 AM the outdoor air dampers (economizer) had apparently closed down considerably. (We say "apparently" because we did not inspect the position of the economizer dampers in the early morning period.) Measurement of building pressure found that NOP had decreased from +4.5 pascals at 9:15 AM to -3.8 pascals at 3 PM. Therefore, the economizers had apparently been operating (opening) during our earlier testing and had therefore been changing the outdoor air flow rates and the building pressure. Based on the blower door test, this change in pressure indicates a reduction in outdoor air from 8840 cfm to 2812 cfm. The fact that the air handlers have economizers, and that they were operating on the day we performed testing, has introduced unexpected complexity to our testing. We could not figure out the controls on the economizers so we did not know what the "normal" damper position would be nor how to change the damper settings. The economizers appear to be controlled by enthalpy sensors, but the actual control strategy and mechanism was not available to us. A summary of building air flows is presented in Table 11.

Table 11
Summary of building air flows.

Air flow based on:	EA	MA	OA	EAnet
measurement @ 9:15 AM	13126	9980	8840	-5694
building dP @ 3 PM	13126[1]	9980[1]	2812	334
TAB report	12846	7436	5698	288
design flows	12450	7500	5625	675

[1] for purposes of estimating OA, we have made the assumption that EA and MA were the same at 9:15 AM and 3 PM

Note that the strong winds that existed during our visit made testing more difficult and the results less accurate. Alternative means are available for measuring exhaust air, make-up air, and outdoor air (e.g., capture tent, tracer gas injection), but these methods require considerably more time. (Note also that the Test and Balance methodology relies upon an assumption of little or no duct leakage to determine the net air flow balance of the building. This assumption is not a good one to make in most buildings.) Given that we had full access to the restaurant only from 8 AM to 10 AM, there was not time for these alternatives. We anticipate re-measuring these building air flows on a less windy day next time.

Inspection of the roof top air conditioning units (RTUs) found four 12.5 ton units and one 10 ton unit. Each is a two-stage Lennox unit with face split cooling coil, with the first stage coil being the (physically) lower of the two (Figure 14). The face split arrangement is an excellent choice for this climate, especially since the air handlers are operated in the fan "on" position. With the face split arrangement, half of the total coil surface is fully cold while the other half remains warm (this is with only first stage active). The portion of the air that passes through the cold side is well dehumidified. By contrast, if the arrangement had been split row (alternating rows are cold and warm), then the entire coil would be cool instead of cold, and therefore little dehumidification would occur.

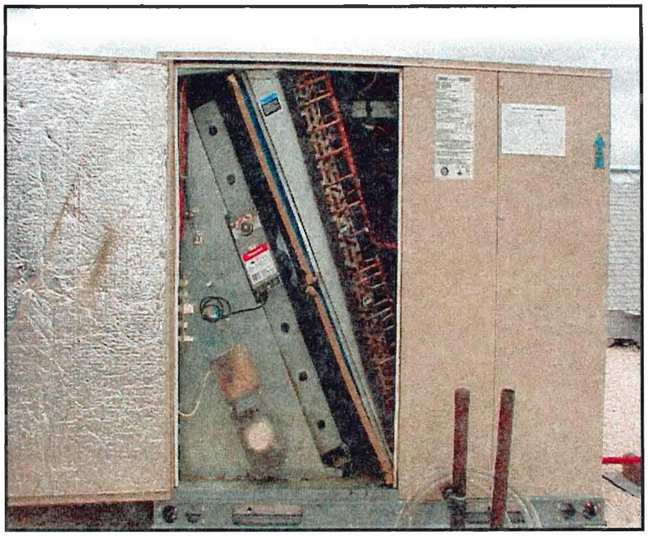


Figure 14. The cooling coil is actually two separate coils, each served by a separate compressor.

Tony Coniglio indicated that it is his understanding and intention that the AC units should be (manually?) switched to "auto" fan at night. When we had checked the thermostats about 7:45 AM, we found four of the five units in the "on" position, and only unit 3 was set to "auto". It was not clear whether there was a master controller that would or could override the manual fan settings.

Because of the uncertainties related to the operation of the economizers and the strong winds, we did not have good confidence in the air flow measurements obtained.

Infiltration testing

The ventilation rate of the building was measured under normal operation. Starting at about 1:45 PM, a tracer gas (sulfur hexafluoride, an odorless and non-toxic gas) was injected into the return air stream to achieve a concentration of about 7 ppm (parts per million) in the room air. For a period of 29 minutes, the tracer gas concentration was recorded at approximately 5 minute intervals. The rate of decline in tracer gas concentration is used to calculate the ventilation rate of the building. The initial concentration was 7.30 ppm at 1:51 PM and had dropped to 2.07 ppm at 2:17 PM when we stopped measurement. Calculation finds that the ventilation rate was 2.91 ach (air changes per hour). Based on the building volume of 82,040 cubic feet, this equates to 3977 cfm of ventilation. We expect that the ventilation rate would have been considerably greater around 8:30 AM when the economizers were apparently operating, because at that time the indicated outdoor air flow rate was 8840 cfm, and this amount of outdoor air should produce a building ventilation rate equal to or greater than outdoor air flow rate.

During the period from 1:48 PM to 2:17 PM we also measured carbon monoxide and carbon dioxide levels in the return air stream. Carbon monoxide was essentially zero. Carbon dioxide was around 815 ppm at the beginning and 700 ppm at the end of this period. The fact that the carbon dioxide level was below 1000 ppm indicates (roughly speaking) that the ventilation rate was sufficient at that time to meet the ventilation requirements of the people in the restaurant. Since outdoor carbon dioxide levels are around 400 ppm in such an urban area, this indicated (again roughly speaking) that the ventilation rate was about 30 cfm per person. (This discussion assumes that there are not significant other sources of carbon dioxide besides people.) To the extent that carbon dioxide from the cooking appliances or soft drink dispensers may be getting into the space, this analysis will be in error. It is expected, however, that most of the combustion gases from cooking are captured by the exhaust hood before mixing into the larger space.

Visual inspections

All five RTUs use economizers (Figure15). The concept of the economizer is to allow the AC units to draw their return air from outdoors when outdoor enthalpy (combination of temperature and humidity) is low enough. When outdoor temperatures are low and outdoor dewpoint temperatures are low, the air brought in by the economizer provides free cooling. We feel that in central Florida economizers are not cost effective because there are few hours during the year when outdoor enthalpy is sufficiently low to obtain free cooling.

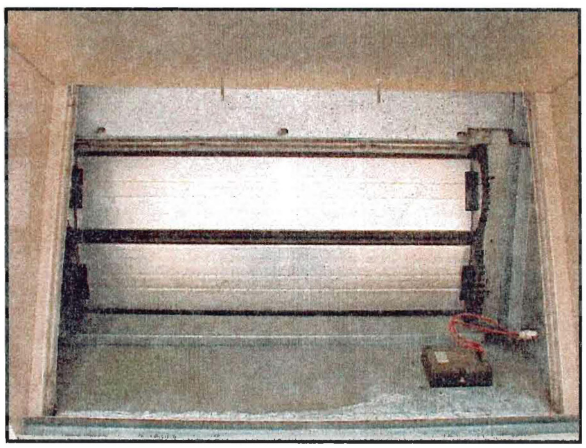


Figure 4. Economizer/outdoor air damper mostly closed.

The economizer dampers open and close based on outdoor conditions, but they (presumably) have a minimum outdoor air setting so that the design outdoor flow rates can be achieved even when the outdoor conditions cannot provide free cooling. When the Test and Balance firm found outdoor air flow rates nearly identical to design, it was July and of course weather conditions would dictate that the AC systems would not be operating in economizer mode. On the day of our testing, outdoor conditions were favorable for the economizer to operate; temperature was about 60F, dewpoint temperature was about 45F, and the resulting enthalpy was about 21.5 Btu per pound of dry air. (This air, when heated to 72F by being placed inside the restaurant, will have a relative humidity of about 42%. Therefore, we can see that this air has both sensible and latent cooling capacity.)

Our testing (as indicated earlier) found outdoor air flows totaling 8840 cfm, considerably more than the design outdoor air of 5625 cfm. By contrast, the indicated outdoor air flow rate that we derived from the blower door test and 3 PM building pressure had fallen to 2812 cfm. Visual inspection of the OA dampers found that they were largely closed at about 2:30 PM. Table 2 presents estimated outdoor air damper opening, measured pressure differential (dP) across the damper, and the estimated outdoor air flow.

Table 2
OA damper opening (visual estimation), pressure differential across the dampers, and estimated OA flow rates based on the an orifice flow equation [1.07 x area (in^2) x dP$^{0.5}$ (pascals)].

RTU	Opening size (in^2)	dP across OA damper (pa)	est. OA flow rate (cfm)
1	84	443	1892
2	98	265	1707
3	49	188	719
4	14	410	303
5	49	250	829
total			**5450**

Since the outdoor air intakes are about 300 square inches when fully open, the dampers were only open about 20% on average when we were testing at 2 PM. Based on the measured pressure differences across the outdoor air dampers, we calculate an estimated total outdoor air flow rate of 5450 cfm. This result is substantially different from that determined based on building pressure measurement at 3 PM of 2812 cfm. However, it is quite close to the 5625 cfm indicated in the construction documents and 5698 found by the TAB firm. Again, we believe the discrepancy is due to the high winds. We will re-measure these air flow rates at our next testing day, hopefully with less wind.

Another observation regarding the AC units is that the filters are "sagging" because of the force of the air flow (Figure 16). Because the filters are being bent by this force, they allow significant air flow bypass. In several of the RTUs, one of the four filters has actually buckled, causing major bypass. This bypassing will lead to rapid dirtying of the coils and reduced cooling performance over time. It appears that a filter with a more rigid frame is required.

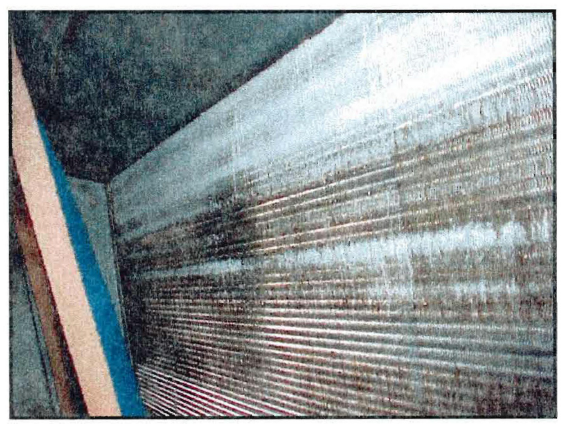


Figure 5. Deflection of filter and frost on coil can be seen.

We also observed that the top coil (second stage) for RTU4 was frosted. This is an indication of low refrigerant charge (Figures 5 and 6).

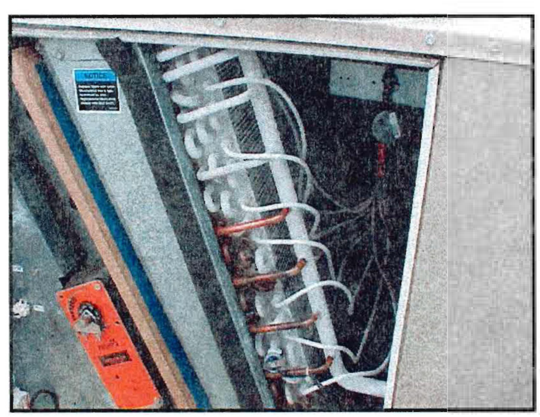


Figure 6. Frost on coil.

On a separate topic, it was reported to us that air flow from supply registers located in front of the exhaust hoods were discharging onto the prepared food and cooling the food. This air discharge may also be disrupting the exhaust hood capture, since it would be blowing toward the face of the exhaust hood. We did not have time to investigate these issues. We have, however, investigated air flow patterns around exhaust hoods in the past using a theatrical smoke generator.

Interestingly, we tested (with smoke) a set-up very similar to this one. The differences were that in the other restaurant the four diffusers were delivering unconditioned make-up air and that the air discharged directly downward from 8" round ducts. Surprisingly, this discharge pattern (jetting down to the floor) worked very well with the hood (that was positioned only 18" away from the make-up air discharge) and the capture effectiveness of the hood was very good in spite of the high velocity air flow (air velocity from the 8" ducts was about 1400 feet per minute).

COMMENTS AND RECOMMENDATIONS

According to Tony Coniglio, it is intended that the AC fans operate in the "on" position throughout the operating hours, and in "auto" position at night. Because of the *two stage* compressor operation and the face split coil, the AC systems can maintain reasonable dehumidification even when the fans are operated in the "on" position. (Note that *single stage* AC systems will produce very poor dehumidification (relative humidities above 70% are common) with fan "on" operation. Also, two stage AC systems with row split coils will produce very poor dehumidification (relative humidities above 70% are common) with fan "on" operation.) Short-term monitoring would help to verify whether in fact humidity levels are being well controlled.

This strategy of running the AC fans in the "on" position during the day (and "auto" at night) should be OK, as indicated because of the two-stage operation with face split coil. As noted earlier, however, four of the five thermostats had air handlers set to "on" when we arrived at 7:30 AM.

Measurements of building air flows indicated at first that the building was at positive pressure. However, later in the day building pressure was negative (exhaust air flows larger than intake air flows), and this indicates to us that the restaurant operates at negative pressure on most days of the year (because the economizer will not be active most days of the year). This negative pressure may be a problem. Extended periods of space depressurization draws moist air from outdoors into wall cavities, and this moist air tends to condense on cool surfaces at the interior plane of exterior walls. The amount of condensation build-up depends upon how cold the walls surfaces are (and this depends largely on the thermostat setting), and how permeable the wall materials are to moisture vapor transport. Cold indoor temperatures during the cooling season combined with vinyl wallpaper, for example, generally cause moisture build-up in the wall board materials and mold and mildew growth. Short-term monitoring of building pressure would verify whether the building is in fact operating at negative pressure a substantial portion of the time.

PART 2 (based on one-week short-term monitoring)

Instrumentation

Instruments were placed in the building on May 27, 1999 and continued to collect 15-minute average data through June 4, 1999. Temperature and relative humidity (RH) were measured in the east dining, west dining, kitchen, and outdoors locations. Indoor air pressure was measured with respect to outdoors. Carbon dioxide levels were measured in the west dining area as an indicator of the adequacy of ventilation.

Findings

Temperature Dining room temperatures remained in a fairly tight range of 74 to 76 degrees during business hours on days with peak outdoor temperatures that averaged about 91 degrees (Figure 18). The hottest days had outdoor temperatures approaching 95 degrees. The fact that indoor temperatures did not experience excursions beyond the 74 to 76 degree range indicates that the AC systems have sufficient capacity to meet load.

The kitchen thermostat was set at 70 degrees. Temperature and relative humidity were monitored near a return intake grill in the kitchen. This temperature represents a blended mixture of kitchen air. It is likely that employees working near the cooking equipment feel hotter due to radiant heat from the cooking appliances and from make-up air that may blow onto them when close to the hoods. From about 7 AM to 11 AM the kitchen temperature remained close to the70 degree setpoint. However, from 11 AM to about 4 PM the kitchen temperature rose steadily (Figure 18). On the hottest days, the kitchen temperature peaked at 76 to 77 degrees. This is quite comfortable compared to kitchens in some other restaurants where we have seen temperatures as high as 90 degrees in the kitchen.

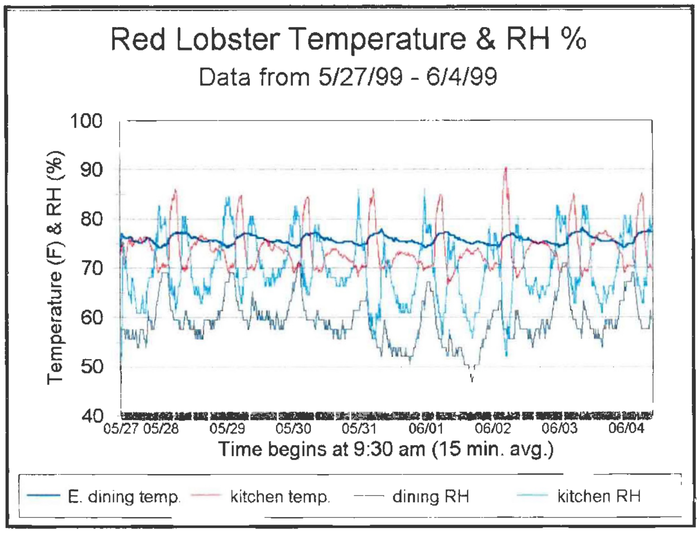


Figure 18. Temperature and RH in the kitchen and dining area

Relative humidity Dining room relative humidity (RH) levels were primarily in the range of 57% to 60% on days with typical summer outdoor dewpoint conditions of 72 to 75 degrees (Figure 18). On two days with outdoor dewpoint temperatures of about 65 degrees, indoor RH levels were primarily in the range of 50% to 54%. The humidity levels found in the dining area -- in the range of 50% to 60% -- are good. The generally accepted rules of thumb are:

- mold and mildew growth is supported above 70% RH
- it is best to maintain RH below 60% most of the time
- 50% to 57% RH is an ideal range
- 40% to 50% RH is comfortable, but additional energy use is required to achieve these lower humidity levels.

Kitchen RH levels were higher than in the dining room. During the majority of business hours, RH was in the range of 65% to 70% (Figure 18). As indicated earlier, these levels are higher than ideal. However, given the considerable rates of humidity generation and high ventilation rates in the kitchen, elevated humidity is not unexpected.

Carbon dioxide People give off carbon dioxide when they breath and it can be used as an indicator of the adequacy of ventilation. ASHRAE recommends 20 cfm per person for dining zones, 15 cfm per person in the kitchen, and 30 cfm per person at the bar. A measured value of 1000 parts per million (ppm) – or about 650 ppm greater than outdoor carbon dioxide levels of 350 to 400 ppm – indicates approximately 15 cfm/person ventilation (assuming a several hour period to reach equilibrium). Measured peak carbon dioxide levels in the dining room were found to be about 1000 ppm (Figure 19). The highest peak was on Saturday May 29 during the evening with a peak of about 1100 ppm. On other days of the week, peak carbon dioxide levels were primarily in the range of 800 to 900 ppm. Therefore, we can conclude that ventilation rates are reasonably on target.

Pressure differential Monitoring found that the building operates at slight negative pressure more than half of business hours. However, the depressurization is generally about -1 pascals with respect to (wrt) outdoors, and during portions of business hours building pressure is in the range of 0 to +1 pascals wrt outdoors (Figure 20). Each day beginning about 11 PM, building pressure goes to about +6 pascals for about 3 hours. Note that during this three hour period the dining room temperature remains steady at about 75 degrees but RH increases steadily from about 60% to about 68% (Figure 21). We do not know the reason for this increase but it may occur because the exhaust fans (with make-up air) are turned off while outdoor air continues to operate (see summary third bullet). It may also occur as a result of the economizers opening up. (We do not know why the economizers would activate at this time, but if they did so the added ventilation air would tend to elevate humidity.) The increase of approximately 7 pascals (from -1 pascals to +6 pascals) indicates a net increase of air flow into the building of

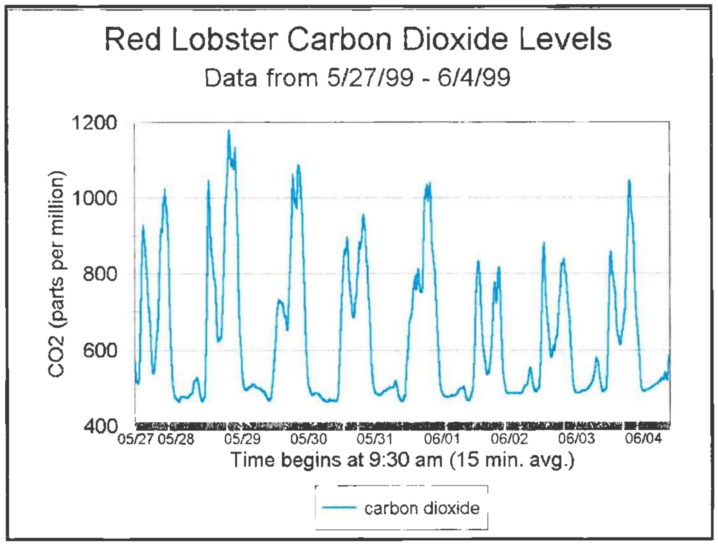


Figure 19. Indoor dining area CO_2 levels

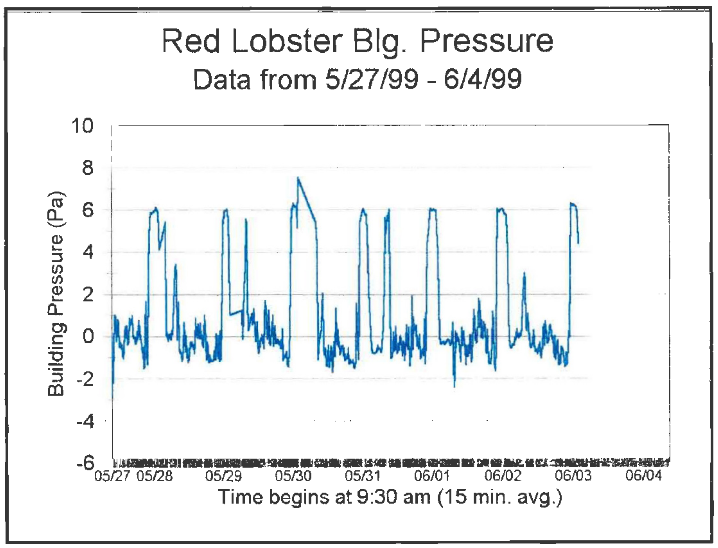


Figure 20. Indoor pressure wrt outside over several days

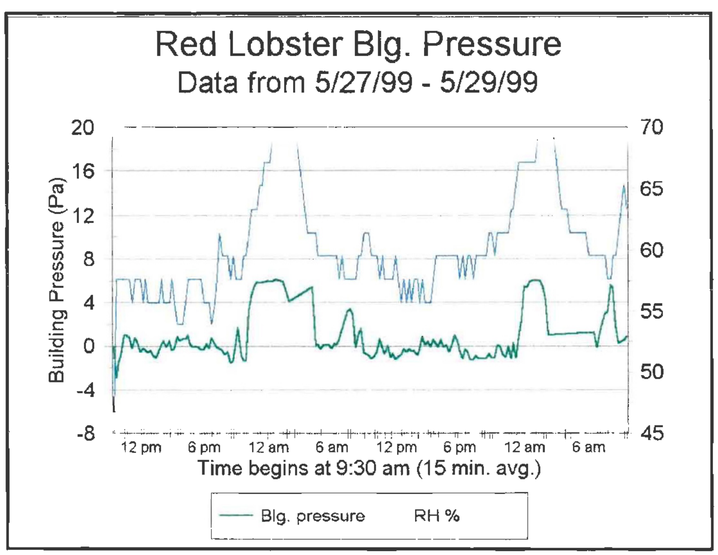


Figure 21. Indoor pressure wrt outside and relative humidity May 27-29

5050 cfm, based on the building airtightness test. On balance, this restaurant is less depressurized than the 15 other restaurants we have tested.

Summary

- The building is moderately airtight, with ACH50 of 9.7. We recommend airtightness to fall in the range of 4 to 8 ACH50. Therefore, it would be useful, in our opinion, to perform some tightening of the building envelope.

- The building is operating at close to neutral pressure most of the time. For just over half of business hours the building is at about -1 pascals and most of the remaining time the building is at about +1 pascals. For 3 hours each night the building is at +6 pascals. Additional investigation could be performed to determine the cause of the +6 pascal events.

- The roof top AC units have two-stage compressor operation with face-split coils (one separate coil for each compressor) with a single speed air handler fan. Cooling output is modulated by switching between first and second stage operation. With first stage, one coil (the physically lower one) is cold. With second stage operation, both coils are cold. This AC system configuration is especially well suited to the Florida climate because normally at least one of the coils is cold and is therefore dehumidifying room air and outdoor air. The AC unit configuration could be improved, however, if the first stage coil was positioned such that the outdoor air would flow through this coil. As configured, the majority of the outdoor air appears to pass over the second stage coil. Since the second stage coil is "off" much more than the first stage coil, the outdoor air tends to enter the building untreated when the second stage is shut down. Dehumidification would be much more effective and efficient if the outdoor air passed through the first stage coil; this would require a design change in the unit.

- The roof- top AC units have economizers which open up the "outdoor air" during periods of cool weather (low enthalpy) to provide free cooling. Testing on a cool day in March 1999 found the building operating at +4.5 pascals at 9:30 AM. However, by 3 PM building pressure had fallen to -3.8 pascals (note that it was a very windy day and strong winds tend to depressurize buildings). We have concluded that the economizers were operating in the morning but had shut down by afternoon.

- Dining room temperatures remained at a fairly constant 75 degrees during business hours.

- Kitchen thermostat temperature was set at 70 degrees. During the beginning portion of the morning, the kitchen operated at 70 degrees but then rose to about 76 degrees in the afternoon. Nevertheless, this temperature should provide reasonable comfort for the kitchen staff.

- Dining room relative humidity was 57% to 60% during business hours during normal summer weather conditions. This is at the high end of acceptable. Ideally humidity would be maintained in the range of 50% to 57%. Relative humidity in the range of 40% to 50% is comfortable, but more energy is consumed to achieve these lower humidity levels.

- Kitchen relative humidity was in the range of 65% to 70% during the majority of business hours. This is higher than ideal, but still below the critical level of 70%. During late evening hours and over night, kitchen humidity rises to 75% to 80%. This may be in response to the phenomenon which is causing the building to go to +6 pascals for a three hour period in the late evening.

- Building ventilation is sufficient for building occupancy, based on the carbon dioxide levels which remained below 1000 ppm on most days. Tracer gas testing on March 15, 1999 found a ventilation rate of 3977 cfm, and this is sufficient ventilation for about 200 persons.

- The roof- top AC units have two compressors and two coils, controlled by two-stage thermostat operation. This equipment is well suited to controlling humidity. Running the AC fans in the "on" position during the day and "auto" at night should work well. During the day, the outdoor air functions as make-up air for the exhaust fans. Therefore, if the air handler fans were to cycle off during periods when the exhaust fans were operating, then the building pressure would go more negative. Having the fans cycle at night should not cause depressurization since the exhaust fans will be off. Leaving the AC fans in the "on" position overnight should be avoided because it will increase indoor RH and could contribute to comfort and possibly building degradation problems.

- As of March 15, 1999, the top coil of AC#4 was frosted indicating low refrigerant charge. This may have been corrected by now.

RECOMMENDATIONS

- Perform airtightening of the building. Building tightness is currently 9.7 ACH50. We suggest a target of approximately 6 ACH50. We can be available to test for leak locations, make recommendations on how tightening could be achieved, and retest after the tightening is complete.

- Adjust exhaust, make-up air, and outdoor air flows to achieve positive pressure (of say +2 pascals) in the building during normal business hours. If the building is made more airtight, then less air will need to be pushed into the building to achieve +2 pascals.

- Based on the information we have obtained about the operation of this restaurant, it appears that in general this building is operating quite well.

7. RIVER FRONT RESTAURANT

Testing was performed on this restaurant 9-21-98.

Building description.

The building is two stories and of block construction. It has a flat roof on corrugated metal roof deck. Ceilings are T-bar throughout the dining areas. The kitchen ceiling is gypsum board. Ceilings are at 10 feet. There is 44 inch clearance between the ceiling and deck above. Total building height is 29 feet.

Three air conditioning systems serve the building – the outdoor units are located on the roof. A ten-ton system serves the downstairs dining room. A ten-ton system serves the upstairs main dining area. A party room upstairs is served by a five ton system. Model numbers for the outdoor units were Trane BTA120C200MA, Trane BTA120C200MA, and Trane BTA060D300AO. Table 13 presents test data from the three cooling systems which serve the building, including return and supply temperature, relative humidity, enthalpy, and calculated cooling output.

Table 13
AC performance test results

location (tons)	return T/RH/h	supply T/RH/h	air flow	Btu/hr
dining up (10)	79.7/50/31.2	58.9/79/23.4	2066	71621
party room (5)	67.8/58/25.2	56.1/82/22.0	1925	27378
dining down (10)	72.1/71/30.4	56.9/91/23.5	2500[1]	76667
[1] Estimated based on measured return air of 2280, and assuming some return side duct leakage.				

Based on preliminary AC performance tests, these systems are putting out considerably less than rated capacity (note that test conditions deviated significantly from ARI test conditions). On average, these three systems cooling output is only about 60% of rated capacity, probably due in large part to restricted air flow.

Building operating pressure was found to be -45 pascals. However, when the kitchen door to outdoors was left open (screen door make-up air), building pressure declined to -2.8 pascals. It turns out that the building is normally operated with the kitchen door open. The owner noted that if the kitchen door is not left open, the water heater has to be turned off to prevent flame roll-out. The pilot light is routinely blown out.

A blower door test found the building airtightness to be 3.8 ACH50 (3722 CFM50). This is a very tight building. When the building was at -50 pascals, pressure in the upstairs ceiling space was found to be -49.4 pascals (wrt outdoors). Therefore, the roof deck (and not the ceiling) is the dominant air boundary.

Exhaust and make-up air flows were measured using the building as a capture tent. First, with all HVAC systems operating, the make-up air (MA) was turned off and masked off. This caused building pressure to go more negative. A calibrated fan (blower door) was used to blow air into the building in order to bring the building back to -45 pascals. This found MA to be 1738 cfm. Then the exhaust air (EA) was also turned off and masked off. Building pressure became more positive. The calibrated blower then was used to pull air from the building in order to return the building to -45 pascals. The flow through the calibrated fan was then equal to net EA which is equal to EA minus MA. Based on this, we determined that net EA was 3580 and EA was therefore 5318 cfm. In order to bring the building back to neutral pressure, additional MA or outdoor air (OA) totaling 3580 cfm would need to be added. Note that there was no OA at the time of testing.

An ventilation test was performed on the building with the air handlers set to continuous operation. The ventilation rate was found to be 0.70 ach which also converts to 1965 cfm.

Carbon dioxide levels were also measured. At 3:03 PM, the level was found to be 855 ppm in the downstairs dining area. Over the lunch hour, there were approximately 25 persons in the building.

Inspection was made of the gas water heater. The water heater is fully exposed to the pressures that exist in the building. This is dangerous since pressures of -5 to -15 pascals can backdraft such appliances (this means the combustion gases including carbon monoxide come into the building). Pressures greater than -20 pascals can produce flame rollout (flames burning outside the burn chamber) which can create a fire hazard. It was also observed that the discharge of one of the exhaust fans was blowing directly onto the termination of the vent pipe and therefore could cause backdrafting of the appliance.

The kitchen was observed to be (and reported to be) very hot. There is no air conditioning being provided to that space. With the kitchen door to outdoors open, little air is drawn from the dining area into the kitchen.

There are many heat sources in the kitchen. There are a number of cooking appliances located under the exhaust hood. There is also a large baking oven located away from the hood. There are heat lamps (to keep the food warm). There are about five refrigeration units (discharging heat to indoors) and an ice machine scattered about the kitchen. In order to enhance the entry of cool outdoor air, two holes have been knocked in exterior walls with total opening of 54 square inches.

Two problems have been identified related to the exhaust system. The hood is located high above the cooking appliances, and is wide open at both ends. Four MA discharge grills are located about 15 inches from the front edge of the hood and are discharging directly downward at approximately 1000 fpm. Upon first inspection, we suspected that this high velocity MA was disrupting the effective capture of cooking heat, humidity, and fumes. We felt that this MA should be directed away from the hood. (Later inspection found that the MA was not disrupting exhaust hood capture.)

RECOMMENDATIONS

Based on this testing, a preliminary set of recommendations were developed and forwarded to the restaurant owner.

1. Enclose the gas water heater in a small combustion room with tight walls and ceiling (gypsum board ceiling or equivalent tightness). Add an additional combustion/dilution air intake stack (probably from the roof) to terminate about 12 inches from the ceiling. Duct the current combustion/dilution vent to approximately 12 inches from the floor. As a consequence, this combustion zone should be well connected to outdoors and poorly connected to the building interior.

2. Add about 6 feet to the water heater combustion stack so that it is well above the discharge of one of the exhaust fans.

3. Add about 3000 cfm of MA. This could be added farther away from the hood.

4. Redirect the existing MA so that it blows away from the hood, or even better, have it discharge from a large plenum/perforated grill downward at a rate of approximately 100 fpm discharge. In order to achieve this, the perforated grill should be approximately 17 square feet in surface area.

5. Add approximately 700 cfm of OA to the downstairs or main upstairs air system. This will provide additional ventilation for staff and customers and help bring the building into air flow balance and neutral or positive pressure.

6. Modify the exhaust hood so that is captures the cooking by-products more effectively. This would involve installation of side panels from the hood to floor. The front face of the hood could be extended down by 12 inches or more. These measures would increase the air velocity at the face of the hood and thereby improve the capture efficiency of the EA.

7. Increase the air flow of the two main AC systems. In the upstairs system, diminish the sharp turns of the return ducts. There are three flex ducts connecting the return grills to the air handler. Because the ducts are too short, the are stretched tightly causing sharp turns and restriction at a total of six junctions. Longer ducts or curved metal turns could be installed. In the downstairs system, add four or more supply ducts and registers to allow substantially greater air flow. Extremely high pressures in the main supply duct indicate that more supply discharge is needed.

8. Clean the cooling coils on the downstairs AC unit. Access is difficult. At least vacuum the coils to remove insulation materials. We could not inspect the coils of the upstairs main system but it is likely that those coils could use cleaning as well.

Additional Field Testing at the River Front Restaurant

Additional testing was performed on the kitchen exhaust and make-up air system at the River Front Restaurant on October 5, 1998. Using a theatrical smoke generator, the capture effectiveness of the exhaust hood system was examined. Smoke was introduced at two levels: 3 feet above floor level (approximating the cooking surface) and right at floor level. A Hi-8 video camera was used to film the capture effectiveness of the exhaust system.

The normal configuration in the kitchen is to operate the exhaust air (EA) and make-up air (MA) concurrently and keep the exterior kitchen door open. Several other configurations were examined as well. The test consisted of introducing a plume of smoke parallel to the face of the exhaust hood within 12 inches of the furthest extension of the hood into the kitchen, first at the floor level and then at 3 feet above the floor. Smoke was ejected from the machine for about 10 to 20 feet along the length of the hood (the distance of ejection depending upon how quickly the smoke was swept up into the hood and the form of make-up air being used). Smoke ejection occurred for several seconds at a time, to create one plum of smoke in proximity to the hood. The tests were performed under four exhaust system operation configurations; 1) EA on, MA on, and kitchen door open (to outdoors), 2) EA on, MA off, and kitchen door open, 3) EA on, MA off, and kitchen door closed, and 4) EA on, MA on, and kitchen door closed. Tables 14 and 15 summarize the visual observations made during these tests.

Table 14
Theatrical smoke testing of exhaust air system (smoke introduced *at floor level)*

Configuratio n	Kitchen Door	Exhaust Air	Make-up Air	# Seconds till Complete Capture	Evaluation of EA Capture of Smoke
1	open	on	on	5	excellent
2	open	on	off	25	good
3	closed	on	off	NA	good
4	closed	on	on	NA	good
NA - Not available					

Table 15
Theatrical smoke testing of exhaust air system (smoke introduced *3 feet above floor*)

Configuration	Kitchen Door	Exhaust Air	Make-up Air	# Seconds till Complete Capture	Evaluation of EA Capture of Smoke
1	open	on	on	5	very good
2	open	on	off	6	very good
3	closed	on	off	8	good - v. good
4	closed	on	on	5	very good

Based on previous testing, this building is very airtight (3.8 ACH50) and operates at -45 pascals when the kitchen (exterior) door is closed. When the kitchen door (to outdoors) is open, the main portion of the building operates at -2.8 pascals. With the door closed, exhaust and make-up air was previously measured using the building as a capture tent. In this method, various fans are turned off (and masked off, to close up the hole they represent when off) and a blower door is used to move air into or out of the building in order to match the building pressure that occurred during normal operation. EA was found to be 5,318 cfm, MA 1,738 cfm, and net EA 3,580 (all flows with exterior kitchen door closed). Considering exhaust hood capture, it was expected that the MA might be causing the hood to not capture well, because the 1,738 cfm of MA was coming from four open round diffusers at very high velocity, straight down from the ceiling (about 9 feet) and impinging on the floor just a matter of inches from the front of the cooking appliances.

Since the cooking and much of the combustion takes place at approximately 3 foot height, it seems that this location is most relevant. All of the configurations yielded good or very good capture at the 3 foot height. Note that having the MA "on" enhances the capture, especially at the floor level, but also at 3 feet above the floor. This is somewhat surprising considering that the MA comes out of 8 inch round ducts at rather high velocity (approximately 1,400 fpm) at a location only 18 inches from the front of the hood (blowing mostly straight down). In configuration 2 with the MA off, it required 25 seconds for the smoke to be captured (but it still was all captured without dispersing into the room). When the smoke was injected at the floor, the MA impinging on the floor appeared to significantly improve the hood capture since it was hitting the floor and pushing air fairly aggressively toward the cooking appliances under the hood. Based on the smoke testing, this high velocity air flow appears to have almost no effect on the air flow into the hood, and at the floor level it drives the combustion fumes (theatrical smoke) aggressively under the cooking appliances and then upward into the hood.

With the kitchen door open, outdoor air flows into the kitchen from outdoors at a fairly good velocity caused by about -3 pascals depressurization. This air flowing into the kitchen seemed to have relatively little impact on exhaust hood's ability to capture the smoke. The air flowing into the kitchen did not create turbulence which diminished the hood capture effectiveness because the air flowing in through the door moved into an area of the kitchen away from the hood.

Additional inspections

Since the exhaust hood appears to be capturing a large majority of the heat, humidity, and cooking fumes for the appliances located under the hood, we inspected the kitchen to characterize other heat sources which cause the kitchen to operate at an elevated temperature and therefore causes discomfort to kitchen staff. There are two walk-in cooling units which discharge their waste heat to outdoors. These are not part of the problem. There are eight other refrigeration units located in the kitchen which discharge heat into the space. These include four refrigerator units (two with glass doors), one counter-top food preparation unit, two freezers, and an ice machine. Other heat generators include a large convection oven, a smaller baking oven, and heat lamps to keep the food hot prior to serving.

There are some problems with the refrigeration systems. First, they discharge all their heat into the kitchen. If there was a way to discharge their waste heat to outdoors, this would yield a cooler kitchen. The ice maker, for example, could be relocated so that the heat blown off the condenser could be ducted to outdoors. Second, they operate in a kitchen which often operates at temperatures well above 90°F and this is a thermally very inefficient environment. Third, some of the units are located in especially hostile thermal environments. One of the glass door coolers is located immediately next to the deep fryers on one side and the large convection oven on the other side (there are insulation panels on the convection oven side but not on the other). One of the freezers (where french fries are stored) is located under the exhaust hood, underneath a steamer and next to the deep fryers on one side and a gas stove on the other side. Dealing with these sources of heat within the kitchen environment was beyond the scope of this project, other than the fact that we brought this information to the attention of the owner.

8. STEAK HOUSE CHAIN RESTAURANT

1. The Building

This four year old restaurant (Figure 22) is located in Orlando, Florida and is the prototype store for this steakhouse chain. Floor area is 6061 square feet excluding the water heater mechanical room. With approximately 8.5 foot ceilings, the building volume is 51,157 cubic feet. [For purposes of tracer gas ventilation testing, we also exclude the walk-in freezers and coolers, yielding 48,955 cubic feet.]

Figure 22. Steakhouse from Kirkman Road

Exterior walls are 6-inch metal studs with 6 inch batts, exterior plywood sheathing, and exterior siding. A "flat" roof slopes from west to east at a 1/4 inch per foot slope. It is constructed of corrugated galvanized metal roof deck covered by rigid insulation (R_value uncertain) and a highly reflective white single-ply roofing membrane. Parapet walls surround the roof area, and a perimeter vented "attic" space extends around all four sides.

A suspended t-bar ceiling is used throughout the building. By design, this ceiling is neither the intended air boundary or the intended thermal boundary of the building. Those are at the roof deck and at the exterior walls above the ceiling. By design, then, the ceiling space is intended to be inside the air and thermal boundaries of the building, and therefore to be quasi-conditioned space. As will be discussed in this report, this does not occur because of unintended air leakage pathways above the ceiling and significant depressurization of the building. According to the store proprietor, there have been no significant humidity problems. However, a significant mold growth event occurred on the ceiling tiles about one year prior to our testing. It was reported that the ceiling tiles had to be replaced (before the mold event?) because of sagging.

Air conditioning is provided by three roof top units (RTUs) in the dining area and two RTUs in the kitchen, with total capacity of 45 tons. Ductwork is located in the ceiling space. It is primarily ductboard trunk ducts and flex to registers. Three exhaust fans serve the cooking area, one serves the dishwasher and turns on only when the dishwasher operates, one serves the two bathrooms, and one small fan serves the kitchen staff bathroom (controlled by light switch). Two make-up air units provide unconditioned make-up air to the kitchen exhaust hoods. The kitchen is reported to be much hotter in the last year, and the AC units "trip" fairly often. Take-out food service (through the south door) has been added to their operations. Food preparation runs at 450 entrees per weekday and 650 per weekend day.

2. Testing of Airtightness, Air Flows, and Pressures

Normal operating pressure

The normal operating pressure (NOP) of the building was determined with all five roof top units (RTUs) operating, both make-up air (MA) units operating, and all exhaust air (EA) systems except the dishwasher operating. NOP was found to be -2.7 pascals (wind < 3 mph). (Figure 23) This means that the building normally operates at -2.7 pascals with respect to (wrt) outdoors, and it is therefore sucking air into the building. When the dishwasher EA is also turned on, building pressure goes to -5.0 pascals.

Figure 23. Eight-channel pressure monitor with computer display helps in diagnostic testing.

Blower door test

A blower door test was done to determine building airtightness (Figure 24). In preparation, all EA and MA units were turned off and masked off. The RTUs remained on but the outdoor air (OA) intakes were masked off. CFM50 was found to be 12,021. [The airtightness curve for the building is defined by an equation $Q = C \times dP^n$ where C = 989.4 and exponent n = 0.64 (see Figure 4). Using this curve, we can project air flow into the building for any given building air

Figure 24. Blower door set-up installed in pick-up doorway.

pressure.] The air flow rate into the building caused by space depressurization of 2.7 pascals is calculated as 1868 cfm [$Q = 989.4\ (2.7)^{0.64} = 1868$ cfm]. At -5.0 pascals (the building pressure with the dishwasher EA also operating), the flow into the building is calculated to be 2772 cfm. This indicates a dishwasher EA flow rate of roughly 904 cfm (2772 cfm - 1868 cfm = 904 cfm). (Note that measurements at the roof discharge found that the dishwasher EA was moving 956 cfm.)

CFM50 is the flow rate (cfm) through all cracks and penetrations in the building envelope when the building is depressurized to -50 pascals (-0.2 inWC). The meaning of this test result is more easily understood (and comparisons between buildings is easier) when converted to ACH50. ACH50 is the flow rate through all cracks and penetrations in the building envelope with the building at -50 pascals expressed in building air volume exchanges (outdoor air replacing indoor air). ACH50 is 14.1 for this building, which indicates a fairly leaky building. For good practice construction, we suggest 6 ACH50 or tighter. On the other hand, making the building excessively tight (ACH50 less than say 3.5) could result in severe building pressure problems if building air flows go out of balance.

ACH50 of 14.1 means that in one hour's time a volume of air equal to 14.1 times the occupied volume of the building moves through building air leakage pathways when the building is under 50 pascals of depressurization. It is important to understand that this has relatively little to do with the actual ventilation rate of the building. We can predict an approximate natural infiltration rate (natural infiltration is based on wind and temperature difference as the driving forces) when all the mechanical air moving systems are turned off, such as during the middle of the night. Based on empirical data from testing in 70 small commercial buildings, we have found that natural infiltration can be predicted by dividing ACH50 by 40. Therefore, the estimated natural infiltration rate would be 0.35 ach (14.1 / 40 = 0.35 ach).

Building ventilation

The actual ventilation rate which occurs during occupied hours has almost nothing to do with the airtightness of the building but everything to do with the mechanical air moving systems. Consider four building air flows:

1) Each of the five RTUs have outdoor air. When the air handlers (blowers) for these units are operating, air is drawn into the return air stream through the outdoor air (OA) intakes. Measurements found a total OA of 2734 cfm for the five RTUs. By itself, the outdoor air would produce a building ventilation rate of 3.35 ach.

2) In addition to OA, return leakage (RL) in the RTUs will draw additional air into the return air stream. We did not measure the amount of RL but it is likely to be 2% to 4% of total air flow. If RTU cabinet return leakage is 3%, then return leak air flow would be about 525 cfm. This RL would be equal to another 0.55 ach ventilation.

3) Additionally, the imbalance between exhaust (EA) and mechanically transported incoming air (MTIA = MA + OA + RL) causes additional ventilation (more accurately, we may call this infiltration which causes ventilation). This difference between EA and MTIA can be referred to as EAnet. When EA is greater than MTIA, as is the case in this building, the result is space depressurization, and this depressurization draws air into the building through all openings in the building envelope.

Building Airtightness

Outback Steakhouse — OUTUS

Building Description

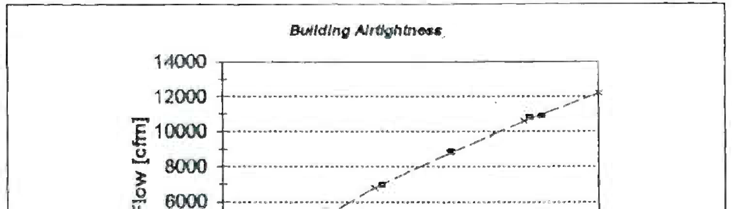
6115	Floor area [sqft]	72	Inside Temp	Date:	19-Jul-99
8.5	Ave Building Height [ft]	87	Outside Temp	Time:	*start*
51977.5	Building volume	1.014	Air Density Factor		
14	Max building height [ft]	a	Wind sheilding (a)ve, (m)in, ma(x)		
0.85	Height factor	23.5	"N"		
	Surface Area		Average number of people		

Blower Door Data

Door Location		takeout door			takeout door											
		Fan Id # 7017			Fan Id # 7018			Fan Id #			Fan Id #					
	Building Press	Fan Press	Fan Config	Fan Flow	Fan Press	Fan Config	Fan Flow	Fan Press	Fan Config	Fan Flow	Fan Press	Fan Config	Fan Flow	Sum Flows	Temp. Comp.	Calc cfm
1	9.5			4004										4004	4060	4222
2	13.4			5348										5348	5423	5259
3	21.0			5681			1189							6870	6966	7006
4	30.2			5556			3185							8741	8863	8835
5	40.8			5500			5144							10644	10793	10705
6	42.4			5312			5400							10712	10862	10972

Calculations

0.64	n	12189	CFM50	14.1	ACH50
1003.28	C	1584.6	FLA	0.6	ach (ACH50/N)
0.9974	r	4363	CFM10	518	cfm (estimated)
		1282.3	EqLA	7.8	SLA
		2431	CFM4		cfm needed for 15cfm / person
		689.1	ELA	303	cfm @ .35 ach



B-55

As indicated previously, the NOP with all HVAC systems operating except the dishwasher exhaust was -2.7 pascals. Based on the building airtightness test, EAnet is indicated to be 1868 cfm. Stated another way, total exhaust air is 1868 cfm greater than the sum of MA, OA, and RL air. This 1868 cfm contributes directly to the bottom-line ventilation of the building, and is equal to 2.3 ach. So far, we have a projected building ventilation rate of 6.2 ach.

4) The last source of ventilation is some portion of the kitchen MA which is not captured by the EA. MA that escapes from the exhaust hood and is drawn into a return grill is then distributed throughout the kitchen space and therefore adds to the ventilation rate of the kitchen (and load on the RTUs). We were unable to perform any testing to examine the MA capture efficiency, so we do not know how much MA is contributing to the building ventilation rate.

Tracer gas ventilation test

A tracer gas decay test was performed to measure the ventilation rate of the building. A tracer gas is well-mixed throughout the indoor air and then the rate of decrease in tracer gas concentration yields the ventilation rate. The faster the gas dissipates, the higher the ventilation rate. The tracer gas test determines the actual ventilation rate occurring during the time of the test. Tracer gas (nitrous oxide) was injected into the return grills of the RTUs and distributed throughout the space. The average space concentration at the beginning of the test was 8.14 ppm (parts per million). In a period of 17.5 minutes, that concentration dropped to an average of 1.64 ppm, indicating a ventilation rate of 5.5 ach. However, since the concentration of tracer gas was not homogenous throughout the occupied space and in fact was lower in the kitchen, the calculated ventilation rate underestimates actual ventilation. Since the tracer gas concentration in the kitchen adjacent to the exhaust hoods (at two locations about 4 feet and 12 feet from the hoods) was only 67% of the building average, and since the air leaves the building through the kitchen hoods, the actual ventilation rate is higher than the calculated 5.5 ach. Dividing 5.5 ach by 0.67 yields a predicted building ventilation rate of 8.3 ach. Expressed in cfm, this is 6750 cfm total ventilation.

To obtain a more accurate assessment of building ventilation would require additional testing. Nevertheless, it appears that the building ventilation rate is in the range of 6.5 to 8.5 ach, or 5285 cfm to 6910 cfm. Since it appears that peak occupancy is on the order of 200 persons and ASHRAE 62-1989 calls for 20 cfm per person, ventilation is on the order of 50% greater than the approximate 4000 cfm necessary. Also, considering that this 200 person occupancy does not occur continuously through the day, there may be opportunities to reduce the ventilation requirement below 4000 cfm based on 3-hour average occupancy.

Elevated indoor humidity

Indoor conditions were running at about 71 degrees F and 70% relative humidity (RH) during the period from 10 AM to 12AM on the day of our testing. Additional data is available for a seven-day monitoring period (Figure 26). Dining room temperatures follow a daily pattern; they rise in the afternoon to about 76 degrees F and fall to about 70 degrees F overnight. RH falls primarily within the range of 65% to 85% which is well above the recommended range of 50% to 58%. The range of 40% to 50% is also considered acceptable, however lower RH levels require greater energy expenditures. On the other hand, RH greater than 70% is considered unacceptable because it is uncomfortable and more importantly can lead to mold and mildew growth.

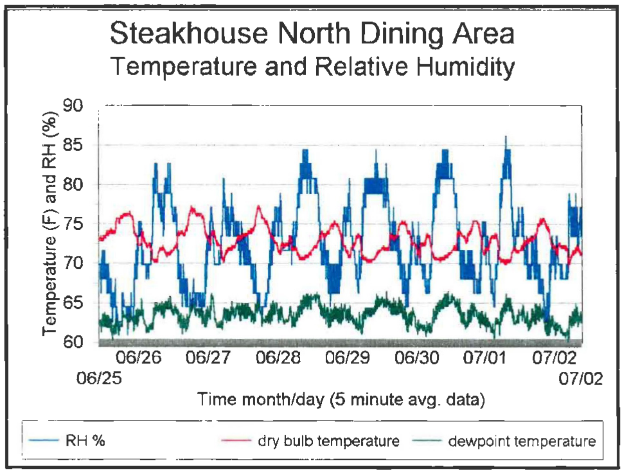


Figure 26

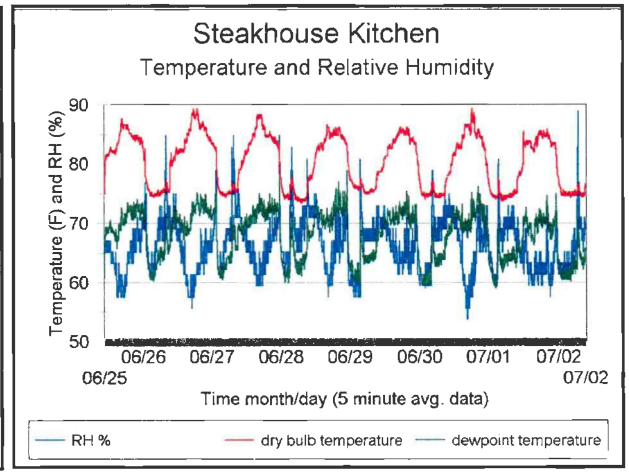


Figure 27

Conditions in the kitchen are less favorable compared to the dining area (Figure 27). Overnight, room temperature fall to about 74 degrees F. However, as soon as the EA is turned on in the morning, the kitchen temperature goes to about 80 degrees F and then rises into the afternoon, often reaching 85 degrees F or higher. RH in the kitchen averages about 65%. Kitchen dewpoint temperature is mostly in the range of 70 to 75 degrees F when the EA is operating, and drops to about 63 degrees during the nighttime hours when the EA is off.

Conditions in the ceiling space are considerably warmer and more humid than in the dining area. During the hours of EA operation and the building is depressurized, the ceiling space dewpoint temperature is about 71 degrees F (Figure 28). On the day of testing, we found ceiling space conditions of 81 degrees F, 67% RH, and 71 degrees F dewpoint temperature immediately on top of the ceiling tiles at 3 PM. During our week of monitoring we found that ceiling space temperatures were peaking at about 85 to 90 degrees F midpoint between the ceiling and roof deck.

Note however, that during the night when the EA was off and the building was at +1 pascal pressure (positive pressure because of OA), the dewpoint temperature in the ceiling space declined abruptly to about 62 degrees F (see Figure 29).

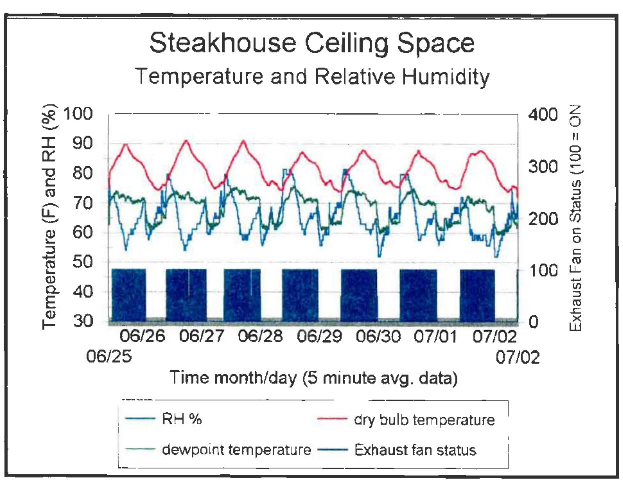


Figure 28

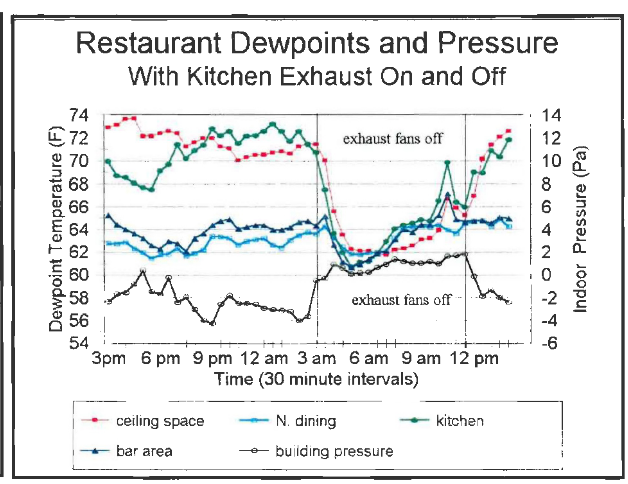


Figure 29

What are the causes of high indoor humidity in the occupied spaces? A high ventilation rate through intended and unintended openings in this building is probably the dominant factor contributing to elevated humidity. It is also possible that air conditioner performance problems, as identified in RTU-4 (see Section 3), may be contributing to poor moisture removal by the cooling systems.

Building air flows

All air flows were measured, including supply, return, EA, MA, and OA. Air flows related to the building air flow balance (air flow across the building envelope) are summarized in Table 16. Total building EA (excluding the dishwasher) was measured to be 10,866 cfm, compared to 10,875 cfm design. Note that we have used the Melink Test and Balance report measurement for EF-1. (We did not use our capture tent to measure the flow rate of EF-1 since considerable grease was being discharged. Instead, we used an air velocity probe to measure an average velocity of 2983 ft/min across the 1.831 square foot EA discharge orifice. This yields a flow rate of 5462 cfm, but we have reason to believe this is too high. If we use our measurement of 5462 cfm, we find that the predicted total EAnet is about 1000 cfm too great. Another way to look at this, if we use this flow rate for EF-1, then the indicated EAnet would produce depressurization of -4.8 pascals when in actual fact the building pressure was only -2.7 pascals. When we compared the TAB measurements done by Melink to our own measurements, we found that in all other cases their measured EA flows were quite close to what we measured (except for the dishwasher), and so we conclude that our measurement for EF-1 is too high. We have chosen, therefore, to use the TAB report flow rate for that exhaust fan of 4279 cfm.) OA filters (screens) were all very dirty and in need of cleaning. Total OA flow was 2734 cfm which is 701 cfm or 13% less than the design OA flow rate of 3160 cfm and 20% less than the 3435 cfm on the Test and Balance report. If the OA flows are increased to design, then building pressure should decrease from -2.7 pascals to about -1.8 pascals.

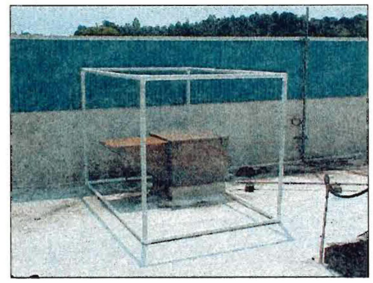


Figure 30. Capture tent frame over MA

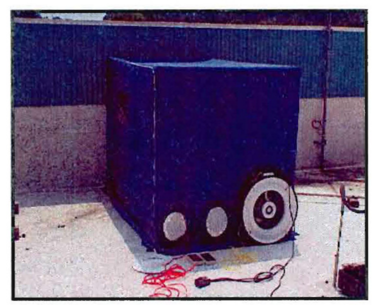


Figure 31. Capture tent measuring air flow rate of MA unit.

Table 16
Measured and design air flows (cfm) related to the building air flow balance.

			make-up air			**exhaust air**			
EA unit	meas.	design	MA unit	meas.	design	RTU	meas. supply	OA meas.	OA design
1 *hood*	4279	4200	1	4091	6615	1	3524	718	720
2 *hood*	2902	3150	2	889	1805	2	3946	561	720
3 *hood*	3080	2850				3	3575	509	720
5 *bathrooms*	605	675				4	3153	507	720
						5	1682	439	280
total	**10866**	10875	total	**4980**	8420	total		**2734**	3160
4 *dishwasher*	956	1000							

EA flow is close to design, OA flow falls somewhat short of design, but MA flows falls far short. Design MA is 8420 cfm while measured MA flow was only 4980 cfm.

Discussion of exhaust and make-up design and air flows

MA should be increased to bring the building to slight positive pressure. Increasing MA flow raises the issue of where and how to discharge this air. Currently, two of the hoods discharge air both internally (inside the hood) and to the room. We feel that internally discharged air does not provide much benefit, and that it would be better to reduce the EA flow rates, increase the MA flow rates somewhat, and discharge all of the MA in front of the hoods (Figure 32). (Testing would need to be done to verify that the exhaust hoods still capture the cooking effluents with the lower flow rate.)

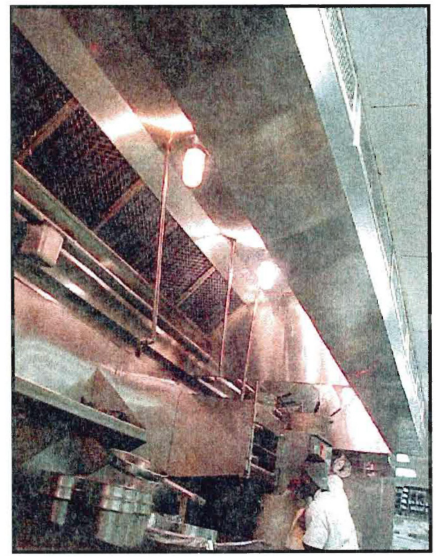


Figure 32. Kitchen hood with both external and internal MA discharge.

This will result in considerably more unconditioned MA being discharged onto and in close proximity to kitchen workers. This air is hot and humid through much of the year. While the velocity of the air provides some moderation of the sense of heat, it nevertheless cannot be considered comfortable. We feel that this discomfort can be largely overcome by four factors.

- Eliminating the unbalanced EA (that is, making total incoming air slightly greater than EA) will reduce the building ventilation rate and thereby reduce the cooling load. This in turn will allow the dining area and the kitchen to operate at cooler temperature.
- Correcting performance and control problems with kitchen RTU-4 will improve cooling output and reduce kitchen temperature.
- Eliminating OA on the kitchen RTUs will increase their cooling capacity and allow cooler kitchen temperature.
- Installing directionally adjustable discharge supply diffusers along the front of the exhaust hood food preparation areas will allow directing supply air onto the workers, thereby improving comfort and providing a sense of control over their environment.

Building air boundary

Note that the ceiling, which is suspended t-bar, is not really an air barrier. Research has shown that suspended t-bar ceilings are about 10 time more leaky than gypsum board ceilings, because of the cracks on all four sides of the ceiling tiles, and because of light fixtures, speakers, and other penetrations (Cummings et al., 1996). At Outback, there are a considerable number of recessed light fixtures, and each of these is vented to the space above the ceiling.

During the blower door test (installed in the drive-through exterior door), we compared the pressure in the ceiling space to that in the occupied space. When the occupied space was at -50 pascals wrt outdoors, the ceiling space was at -42.9 pascals wrt outdoors. Another way to look at this is that the pressure drop across the roof deck was 42.9 pascals and the pressure drop across the ceiling was 7.1 pascals. Therefore, the ceiling is about 6 times more leaky than the roof deck (by roof deck we mean the combination of roof deck and exterior walls that are above the ceiling). This does not mean that the roof deck and walls above the ceiling are particularly airtight (Figures 33-35). It means rather that the roof deck and walls above the ceiling are tight in comparison to the very leaky ceiling. In fact, there is substantial leakage between the ceiling space and outdoors. Leakage locations have been identified. One large leak pathway exists at the gap between the corrugated roof deck and the structural beams. In some areas, fiberglass insulation batts have been wedged into this space. Fiberglass batts, however, do not produce a good air barrier since air can pass through the fibrous portion. In other areas, the gap is open. There are also large leaks between the dining room ceiling space and the eave space adjacent to the entry.

Figure 33. Ceiling space showing gap at roof deck to wall intersection.

Figure 34. Gap above structural beam

Figure 35. Air leak pathway where metal truss sits on top of structural beams.

Because the building operates under negative pressure and because there are substantial air leaks between the ceiling space and outdoors, humid outdoor air is drawn into the ceiling space. Therefore, the ceiling space is at an elevated relative humidity compared to the occupied space below. The temperature and relative humidity were running about 72 degrees F and 69% in the occupied space and 81 degrees F and 67% in the ceiling space. These numbers mean that the dewpoint temperature in the occupied space was running about 61 degree F while the dewpoint temperature in the ceiling space was 70 degrees F. This elevated ceiling space humidity is contributing to the sagging of the ceiling tiles in the dining area. This problem can be remedied by sealing the leak pathways between the ceiling space and outdoors <u>and</u> operating the building at positive pressure.

Note that positive pressure by itself can greatly reduce ceiling space humidity levels during the hours of the day that the mechanical systems are creating positive pressure. Referring back to Figure 5, note that during the nighttime hours when the EA was turned off and the building went to positive pressure, the dewpoint in the ceiling space dropped markedly from about 71 to 62 degrees F. However, during periods when the AC systems cycle off (mostly during the night), the leaky building shell will allow humid outdoor air to enter the ceiling space. Also, if the building is quite leaky, then the amount of air required to produce significant positive pressure would be much greater than it would otherwise be. For example, to produce +2 pascals of pressure in the building would require that incoming air (MA + OA) would have to be 1542 cfm greater than EA. If the building were tightened to say 5 ACH50 (from current 14.1 ACH50), then the required incoming air to produce +2 pascals would only be 547 cfm, or about 1000 cfm less. The energy requirements to cool 1000 cfm for 18 hours/day for say 300 days per year would be about $1500 per year. Additionally, it will be easier to control indoor humidity to within the optimum range of 50%

to 55% with less ventilation, and it will be easier to maintain comfortable temperature conditions on design cooling days.

Building thermal boundary

Above the suspended ceiling, insulation batts are partially held in place against the plywood exterior sheathing and the beams by duct tape (Figure 36). In some areas, the batts are falling off (Figure 37). The roof thermal system has some deficiencies. The white roof is an excellent feature, helping to shed much of the solar radiation. However, portions of the roof are stained by grease and dirt and are therefore considerably hotter than they otherwise would be. Cleaning will make the roof more energy efficient. The insulation in the roof system is apparently only 1 inch in thickness, so considerable heat can pass through. Also, air can flow freely in the corrugated grooves just below the insulation and just above the roof deck, and in some cases the hot air in these grooves is drawn in through openings around roof curbs by the building depressurization.

Figure 36. Wall insulation above ceiling

Figure 37. Insulation fallen from beam.

3. The Air Conditioning Systems

On Tuesday July 20, testing was done on two air conditioning systems; RTU-1 and RTU-4.

RTU-1 was supplying 61 degree F air with both compressors operating when outdoors temperature was in excess of 95 degrees F, and this 61 degree F supply temperature is acceptable. Air flow was about 600 cfm below design value, but this should not be considered a major problem. One of the two compressors (second stage), however, had abnormally high superheat. Normal superheat is in the range of 10 - 14 degrees F, but in this case it was 30 degrees F.

RTU-4 air flow was about 800 cfm below design value. This is somewhat more of a concern, but still not a major problem. In fact the lower flow rate will tend to cause improved dehumidification of room air. Supply air temperature was 68 degrees F with both stages operating and with outdoors only about 85 degrees F, and this indicates a significant problem. Refrigerant levels were not far out of line but superheat was 22 degree F, somewhat over the 10 to 14 degrees F normal range. The high superheat value does not, however, account for the extreme capacity shortage and therefore further investigation is warranted.

Controls for RTU-4 also have problems. Experiments were performed to evaluate thermostat control. We found that the second stage of cooling did not come on until the room air was 11 degrees above the setpoint for stage 2 (it normally should be about 2 degrees F). This experiment was repeated and the second time second stage came on 6 degrees above the setpoint for stage 2. We waited five minutes between the experiments and adjusted the thermostat setpoints very slowly (less than 1 degree per minute) to make sure that any anti-cycling timers would not influence the tests. The problem with this control malfunction is that the second stage compressor is not turning on until the room reaches 6 to 11 degrees above the thermostat (second stage) setpoint. Since the first stage thermostat control operates at about 74 degrees, and if we assume that the second stage setpoint is 76 degrees, then the second stage compressor would not turn on until room air reaches somewhere in the range of 82 to 87 degrees F. This second stage control anomaly may be partly responsible for the rapid rise in kitchen temperature in the morning.

4. Retrofit Recommendations

1. Building air flows:

While total exhaust air flow is close to design and total OA is only slightly less than design, MA is 41% below design (4980 cfm measured versus 8420 cfm design). We recommend reducing EA flows as much as possible while still maintaining effluent capture and increasing MA to the level required to create slight positive pressure in the building. Testing would be required to determine the optimal level of EA and MA to achieve both effective capture of cooking effluents and minimization of energy consumption. We recommend that MA be rerouted so that it is delivered outside of the hood in order to optimize the ability of the MA to capture cooking effluents on its journey into the exhaust hood. Use of theatrical smoke generation is recommended to optimize the MA discharge pattern.

We suggest that outdoor air for kitchen RTUs be closed off because there are relatively few persons in the kitchen (and therefore less ventilation requirement) and because ventilation air in effect flows from the dining area to the kitchen. Additionally, some of the MA escapes from the hood and provides ventilation. Closing off these outdoor air intakes would increase the cooling output of the kitchen air conditioning systems and improve the ability of these cooling systems to maintain comfort conditions in the kitchen.

2. Building airtightness:

We suggest that the major leak pathways from outdoors into the ceiling space of the building be sealed. One successful approach is application of a spray foam insulation material called Icynene. This product both airtightens and insulates at the same time. Icynene could be applied in a 3 inch or 5 inch thick layer (with embedded hardware cloth as rodent barrier) at the top of beam, truss mounts, curbs, etc. to stop air flow. Additionally, this would have the effect of stopping thermal bridges. The end result would be a more airtight building and reduced heat gain into the ceiling space. Note that heat and humidity that enters into the ceiling space also flows into the occupied space because of space depressurization. Reducing this air flow would also reduce cooling costs, lower indoor humidity, improve indoor comfort for staff and customers, eliminate condensation drippage onto ceiling tiles from cold ducts, and yield longer life for the ceiling tiles.

3. Service AC systems:

Perform AC performance tests on the remaining three RTUs including controls. Service units with identified deficiencies. We suggest that FSEC staff visit the site with the service company to explain our findings and assist with measurements and diagnostics as necessary. FSEC's cost for this would be roughly $50/hr-person plus travel expenses.

4. Improve roof thermal performance:

Clean grease, charcoal, and dirt from the roof on a regular maintenance schedule, perhaps once each spring. Keeping the roof white will reduce heat flux into the building and perhaps reduce roof degradation. When the roof requires replacement, consider installation of more insulation. Our reading of the plans indicate 1 inch of insulation board under the roof membrane. Increasing insulation would reduce total building cooling load and lead to improved comfort conditions for customers and staff.

5. Improved insulation of walls above ceiling:

Currently many insulation batts are held in place by duct tape which generally loses its adhesion over time and fails. As a result, insulation batts will fall and some have already fallen. Vertical insulation above ceiling should be fastened in place with durable fasteners. Alternatively, covering these insulation batts with Icynene foam insulation would secure them in place and create a very effective thermal barrier.

6. Kitchen comfort:

In addition to closing down the OA to the kitchen RTUs, we suggest that adjustable supply diffusers be installed at strategic locations where workers are likely to spend considerable time. If these diffusers are user adjustable, perhaps like air diffusers on air planes, then the direct flow of cool air onto the employees plus the sense of control over their environment may yield significant perception of improved comfort.

5. Recommendations for Future Activities

We recognize that Outback Steakhouse Inc. will be constructing 200+ buildings over the next few years. Therefore, we think it will be in your financial best interest to take steps to optimize these buildings that will be with you for many years to come. This initial report has been provided by the State of Florida. We think, however, there are additional steps that would be beneficial to Outback Steakhouse. The Florida Solar Energy Center would like to provide some or all of the following services to Outback:

1) *Retrofit of Kirkman Road restaurant.* FSEC would assist Outback Steakhouse staff in planning for and arranging a variety of retrofits intended to control humidity, reduce cooling loads, reduce utility costs, and improve comfort. FSEC would assist Outback Steakhouse staff in oversight of the retrofits. FSEC would provide testing of the building and HVAC systems after the retrofits to assess their effectiveness. FSEC would monitor building energy consumption, HVAC system performance, and indoor conditions for a period prior to and a period after the

retrofits in order to characterize the benefits of the retrofits. The information gained from these retrofits and monitoring can be extrapolated to other stores in this region, so that the results of these investigations will be multiplied times a number of buildings.

2. *Prototype design optimization.* FSEC would provide design assistance in optimizing the construction details of the Outback Steakhouse prototype buildings. FSEC would make recommendations for enhancements to the current design in order to optimize the life-cycle durability, functionality, operating energy costs, and initial cost of the facilities, with regional climatic variations taken into consideration. FSEC has advanced building modeling capabilities, including uncontrolled air flow issues. We propose to model the Outback Steakhouse prototype building as is, and then with a range of retrofit options. Modeling will indicate reductions in cooling loads, uncontrolled ventilation rates, indoor relative humidity, and indoor temperatures. The largest savings that result may be in terms of improved comfort for customers and staff, followed by reduced building materials degradation (such as ceiling tiles), elimination of pipe freezing in colder climates without the use of auxiliary electric heat, possible reduced first-cost as a more efficient building envelope and optimized HVAC system operation can lead to reduced cooling loads and downsized cooling equipment, and lastly energy savings. This modeling analysis would yield a prioritization of and recommended package of improvements.

3. *Design, construction, and commissioning manual.* FSEC proposes to develop a design, construction, and commissioning manual for Outback Steakhouse that would incorporate the optimized prototype design elements from step 2 (above) plus step-by-step procedures for optimizing the design, construction, and acceptance phases of the facility development. This document would serve as the "blueprint" by which new facilities are developed, constructed, brought on-line, and operated.

4. *Testing and consulting.* Further testing and consulting regarding other stores around the country as need arises. Testing and consulting services would be provided on an as-needed basis.

References

Cummings, J.B., C.R. Withers, N. Moyer, P. Fairey, and B. McKendry. 1996. "Uncontrolled Air Flow in Non-Residential Buildings; Final Report", FSEC-CR-878-96, Florida Solar Energy Center, Cocoa, FL, April, 1996.

9 - 15. SEVEN PORTABLE CLASSROOMS AT TWO SCHOOLS

Testing was performed on a total of 7 portable classroom buildings located at two central Florida middle schools. Tests were performed at Kennedy middle school (four portables) on April 6, 1998 and at McNair middle school (three portables) on April 7, 1998.

The following tests were performed:

- blower door test – multi-point
- building air boundary test
- air flow rates of supply, return, and outdoor air (OA)
- AC performance test (supply and return air enthalpy)

- duct system airtightness
- infiltration rate: TG decay – w/OA open
- infiltration rate: TG decay – w/OA sealed
- infiltration rate: TG decay – HVAC off
- return leak fraction (RLF)
- outdoor air fraction (OAF)

The air flow and airtightness test results are presented below.

Floor area: Most of the portable classrooms that were tested had exterior dimensions of about 20' x 36' and ceiling heights of 7.5'-8' above finished floor. One portable class was a "single wide" unit and had dimensions of 13' x 50' with a ceiling 8' above finished floor.

Building Airtightness

Blower door tests were performed to determine the airtightness of the building envelope (Tables 17 and 18). On average ACH50 was 22.5 indicating very leaky structures. One building, which had a rigid panel ceiling, was quite tight with 6.8 ACH50.

Table 17
Building airtightness test results

Building	CFM50 (OA sealed)	n	C	r
School #1, Portable "B"	2080	0.603	196.6	0.9940
School #1, "Single wide"	588	0.726	34.42	0.9963
School #1, Portable "D"	1945	0.498	277.1	0.9800
School #1, Portable "E"	4217	0.742	231.6	0.9893
School #2, Portable "#1"	1623	0.698	105.7	0.9983
School #2, Portable "#6"	1871	0.687	127.2	0.9960
School #2, Portable "#7"	2680	0.614	243.1	0.9860
average	**2143**	**0.653**	**173.7**	**0.9914**

Table 18
CFM50, ACH50, and zonal pressures in seven portable classrooms with space at -50 Pa

Building	Floor area (ft^2)	CFM50 (OA sealed)	Attic to building. pressure difference with bldg. at -50 pa wrt outside (pascals)	Attic to outside pressure difference with bldg. at -50 pa wrt outside (pascals)	ACH50
B	706	2080	28	-22	20.6
single wide	650	588	49	-1	6.8
D	756	1945	27	-23	19.3
E	756	4217	33	-17	44.9
#1	720	1623	40	-10	16.9
#6	720	1871	23	-27	20.8
#7	720	2680	17	-35	27.9
average	**718**	**2143**	**31.0**	**-19.3**	**22.5**

Building air boundary

Tests were performed to identify the primary building air boundary. Two classrooms had rigid board ceilings; these are School #1, "single wide" and School #2, Portable #1, and these units were more airtight than the others. Suspended t-bar ceilings are quite leaky. Five of the seven portables had suspended t-bar ceilings. In general testing has found that this ceiling construction has approximately 5 CFM50 per square foot of ceiling area. Pressure drops across the ceiling indicate the airtightness of the ceiling relative to the ventilation openings of the attic. The two buildings with rigid ceilings showed pressure drops of 49 and 40 pascals across the ceiling, and only 1 pascal and 10 pascals across the roof deck. Thus the ceiling is the primary air boundary of the building. The five with suspended ceilings had pressure drops of 17 to 33 pascals across the ceiling with an average of 25.6 pascals. Therefore, in these five buildings, the ceiling provides approximately the same resistance to air flow as the vented roof (roof venting included eave and ridge vents).

Duct Airtightness Testing

The airtightness of the air distribution system was tested in each building (Table 19). The package AC units hang on exterior walls. Five of the seven portables had essentially no ductwork. In these, a single return grill and a single supply register transfer air directly to and from each air handler. Therefore, essentially all of the "duct leakage" exists in the air handler and the connections of the air handler to the building in these five buildings. Two of the classroom buildings had supply ducts and multiple diffusers and a single through-the-wall return.

Considering the absence of ductwork, the amount of duct leakage is substantial. (Leakage is not split between return and supply.) The return leaks, however, do not represent a significant problem since they draw directly from outdoors (and not a hot attic or musty/moldy crawl space) and therefore act like outdoor air (ventilation air). The supply leaks should be repaired. Note that the CFM25 duct leakage numbers were obtained with the outdoor air intakes sealed off so the outdoor air intake does not show up as a duct leak.

Table 19
Duct airtightness test results in seven portable classroom buildings.

Building	CFM25 total	n	C	r	CFM25 out
B	80	0.76	6.85	0.9114	66
Single wide *	148	0.64	18.68	0.9896	129
D	102	0.82	7.29	0.9255	79
E	157	0.64	19.98	0.9982	78
#1	157	0.67	18.24	0.9948	132
#6	167	0.74	9.30	0.9868	87
#7**	376	0.44	90.04	0.8633	266
average	**170**	**0.67**	**24.34**	**0.9528**	**120**

* Contains supply duct system in attic with four supply registers
** Contains supply duct system in attic with six supply register
All other portables have single supply register discharging directly from AHU and no duct system.
CFM25 total = air leakage into ductwork when depressurized 25pa with respect to the conditioned space.
CFM25 out = air leakage into the ductwork from outside the exterior air boundary of the building when ductwork at -25pa with respect to conditioned space.

Building Ventilation Rates

Tracer gas testing was done to measure the classroom ventilation rates in normal operating mode (Table 20). Ventilation rates with the air handler on and the outdoor air open average 1.8 air changes per hour (ach), and ranged from 1.3 to 2.5 ach. Given an average volume of 5744 cubic feet, average ventilation is about 170 cfm. Given an average occupancy of 27 persons (26 students and one teacher), this yields 6.3 cfm per person. Even if an occupancy factor of 0.77 is used (as was the case with Polk county school district), ventilation is nearly 50% lower than required by ASHRAE 62-1989 (in force at time of testing). With the outdoor air sealed, ventilation averaged 1.2 ach and ranged from 1.0 to 1.7 ach.

Table 20

Ventilation rates in seven portable classroom buildings expressed as air changes per hour (ach) and cubic feet per minute (cfm) with the air-conditioner "off", with the air-conditioner "on with OA open", and with the air-conditioner "on with OA sealed"

Building	ach off with OA open	ach on with OA open	ach on with OA sealed	cfm off with OA open	cfm on with OA open	cfm on with OA sealed	occupancy (# of desks)
B	n/a	2.26	1.34	n/a	217	129	28
single wide	n/a	1.45	1.10	n/a	126	96	13
D	n/a	1.47	1.04	n/a	141	99	35
E	n/a	2.51	1.17	n/a	240	113	30
#1	0.59	1.57	1.26	57	151	121	25
#6	0.49	1.27	1.01	47	122	97	26*
#7	1.03	1.85	1.74	99	178	167	26*
average		**1.77**	**1.23**		**168**	**117**	**26.1**
* Number of desks estimated.							

Tracer gas tests were also performed to determine the fraction of air entering the air handler that originates from outdoors. This is done by comparing the concentration of tracer gas in the room air as it enters the return grill to the concentration of tracer gas in the supply air. Equation 1 is used to calculate the fraction coming from outdoors.

$$\text{OA fraction} = (A\text{-}B/A\text{-}C) \qquad \text{Equation 1}$$

where A is the tracer gas concentration at the return grill (ppm)
B is the tracer gas concentration at a supply (ppm)
C is the tracer gas concentration in the air entering from outdoors (usually zero ppm)

The test is performed once with the air conditioning system operating normally. Then the test is repeated with the OA intake masked off. The first. The test gives the fraction of air entering the air handler that originates from outdoors by means of both outdoor air and return leakage. The second test gives the fraction of air entering the air handler that originates from outdoors only by means of return leakage. Subtracting the results from the second test from the first test yields the fraction entering the air handler by only the OA intake. These results are summarized in Table 21.

Table 21
Outdoor air (OA) and return leakage (RL) expressed as a percentage of total airflow and expressed as cfm. Cfm is calculated based on OA and RL percentages.

Building	RL + OA	RL	OA	RL +OA cfm	RL cfm	OA cfm
B	13.1%	2.3%	10.8%	152	27	125
single wide	8.3%	3.9%	4.4%	73	34	39
D	11.5%	3.3%	8.2%	114	33	81
E	21.8%	na	na	241	na	na
#1	12.9%	8.1%	4.8%	115	72	43
#6	7.3%	7.1%	0.2%	61*	59*	2*
#7	18.9%	13.7%	5.2%	167	121	46
average	**13.4%**	**na**	**na**	**132**	**na**	**na**
* airflow calculation based on assumed (not measured) total airflow rate of 834 cfm						

Visual inspection found that in most cases the OA dampers are partially or completely closed. As a consequence, OA fractions ranged from a low of 0.2% to 10.8% of air handler flow. Opening of the OA dampers would increase classroom ventilation rates to levels more in line with the ASHRAE ventilation standard.

Units 1, 6, and 7 were tested on the same day. Note that unit 7 has much higher air change rates with the outdoor air masked off. Two factors may explain the difference. First, of the three systems tested at school #2, unit 7 is the only one with an air conditioner mounted on the east wall. Winds of about 8 to 15 mph were coming out of the east during the time of testing. Wind may have been able to drive more air into the building through leaks in the a/c system even with the AC system cycled off. Duct leaks are another significant difference that may help explain the higher air exchange rates, since unit 7 had by far the most leaky ductwork.

16. BAR AND GRILL

This building was originally tested in 1994 during a prior research project. Duct repair retrofits were performed in 1995 and energy savings were observed, as indicated in the following *italicized* paragraphs from a 1996 report (Cummings et al., April 1996).

Two four-ton air conditioners serve this rather leaky (17.5 ACH50) 2400 square foot restaurant/bar. The air handlers and ductwork are located in the attic space. The ductwork was very leaky; 655 CFM25. Large return leaks (505 cfm) were drawing hot attic air into the building. There were substantial supply leaks as well, but return leaks dominated. A kitchen exhaust fan pulled 987 cfm of air from the building throughout most of the day, and there was no make-up air. The building was depressurized to -0.8 pascals when both exhaust and air handlers were operating.

Based on the size of the duct leaks and the fact that they were drawing in hot attic air, cooling energy savings of 30% or more could have been expected. To our surprise, monitored savings were only 10.6%. In hind-sight two problems were identified. First, only 58% of the duct leaks were sealed. Second, we had mis-diagnosed the building UAF problems, or more accurately we only got it half right. We believed that sealing the large return leaks would reduce the amount of hot attic

being brought into the building. It turned out that for the most part this was not true because the exhaust fan would continue to draw nearly 987 cfm of hot attic air into the building for 8 to 10 hours per day. (Most of the air would come from the attic since nearly all of the shell leakage of this slab-on-grade, concrete block building existed in the ceiling.) As a result of duct repair, building pressure went from -0.8 pascals to -2.0 pascals. After repair, less attic air was being drawn into return leaks, but more attic air was being pulled into the building by space depressurization.

We believe that substantially greater energy savings could have been realized if make-up air were installed in the kitchen. Alternatively, the large leak paths between the attic and occupied space (above the bar) could have been sealed. This would have reduced the amount of air drawn from the hot attic and increased the amount drawn from the relatively cooler outdoors. To make this second approach even more effective, a passive make-up air vent could be installed in the kitchen, in the proximity of the cooking area and the exhaust fan. This would minimize the impact of the make-up air by causing it to "short circuit" -- go almost directly from the make-up grill into the exhaust intake. This case illustrates how important correct diagnosis is and the importance of taking all uncontrolled air flows into account when specifying repairs.

As a result of these repairs, duct CFM25 decreased 59% from 655 to 272. Relative humidity in the occupied space declined from an average 63% to 57%. Cooling energy use decreased from an average 142.4 kWh/day to 127.3 kWh/day, or 10.6%. Repair time was 18 person-hours and materials totaled \$75. Energy savings pay for the estimated \$975 repair cost in 4.2 years. Calculations indicate that installation of make-up air could save over \$1000 per year.

An additional reason why energy savings from duct repair was less than expected

Another reason has subsequently been identified explaining why energy savings from duct repair were less than anticipated, and this has to do with the fact that the AC systems <u>cannot</u> meet the cooling load during peak hours even after the duct repair. Therefore, the reduction in cooling energy savings occurred only during non-peak hours -- that is, in the morning and late evening periods. Furthermore, during the coolest four months of the year (November through March), when the AC systems would be able to meet cooling load and therefore there would be more substantial energy savings, the owner ventilates almost exclusively during this period.

There are three principal reasons why the cooling load exceeds cooling capacity, even after the duct repair retrofits were implemented. First, there are large internal heat sources related to cooking and refrigeration/freezer units. Second, the exhaust system operates without make-up air so approximately 1000 cfm of air is drawn into the restaurant for about 8 to 10 hours per day, and this air comes largely from the hot and humid attic space. Third, the cooling capacity is only 3.3 tons per 1000 square feet of floor area or about half the average for other restaurants.

Field testing

Blower door testing at the beginning of this project found that the building airtightness had remained essentially unchanged -- 17.4 ACH50 in 1997 versus 17.5 ACH50 in 1995. Duct leakage, however, had increased substantially since 1995. Duct repairs performed in 1995 reduced CFM25 from 675 to 272. When we tested again in 1997, this had increased to 587 CFM25. One major new leak was found. *An air conditioning contractor had apparently cut a 10" x 10" opening into the return duct* *of one of the AC systems to gain access for coil cleaning and had closed it with duct tape, and this closure had failed.* We estimate approximately 30 square inches of opening where the flap was being sucked into the return duct. This duct leak site was repaired prior to beginning monitoring in this project and was therefore not considered one of the project retrofits.

The building was operating at about -2.0 pascals as a result of the 954 cfm kitchen exhaust fan. There was no make-up air.

Retrofit

Based on this initial testing, three retrofits were identified that make sense:

- addition of unconditioned make-up air to the kitchen
- sealing of the leaky ceiling
- adding insulation where portions of the ceiling insulation was missing

Following is a retrofit and monitoring plan that was submitted to the owner of the bar and grill. The owner subsequently authorized our implementation of the make-up air and spray foam insulation retrofits and the monitoring.

Retrofit/Monitoring Plan Submitted to Owner

Background

Testing was performed on this building in 1994. Repair of duct leakage was done in 1995, reducing return leakage (from attic) from about 20% to about 3%. Unbalanced exhaust air was not fixed. A kitchen exhaust fan operates for about 8 hours per day, Monday through Saturday, drawing 987 cfm from the kitchen. There is no make-up air; consequently the building is depressurized to -2.0 pascals during exhaust fan operation. The energy impacts of this unbalanced air are increased by the fact that the largest air leakage pathways are from the attic space to the bar area. In addition, there are significant energy penalties associated with insulation problems.

Proposed retrofit

We propose that make-up air equal to 80% of exhaust air be installed in the kitchen, to operate on the same control as the exhaust air. This could be added in the next 3 to 4 weeks, and then "flip flop" experiments could be run -- alternating, one week without make-up air followed by one week with make-up air.

We propose to airtighten the ceiling near the bar (and other areas if indicated), fix insulation that has fallen in the attic, and add insulation at horizontal and vertical surfaces separating the attic from the dining area near the bar.

We propose that no outdoor air be installed. Since the building leakage to the attic will be diminished, the passive make-up air will come primarily from outdoors. The advantage of not providing make-up air by means of outdoor air is that this ventilation will therefore not occur when the exhaust fan is not operating.

We propose that the building whole house fan be put under control of a CO_2 controller, so that it will cycle on at say 1000 ppm and shut off at 800 ppm. We believe this will be a cost-effective means to maintain adequate ventilation because high occupancy is intermittent and represents only a small fraction of the time (perhaps 10% or less of time). By contrast, outdoor air would be provided whenever the air conditioners operate, which is much of the time, and therefore would consume much more energy.

Monitoring

We propose that a datalogger with multiplexer be installed with various probes to measure energy, temperature, humidity, pressure, and carbon dioxide levels for a period of approximately 12 months. The following parameters will be monitored:

energy	* building (1 meter)
	* east AC unit (1 meter)
	* west AC unit (1 meter)
	* exhaust air status (relay or meter)
	* status of whole house exhaust fan (relay)
pressure	* building pressure wrt outdoors
carbon dioxide	* CO_2 in the dining area
temperatures	* in kitchen (cooking)
	* in kitchen (food preparation)
	* in dining room near bar
	* in dining room near TV
	* in attic near bar
	* in attic near air handlers
	* return air location between two return grills (run time)
	* supply air for system 1 (run time)
	* supply air for system 2 (run time)
	* exhaust fan (run time)
relative humidity	* in kitchen
	* in dining room
	* in attic
outdoors	* temperature
	* relative humidity
	* top of ridge shingle (solar)

Two retrofits were implemented; 1) addition of make-up air and 2) airtightening and improving insulation by means of a spray foam application. Cooling energy use, indoor temperature, attic temperature, indoor relative humidity, attic relative humidity, building pressure, and indoor carbon dioxide were monitored to detect changes resulting from the retrofits.

In the first retrofit, exhaust air flow was increased and make-up air was installed. Exhaust air flow was increased to 1135 cfm. After installation, make-up air was set at 933 cfm or 82% of exhaust air flow. Getting the make-up air system installed and adjusting the air flow to the desired rate was no trivial task. Installation began in October. Installation was complete in January. Our request for bid contained the following specifications:

"MAKE-UP AIR FAN: capable of delivering 1000 cfm or more of unconditioned air from outdoors to the kitchen, taking into account the resistence of the filter, ductwork, and diffuser. It will have the capability (by means of damper or similar) of adjusting air flow to 700 cfm or less. Fan shall be selected and installed to achieve low noise operation, such that operation of the make-up air system will not be objectionable to kitchen staff or clientele. This could include "low noise" fan, vibration isolators, etc."

The contractor initially installed the make-up air unit with a ½ HP motor, 1725 rpm motor with 2400 cfm capacity. We found that the air flow rate was much too great. We requested that a smaller motor be installed. As a final adjustment, he then installed a 1/3 HP motor, also with 1725 rpm. Testing found that the air flow rate had not changed. He then installed a 1/3 hp motor with 1140 rpm. Even with the reduced motor capacity, we had to block 70% of the make-up air intake grills in order to reduce flow to 933 cfm.

MA duct airtightness was tested. Leakage was found to be 122 CFM25.

There was no outdoor air on the AC units. As a result of installing make-up air, normal building operating pressure went from -2.0 to -0.2 pascals. Note that using the building airtightness curve ($Q = 645.6\ (dP)^{0.59}$), we find that -2.0 pascals is equivalent to 972 cfm of net exhaust and -0.2 pascals is equivalent to 250 cfm. 972 cfm is close to the measured exhaust flow rate and 250 cfm is close to the difference between exhaust and make-up air flows.

In the second retrofit, a spray foam insulation material was applied to the bottom of the roof deck and to attic vent openings. This foam was applied at an estimated average thickness of 4.5 inches over the bottom of the plywood roof deck material, over eave vents, and over gable vents. The end result was to make the building more airtight, move the primary air boundary to the roof deck, and improve the insulation system. This was deemed to be the best solution because airtightening of the soffitted area above the bar would be very difficult, in part because of recessed lights and because many of the insulation batts suspended from the trusses had been knocked down by people moving about in the attic and we assumed that this would likely happen again.

A building air tightness test was conducted September 8, 1999 after the spray foam retrofit.

	CFM50	n	C	r	ACH50	dP, all HVAC on
Before Icynene	6496	0.58	684.0	0.9978	17.1	-0.2
After Icynene	5120	0.58	524.02	0.9989	13.5	-0.3
difference	1376	---	---	---	3.6	-0.1

The building became substantially more airtight; CFM50 declined by 21% as a result of the spray foam retrofit.

Following is the retrofit and monitoring plan we submitted to the owner of the bar and grill.

Subsequently, the retrofits recommended in this proposal (except for the CO_2 control of the building exhaust fan) and monitoring were implemented. Discussion of the energy and humidity impacts resulting from the retrofits are presented Appendix C.

17. CONVENIENCE STORE WITH FAST-FOOD SERVICES

Testing was performed at this site on May 5, 1998.

The building floor area is 4,320 square feet. Building volume is 41,904 cubic feet. General construction is slab-on-grade, block wall construction, suspended t-bar ceiling over approximately 60% of the floor area, rigid gypsum board ceiling over about 20% of the floor area, no ceiling above the approximately 20% of the floor area covered by walk-in coolers, insulation batts on the ceiling, sloped plywood roof decking, and light gray colored metal roof.

Building airtightness: a blower door test found 10,776 CFM50 (this indicates that 10,776 cfm of air are drawn into the building when it is depressurized by 50 pascals). This may also be expressed as 15.4 ACH50 which indicates that the building is quite leaky. Most of the leakage is in the ceiling, since suspended t-bar ceilings are inherently leaky. The attic space is subdivided into an east zone and a west zone by a fire wall. When the building was at -50 pascals (with respect to outdoors), the east ceiling space was at -21 pascals with respect to outdoors, indicating that the ceiling is the primary air boundary (59% of the pressure drop is across the ceiling and 41% is across the roof deck). We did not measure the pressure in the west ceiling space. However, based on previous measurements collected from this site in 1996, we project that the west ceiling space would be at approximately -26 pascals when the occupied space was at -50 pascals. Therefore, the ceiling and roof deck are approximately equal as air boundaries. The fact that a vented roof is approximately as tight as the ceiling indicates that the ceiling is very leaky.

Kitchen exhaust hood air flow is low. When we previously tested the building in 1996, the exhaust air flow rate was 1546 cfm. In May 1998, we found only 656 cfm. How much exhaust air is required? There are two general criteria for determining necessary air flow. First, there is the general requirement of approximately 200 cfm per lineal foot of hood (depends on hood type, operating temperature and types of appliances being exhausted, types of contaminants being generated: steam, heat, grease). The exhaust hood is 5.5 feet in length, so the flow requirement would be around 1,100 cfm. Second, there is a requirement for a minimum of 1500 feet per minute (fpm) air velocity in the exhaust ductwork for grease-laden exhaust air. Given the dimensions of the existing exhaust duct are 10" x 10", the air flow rate through this duct should be 1042 cfm or greater.

Tests were done to measure the **overall ventilation rate of the building**. With all mechanical systems operating in their normal mode, the building ventilation rate was 1.0 ach. This can also be converted to a ventilation rate of 694 cfm. This is consistent with the air flow rate of the dominant building air flow driver, namely the kitchen exhaust fan which had a measured flow rate of 656 cfm. The building ventilation rate was measured at 1.90 ach in 1996 which equates to 1327 cfm, and this roughly correlates with the 1546 cfm of exhaust fan flow which was measured at that time.

The **air handler** (AH) for the south AC unit runs even when the unit is shut off. A contractor should be contacted to correct this problem. (This problem was subsequently corrected.) Having the air handler running continuously has several negative consequences, including: 1) the air handler fan will wear out and/or become dirty (and move less air) more quickly, 2) the fan will use more energy, 3) the ductwork will pick up additional heat (by conduction) from the hot attic and move it into the occupied space, 4) duct leaks will draw more outdoor and attic air into the building, and 5) the air conditioning system will not dehumidify the building as well.

Outdoor air (air drawn into the return side of the air handling system to provide ventilation) has been provided to the two air conditioning systems. However, there is very little outdoor air flow. By design, outdoor air should be 650 cfm. Measurements found outdoor air flow rates of 141 cfm on the 10-ton system and only 13 cfm on the 7.5-ton system. Tracer gas tests also found return leaks of 26 cfm and 31 cfm, respectively. The outdoor air ducts appear to be severely undersized, and the outdoor air filters are "hidden" behind outdoor intake grills under the east eave and are very dirty. The store owner was not aware of the outdoor air filters and the need to clean these filters on a regular basis.

Air handler flow rates are low. The 10-ton system which serves the south side of the store has a measured air flow rate of about 2700 cfm, or about 32% below the standard air flow rate for this unit. The 7.5-ton system which serves the north side of the store has a measured air flow rate of about 2000 cfm, or about 33% below the standard air flow for this unit.

The air conditioners fall far short of rated capacity. The cooling output of each AC unit was tested. Outdoor conditions were not severe, only 82°F. The 10-ton system which serves the south side of the store had a cooling output of 64,100 Btu/hr, or about 53% of the rated capacity. The 7.5-ton system which serves the north side of the store had a cooling output of 34,700 Btu/hr, or about 39% of the rated capacity. The rating conditions for AC capacity and efficiency are 95°F outdoors, 80°F/50% RH indoors and air flow of about 400 cfm/ton. Manufacturer's performance data indicates that the total delivered capacity, at the <u>actual</u> temperature and humidity conditions measured during this site visit and with airflow of 400 cfm/ton, would be roughly the same as the rated capacity. Therefore, the substantial shortfall of capacity is probably due in part to the severely reduced air flow, and low refrigerant charge may also be an issue.

Attic insulation is missing over approximately 50% of the ceiling surface area, considerably increasing heat entry into the building.

Temperatures were measured in various locations about the building using a spot radiometer which measures the surface temperature of materials:

roof surface	117°F
bottom of roof deck	109°F
ceiling in retail area	78°F
ceiling in kitchen	90°F
floor	76°F
south window	82°F
west window (shaded)	87°F

Duct leakage was not measured during the most recent testing. However, tests from 1996 found 479 CFM25 in the north system and 867 CFM25 in the south system. Specifically, these numbers mean that when the ductwork is depressurized by 25 by pascals, a total of 1346 cfm of air is drawn in through leak sites. This means that the ductwork is significantly leaky.

Discussion:

There are a number of problems at this store.

The kitchen exhaust fan draws 656 cfm of air from the building. This depressurizes the building which in turn draws air into the building primarily through the ceiling (the leakiest building plane) from the attic. This hot and humid attic air adds substantially to the cooling load. If the exhaust flow rate is increased to 1050 cfm to meet the minimum duct velocity of 1500 fpm, then the cooling load from attic air will increase substantially.

A number of additional factors are working to increase utility costs and reduce indoor comfort. 1) The air conditioning systems are providing only 50% or less of their rated cooling capacity. 2) Insulation is missing over 50% of the ceiling area, adding to the cooling load. 3) The ductwork is significantly leaky and therefore losing a substantial portion of the available cooling capacity.

Lighting and equipment audit, and HVAC performance

Following are results from audit of lighting and HVAC equipment and systems.

Audit Date: 5/6/98

Sales Area Equipment:

Soda Display Case:	True Manufacturing Mod: GDM-45 2 sliding glass doors, ~33°F O'Fallon, MO 63366 Comp: 115V, ½ hp, 1Ph, 8.9 FLA, 312Dis/140Suc, R-134A, 16 oz. Refrig. Unit AK4476Y, Serial# 11510073 Heat rejected to space below unit.
Soda Display Case:	True Manufacturing Mod: GDM-45PE5182 2 sliding glass doors, ~33°F O'Fallon, MO 63366 Comp: 115V, ½ hp, 1Ph, 8.9 FLA, 312Dis/140Suc, R-134A, 17 oz. Refrig. Unit AK4460Y, Serial# 11805354 Heat rejected to space below unit.
Frozen Food Case:	Pinnacle Mod: URF3-30 Ser: 75349 3 glass doors, -20°F Fleetwood, PA Mfd: 88D Lights: 4.6A Anti-Sweat: 678 watts Refrig: R-502 250Dis / 150Suc Fans/Defrost: 230V/6.5A Ceiling mount sensor, front of case: SUPCO TA-2 Temp alarm Freezer Zone 5°F
Ice Display Case:	Leer 2-door display, condenser heat rejected to space above unit.
Reach in GD Display:	Ardco Model SX-WI Scan X Energy Control System 3, 5-door displays (GD = Glass Door) Chicago, IL 115V/15A Model ASK-5, Serial 1212Q04, B/M# 27259G015
Northeast GD Display:	Lights 1.92A, Heater 3.5A, 120V (displays at northeast corner of sales area)
Middle:	Lights 1.26A, Heater 3.5A, 120V
Northwest GD Display:	Lights 1.92A, Heater 3.5A, 120V
Chicken Display:	Alto Shaam Model DCD-48 188 Serial 4686-71-1 3.4 kW, 120/240-208V BUE-48 3.4 kW

Frigidaire Igloo Cooler: Vestfrost Denmark Model IK403 115V/6A, 7.6 oz R134A, 215 watts, 285Dis/90Suc

Self-contained GD: Beverage-Air Model MT12 5.5 oz R-134A, 250Dis/250Suc 115V/5A/1Ph, 15w lgt

Capuccino Machine: Cecilware Corp. Model GB3K-LD Serial 275401 120V, 1.8 kW

Gemini System 5: Curtis Model GEM-5 coffee machine, 115V/1A, 100 watts, Date 5/88

Gemini System 120A: Curtis Model GEM-120A Serial 11139 coffee machine, 115V/18A, 1975 watts

Connolly Roll-A-Grille: Model C-360 Serial 3395 120V/13A 3-31-95 (hot dog server)

Pastry Display Wilshire Citrus Cooler: 2 ea. Very small drink dispensers

Frosty Beverage Mach: 1-800-962-4407, 6-hr defrost

Med. Temp. Reach-in: Pinnacle Model PDU866, Serial 74023, Mfg 88D (northwest corner)
115/230V, 4 oz. R-12, 250Dis/150Suc, Lgt 2.16A, Evap Fan 1.34A, FLA 5.4A

ATM machine: Tidel, Hanco Systems Inc 1-800-426-7118 (southwest wall)

Copy machine: Sharp Model SF-2114

Front counter: 2 cash registers, 1 lotto machine

Ceiling fans: 3 equally-distributed fans in sales area

Lighting: Fluorescent: sales area 68 ea, 2-light fixtures, 4 ea. recessed can lighting above dining tables, bathroom hallway 1 ea. 2-light fixture, cove fluorescent: 14 ea. 2-light fixtures at perimeter of sales area

Front Kitchen Equipment:

Bunn Coffee Maker: Model H5X-40-208 (212F) Serial HX00010069 208V/19A/4050 watts

Pitco Frialator: Steamer Model RTE14S Serial E97EA01018 208V/38A/8kW/1Ph 196°F

Bevels Cabinet: Hot holding cabinet Model CS-82-CH8 Serial CH-1V-TB-13613 120V/16A/2kW

Traulsen: Model G10010 Serial T18950A96 115V/8A 18oz R-134A 186Dis/88Suc

Taco area: Infra Corp Randell CC101.001 32oz R134-A 120/208/1Ph/3wire 11.3kW heat 54A
Cool 120V/6.5A Steam table 208V/36A/1Ph

Deerfield freezer:

Back Kitchen Equipment:

Tyler Exhaust Hood: Fan belt slightly loose and was adjusted, motor shieve set for max rpm.

Deep Fryer: Henny Penny Corp Model 500 Serial JB246HA Product 02202
208V/32A/3Ph/11.25 kW 4.5A motor

Blodgett Oven: Model CTBR-1 Serial 0886M7803102, fan motor 208/230V/2.5A
208/230 VAC, 24A (L1&L3, 0 Amps on L2)

Amana radar range: Model RCS/710

Stockroom:

Ice machine: Manitowoc Series 1200 ice cube maker (outdoor condenser)

Electric Service Meter and Main Electric Panel:

Building Power Meter:	Meter Factor = 80, Max = 89.6 kW, Current reading 62.4 kW (3:30 pm)
Main Panel:	Siemens FC-1 Series 6 208Y/120 3Ph/4wire 800A 4 panels/42 breakers ea.

Walk-in Freezer Condensers:

Walk-in Freezer: Larkin Model OSH0150L5 Serial 8885184F -40°F to -5°F
Compressor 208-230V/3 Ph, 1 ½ hp, 5.1 RLA 35.5 RLA 400Dis/150Suc R-502
Condenser 208-230V/1 Ph, 2A

Walk-in Cooler: Larkin Model OWH030OH2 Serial 8886684B , 16.3 FLA 0°F to +45°F
Compressor 3 hp, 14.3 RLA 74 IRA R-22
Condenser 1/4 hp, 2 FLA

Walk-in Cooler: Larkin Model CWH030OH2 Serial 8886949A 1, 16.3 FLA 0°F to +45°F
Compressor 3 hp, 14.3 RLA 74 IRA R-22
Condenser 1/4 hp, 2 FLA

Sub-zero Freezer: Larkin Model O6H0200L5 Serial 8886949A -40°F to -5°F
Compressor 208-230V/3 Ph, 2 hp, 6.6 RLA 46 RLA 400Dis/150Suc R-502
Condenser 1/4 hp, 2 FLA

Ice Machine: Condenser only, Model AC1295A 220V/1.2A, Use with Ice Maker series G-1200

Radiometer measurements of various surface temperatures:

Rear sales area floor:	76°F
Inside ceiling tile:	78°F
Space temperature:	76°F
Bottom of roof deck:	109°F
Top of roof deck (e=0.92):	109°F
Top of roof deck (e=0.7):	119°F
(thermocouple):	117°F
Top of cooler (metallic)	95°F
GD display doors?	68°F
South facing windows?	82°F
Bottom of S. facing windows (black)	80°F
West shaded window	87°F
Floor below shaded window (lit by sun)	79°F
Chicken display?	107°F
Kitchen floor near hood	79°F
Front kitchen ceiling	90°F
Back kitchen ceiling (East)	90°F
Back kitchen ceiling (West)	85°F

Air Conditioning equipment:

Indoor Air Handlers:

Indoor Unit #1: Trane Model# BWE120C400GA, Serial# C40170634, Date 10/88
Refrigerant R-22, Motor FLA 6.6/3.3, Volts 200-230, 3Ph, 2 hp
Min. Airflow (heating) 4000 cfm, Max. Airflow (cooling) 4000 cfm, electric strip heat
Thermostat on south wall, setpoints = 72°F cool / 54°F heat, auto fan

Indoor Unit #2:	Trane Model# BWE090C400GA, Serial# C50172730, Date 12/88 Refrigerant R-22, Motor FLA 5.0/2.5, Volts 200-230, 3Ph, 1½ hp Min. Airflow (heating) 3375 cfm, Max. Airflow (cooling) 3375 cfm, electric strip heat Thermostat on front kitchen wall, setpoints ??

Condenser units:

AC Unit #1:	Trane Model# BTA120C300MB, Serial# C23197061, June 1988, 80 amp breaker Compressor 37 RLA, 200-230V/3 Ph/200 IRA, 420Dis/150Suc R-22 2 condenser fans, 3.8 FLA, 200-230 V/1 Ph, Min. Circuit amps = 56
AC Unit #2:	Trane Model# BTA090C300, 60 amp breaker Compressor 27 RLA, 200-230V/3 Ph 400Dis/150Suc R-22 2 condenser fans, 3.8 FLA, 200-230V/1 Ph

AC Performance Measurements:

AC#1:

Power: 3-phase cond. unit: 32.5/35.1/35.2 Amps Avg = 34.27 Amps at 208 VAC
12.35 Kva 208/208/208 VAC

Refrigerant pressures: 265 psig Dis / 63 psig Suc at 83°F ambient

3-phase supply air fan: 3.5/3.8/3.8 Amps Avg = 3.7 Amps at 208 VAC 1.33 kVA
208/208/208 VAC

Total: **13.68 kVA / 10.94 kW (at .8 PF)**

Capacity:	RA grille: 75.5°F (TC)	Enthalpy:	27.59 Btu/lb (at 46% RH)
	73.5°F / 46 % RH (Solomat)	26.49	
RA at unit:	77.4°F (TC)	27.44 (at RA grille W)	
SA at unit:	54°F (TC)	22.11 (at 95% RH)	
Ambient:	82°F		
Airflow:	2569 cfm SA using hood		(265 avg. cfm/ton)
	2732 cfm RA using hood		
	2650 avg cfm using hood Density:	0.0748 lb/cu. ft.	28 cfm OA
Total capacity:	**64.1 kBtu**		

Unit Efficiency (EER): **5.8** at 75.5°F/46%RH inlet, 82°F ambient

Air handler pressure map: Return plenum: - 90 Pa
After coil: - 155 Pa
After fan: + 105 Pa

AC#2:

Power:	3-phase cond. unit	28.0/26.5/24.5 Amps	26.3 Amps at 208 VAC	9.48 kVA
	208/208/208 VAC			
Refrigerant pressures:	210 psig Dis / 72 psig Suc at 83°F ambient			
3-phase supply air fan:	2.6/3.1/3.2 Amps	2.97 Amps at 208 VAC	1.07 kVA	
	208/208/208 VAC			
Total:	**10.55 kVA / 8.44 kW (at 0.8 PF)**			
Capacity:	RA grille: 74.7°F (TC)	Enthalpy:	27.35 Btu/lb (at 47% RH)	
	73.0°F / 47 % RH (Solomat)		26.41	
SA at unit:	56.5°F (TC)		23.61 (at 95% RH)	

Airflow:	2055 cfm SA using hood		(275 cfm/ton)
	1577 cfm RA using hood		
	Leak at NE RA grille at fire damper above filter		
	2055 cfm using SA	Density:	0.0752 lb/cu. ft.
	0 cfm OA		
Total capacity:	**34.7 kBTU**		
Unit Efficiency (EER):	**4.1** at 74.7°F/ 47%RH inlet, 82°F ambient		

Air handler pressure map:

Underside of filter	- 25 Pa
Top of filter	- 80 Pa
Return plenum:	- 142 Pa
After coil:	- 170 Pa
After fan:	+ 48 Pa

RETROFIT RECOMMENDATIONS

The following retrofit recommendations and improvements, with projected dates, were provided to the store owner.

Phase 1: fix AC equipment (late August 1998)
Phase 2: increase exhaust air (EA) flow to 1050 cfm plus add 800 cfm of make-up air (MA) and 300 cfm of outdoor air (OA) through the AC units; alternatively install a smaller exhaust duct, keep EA at current level, add 500 cfm of MA, and add 200 cfm of OA (January 1999)
Phase 3: add insulation batts (Icynene on vertical attic surfaces) and fix duct leakage (June 1999)
Phase 4: remove attic insulation batts, install Icynene insulation at roof deck, and seal attic venting (August 1999)

The owner agreed to have these retrofits performed with the project to pay for their cost. Phase 1 retrofits were implemented in 1998. However, within just a few weeks of these retrofits, the property was sold to a new owner. The new owner agreed to allow the monitoring to continue. However, within a couple weeks of taking over the store, a complete renovation of the building took place. This included changing the metal roof color from light gray to dark green, replacing west windows, replacing the entire suspended ceiling, and changing the ceiling insulation. Some changes were also made to interior wall layout. Based on these dramatic changes to the building, we concluded that we could not go forward with the experiments, because the comparison of before and after energy use would be greatly affected by the changes in the building envelope. As a consequence, we have no long-term monitoring data to report for this building.

18. OCEAN FRONT RESTAURANT

Testing was performed on October 13, 1998.

This building is located on a pier which extends several hundred feet into the Atlantic Ocean. The restaurant has floor area of 5977 square feet and is about 20 feet above the ocean level. Average ceiling height is 10.3 feet yielding a volume of 61,563 cubic feet. The ceiling above dining area is suspended t-bar ceiling with a vented attic space above. The ceiling above the kitchen is gypsum board with vented attic space above.

A total of six air conditioners serve the dining and kitchen areas. Their combined nominal cooling capacity is 36 tons, or just over 6 tons per 1000 square feet of floor area. This is rather high cooling capacity even by restaurant standards. Owners report that the building is sometimes uncomfortable and utility bills are high. Air flow rates at supply registers and return grills were measured using air flow hoods. Supply and return air flows were measured for all six air handlers. These numbers are summarized in Table 22.

Table 22
Air flow rates (cfm) and pressure pan test results (pascals) for six air-conditioning units.
Pressure pan measurements are an indication of the amount of duct leakage

	capacity	supply air flow	return air flow	pressure pan
1. West dining	7.5 ton	2158	1537	5.9
2. East kitchen	7.5 ton	NA	~ 1300	NA[2]
3. Kitchen/party	10 ton	NA	NA	5.3
4. Party room	3.5 ton	840	1028	0.4
5. East dining	5 ton	1510	1627	4.8
6. Entry	2.5 ton	926	312[1]	NA[3]
Total	**36 tons**			

1. large leakage at return grill.
2. could not access some of the supply grills in this portion of the kitchen
3. this air handler would not shut off

We were unable to obtain a complete set of air flow numbers. However, based on what was measured, air flow rates appear to be very low. Normally, air conditioners should have about 400 cfm per ton (12,000 Btu/hr) capacity to achieve good efficiency. Based on the measurements, a reasonable estimate of total air flow for all 6 air handlers is about 7000 cfm, or just under 200 cfm per ton. This low flow rate reduces the energy efficiency of the air conditioning systems but improves AC system dehumidification performance.

Duct leakage was characterized by means of pressure pan testing. In this test, air handlers were turned off and a blower door was used to depressurize the space, to -50 pascals wrt the attic. A pressure pan (essentially a cake pan on a pole, with a pressure tap) was placed over each register, one at a time, and the pressure inside the pan (which also represents the pressure in the ductwork) was measured. If the ducts are leak free, then the pressure difference from inside the pan to the room will be 0.0 pascal. If there are large leaks, then the pressure pan reading could be 5 pascals, 10 pascals, or even higher. If there was no duct connected to a specific register (if you looked up through the register, you would see only attic), then the pressure pan would "see" the pressure in the attic and would register a reading of 50 pascals. Table 23 provides guidance in interpreting pressure pan test results (Duct Doctoring Manual, FSEC, 1993).

Table 23
Interpretation of Pressure Pan Test Results

Pressure difference (pa)	Condition of duct system
0.0	completely airtight
0.5	Very small duct leakage
1.0	Small duct leakage
3.0	Moderate duct leakage
8.0	Large duct leakage
15.0	Very large duct leakage
30.0+	Open to the world!

The pressure pan testing found very substantial duct leakage. The average pressure pan reading for all of the dining areas was 5.9 pascals, indicating that there are moderate to large duct leaks throughout the duct systems of this building.

Tracer gas decay tests were performed to characterized the building ventilation rate. With the door opening to the adjacent Marlins restaurant closed, the ventilation rate (with HVAC system operating normally) was 2.81 ach. Given the volume of the restaurant, this air change rate is equivalent to 2883 cfm of air flow into the building.

With the door to the adjacent Marlins restaurant open, the ventilation rate (with HVAC system operating normally) was 4.05 ach. Given the volume of the restaurant, this air change rate is equivalent to 4156 cfm of air flow into the building. The increased ventilation rate occurs because the Marlins restaurant (which we did not test) is more depressurized than the Pier restaurant, and opening the door between the two further depressurizes the Pier restaurant.

The two kitchen exhaust fans were measured at 899 cfm and 550 cfm.

A blower door test was performed to characterize the airtightness of the building (see attached figures). The test was performed twice. With the air handlers on and the exhaust air and passive make-up vents masked off, ACH50 was 16.3. With the passive make-up air vents unmasked (they are always open, so this is the normal operating conditions of the building), ACH50 was 17.5. Both tests indicate a substantially leaky building. With the occupied space at -26.8 pascals, the vented attic space above the ceiling was +14.8 pascals wrt to the occupied space. This indicates that the ceiling is, by a small margin, the primary air boundary of the building. It also indicates that the suspended t-bar ceiling is essentially as leaky to the attic as the attic is leaky (vented) to outdoors!

Pressure mapping was performed. With the kitchen exhaust fans operating and all air handlers on, the main dining area was at -0.2 pascals wrt outdoors and +1.1 wrt the adjacent Marlins restaurant. The west dining area was -1.7 pascals wrt the main dining area and the kitchen was -7.8 pascals wrt main dining area.

Summary of Problems

Two major problems exist in this restaurant; 1) large duct leakage and 2) low air handler flow rates. A third but lesser problem is that the building is quite leaky. This leakage is concentrated primarily in the suspended ceiling which separates the occupied space from the attic.

RECOMMENDED RETROFITS

Duct leaks should be repaired. This will not be an easy task because the attic is shallow and it very full of ducts.

The solution to the second major problem, low AH flow rates, may be related to the solution to the duct leakage. That is, the low air flow rates probably are the result of poorly sized and designed ductwork. The most cost-effective solution, therefore, may involve solving the duct system design problem and duct leakage problem together.

One solution would be to fix only the duct leaks that are readily accessible (without major building renovation), modify some of the ductwork to increase air handler flow rates, and then move the attic inside the building air boundary and thermal boundary. In practical terms, this would mean getting rid of the attic ventilation and insulating the attic at the roof deck (either attaching insulation to the bottom of the roof deck or applying an insulation system on top of the roof deck with a roofing membrane or roofing system on top of that).

Further analysis and testing is required before a definitive retrofit solution can be offered.

19. GREEK RESTAURANT

Tested July 9 and 14, 1997

Building

This is a one-story concrete block building with a flat roof located in Palm Bay. It is located at the east end of a strip mall containing about 8 shops, and shares a common wall with a retail shop to the west. This unit has a total of 1500 square feet of floor area and total volume of 13,500 cubic feet below the T-bar ceiling. The volume, including the space between the ceiling and the roof deck, is 18,000 cubic feet. The kitchen/food preparation area is connected to the dining area by an approximate 3' x 6' window and an open doorway. There are two bathrooms, each with an exhaust fan that operates on the light switch. There is one small office/storage room that remains closed most of the time.

A blower door test was performed, and the results found CFM50 equal to 7736. This means that when the blower door draws 7736 cubic feet per minute of air out of the building, the interior space is depressurized to -50 pascals (-0.2 inWC). [Note that the building airtightness can also be expressed by the following formula; $Q = 851.9\ (dP)^{.56}$ where Q is air flow rate (cfm), 851.9 is a flow coefficient approximately representing cumulative hole size, dP is the pressure differential between indoors and outdoors, and 0.56 is an air flow exponent indicating the shape of the curve.] This can also be expressed as 34.4 ACH50. This means that 34.4 building volumes of air are exchanged during a one-hour period when the building is at -50 pascals. It also means that this building is considerably more leaky than the average commercial building. We recommend that commercial building airtightness generally fall in the range of 4 to 8 ACH50.

During the blower door test, it was determined that the suspended, T-bar ceiling is the primary air barrier and that the ceiling space is quite leaky to outdoors. This leakage occurs primarily through a long opening between the ceiling space and the large eave cavity on the south side of the building. Some additional air leakage exists where the ductwork penetrates the roof. When the building was at -50 pascals, the ceiling space was at -10 pascals with respect to outdoors and the eave space was at -5 pascals with respect to outdoors. The 40 pascal drop across the ceiling indicates that the ceiling is the primary air barrier.

Insulation batts are located on top of the ceiling tiles. Therefore, the primary air barrier and the primary thermal barrier are located in the same plane, at the ceiling. Usually, it is good practice that the air barrier and thermal barrier are located together. The problem in this case, however, is that suspended T-bar ceilings are inherently quite leaky, and while the ceiling is more of a resistor to air flow than the roof assembly, the leaky suspended T-bar ceiling cannot be considered an acceptable air boundary. The fact that the ceiling is the primary air barrier is a source of concern. The ceiling space is excessively leaky to outdoors (much of that leakage to the eave space). This has important implications regarding the building cooling load, because this eave space becomes very hot on summer afternoons.

All of the supply and return ductwork, except the air handlers themselves, are located within the ceiling space of the building. Note, however, that the ductwork is outside the primary thermal barrier of the building, since the insulation batts are on top of the ceiling tiles. The estimated R-value of the roof is R5; it is composed on 2 inches of light-weight concrete and 1 inch of polystyrene insulation bead board.

HVAC Systems

At the time of initial testing (July 9, 1997), a single 5-ton cooling system served the restaurant. Because that unit could not achieve and maintain comfortable cooling conditions in the restaurant, a second 3-ton cooling system was added by the restaurant owner. Before the new unit was added, indoor temperatures were about 84F throughout the restaurant in the early afternoon of July 9. After the new unit was added, indoor temperatures were about 78F in the central part and 76F in the front part of the store in the early afternoon of July 14 (thermostats set at about 70F). Even with 8 tons of air-conditioning, the space could not be cooled below these temperatures. The two air-conditioners ran continuously from about 10:20 AM to 1:30 PM (we departed at 1:30 PM).

The return and supply ductwork for both systems is located in the ceiling space. The return air is hard-ducted (that is, the ceiling space is not used as a return plenum). Neither air-conditioning system provides outdoor air to the conditioned space. Thermostats are located in the middle of the dining area (3 ton system) and at the back of the dining area (5-ton system).

There is an exhaust fan which serves the kitchen cooking appliances. Using a capture tent with a duct tester fan, we measured exhaust air flow of 871 cfm. The restaurant owner operates the exhaust fan day and night throughout the week. Make-up air is provided to the kitchen through a 7.5 inch round metal duct which extends about 15 inches down from the ceiling about two feet out from the front of the exhaust hood. Measurements with a hot wire anemometer found the make-up air flow rate to be 167 cfm. Make-up air control by means of the kitchen light switch.

The difference between the exhaust air and the make-up air is **net exhaust air**. Net exhaust air is 704 cfm (871 cfm - 167 cfm). Stated another way, the exhaust fan is drawing 704 cfm more air out of the building than the make-up air is drawing into the building. Therefore, the building operates at negative pressure since no outdoor air is provided through the air-conditioning systems. Measurements found building pressure of -1.2 pascals with respect to outdoors. If the building was more airtight, the air flow imbalance would produce much greater depressurization. If, for example, building airtightness was 4 ACH50, then the building would experience about -30 pascals.

When compared to the building airtightness curve ($Q = 851.9\,(dP)^{.56}$; determined from the blower door test), we find that -1.2 pascals indicates net exhaust flow of 943 cfm. This is somewhat more than the 704 net exhaust, but these numbers are in the ballpark.

As a consequence of this air flow imbalance, 704 cfm of air must be drawn into the building (by the -1.2 pascals of negative pressure) to make up for the net exhaust air flow. Inspection of the building envelope identified that most of the building leakage is located above the ceiling at the wall separating the ceiling space from the south facing eave space. The air that enters the building from this attic-like eave space is very hot and adds considerably to the cooling load of the building.

Inspection of the make-up air found that the make-up air duct was disconnected about 1 foot above the suspended ceiling level. Since the make-up air fan was located below this disconnect, virtually all of the make-up air being drawn into the kitchen was coming from the ceiling space. This has the effect of depressing the ceiling space pressure with respect to room, thus increasing the air flow rate from the eave space to the ceiling space.

Considerations of Cooling Load Calculation

We received a copy of load calculations from a mechanical engineer R. J. Vadminsky of Relco Unlimited. His calculations indicate a total cooling load of 7.4 tons. Based on his calculations, the 8-tons of installed cooling is a good match to this facility. Based on the room temperatures observed in the restaurant on July 14 (76F in front and 78F in the back dining area), it appears that the 8 tons did not meet the cooling load, especially considering that several of the refrigeration appliances had been turned off and that this was not an especially hot day.

There are several factors which might account for the fact that 8 tons of air-conditioning is not meeting the entire cooling load. First are the exhaust and make-up air assumptions made by Vadminsky. He assumes exhaust flow of 4000 cfm, make-up air of 3600, and 400 cfm from the room. He then calculates the associated cooling load for the 400 cfm of make-up air based on 15F temperature difference; load from moisture vapor is also calculated. Our observations and measurements, however, found that most of the air drawn into the building originates from the eave space (south end of store) which we suspect is much hotter than outdoors during the afternoon (indicating an estimated 40F temperature difference) and that actual net exhaust flow rate is 700 cfm rather than 400 cfm. Given the larger net exhaust air flow rate and the greater temperature difference, we calculate a cooling load of 62,200 Btu/hr from make-up air, or 2.5 times his calculations. This load from make-up air is five tons of cooling by itself. A significant reduction in cooling load (perhaps as much as three tons) associated with this unbalanced air flow could be achieved by increasing the make-up air flow rate.

Ventilation Testing

Tracer gas tests were done to characterize the ventilation rate of the building. With all air handlers operating, the ventilation rate of the building was 3.28 air changes per hour (ach). This works out to 738 cfm of air entering the building. This air flow rate is approximately equal to the 704 cfm of net exhaust.

We did not measure the infiltration rate with all HVAC equipment turned off. Based on the blower door test results, we would expect about 0.90 ach with all the air moving equipment turned off (passive building) under average outdoor weather conditions (this passive infiltration prediction based on "divide by 38" identified in Cummings et al., 1996, where 34.5 ACH50 divided by 38 = 0.90 ach).

After the initial tracer gas tests, the make-up air duct was reconnected so that the make-up air was in fact coming from outdoors. Repeat of the tracer gas test found that the building ventilation rate decreased to 2.64 ach. This is equivalent to 594 cfm. Building pressure changed to -0.9 pascals (from -1.2 pascals) with the make-up air duct attached.

A third tracer gas test was done with make-up air increased to 505 cfm using a duct tester fan. In this configuration, ventilation was 2.73 ach. Building pressure increased to about +0.3 pascals with the additional make-up air.

Return leak fraction was 9.4% for the 5-ton system and 4.8% for the 3-ton system. Measured supply air flow rates were 1740 cfm for the 5-ton system and 1177 cfm for the 3-ton system. Total return leak air flow is then 220 cfm. This will probably underestimate total return leak air flow because the total air handler flow rate may be higher because supply leaks were not accounted for and because some of the return leaks were located in the ceiling space where some tracer gas was located.

Duct System Airtightness Tests

Duct airtightness tests were done. An airtightness test was done on the 3-ton system. We found CFM25 to be 179. [CFM25 is the air flow through the duct system leaks when the registers are sealed and the ductwork is depressurized to 25 pascals. CFM25 of 179 indicates a substantial amount of duct leakage.] If the leaks are equally distributed between the supply and return side of the system, and the ductwork pressure is about 25 pascals, then about 8% of the supply air and about 8% of the return air is leaking.

An airtightness test was done on the 5-ton system. After the test was completed, we found that we had failed to seal one of the supply registers. Therefore, that open register would look like a duct leak to the test equipment. Consequently, we could not use those test results. A second type of duct test was done on this system, however. That test is called a "pressure pan" test. With the building depressurized to -50 pascals and the air-conditioner turned off, a pressure pan (like a cake pan on a pole with a manometer [pressure gauge] attached) is placed over each register and grill, one at a time. The pressure in each duct is measured with respect to the room. If there are no duct leaks near that register, then the pressure measurement will be near zero. If there are large leaks, then the pressure measurement might be 10 pascals or greater. Pressure pan measurements ranged from 0.6 pascals (tight) to 7.4 pascals (quite leaky). The average value for the supply ducts was 2.6 pascals. The average value for the return ducts was 4.1 pascals. These test results indicate substantial leakage in the ductwork of the 5-ton system as well.

Exhaust System

The air flow rate of the exhaust fan is less than expected. Mr. Vadminsky based his calculations on an expected 4000 cfm of exhaust air while in fact the air flow was less than 1000 cfm. The problem with low exhaust air flow rates is that the air velocity entering the space between the hood and the cooking surfaces is too low, thus allowing cooking vapors to escape. While there, we observed cooking smoke escaping from the hood and getting into the kitchen, where we expect that it became entrained into the return air of the 5-ton system. Incomplete capture of cooking gases and heat may be adding significantly to the building cooling load. It may be necessary to increase the exhaust air flow rate or close in the hood more completely, or both, in order to maximize the capture of cooking by-products.

RECOMMENDATIONS

Based on our observations and reports from the store owner, three problems have been identified: 1) the restaurant is warm, 2) the exhaust system does not appear to capture all of the cooking by-products, and 3) energy use is high.

Solutions to these problems are interrelated. Following are retrofit recommendations.

1) Consider making modifications to the exhaust system. These modifications could include modifying the exhaust hood to improve capture efficiency of unwanted cooking by-products (including heat, water vapor, odors, and combustion gases). Some possibilities include removing the vertical panel which separates the two sections of the hood. Consider lowering the front and side lip of the hood (perhaps 10" or more) to increase the capture of vapors and radiant heat from the roasters. A qualified engineer should assess your current exhaust system and make final recommendations for exhaust system modifications. It may be necessary to increase the exhaust fan flow rate as well. Standard exhaust fan design practice would call for about 4000 cfm of exhaust flow rate for this system. We are reluctant to recommend going to standard practice because of possible high equipment costs and on-going energy costs -- rather it would be better to see if hood modifications could achieve acceptable capture rates at the current flow rate or at moderately increased flow rates.

2) Consider increasing make-up air to 80% of exhaust air. If exhaust air remains at about 900 cfm, then increase make-up air to 700 cfm. This measure may reduce total cooling load by as much as 3 tons. If exhaust air is increased to 1600 cfm, for example, then make-up air should be increased to 1300 cfm. Consider installing a new make-up air duct that would accommodate the necessary air flow rate and that would discharge the air beneath the cooking appliances. This later modification would improve the capture rate of the unconditioned make-up air and decrease the amount of room air that is drawn into the exhaust system.

3) Consider adding outdoor air to both the 3-ton and 5-ton air-conditioners to bring the building to positive air flow balance and positive pressure. If the exhaust flow remains at 900 cfm with 700 cfm of make-up air, then the total outdoor air should be about 250 cfm. It may be possible to put all of the 250 cfm onto the 3-ton unit. If the exhaust flow is increased to say 1600 cfm with 1300 cfm of make-up air, then the total outdoor air should be about 350 cfm. With more air being brought into the building than is being exhausted, the building will operate at positive pressure. Thus room air will flow out of the building and hot air will not enter from the eave space.

4) Consider sealing up the approximate 12 inch x 30 foot opening between the ceiling space and the eave space. This should make the building fairly airtight, which will make it easier to achieve positive pressure in the building. It will also reduce the amount of hot attic-like air being drawn into the building.

5) Consider insulating the wall separating the eave space from the ceiling space. If batt insulation is used, it should be attached so that it is held firmly against the wall surface. Gravity will try to pull it away from the wall surface and allow heat to enter the ceiling space by means of convective air flow. If rigid board insulation is used, it would be good to use foam at the bottom and top joints to insure an airtight seal against the wall surface. A spray foam insulation could be applied.

6) Consider repairing duct leaks. There is a considerable amount of duct leakage. The fact that the ducts are primarily within the ceiling space, which is partially conditioned, means that duct repair will save less than if the ducts were in a hot attic. Nevertheless, repairs will increase the efficiency of your air-conditioning systems and get cooling more directly to your customers. It is not essential that leaks that are in the air handler cabinet (on the roof) and are return side leaks do not need to be sealed except to keep out rain, dust, and insects, because the leaking air will act like "outdoor air" which is desirable. (Note, however, that when sizing the outdoor air, you should take into account the return leaks.) If you want to seal cabinet penetrations and cracks, they can be sealed by mastic (and imbedded fiberglass mesh if the opening is 1/4" or more) on panels that do not have to be opened for service. Those that need to be opened for service can be sealed by foil tape [note that the foil tape will need to be reapplied every so often].

7) Consider applying a white roof coating system to the roof surface. While the silver painted roof currently reaches temperatures around 135F, a white elastomeric roof coating material would remain below 105F. Thus the cooling load from the roof would be reduced from about 18,000 Btu/hr to about 9,000 Btu/hr. Since all this heat entering through the roof is sensible heat, this 9,000 Btu/hr sensible load reduction could actually reduce the total cooling load by nearly 1 ton. **Alternatively,** a couple inches of insulation could be sprayed on the bottom side of the roof deck to increase the effective insulation value of the roof from about R-5 to about R-15. This would be beneficial even though there are insulation batts on the ceiling tiles, because the ducts, which are located in the ceiling space, would be exposed to less heat and any duct leaks would draw cooler air into the system. Wet-spray cellulose, urethane foam, and Icynene are materials that could be applied to the bottom of the roof deck. This would move the primary thermal barrier to the newly established air barrier (after sealing the opening to the eave space, the roof deck will be the primary air barrier), and create a more effective total building envelope.

Short-term monitoring at the Greek restaurant

Short-term monitoring on the Greek restaurant was performed during the third quarter of 1997. Two types of data loggers were employed; 1) a Campbell logger with 8 channels of data and 2) one-channel stand-alone loggers. The Campbell logger measured temperatures at top and bottom of roof, in exhaust air, in the eave, plus CO, CO_2, building pressure., and RH in the eave. Twenty-four Stowaway loggers measured temperatures and humidity before the coil on two systems, at two return grills, at two supply diffusers, at two thermostats, and in the make-up air. Stowaways measured temperature only after the coil. Energy consumption for the building and each AC unit was measured. On/off status of the make-up air was also monitored.

Monitoring had been expected to run for 14 days but was extended to 28 days due to data loss in the Campbell logger (memory overwrite). In addition, 4 Stowaway loggers were damaged and stopped working after about five days, due to moisture accumulation within the casing. (We do not know the cause of the moisture accumulation, however they were located very near the evaporator coil and condensate drain pan.)

The short-term monitoring looked at temperature and dewpoint drops in two AC systems, operation/cycling behavior of AC systems, occupied zone temperature and humidity, depressurization of the occupied zone with respect to a large (and hot) eave space, operation of the kitchen exhaust system, temperature and humidity of entering make-up air, and temperature of the top and bottom of the roof assembly. A short-term monitoring plan was developed before installation. One important finding is that about 20% of the cooling capacity of the two rooftop package AC units is lost between the coil and the diffusers. This prompted us to return to the store to do additional testing (temperature mapping of the duct systems) in order to verify the approximate 6 degree F temperature rise from coil to diffuser and determine the cause. The largest portion of the temperature rise is attributable to return leakage that occurs between the coil and blower. A retrofit plan was developed which includes seven specific modifications to the building, AC, exhaust, and make-up air systems. This retrofit plan was presented to the store owner. No retrofits were implemented at this store.

20. CHARITY OFFICE / THRIFT SHOP

Results from Testing
March 23, 1998

A number of tests were done on the North Brevard Charities building.

1) Building shell
   a. Airtightness – the building is somewhat tighter the average commercial building, but the building is still rather leaky. ACH50 = 12.5 (6841 CFM50; see blower door test form on the following page). We recommend building airtightness in the range of 4 to 8 ACH50. Therefore, it would be useful to perform airtightening of the building envelope. Note, however, that tightening the building envelope from 12.5 to say 6.0 ACH50 would not be expected to save a great deal of energy.
   b. Air boundary -- importantly, the ceiling is not the primary air boundary of the building; rather the roof deck is the primary air boundary. Suspended ceilings are quite leaky and therefore allow considerable air flow through them when exposed to a pressure differential. With the occupied space at -50 pascals, pressure drop was 11.4 pascals across the ceiling (2' x 4' suspended ceiling tiles) and 38.6 pascals across the roof deck (ceiling space is unvented).

2) Building insulation
   a. Insulation is located in the walls and on top of the ceiling tiles
   b. There is also a thin layer of insulation at the bottom of the roof deck, but heat comes readily through the insulation from the hot roof and causes the ceiling space to be hot during sunny periods
   c. Calls above the ceiling have no insulation – they are simply bare metal on the east side (2 feet high) and the south side (average 3 feet high), block on the west side, and gypsum board on the north side (to warehouse); therefore the ceiling space is very hot. For example, on a cold

Friday March morning (about 45F outdoors at 9:30 AM), the temperature in the ceiling space had exceeded 90F by 9:30 AM and 105F by 11:20 AM. Actual monitored ceiling space temperatures have commonly exceeded 130F.

3) Air conditioning systems – there are two air conditioning systems. One is 5 tons and the other is 2.5 tons. 7.5 tons of air conditioning should be adequate for a thermally efficient building with 3700 square feet. However, the building is not thermally efficient. Several problems have been identified:
   a. The cooling coils for both systems are caked with dirt on the bottom side, and thus are reducing the cooling capacity and efficiency of the systems
   b. The smaller system has moderate duct leakage (157 CFM25) and the larger system is more leaky (401 CFM25). Good practice would call for a maximum of 50 CFM25 for the small system and 100 CFM25 for the large system.
   c. There are return air imbalance problems
      1) The return of the smaller system is located in the food storage room and the doors to this room are reportedly closed most of the time. With the doors to this room closed, the room is depressurized (-4 pascals) and therefore draws air from the attic space. As a consequence, we estimate that more than 50% of the total return air flow for the small system originates from the attic. Since this air is quite hot, it reduces the ability of the AC system to cool the building.
      2) The return of the larger system is located in the hall. The office suite has no return air and can be closed off from the hallway. When closed it operates at +2 pascals with respect to the hallway. This pressure pushes cool room air into the hot attic space. Since this air is not returning directly to the return grill in the hall, the central zone of the building is depressurized slightly and thus draws hot air from the attic. The office suite is reportedly closed 50% of the time.
   d. Supply air flows are out of balance in the office suite. Air flow into the two offices is too great and flow to the conference room is too small. This causes poor temperature control. There are dampers on each of the duct runs we have inspected, so the air flows to the office suite supply diffusers can be adjusted. It may be necessary to add an additional duct to the conference room if the air flow rate cannot be increased sufficiently.
   e. Lighting is inefficient.

Proposal

[This is the proposal which the Florida Solar Energy Center submitted to the North Brevard Charities board of directors. The board approved this proposal and the retrofits and monitoring activities described in this proposal were implemented.]

The Florida Solar Energy Center (FSEC) is performing testing, retrofits, and monitoring of buildings as part of a project sponsored by the Florida Energy Office. The testing we have done at North Brevard Charities is part of that research.

We would like to propose retrofits be done to the building during the summer of 1998 and that FSEC monitor the energy use, temperature, and humidity within this building for a period of 1 year. There are several items that would be best done at the start of the monitoring period; these items are labeled "baseline". In the summer, phase 1 and phase 2 repairs would be done, probably in late June and early

August, respectively. We would ask North Brevard Charities to perform some of the retrofits and FSEC would perform some of the retrofits.

Baseline (March)

1. Decrease air flow to two offices
2. Increase air flow to conference room (this may require an additional duct)
3. Clean AH fans and cooling coils, and verify refrigerant levels

Phase 1. (June)

1. Repair duct leaks
2. Balance return air
   a. Small system – add a return grill directly from return plenum into the hallway, and reduce the size of the return grill in the food storage room to an amount equal to the supply air flows to that storage room
   - Large system – add transfer grills (and ducts) from office suite to retail area

Phase 2. (August)

1. Put white elastomeric roof coating on metal roof.
2. Insulate walls above ceiling, or paint the exterior of the south and east sides of building with white paint, most specifically above the level of the ceiling.

Benefits that would result from retrofits:

- Baseline retrofits would improve comfort in the office suite and improve AC system performance.
- Phase 1 retrofits would reduce the air flow rate across the ceiling (and through the insulation) by about 80% or 500 cfm. This would reduce the cooling load on the building, and probably improve comfort during hot weather. It is likely that this will decrease cooling energy use by 15% or more.
- Phase 2 retrofits would reduce the entry of heat into the building through the roof and exterior walls above the ceiling. It is likely that this will decrease cooling energy use by an additional 15% or more.
- In total, it is anticipated that total cooling energy use will decline by 30% or more.

The Florida Solar Energy Center would offer to do or pay for the following:

1. Adjust air flows to offices (with feedback from NBC staff regarding comfort).
2. Repair duct leakage.
3. Add return air transfer to the office suite.
4. Install a white elastomeric coating on the roof.

We would ask North Brevard Charities to do or pay for the following:

1. Add an additional duct to serve the conference room if the air flow rate cannot be adjusted upward to the required level.
2. Have cooling coils cleaned and refrigerant levels checked and adjusted if necessary
3. Have a return grill installed directly from the return plenum of the small AC system into the

hallway, and reduce the size of the return grill in the food storage room to an amount equal to the supply air flows.

4. Insulate walls above ceiling, or paint the exterior of the south and east sides of building with white paint, most specifically above the level of the ceiling.

RECOMMENDATIONS

Baseline

1. Decrease air flow to two offices (NBC)
2. Increase air flow to conference room (NBC)
3. Clean AH fans and cooling coils, and verify refrigerant levels (NBC)

Phase 1.

1. Repair duct leaks (FSEC) (18 hours + $50 materials)
2. Balanced return air
   a. Small system – add a return grill directly from return plenum into the hallway, and reduce the size of the return grill in the food storage room to an amount equal to the supply air flows (NBC) (contractor = $200)
   b. Large system – add transfer grills (and ducts) from office suite to retail area (FSEC) (10 hours + $45) (second unit 7 hours + $40) = (17 hours +$85)

Phase 2.

1. Put white elastomeric roof coating on metal roof (FSEC) (20 hours + $50 materials)
2. Insulate walls above ceiling, or paint the exterior of the south and east sides of building with white paint, at least above the ceiling level. (NBC)

Proposal to Perform Additional Retrofits Beyond the Scope of This Project

Written September, 1999

[This proposal was developed after completion of the three retrofits of this project and submitted to the Florida Energy Office. No funding for this follow-on work has been obtained.]

Background

North Brevard Charities is a community charitable organization which serves the needs of the poor in Titusville, Florida. It is a 4000 square foot metal building (metal skin walls on three sides and roof, fourth wall is stucco). It has been the subject of three energy retrofits over the past two years as part of a study of uncontrolled air flow and HVAC problems. Those retrofits have focused on improved thermal envelope, balancing of return air, and duct repair.

Prior to these retrofits, interior conditions would often exceed 85F on summer afternoons (9 degrees above the thermostat setpoint). Covering the roof with a white elastomeric coating and exterior walls with white paint yielded a four degree reduction in indoor temperatures. Balancing of return air and implementation of duct repairs has further reduced indoor temperatures and yielded cooling energy savings. During a majority of hot summer days, peak indoor temperatures now remain below 80F.

Proposed additional work

Our proposal: to implement four additional retrofit measures at this facility, and monitor the change to comfort and total energy use. These proposed retrofit measures are:

1. High efficiency lighting
2. High efficiency refrigeration
3. Window shading
4. High efficiency air conditioning

Our objective is to demonstrate the application of a range of energy conservation measures. As indicated, several retrofits have already been implemented. When we first began, the building was difficult to inhabit during a portion of summer days because indoor temperatures often exceeded 85F. *First*, we coated the roof and walls with a white coating. We would normally expect cooling energy use reduction of 30+% in this type of building. However, based on the monitored data, yearly cooling energy use was projected to be less than 5%. The major impact was to significantly reduce average indoor temperatures and the number of hours per day that indoor temperatures were uncomfortable. On average, afternoon indoor temperatures declined by about 4 degrees F as a result of the white coatings.

Second, we balanced the supply versus return air flows so air was no longer being driven from indoor zones into the hot ceiling space and then pulled back into the occupied zones where the returns were located. This resulted in further reduction in peak indoor temperatures and about 15% reduction in cooling energy use. Duct repairs have also been performed but the savings from this retrofit are not known at this time.

We would like to complete the story and demonstrate the extent to which cost-effective energy retrofits can create a true success story. We would like to illustrate the overall cost-effectiveness of an energy retrofit tour-de-force.

Tasks to be carried out in this project would be:

1. Implement four additional retrofits
2. Increase our monitoring capability at the facility to gain a full picture of the results of these retrofits (including adding power meters to the monitor electrical circuits serving the lighting systems and the plug circuits which serve the refrigerator/freezers)
3. Collect and process the data during the monitoring period
4. Analyze the data
5. Prepare a final report
6. Develop an approximate 90 minute presentation (lecture with visuals) of the results of this project
7. Prepare a professional paper on the results of this project
8. Post the results of this project on the internet.

Lighting Retrofits

An initial assessment of potential energy savings from installation of high efficiency lighting has been performed. Currently, there are 32 light fixtures consuming 106 watts (2 lamps) each and 7 light fixtures consuming 186 watts (4 lamps) each. Total energy consumption is currently about 4700 watts or 42 kWh each 9 hours day. We anticipate installing high efficiency reflectors and two high efficiency lamps in each fixture. After retrofit, these fixtures will produce more light and better quality (whiter) light than is currently available, and consume only about 55 watts each. Total energy consumption will be about 2150 watts or 19 kWh each 9 hour day. Annual electricity savings are anticipated to be about 7000 kWh, or about $500 per year.

More efficient lighting will also reduce the building cooling load substantially, because it will give off less heat. The two air conditioning systems operate at an EER (energy efficiency ratio) of about 8, which is equivalent to a COP (coefficient of performance) of about 2.3. If we assume that 90% of the heat from the lighting system must be removed by the cooling system (we assume 10% of the time is heating season when cooling is not required), then annual cooling energy savings can be projected to be about 3000 kWh, or about $225. The combined savings from improved lighting and reduced cooling load from lights are expected to total about $725 per year.

Importantly, we anticipate that the reduced lighting load will reduce peak cooling loads (because the lights give off considerable heat) and achieve better indoor comfort (diminish the length of time that indoor temperatures exceed the thermostat set point).

The potential for lighting energy use reduction in the bathrooms, exit lighting, and adjacent (attached) warehouse will also be examined. Currently the lights in the bathrooms and exit signs are incandescent, and fluorescent change outs can be performed. In the warehouse, there are 11 light fixtures each consuming 178 watts. If a cost-benefit analysis looks promising, we may also implement a lighting retrofit in the warehouse.

Refrigeration retrofits

We also propose to examine the efficiency of the refrigeration units in this facility. There are a total of seven refrigeration units (freezer and refrigerator/freezer units) located inside the conditioned space. These are used to store food for their food distribution services. Most of these units are 10 to 20 years old and therefore are likely to be less energy inefficient than current models. We propose to monitor the energy use of each unit for a period of several days or more, and then investigate the cost-effectiveness of replacing these units with newer high efficiency residential style units or commercial style split systems (compressor/condenser located outside). In the latter case, the refrigerator would in effect be acting as an air conditioner to the building, pumping heat out of the building. Depending upon the cost and efficiency of commercial units, this might be an effective means to reduce utility costs and reduce the peak cooling load on the building.

Depending upon the results of this analysis, we propose to replace some or all of the refrigeration units. Higher efficiency refrigeration would save energy and would reduce the cooling load on the building. Split system refrigerators/freezers would even more greatly reduce the building's cooling load. Utility bill savings and improved indoor comfort would result.

Window Shading

Two large windows and two glass exterior doors are located on the west side of the building. Because they are unshaded, the sunlight coming through these windows causes spikes in indoor temperature (and cooling load) after about 3 PM. We propose to install shading devices which should reduce the solar-induced window cooling load by about 70%. Currently there are 165 square feet of west facing glass (there are no other windows in the building) and under bright sunshine conditions during the afternoon, this represents about 50,000 Btu/hr of cooling load. With shading, this could be reduced to about 15,000 Btu/hr. This will primarily reduce cooling load from 3 PM to 6 PM (6 PM is when the AC is turned off for the day). It will produce some cooling energy savings, but more importantly it will reduce the spike in room temperature (and occupant discomfort) which occurs daily from 3 PM to 6 PM in the west portions of the retail and office areas.

Cooling system retrofits

We also propose to replace the older inefficient air conditioning systems. Currently, there are two systems; one 2.5 ton system and one 5.0 ton system. We are currently monitoring the energy use and cooling output of these systems. Based on our data, these units are operating at EERs of 8. It is anticipated that new systems with rated efficiency of approximately 14.0 SEER would be installed and that total cooling energy use would be reduced by approximately 45%. Currently, air conditioning energy use runs at about 21,000 kWh per year, or about $1550 per year. If we assume that lighting, refrigeration, and shading retrofits reduce cooling energy consumption by 30% from current levels, then 45% savings would about $500 per year. If the change-out price for the two systems were $5000, then the simple payback for this retrofit would be 10 years.

Because these retrofits will improve comfort to customers of the thrift shop and enhance the quality of indoor light, these retrofits may also increase retail sales.

Deliverables

Final report. A final report will be prepared, summarizing the energy retrofits implemented on this building and the energy savings and improvement in indoor comfort conditions resulting from the retrofits. A cost-effectiveness analysis will also be included in the report.

Presentation. We will develop a presentation, including a variety of photographic and graphic materials, suitable for presentation to both layman and professional groups.

Internet. We will place the final report on the internet to effectively transfer project findings and achieve exposure.

Professional paper. We will publish a professional paper covering the results of this research and present it at a national conference.

Newspaper article. We will prepare an article suitable for distribution to Florida newspapers.

AC TONS

TOTAL BUILDING INSTALLED COOLING CAPACITY

TONS PER 1000 SF

IS COOLING CAPACITY PER 1000 SQUARE FEET OF FLOOR AREA

AH LOCATION

A = ATTIC
CL = CLOSET
CS = CEILING SPACE
EG = EXTERIOR GROUND
EMR = EXTERIOR ACCESS MECHANICAL ROOM
EW = EXTERIOR WALL
MR = MECHANICAL ROOM
OS = OCCUPIED SPACE
R = ROOF TOP
W = WAREHOUSE

DUCT LOCATION

0 = NO DUCTS
1 = IN CONDITIONED SPACE
2 = IN UNCONDITIONED SPACE
3 = OUTDOORS

DUCT TYPE

D = DUCTBOARD
F = FLEX DUCT
M = METAL
N = NONE

BUILDING CAVITY DUCT

CH = CHASE USED AS DUCT
CL = CLOSET USED AS PLENUM
CS = CEILING SPACE USED AS PLENUM
MR = MECHANICAL ROOM AS PLENUM
N = NONE
O = OTHER CAVITY USED AS DUCT
SP = SUPPORT PLATFORM ENCLOSED AS PLENUM
WCD = WALL CAVITY AS DUCT

OCCUPANCY

TYPICAL FULL UTILIZATION, NOT AVERAGE OCCUPANCY; in a church, for example, typical full utilization might be 200 persons on a typical Sunday morning, while the average over all hours of the week might only be 5 persons.

FLOOR AREA

BUILDING FLOOR AREA (square feet), INCLUDING ALL OCCUPIED FLOORS

OCCUPIED VOLUME

IS VOLUME BETWEEN FLOOR AND CEILING IN PORTIONS OF THE BUILDING NORMALLY CONDITIONED AND OCCUPIED BY PERSONS (cubic feet)

THERMAL VOLUME

IS THE VOLUME BETWEEN THE FLOOR AND THE CEILING/ROOF THERMAL BARRIER (cubic feet); if the thermal barrier is located at the ceiling the thermal volume will be equal to the occupied volume; if the thermal barrier is at the roof deck or some place in between the roof deck and the ceiling, then the thermal volume will be larger than the occupied volume.

AIR VOLUME

IS THE VOLUME INSIDE THE BUILDING'S PRIMARY AIR BARRIER, THAT IS THE VOLUME BETWEEN THE FLOOR AND THE PRIMARY AIR BARRIER (cubic feet); if the ceiling is the primary air barrier then the air volume will be equal to the occupied volume; if the air barrier is at the roof deck then the air volume will be larger than the occupied volume.

ROOF SLOPE

F = FLAT
S = SLOPED

ABOVE CEILING (description of space above the ceiling)

A = ATTIC
C = CEILING SPACE
N = NO SPACE ABOVE CEILING
W = WAREHOUSE

CEILING MATERIAL

S = SUSPENDED CEILING TILES
P = CEILING PANELS, NOT ON SUSPENDED T-BAR
G = GYPSUM BOARD
N = NO CEILING OTHER THAN THE BOTTOM OF THE ROOF DECK

CEILING SPACE BARRIER CONFIGURATION (AB = air barrier, TB = thermal barrier)

1 = PRIMARY AB AND TB ARE LOCATED AT THE ROOF DECK WHICH IS ALSO THE CEILING
2 = PRIMARY AB AND TB ARE LOCATED AT THE ROOF DECK; SUSPENDED CEILING
3 = PRIMARY AB IS LOCATED AT THE ROOF DECK; PRIMARY TB IS LOCATED AT CEILING
4 = PRIMARY AB AND TB ARE LOCATED AT THE SUSPENDED CEILING WHILE THE SPACE BETWEEN THE CEILING AND ROOF DECK IS GENERALLY WELL VENTILATED
5 = PRIMARY AB IS LOCATED AT THE CEILING; PRIMARY TB IS LOCATED AT THE ROOF DECK
6 = PRIMARY AB IS LOCATED AT THE ROOF DECK; PRIMARY TB IS LOCATED BETWEEN THE CEILING AND THE ROOF DECK
7 = PRIMARY AB IS LOCATED AT THE CEILING; PRIMARY TB IS LOCATED BETWEEN THE CEILING AND THE ROOF DECK

8 = PRIMARY AB AND TB ARE LOCATED AT GYPSUM BOARD CEILING WHILE THE SPACE BETWEEN THE CEILING AND ROOF DECK IS GENERALLY WELL VENTILATED

CEILING SPACE CONDITION (AB = air barrier, TB = thermal barrier)

0 = NO CEILING SPACE
1 = CEILING SPACE IS PLENUM; cool and dry
2 = INSIDE AB AND TB; warm and dry
3 = INSIDE AB, OUTSIDE TB; hot and dry
4 = OUTSIDE AB AND TB; hot and humid
5 = OUTSIDE AB AND INSIDE TB; warm and humid

CFM50

THE AIR FLOW RATE INTO THE BUILDING WHEN THE BUILDING IS DEPRESSURIZED TO -50 PASCALS (by a calibrated fan), WITH ALL MECHANICAL AIR MOVING SYSTEMS TURNED OFF AND EA, MA, AND OA OPENINGS SEALED OFF.

CFM50 PER SF

CFM50 DIVIDED BY THE SQUARE FEET OF FLOOR AREA.

ACH50

THE AIR EXCHANGE RATE OF THE BUILDING (exchange with outside air) WHEN THE BUILDING IS DEPRESSURIZED TO -50 PASCALS (by a calibrated fan), WITH ALL MECHANICAL AIR MOVING SYSTEMS TURNED OFF AND EA, MA, AND OA OPENINGS SEALED OFF.

SURFACE AREA

THE EXTERIOR SURFACE AREA OF THE BUILDING, INCLUDING EXTERIOR WALL SURFACES UP TO THE HEIGHT OF THE CEILING/ROOF PRIMARY AIR BARRIER, AND THE SURFACE AREA OF THE CEILING/ROOF PRIMARY AIR BARRIER. THE FLOOR SURFACE AREA IS NOT INCLUDED.

C BUILDING AIR FLOW COEFFICIENT, DERIVED FROM A MULTI-POINT BLOWER DOOR TEST.

n BUILDING AIR FLOW EXPONENT, DERIVED FROM A MULTI-POINT BLOWER DOOR TEST. AIR FLOW THROUGH THE BUILDING ENVELOPE IS DEFINED AS A FUNCTION $Q = C\ (dP)^n$, WHERE Q IS AIR FLOW IN CFM AND dP IS PRESSURE DIFFERENTIAL BETWEEN INDOORS AND OUTDOORS IN PASCALS.

r CORRELATION COEFFICIENT; INDICATES THE GOODNESS OF FIT OF THE PRESSURE VS AIR FLOW DATA POINTS WITH THE BEST FIT LINE DEFINED BY $Q = C\ (dP)^n$.

DUCT CFM25

AIRTIGHTNESS OF THE DUCT SYSTEM. THIS IS THE AIR FLOW RATE THROUGH LEAKS IN THE AIR DISTRIBUTION SYSTEM, INCLUDING SUPPLY DUCTS, RETURN DUCTS, SUPPLY PLENUMS, RETURN PLENUMS, AND AIR HANDLERS WHEN THE AIR HANDLERS ARE TURNED OFF, THE REGISTERS AND GRILLS ARE MASKED CLOSED, AND THE SYSTEM IS DEPRESSURIZED TO -25 PASCALS.

ach AH ON

TRACER GAS DECAY MEASUREMENT OF BUILDING AIR EXCHANGE (WITH OUTDOORS) WITH AIR HANDLERS ON CONTINUOUSLY AND EXHAUST FANS IN NORMAL MODE

ach AH OFF

TRACER GAS DECAY MEASUREMENT OF BUILDING AIR EXCHANGE RATE (WITH OUTDOORS) WITH AIR HANDLERS, EA, AND MA OFF

OA

THE OUTDOOR AIR FLOW RATE (CUBIC FEET PER MINUTE) INTO THE BUILDING; note that OA flow may vary in cases where the air handler cycles on and off with the operation of the AC compressor.

MA

THE MAKEUP AIR FLOW RATE (CUBIC FEET PER MINUTE) INTO THE BUILDING; note that MA flow is normally continuous during EA operation.

RL flow

THE FLOW (CUBIC FEET PER MINUTE) OF AIR INTO LEAKS IN RETURN DUCTS; this air may be drawn from outdoors or unconditioned zones of the building

EXH

THE FLOW RATE (CUBIC FEET PER MINUTE) OF AIR LEAVING THE BUILDING THROUGH EXHAUST FANS

EXH PER 1000SF

THE FLOW RATE OF AIR LEAVING THE BUILDING THROUGH EXHAUST FANS PER 1000 SQUARE FEET OF FLOOR AREA.

dP ON

PRESSURE IN THE BUILDING WITH RESPECT TO OUTDOORS WHEN THE AIR HANDLERS ARE TURNED ON AND THE EXHAUST AND MAKE-UP AIR FANS ARE IN THEIR NORMAL OPERATING MODE

BUILDING	YEAR BUILT	AGE BLDG	BLDG USE	STAND ALONE	TYPE CONSTR	NUMB STORY	SLAB CRAWL	ROOF STRUC	CONVRT USE	# AC UNITS	AC TONS	TONS / 1000SF	AH LOC	DUCT LOC	DUCT TYPE	B_CAV DUCT	OCCU-PANCY	FLOOR AREA	OCCUP VOL	THERMAL VOL	AIR VOL
1 ELEM. SCHOOL 1	1966	34	6	0	MASONRY	1	0	MT	N	40	112	2.23	CS	1		N	932	50139	440930	601668	601668
2 CHICKEN REST.	1993	7	7	0	MASONRY	1	0	MT	N	4	35	11.07	R	2	D/F	N	30	3161	28330	28330	41093
3 COMMUN. CENTER	1991	9	8	0	MET/FRAME	1	0	W	N	5	100	4.18	CS/MR	2	D/F	N	300	23934	292234	292234	388688
4 ELEM. SCHOOL 2	1998	2	6	0	MET/MAS	1	0	MT	N	4	15	3.00	PTHP	2	D/F	N	100	5000	40000	50000	73040
5 ELEM. SCHOOL 3	1998	2	6	0	MET/MAS	1	0	MT	N	4	15	3.00	PTHP	2	D/F	N	100.	5000	40000	50000	73040
6 SEAFOOD REST.	1998	2	7	0	FRAME	1	0	MT	N	5	58	7.07	R	1 & 3	M	N	250	8204	82040	82040	82040
7 RIVER FRONT REST.	1986	14	7	0	MASONRY	2	0	MT	N	3	25	4.27	CS	1	D/F	N	120	5850	58500	58500	84825
8 STEAKHOUSE REST.	1995	5	7	0	MET/FRAME	1	0	MT	N	5	45	7.36	R	1	D/F	N	180	6115	51157	79495	79495
9 MDL. SCHOOL 1 B	1996	4	6	0	FRAME	1	1	W	N	1	3	4.25	EW	0	M	WCD	28	706	5370	5841	5370
10 MDL. SCHOOL 1 SW	1991	9	6	0	FRAME	1	1	W	N	1	2.5	3.85	EW	2	M	WCD	13	650	5200	5200	5200
11 MDL. SCHOOL 1 D	1991	9	6	0	FRAME	1	1	W	N	1	3	3.97	EW	0	M	WCD	35	756	5859	5859	5859
12 MDL. SCHOOL 1 E	1991	9	6	0	FRAME	1	1	W	N	1	3	3.97	EW	0	M	WCD	30	756	5859	5859	5859
13 MDL. SCHOOL 2 1	1997	3	6	0	FRAME	1	1	W	N	1	2.5	3.47	EW	0	M	WCD	25	720	5760	5760	5760
14 MDL. SCHOOL 2 6	1997	3	6	0	FRAME	1	1	W	N	1	2.5	3.47	EW	0	M	WCD	24	720	5400	5880	5400
15 MDL. SCHOOL 2 7	1997	3	6	0	FRAME	1	1	W	N	1	2.5	3.47	EW	2	M	N	26	720	5760	5880	5400
16 BAR AND GRILL	1985	15	7	0	MASONRY	1	0	W	N	2	8	3.33	A	2	D	N	25	2400	22800	25200	22800
17 CONVEN. STORE	1988	12	3	0	MASONRY	1	0	W	N	2	17.5	4.05	A	2	D/F	N	8	4320	41904	41904	41904
18 OCEAN FRONT REST.	1962	38	7	0	FRAME	1	1	W	N	6	36	6.02	R	2	D/F	WCD	90	5977	61563	61563	61563
19 GREEK REST.	1960	40	7	1	MASONRY	1	0	MT	CC	2	8	5.33	R	2	F/D	N	30	1500	13500	13500	13500
20 CHARITY OFFICE	1984	16	3	0	MET/MAS	1	0	MB	CC	2	7.5	2.00	OS	2	D/F	MR/SP	16	3741	32921	32921	56115
average		11.8				1.1				4.55	25.05	4.47					118	6518	62254	72882	82931

BUILDING	ROOF SLOPE	ABOVE CEIL	CEIL MAT'L	CEILING BARRIER CONFIG	CEILING SPACE COND.	CFM50	CFM50 PER SF	ACH50	SURF AREA	C	n	r	DUCT CFM25	ach AHon	ach AHoff	OA cfm	MA cfm	RL cfm	EXH cfm	EXH PER 1000SF	dP ON
1 ELEM. SCHOOL 1	F	C	S	2	2	63200	1.26	8.60	10483	4989.0	0.65					4236	0		10184	203	-1.6
2 CHICKEN REST.	F	C	S	3	3	3165	1.00	6.70	5616	312.7	0.59	0.997		3.000		2940	6940	868	9534	190	-22.8
3 COMMUNITY CENTER	S	A	S	4	5	155047	49.05	31.83	10126.	6516.6	0.81	0.975	3700	0.928		1800	3800	0	5900	1117	-0.2
4 ELEM. SCHOOL 2	S	A	S	6	3	16644	5.27	24.97	2640	1641.9	0.59.			1.000	0.030	0	0	0	0	0	
5 ELEM. SCHOOL 3	S	A	S	6	3	18509	5.86	27.76	2640	1649.5	0.62			1.325	0.070	0	0	0	0	0	

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Appendix C
Long-term Monitoring of the Impacts of Retrofits on Six Buildings

Table 1
Retrofits and long-term monitoring were done on six buildings.

building	retrofit description
convenience store	1) improve HVAC systems
bar and grill	1) add make-up air, 2) improve air/thermal barrier
charity thrift store	1) change exterior to white, 2) balance return air, 3) duct repair
classroom building	1) improve air/thermal barrier
classroom building	1) improve air/thermal barrier
community center	1) improve air/thermal barrier 2) reduce operation time 3) dehumidify OA

Convenience Store

The convenience store did not have a successful retrofit. The circumstances unfolded as follows. We set up a data acquisition system to monitor temperatures, humidity, air-conditioning system energy use, building air pressure, solar radiation, and outdoor temperature and humidity. After several months of monitoring (pre-retrofit data), we arranged to have air-conditioning system retrofits done. These included bringing the air-conditioning system air flows (including outdoor air) and refrigerant charge as close as possible to design specs and modifying the thermostat of one of the two air-conditioning units so that the fan would not run continuously (this fan could literally not be turned off by the thermostat). Within two weeks of completing these retrofits, the store was sold (without warning and to our surprise) to another owner (February 26, 1999). We spoke with the new ownership and received permission to continue our monitoring, but we did not know the extent of the changes which were coming. The roof, which had been light gray color, was painted a dark green. Some of the exterior walls were painted. The suspended ceiling and the ceiling insulation were removed and replaced by new product. The west facing (tinted) windows were replaced. Additionally, all condenser units for both air-conditioning and refrigeration systems were replaced. Given the change in ownership and all the retrofits, there was no opportunity to collect data reflecting changes resulting from the retrofits alone. We therefore abandoned monitoring at this building.

Bar and Grill

This 2400 square foot building is used primarily as a restaurant that is independently owned and operated and is not part of any restaurant chain or organization. The restaurant would be very similar to residential building construction properties if it was not for a lack of windows or a soffit area located above the bar (consists of rustic wood shingles with gaps and knot holes and can be seen in Figure 3). The entire ceiling area is covered with sheet rock with exception to this bar overhang.

Duct tightness testing following duct repairs made during a previous project in 1995 on the two air conditioning systems indicated the east system had a tightness of 134 $CFM25_{total}$ and the west system was 138 $CFM25_{total}$. However, testing during November 1997 indicated the tightness of the east system had become much worse with a tightness of 485 $CFM25_{total}$ and 328 $CFM25_{out}$. The west system test showed 102 $CFM25_{total}$ and 39 $CFM25_{out}$. Inspection of the east system revealed a cut-out section in the return duct near the air handler that had not been sealed. This was apparently done by someone to provide access for coil cleaning, as had been done in the past. (It is unlikely that depressurization during duct or building tightness testing produced the leak since the operational pressure in the return is -78 pa and the maximum pressure observed during any airtightness test was only -36 pa.) Before proceeding with retrofits slated for this project, we sealed this duct leakage by replacing the cut out section, then sealing it using duct sealant mastic. After repair, both duct systems were considered to be tight enough that there would be very limited benefit from additional sealing.

Kitchen Make-Up Air Retrofit

The first retrofit was installation of make-up air of the kitchen exhaust hood.

Further testing and inspection indicated that this building had unbalanced exhaust and significant envelope thermal inefficiencies. Testing showed that the single kitchen exhaust fan in this restaurant moved 980 cfm out of the building, causing it to be depressurized to about -1.8 pa. This caused air to move from the attic space into the occupied space through many visible pathways located over a bar area. Further investigation of the heating appliances and exhaust hood dimensions revealed that the exhaust flow should be increased to maximize the capture of heat, steam and grease.

A request for bids to increase exhaust air to about 1200 cfm and install make-up air (MA) capable of delivering 1200 cfm began June 1998. Two contractors expressed interest in doing the work, but were too busy to complete it during the time required. By October 15 a contractor was under contract and completed most of the work. However, due to a delay in obtaining parts, the MA was not fully operational until the middle of December 1998.

Make-Up System Duct Leakage

The MA system airflow and tightness were tested during January 1999. Tightness results were measured using a Duct Blaster mounted on one of the four intake grills of the unit on top of the roof. The remaining three intake grills and three discharge grills were sealed using a masking material. A multi-point airtightness test found the total tightness of the system to be 122 CFM25. Visual inspection revealed some leakage on the return side of the fan unit in the bottom corners and a small hole in the duct board plenum where electric conduit penetrated it on the supply side of the system. The return and supply system tightness were not tested separately, however, based on visual inspection it is estimated that 40% of the total system leakage is on the return side and 60% is on the supply side. The total leakage and estimated leakage on each side of the unit are summarized in

Table 2. Most of the supply side leakage is difficult to detect in any specific location since it is spread out over the entire supply side system.

Table 2
Make-up air system leakage

Make-up air system	CFM25
Total C =19.52 n = 0.57 r = 0.9958	122
Return leakage (est. 40% of total)	49
Supply leakage (est. 60% of total)	73

Make-Up System Airflow

Upon completion of the MA system, we found building pressure at +2.0 pa (with exhaust air, make up air, and air conditioners operating). This indicated excessive MA flow; in fact the measured airflow was about 1911 cfm through the intake grills located on the rooftop mounted unit. The amount of air that came into the system through return leaks can be estimated by using the measured operational pressure in the return, the flow variables measured for the total system, then multiplying that quantity times 40% which adjusts the amount to reflect the return portion only. Supply leakage airflow can also be calculated by replacing 40% with 60%. The calculation for return side MA leakage is shown below:

$(8.9^{0.57} \times 19.52) \times 0.4 = 27$ cfm (return side leakage)

Estimated total MA flow is 1911 cfm + 27 cfm = 1938 cfm.

Using another measurement technique, the building as a capture tent method, the make-up and exhaust air flow were measured. First the normal operating building pressure is measured with all equipment on. Then a calibrated blower door fan is used to measure the flow of the exhaust by operating the blower in an exterior doorway at a speed that reproduces the normal operating pressure of the building. The exhaust fan is turned off and exhaust fan openings are sealed during the measurement. The MA flow measured was 1970 cfm which is reasonably close to the 1938 cfm measured using the flow-hood and estimated return leakage.

The exhaust flow rate was measured and found to be 1135 cfm, an increase of 19% from 954 cfm measured prior to the retrofit. The target MA amount of 80% of the exhaust air could not be reached until 85% of the total intake grill area was sealed. Figure 1 shows test results from the MA unit using the original motor. (The airflow was measured using a flow-hood over the intake grills and does not include any return leakage in the unit.)

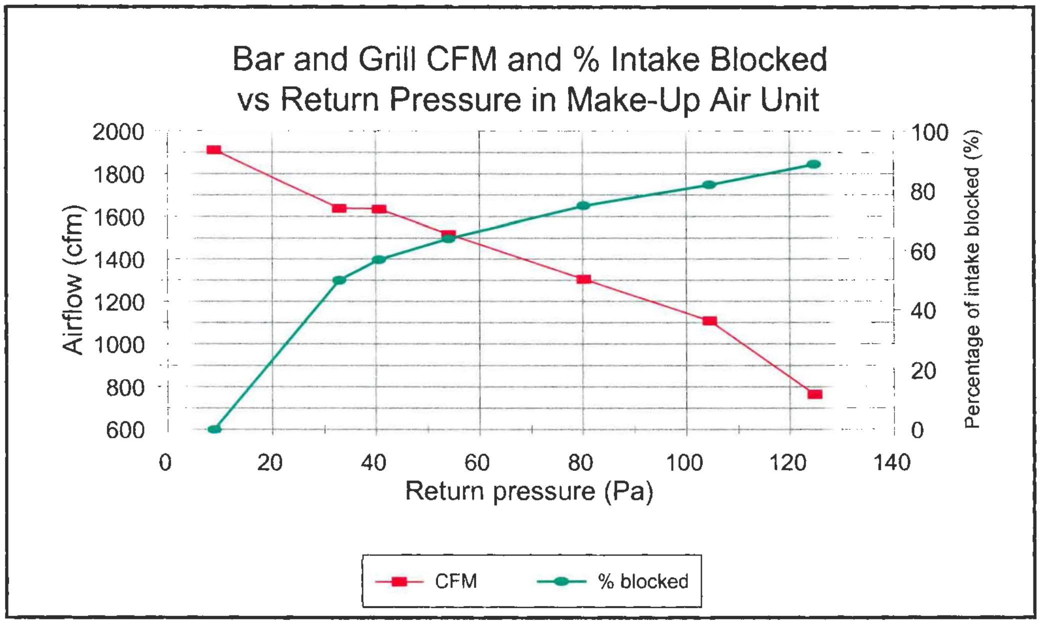

Figure 1 Airflow vs pressure on return side of MA unit

Sealing off 85% of the total intake grill area to get the correct amount of airflow indicated that the current ½ horsepower motor was oversized and that using a smaller motor could still deliver the airflow needed and use less energy. A replacement motor was specified and the original motor was replaced by the contractor. Follow-up measurements on February 26 found that the airflow had not declined because the new motor had lesser horsepower (⅓ horsepower) but the same rotational speed (1725 rpm). Finally, after installing a ⅓ horsepower, 1140 rpm motor, we found that we could achieve the target airflow of about 933 cfm through the intakes with 73% of intake grills sealed off.

Since there are duct leaks, the delivered MA does not equal the measured airflow at the intake. Return leak airflow must be added to the intake amount and the supply leak amount must be subtracted. Using the previous calculation for estimating duct leakage and the measured operational duct pressures, the delivered MA into the space can be calculated.

Return leakage	=	$(30.3^{0.57} \times 19.52) \times 0.4$	= 55 cfm
Supply leakage	=	$(25.8^{0.57} \times 19.52) \times 0.6$	= 75 cfm
Delivered airflow	=	933 + 55 - 75	= 913 cfm

Delivered airflow is approximately 80% of the 1135 cfm exhaust airflow.

Table 3
Summary of the characteristics of the single phase motors used for MA unit.

Motor	Horsepower	RPM	Volts	Amps
Dayton	½	1725	115	7.6
General Electric	⅓	1725	115	6.2
Magnetek	⅓	1140	115	4.3

The restaurant owner commented that he liked the smaller motor observing that it was more quiet.

Energy Savings from Installing a Smaller Motor

A Dranetz power platform 4300 analyzer was placed on the specified Magnetek motor to measure power consumption. The measured power factor was 0.50 with 409.5 volt amps and measured wattage of 204.7. The actual power consumption of the original Dayton motor was not measured, however if a power factor of 0.50 is assumed, then an estimate can be made for the energy saved from using the smaller motor. The VA of the Dayton motor would be about 115V x 7.6A = 874 VA. Using 0.5 power factor would result in 437 watts. The make-up air fan is in use about 12.5 hours per day six days a week and the energy and cost per day in Table 4 below reflects this. Sunday use is much more sporadic and is not included in this analysis.

Table 4
Energy and dollar savings from reducing size of motor on MA unit

Motor	Volt Amps (VA)	Power (Watts)	Energy per day (kWh)	Cost per day @$0.08 / kWh
Dayton	874	437	5.463	$0.437
Magnetek	409.5	204.7	2.559	$0.205
Difference	464.5	232.3	2.904	$0.232

The smaller motor may be saving as much as 53% of the original MA motor energy. Using the average operation of 12.5 hours per day six days a week less 3 days for non-operational holidays, the estimated yearly energy savings are 900 kWh. Assuming electricity cost of $0.08 / kWh, projected savings are $72 per year.

Energy Savings from Installing Make-Up Air

Analysis was performed to assess energy savings from installation of make-up air for the kitchen exhaust hood. Energy monitoring of the MA retrofit impact began with the pre-retrofit period in September 1997 and continued through August 24 1999 when the post-retrofit period concluded. Unknown to us, a change occurred in the building that caused significant impact on our analysis. Late in the project, we discovered that a large refrigeration unit with sliding glass doors had been installed in the kitchen sometime after October 6, 1998. This new unit would inject considerable heat into the building. In order to overcome this problem, we turned the MA unit on and off during periods of 1999 in order to facilitate accurate energy savings comparison.

Figure 2 shows the building total cooling energy used versus the daily average temperature difference between indoors and outside. Regression analysis was used to create a linear equation that best represents the data and is shown as a solid line. The red line represents cooling energy used without the MA unit operating. The light blue line represents cooling energy used with the MA unit operating. A third solid line (dark blue) is shown for the MA on data that represents the cooling energy used plus the added energy to operate the make-up air fan. Based on monitoring data, the exhaust and MA operate an average of 12.5 hours per day. Using the measured power of the MA motor, 204.7 watts, over 12.5 hours results in 2.6 kWh/day of MA fan energy use.

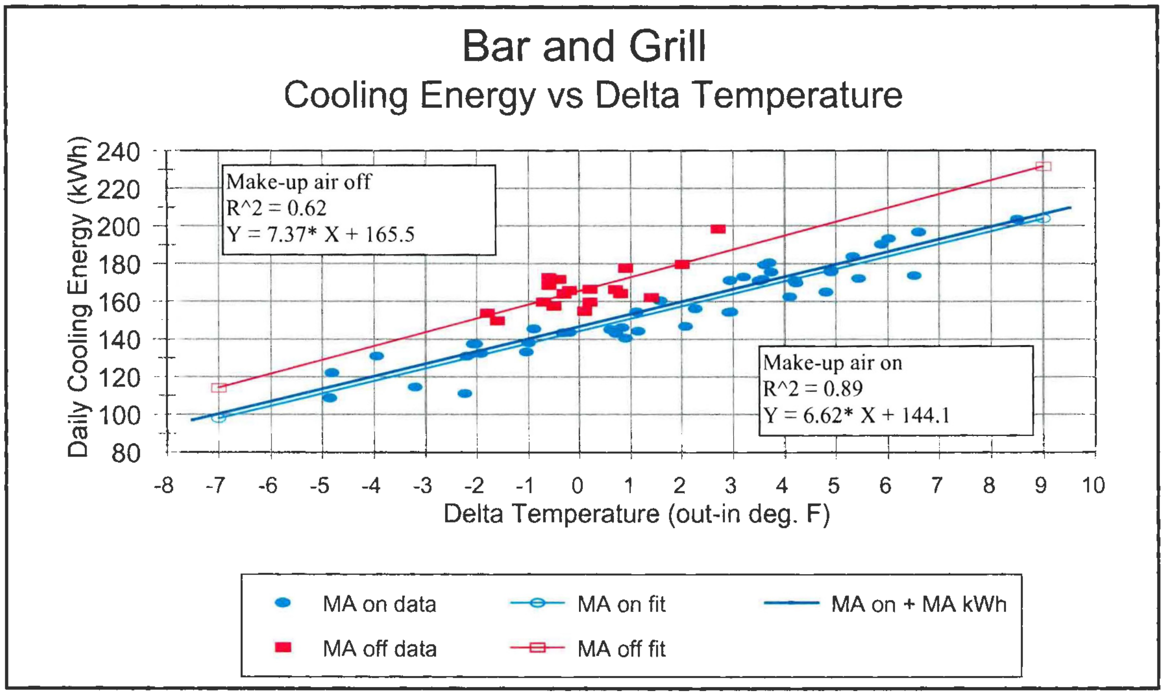

Figure 2 Cooling energy used before and after make-up air retrofit

Data was screened so that compared periods have similar weather and occupancy factors represented. Occasionally, exhaust or MA fans were left on all night or air conditioner thermostat settings were changed. Such unusual operation days were also eliminated from the analysis data.

During most of the winter period (November through March), little cooling is used, and the building is often ventilated with open doors much of the time. Therefore cooling energy analysis uses an eight month period from March through October to calculate the energy savings for a typical cooling season.

Yearly energy savings from the MA retrofit were calculated for an eight month period by means of the best-fit lines in Figure 2 and TMY2 weather data for Tampa (Tampa has the closest climate of the possible choices of TMY2 weather cities). The 2.6 kWh/day of MA fan energy use was taken into account in the calculation and was subtracted from the measured cooling energy savings.

Energy savings were calculated using TMY data to represent meteorological conditions for a typical cooling season and the linear equations obtained from regression analysis. The daily average indoor temperature was subtracted from the daily average TMY outdoor temperature to get the daily average delta temperature. Then the delta temperature was used with the linear equations to calculate a daily total cooling energy for each day of the year. Since the exhaust equipment was seldom used on Sundays, Sundays were excluded in the energy savings calculations. The cooling energy with the MA on was subtracted from the cooling energy with MA off to get a daily cooling energy savings (Table 5).

Table 5
Energy and dollar savings over an eight month cooling season from installing MA.

	Make-up off	Make-up on	Savings over 8 months	Percent savings
Cooling energy (kWh)	32,893	29,113	3,779	11.5
Energy cost @ $0.08 /kWh	$2,631	$2,329	$302	11.5

Retrofit Cost and Simple Payback.

The total MA retrofit cost, including adjustment of the exhaust flow rate, was $3500. Under the assumption that the MA system alone was $3300, then seasonal savings of $302 indicates that this retrofit has a simple payback of 11 years. Roofing work and the very restrictive work environment in the attic helped make the costs more in this restaurant than may be found in many chain type restaurants that already have roof curbs available or perhaps even unused or inoperative MA units.

The bar and grill did not use any space heating during the entire monitoring period. Monitored heating appliance energy and supply air temperatures confirm this. Even during one of the coldest winter days on January 6, 1999, there is enough waste heat in the building from refrigeration and heating appliances that the heating systems were never activated. The outdoor temperature on this day ranged from a low of 33.5 F to a high of 61.2 F with a daily average temperature of 46.7 F. The dining room temperature ranged from 61.2 F to 73.9 F with an average of 66.7 F and conditions in the kitchen ranged from 67.5 F to 87.1 F averaging 77.8 F.

Air Tightening and Insulation Retrofit at Bar and Grill

The second retrofit was improvement of the thermal and air boundaries of the building. Testing and visual inspection identified significant problems with the insulation system and with the airtightness of the building envelope.

Major deficiencies were identified in the thermal barrier of this restaurant on the south side of the attic. R-19 batt insulation attached to the trusses had fallen in several places and insulation on the ceiling areas over the kitchen had been kicked around leaving a void of about 60% on the south half of the attic. Replacing the insulation batts would likely be a temporary repair since they must be removed to allow access to service the air handlers which are located in the attic, and service personnel in the past have not replaced them.

Major air leakage pathways were identified from the south side of the attic into the building. Air tightening and insulating the area over the bar (Figure 3) would have been difficult and time consuming due to many obstructions and there were many recessed lighting fixtures in this area that could not be air tightened or insulated for safety reasons. The bar runs east to west with the south attic section directly above and behind it. The kitchen area is behind the bar and has a horizontal drywall ceiling 8 feet above the floor.

Figure 3 Ceiling over bar area was very leaky and poorly insulated.

An insulating foam sealant, Icynene™, was sprayed on the bottom of the roof deck as part of this project. Vents at the eave and gable were also sealed by the spray foam application. With this retrofit, the air boundary and the thermal boundary would both be located at the roof deck. In other words, this allowed a thermal barrier to be placed at the air barrier that would not have to be moved to allow service access to equipment. The retrofit began at 4 AM August 24, 1999 and was completed by 7:30 AM on the same day. The retrofit involved only the south side attic since the north side was well insulated and had no significant air leakage problems. A fabric material was stapled across the vertical truss members located in the center of the attic to separate the north and south sides. This created a backing material on which the Icynene™ could be sprayed. The insulation was then sprayed from the top of the sheet rock ceiling (attic floor) up on to the vertical backing, and continued to the plywood decking of the roof (Figure 4). The roof deck was covered all the way down to the eave soffit vent. The ridge vent was isolated from the south attic but not sealed up. It was left open to vent the north side attic.

In Figure 4, the vertical section is the constructed wall in the middle of the attic that separates the north and south sides. The supply duct is on the west air conditioner system and runs from the air handler (not in picture) on the left side to the north side attic where it branches off to the supply registers. Large voids in batt insulation can be seen in the bottom of the picture. Overall the insulation ranged in depth from about 4- 8 inches thick, with average depth of about 4½ inches.

Figure 4. View of attic air / thermal retrofit with void in original batt insulation shown at the bottom of picture.

Airtightness Test Results

A building air tightness test was conducted September 8, 1999 after the air / insulation retrofit. This building is still not very tight. However, building tightness improved 21 % as a result of this retrofit as is shown in Table 6.

Table 6
Building air tightness and normal operating pressure before and after insulation retrofit

	CFM50	n	C	r	ACH50	dP, all equipment on
Before Icynene	6496	0.58	684.0	0.9978	17.1	-0.2
After Icynene	5120	0.58	524.02	0.9989	13.5	-0.3
Difference	1376	-------	-------	-------	3.6	-0.1

Energy Savings from Air Barrier/thermal Barrier Retrofit

Cooling energy analysis was performed in a manner similar to that described in the MA energy savings section. Note that the "MA on" line in Figure 5 represents only cooling energy use and does not include MA fan energy use. Cooling energy savings for an eight month season are summarized in Table 7. This analysis includes Sundays since the air conditioning is controlled by thermostats at fixed settings 24 hours a day 7 days a week. This explains why the pre-insulation "MA on" energy use for 8 months (Table 7) is higher than in the "MA on" energy use shown in Table 5. During pre and post insulation monitoring, the MA unit was operated simultaneously with the exhaust fan throughout both monitoring periods.

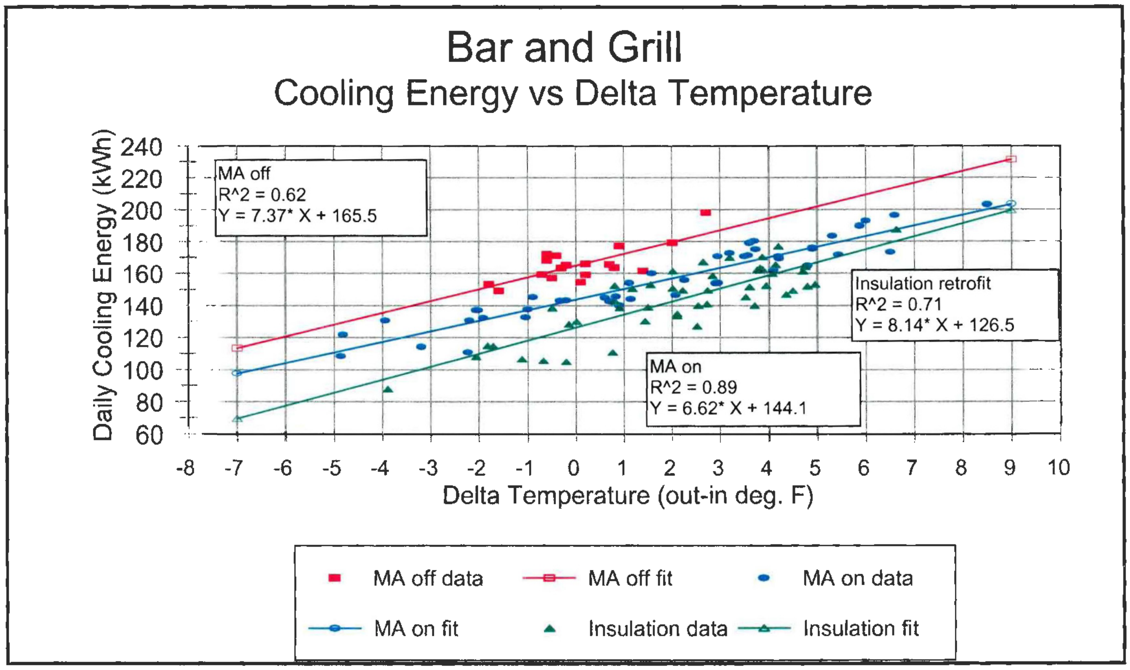

Figure 5 Cooling energy used before and after MA and insulation retrofits

Table 7
Energy and dollar savings over an eight month period from improved airtightness and insulation retrofit.

	Make-up on pre insulation	Make-up on post insulation	Savings over 8 months	Percent savings
Cooling energy (kWh)	34,007	29,378	4,629	13.6
Energy cost @ $0.08 /kWh	$2,721	$2,350	$370	13.6

Cost and Simple Payback

The total air and thermal boundary retrofit cost was $1974. Seasonal savings of $370 indicates that this retrofit has a simple payback of 5.3 years.

Comparison of Zonal Environment Conditions

One very important result from the study carried out in this building was that the owner and two employees gave unsolicited comments stating they noticed thermal environment improvements after the insulation retrofit. Tables 8-11 show significant improvement in indoor conditions after retrofits. Four to six typical summer days from each monitoring period in 1999 were chosen with similar outdoor conditions during 24-hour (Tables 8 and 9) and 3-7 PM peak periods (Tables 10 and 11).

In Tables 9 and 11, the "E. zone" conditions are in the east side of the kitchen near the cooking appliance area and the "W. zone" conditions are in the west side of the kitchen near a food preparation area that has three large refrigeration units (with internal heat discharge). In the attic, the upper conditions are 3 feet down from the roof deck and the lower temperature is 5 feet down from the roof deck, which is also in the vicinity of the east air handler . Solar T is an indicator of solar energy measure using a thermocouple on the roof surface.

Table 8
Daily conditions: 24-hour average for outdoors and indoors (dining room)

	Outside			Indoors				
	Out T	Out RH	Solar T	In T	In RH	Bar T	CO_2 (ppm)	dP (pa)
Pre MA	80.0	70.4	95.7	78.7	52.4	80.5	978	-1.4
Post MA	79.6	69.6	94.7	76.9	50.9	78.5	1211	-0.3
Post Insulation	79.5	38.1	93.6	76.3	53.2	77.9	1143	-0.2

Table 9
Daily conditions: 24-hour average for kitchen and attic

	Kitchen			Attic		
	E. zone T	E. zone RH	W. zone T	Upper T	Upper RH	Lower T
Pre MA	88.2	33.7	86.6	91.7	47.2	88.3
Post MA	84.4	36.6	85.1	88.8	49.3	84.9
Post Insulation	84.9	35.9	85.3	83.4	52.1	81.3

Table 10
Peak conditions: average for outside and indoors (dining room) for 3 -7 PM

	Outside			Indoors				
	Out T	Out RH	Solar T	In T	In RH	Bar T	CO_2 (ppm)	dP (pa)
Pre MA	82.8	66.3	109.8	83.9	53.6	85.7	650	-2.3
Post MA	82.8	64.1	108.3	79.3	53.0	80.8	840	-0.3
Post Insulation	82.7	43.3	105.3	79.0	54.3	80.6	1014	0.0

Table 11
Peak conditions: average for kitchen and attic for 3 -7 PM

	Kitchen			Attic		
	E. zone T	E. zone RH	W. zone T	Upper T	Upper RH	Lower T
Pre MA	92.6	34.9	92.1	106.2	31.8	99.0
Post MA	87.6	38.6	88.8	102.5	33.2	94.7
Post Insulation	87.8	37.2	89.4	90.5	44.9	86.6

The data summarized in Tables 8-11 point to significantly more comfortable indoor conditions. Kitchen temperatures dropped slightly following retrofits. 24-hour kitchen temperatures dropped by 1.3 F in the west end and 3.3 F in the east end (near the cooking appliances). During the peak period, kitchen temperatures declined even more, by 2.7 F in the west end and 4.8 F in the east end. This may be partially attributed to the extra supply that was added when the MA was installed. Conditions in the dining area also improved. 24-hour dining room temperatures dropped 2.4 F. During the peak period, dining room temperatures declined even more, by about 4.9 F in the main area and about 5.1 F at the bar. *We may conclude, therefore, that not only is there approximately a cumulative 25% reduction in total cooling energy use as a result of both retrofits combined, there is also a substantial improvement in the quality of indoor conditions.*

Energy Savings Conclusion

Additional savings could have been observed if there had been more air conditioning capacity. This restaurant had 3.3 tons of cooling capacity per 1000 square feet compared to an average of 5.5 tons/1000 square feet found in a group of 10 buildings with large exhaust systems (Cummings et al 1996). This is 40% lower than the average of the larger group. There was no measured reduction in air conditioning energy use during peak period from 3 - 7 PM. This can be attributed to a cooling capacity that is too small to maintain 76 F indoors. While there was no measured reduction in peak energy usage, interior conditions improved significantly due to retrofits indicating a reduction in cooling load. This improvement in temperature resulted in improved comfort as indicated by employees.

Although the cooling capacity was much lower than in other restaurants, providing 80% MA resulted in calculated energy savings of 11.5% for a typical 8-month cooling season, and insulating and air tightening half the attic space resulted in 13.6% savings with simple paybacks of 11.6 and 5.3 years, respectively. The improved temperature conditions in the kitchen, the dining room, and at the bar no doubt has considerable economic value. Turnover among employees is likely to decline. Customers are more likely to be content, and therefore come more often and stay longer. It is, of course, beyond the scope and ability of this project to put a economic value on those improvements.

Charity Office/Thrift Shop

Three retrofits were implemented at this building.

Building description: metal building (metal skin walls on three sides and roof, fourth wall is block with stucco) with suspended t-bar ceiling and R-19 insulation batts on the ceiling tiles. One 5-ton and one 2.5 ton air-conditioner serve this approximately 3700 square foot facility which houses office space, a thrift shop retail floor, and inventory storage and preparation. The return plenums (enclosed support platforms) and the air handlers for both systems are located in a fairly large storage room. The return air (grill) for the 2.5 ton system was located in this storage room. The return grill for the 5 ton system was located in a hallway that is open to nearly the entire building. The supply air ductwork (ductboard mains with flex duct branches) was located in the unvented ceiling space above the insulation (R-19 insulation batts lie on the ceiling tiles) and below the essentially flat metal roof deck. R-3 insulation was also located at the roof deck.

Problems noted:

- the building could not be kept cool on summer days. On many days, indoor temperatures would exceed 85F in the late afternoon.
- utility bills were very high.

Several factors contribute to these problems.

1) The metal building envelope gets quite hot in the summer sun and conducts considerable heat into the space. The ceiling space routinely reaches 120 degrees F on summer afternoons, in spite of the fact that there is insulation at the roof deck and unbalanced return air cools that space by pushing conditioned air into it. Wall insulation seems to be fairly ineffective because of thermal bridging.

2) Return air for the 2.5-ton system is located entirely in a (fairly large) storage room that can be closed and is often closed for inventory security reasons. As a consequence, this room is depressurized (about -4 pascals) with respect to the hallway. The room has a suspended t-bar ceiling which is very leaky, and as a consequence about 600 cfm of air is drawn from the hot ceiling space through the ceiling. Simultaneously, room air from the main zone of the building is pushed into the ceiling space. We estimate that this storage room is closed 60% of the time.

3) Return air for the 5-ton system is located entirely in the main retail hallway. An office suite, containing three offices and a small internal hallway can be closed off from the main hallway.

Since there is no return air to this suite, it operates at positive pressure when the suite doors are closed. Even if a suite door is open, individual offices within the suite operate at positive pressure when their doors are closed. This positive pressure pushes air through the ceiling, and air is then drawn from the ceiling space into the hallway area in order to return to the return grill. We estimate that this office suite is closed 30% of the time, and that individual offices are closed another 10% of the time when the suite door is open.

4) There are significant leaks in the air distribution system.

5) Total lighting energy consumption is currently about 4700 watts or 42 kWh each 9 hours per day (32 light fixtures consuming 106 watts (2 lamps) each and 7 light fixtures consuming 186 watts (4 lamps) each).

6) There are 165 square feet of west facing glass with essentially no shading (there are no other windows in the building). Under bright sunshine conditions during the afternoon, this represents about 50,000 Btu/hr of cooling load from 3 PM to 6 PM (6 PM is when the AC is turned off for the day).

7) There are a total of seven refrigeration units (freezer and refrigerator/freezer units) located inside the conditioned space. These are used to store food for their food distribution services. Most of these units are 10 to 20 years old and are therefore less energy inefficient than current models.

Retrofits were developed to address items 1, 2, 3, and 4.

The first retrofit was to change the reflectivity of the building envelope. In July 1998, the roof and exterior walls were painted white. The roof paint was an elastomeric roof coating material. Before the retrofit, the manager explained that there were roof leaks occurring so the retrofit needed to change the reflectivity as well as address tightening the roof where needed. The retrofit involved:

- an acid wash and pressure cleaning
- tightening fasteners and replacing bolts as necessary
- application of rust inhibiting metal primer
- caulking of fasteners
- application of 2 coats of solar reflective paint

The second retrofit, to solve unbalanced return air problems, was implemented at this site in October 1998. Unbalanced return problems existed for both air-conditioning systems, however, by far the largest problem was with the 2.5-ton system. The retrofit involved two elements. 1) A second return grill was installed for the 2.5-ton system in the hallway. 2) The return for the 2.5-ton system located in the storage room was constricted by means of a template inserted behind the filter at the grill so that the return air in that room matched the supply air in that room (this took room pressure to neutral with respect to the hallway). Regarding the 5-ton system, we had considered installing a return transfer duct in the office suite to allow return air to pass from the office suite to the hallway where the air-conditioner return is located, when the office suite is closed. However, after monitoring the status of the office suite door (using a door closure status switch), it was determined that the office doors were not closed a sufficient fraction of the time to make the retrofit worthwhile. Furthermore, most of the time when the suite was closed was from 3 AM to 8 AM. This is when the air conditioners are manually turned off and the ceiling space would be at its coolest, and consequently the energy impacts of door closure were considered to be relatively small under these circumstances.

The third retrofit was repair of leaks in the duct system. The majority of the leaks were located at the connections between the flex duct and the main ductboard ducts and at the connections of the flex duct to the diffusers. Because the inner liner of the flex ductwork would essentially fall apart when repairs were attempted, we were able to only seal about 30% of the total leakage.

Analysis of Energy Savings

Retrofit One – White Roof and Walls.

Monitoring of energy use in this building began May 4, 1998. The roof was painted white on July 10 (elastomeric roof coating material) and the exterior walls were painted white on July 26. There were no measured cooling energy savings from changing the color of the roof and exterior walls. However, a significant drop in indoor temperature indicates that cooling energy savings would have occurred if the AC systems could have met the cooling load prior to the retrofit.

Figure 6 compares temperatures before and after the exterior walls and roof were painted white. Days with similar outdoor temperature, relative humidity, solar energy as well as similar indoor operating conditions were chosen for the comparison. The sharp dip and rise in the roof deck temperature before paint was applied shows the rapid response of the unpainted corrugated metal roof has to sunlight. When the solar radiation upon the roof would change significantly, such as from very sunny to very cloudy, the roof deck temperature could be observed to drop every 10 seconds as data was collected by the datalogger.

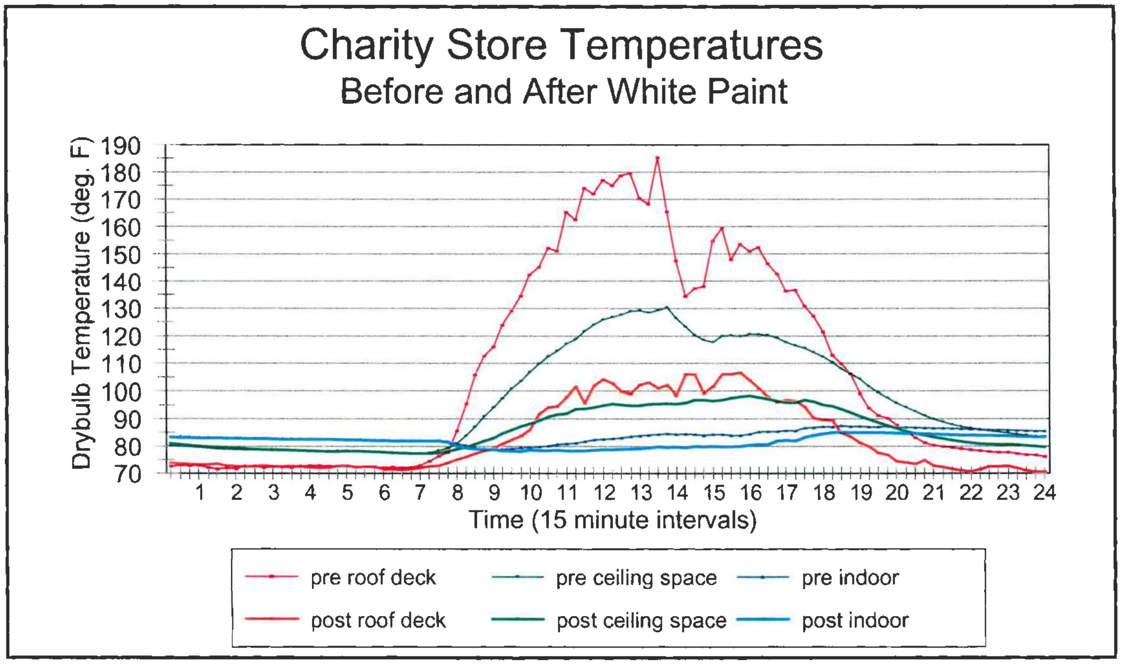

Figure 6 Charity thrift store temperatures for one day before and one day after the building was painted white.

Prior to the white wall and roof painting, the cooling systems would run non-stop for the entire business day. At about 7:30 AM each day, the manager would turn on the air-conditioning units. With the room temperature at about 82 F and the thermostats set at 76 F, the AC units would run 100 % of the time. By 9:30 AM, the interior temperature came down about 3 F degrees, but after this the load on the building would increase beyond the air-conditioner capacity and the interior temperature would begin to rise. Interior temperatures would continue to rise above 87 F a few hours after closing. Prior to beginning monitoring, we had the air-conditioning units serviced. Therefore, we believe that the air-conditioning units were operating correctly and were not the reason for rising temperatures indoors during the day.

After the white coating retrofit, the air-conditioning systems ran the entire day also. The air-conditioning systems did not have sufficient capacity to cool this building, and therefore there was no reduction in cooling energy use.

One factor which made detection of energy savings more difficult was the extreme nature of the summer weather. Starting the third week of May 1998 and continuing into August, central Florida had its hottest three month period ever. In Melbourne, Florida (30 miles south of this building's location), 21 of the 30 days of June either tied or broke record high temperatures. This was also the summer of wide-spread forest fires up and down the east coast of Florida. Because of the persistent heat, there were few cooler days when cycling of the air-conditioning systems would be more likely. As a consequence, the air-conditioning systems ran continuously during most business hours of the summer of 1998.

Zonal Environment Condition Comparison

While the first retrofit did not cause a reduction in cooling energy use, it did produce a reduction in indoor temperatures. Table 12 compares thermal conditions in various zones before and after the paint retrofit for business hours. Days with similar summer-like meteorological conditions were chosen for comparison (notice the similar outdoor temperature and solar radiation indicator values). Another important consideration was similar use and activity in the store. The amount of time and the time of day that interior doors were closed were also considered.

Table12
Average indoor and outdoor conditions before and after paint retrofit

	Outside			Inside				
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	Roof deck T
Pre paint	94.6	53.8	5870	83.4	52.2	114.0	26.8	145.9
Post paint	94.7	55.1	5961	80.2	53.5	91.9	51.7	94.7

Note: Conditions during normal business hours.

Over the 7:30 AM to 5:00 PM business day, average indoor temperature declined from 83.4 F to 80.2 F, a substantial 3.2 F decrease in indoor temperature. Since the thermostats were set to 76 F, the decline in temperature is a function of the decline in total cooling load, and this was directly related to a dramatic decrease in the exterior surface temperature of the building envelope. Roof deck average temperature from 7:30 AM to 5 PM declined by 51.2 F As a result of lower roof deck temperature, average temperatures in the ceiling space cooled down 22.1 F during the same period. Building staff reported improved comfort after the white building retrofit.

Table13 shows change in indoor conditions during peak hours (3 PM to 5 PM) before and after paint retrofit.

Table 13

Peak period (3 PM to 5 PM) average conditions before and after the paint retrofit.

	Outside			Inside				
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	Roof deck T
Pre paint	98.2	46.9	844	86.2	48.1	119.8	15.8	148.6
Post paint	99.1	44.3	1106	82.0	50.5	97.2	38.3	101.6

Over the 3 PM to 5 PM peak period, average indoor temperature declined from 86.2 F to 82.0 F, a substantial 4.2 F decrease in indoor temperature. Roof deck temperature from 3 PM to 5 PM declined by 47.0 F As a result of lower roof deck temperature, average temperatures in the ceiling space were reduced 22.6 F. Notice that these declines in roof, ceiling space, and indoor space temperatures occurred in spite of the fact that the post retrofit days were hotter and had much higher levels of solar radiation.

Cost and simple payback

The total retrofit cost was $3639. Preparation and painting of the roof cost $2289 and painting the exterior walls cost $1350. The majority of the expense was for the labor involved in cleaning, priming, and applying two coats of an elastomeric white paint to the roof. There were no measured cooling energy savings, so a simple payback can not be calculated. Some research exists, however, indicating that for each degree Fahrenheit increase in thermostat setting there are cooling energy savings of about 8% (Parker et al., 1996). Parker found that in 20 low-income homes, a 1 degree Fahrenheit increase in thermostat setpoint caused a 14% increase in cooling energy use. In another study, computer modeling was used to characterize cooling energy use in single family homes as a function of room temperature. Decreasing thermostat setpoint by 1 degree Fahrenheit increased cooling energy use by about 13% in Orlando, Florida (Fairey et. al., 1986).

In another study, cooling energy use was characterized in 17 central Florida small commercial buildings (Cummings et. al., 1996). Daily cooling energy use was compared to delta-temperature (temperature difference between indoors and outdoors). For each 1 degree Fahrenheit decrease in delta-temperature (which can also be thought of as 1 degree Fahrenheit increase in room temperature for a given outdoor condition), cooling energy use declined by an average 8.2%. The same analysis was performed for this building (the Charity Office/Thrift Shop) based on daily cooling energy use versus outdoor-minus-indoor temperature difference (see Figure 2 in a following section of this report). For each degree increase in indoor temperature, there is a 7.5% decrease in cooling energy use.

For purposes of the following discussion, cooling energy use is assumed to decrease by 8% for each degree of indoor temperature increase (other factors being held constant). (Note that this can also be stated conversely, that cooling energy use is assumed to increase by 8.7% for each degree of indoor temperature decrease.) The 7:30 AM to 5 PM indoor temperature decrease caused by the paint retrofit was 3.2 F. This decrease in indoor temperature did not occur as a result of lowered thermostat setting or increased cooling output. Rather the temperature decrease occurred as a result of reduction in cooling load.

The question that we want to examine is: "What amount of additional cooling energy use would be required to decrease building temperature by 3.2 F in the absence of the retrofit?" Since the evidence suggests approximately 8.7% increase in cooling energy use for each one degree Fahrenheit decrease in indoor temperature, we project that an increase in cooling energy use of 28% would be needed to affect that indoor temperature reduction. Therefore, if the building cooling systems had been sufficiently large so they could maintain 76 F indoor temperature prior to any retrofits, then we would expect cooling energy savings of 22% as a result of application of white roof and wall coatings. [Note that the energy savings would be 28% of the current measured cooling energy use or 22% of the higher cooling energy use which would have occurred if the cooling systems had been much larger.] This indicates approximately 6050 kWh of cooling energy savings potential from the white paint retrofit. At $0.08/kWh, the savings of $484 per year yields a 7.5 year simple payback period. Note that in addition to potentially reducing cooling energy use, the white roof coating also will extend the life of and improve the water tightness of the roofing system.

Retrofit Two – Balancing Return Air

In order to more clearly observe the impacts of the return air retrofit, we adjusted the original air-conditioner operation schedule from 7:30 AM - 5:00 PM to3 AM - 6 PM. The earlier start to the cooling day would give the cooling systems greater opportunity to meet building cooling load later in the day, and therefore there would be a greater opportunity to see variation in cooling energy use as a function of reduction in building cooling load. Control over the schedule was provided by the monitoring datalogger.

Balancing the return air for the 2.5-ton system was completed October 21, 1998. Implementation of this retrofit was delayed well into the autumn because we wanted sufficient cool weather prior to retrofit in order to obtain days in which the air-conditioning systems cycled on and off. Data was screened to make sure that days with unusual events that might skew results were not used. For instance, we compared the amount of time that the storage room doors were closed (we used door closure detectors) and even the time of day for that closure, to make sure that both the weather and building use parameters were comparable.

Analysis of cooling energy consumption data predicts cooling energy savings of 14.8% as a result of this retrofit for a typical 8 month cooling season. Figure 7 shows the building cooling energy use impacts of the return balance retrofit on the 2.5-ton AC unit.

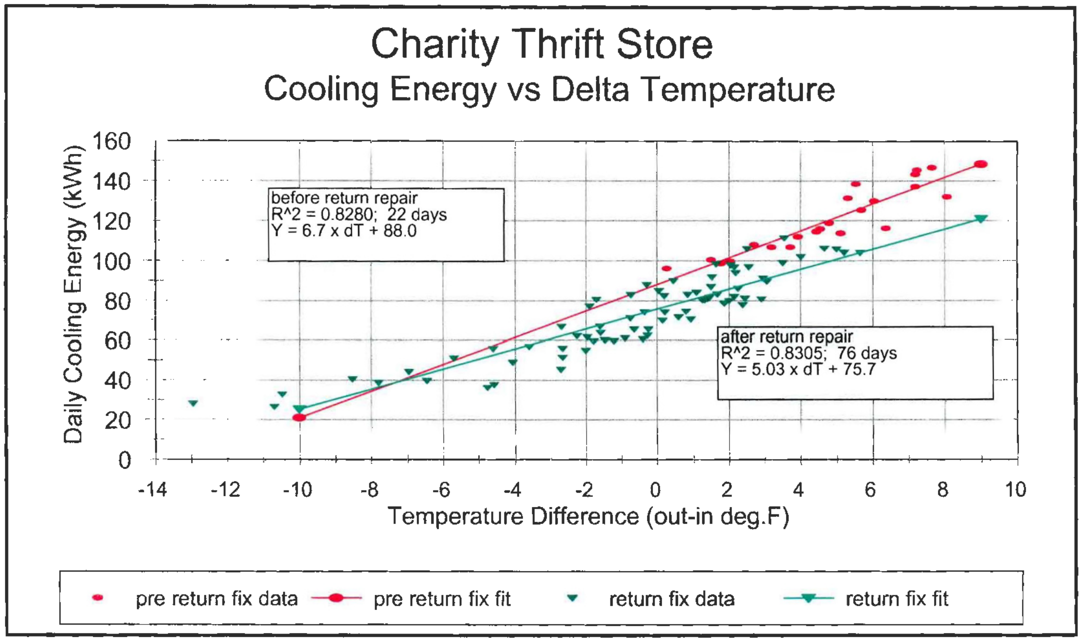

Figure 7 Cooling energy before and after return air balance repair to 2.5 ton unit

Zonal environment conditions

In addition to cooling energy savings of 14.8%, indoor temperature during business hours also declined by 1.8 F (Table 14). Days with similar summer-like meteorological conditions were chosen for comparison. In selecting days for comparison, similar use and activity in the store was considered. The amount of time and also the time of day that interior doors were closed was considered. Notice that "% door closed" is nearly identical for the pre-retrofit and post-retrofit days selected for analysis.

Table 14

Average business hours conditions outside and inside before and after return retrofit

	Outside			Inside					
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	% door closed
Pre return	90.3	39.4	5270	77.6	39.4	85.2	50.8	43.6
Post return	90.3	41.9	5885	75.8	41.9	88.7	48.9	43.4

Since the thermostats were set to 76 F, the average indoor temperature of 75.8 F post-retrofit indicates that the AC systems can meet cooling load throughout essentially the entire day. Prior to the retrofit, the average temperature of 77.6 F indicates substantial periods of the day when indoor temperature drifts upward due to load to cooling-capacity imbalance. Notice that the 1.8 F reduction in indoor temperature occurred while the ceiling space temperature was 3.5 F warmer than the pre-retrofit period. It is not clear whether the cooler ceiling space temperature was the result of slightly lower solar radiation levels or the unbalanced return air being pushed into the ceiling space prior to retrofit. It may be a combination of both factors.

Peak indoor temperatures (3 PM to 5 PM) declined by 2.4 F (Table 15). The fact that indoor temperatures *after retrofit* were about 1.7 F warmer than set-point suggests that during the peak period the cooling systems could not quite meet load. By contrast, the fact that indoor temperatures *before retrofit* were about 4.1 F warmer than set-point clearly indicates that building cooling load greatly exceeded cooling capacity. Storage room door closure of 100 % is typical during the period from about 1 PM - 8 AM the next day.

Table 15
Average business peak hours conditions before and after return balance retrofit

	Outside			Inside				
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	% door closed
Pre return	92.4	36.3	767	80.1	34.8	88.9	39.4	100
Post return	93.7	35.6	879	77.7	52.5	93.9	35.2	100

Cost and simple payback

Measured cooling energy savings from return balance results in a total calculated savings of 2450 kWh (14.8%) over a typical 8 month cooling season. The total retrofit cost was $100. Seasonal savings of $196 indicates that this retrofit has a simple payback of about 6 months.

Considering that average daily indoor temperature also declined by 1.8 F, we may also project additional cooling savings which would have occurred if the cooling systems had been larger. Assuming 8% decrease in cooling energy use per one degree Fahrenheit increase in room temperature, cooling energy use would have been 14.4% higher if the 76 F setpoint had been maintained during both pre- and post-retrofit periods. Based on this, we can project additional savings of approximately 2384 kWh. Thus there would be total energy savings of 4834 kWh, or 29% of total cooling energy use. Projected annual savings of approximately $387 would mean a simple payback of less than 3 months.

Retrofit Three – Duct Repair

The last retrofit, sealing of duct leaks, was implemented on May 13, 1999. Data collected before and after duct repair shows no energy savings, and in fact shows energy use 8.5% higher after duct repair (Figure 8). If there had been little or no savings, this would not have been a large surprise because relatively little of the total leakage was repaired. As indicated earlier much of the leakage was at flexible duct connections. These connections were originally sealed with fabric duct tape that was wound very tight around the outer thermal liner at the connection collar. This tape was found to be in reasonably good shape and the several windings at each connection were very tight and not easy to remove on several of the connections. When the tape was removed and the outer liner was pulled back to expose the inner liner, the inner duct would start to unravel as it was disturbed. It was made up of a very thin cellophane type of material 2-1/4 inches wide that was spiral wound. After encountering problems with the inner liner coming apart at four connections, it was decided to apply mastic around the outer jacket of flex duct connections. This would not eliminate any leakage from the inner duct to the outside jacket. There were also several air flow dampers on the branch ducts that were leaky. The dampers were constructed of light sheet metal and had the appearance of poor quality fit and finish. These could not be sealed (with mastic) completely without making the damper inoperable. Mastic would seal the damper in place but it would become inoperable in the future.

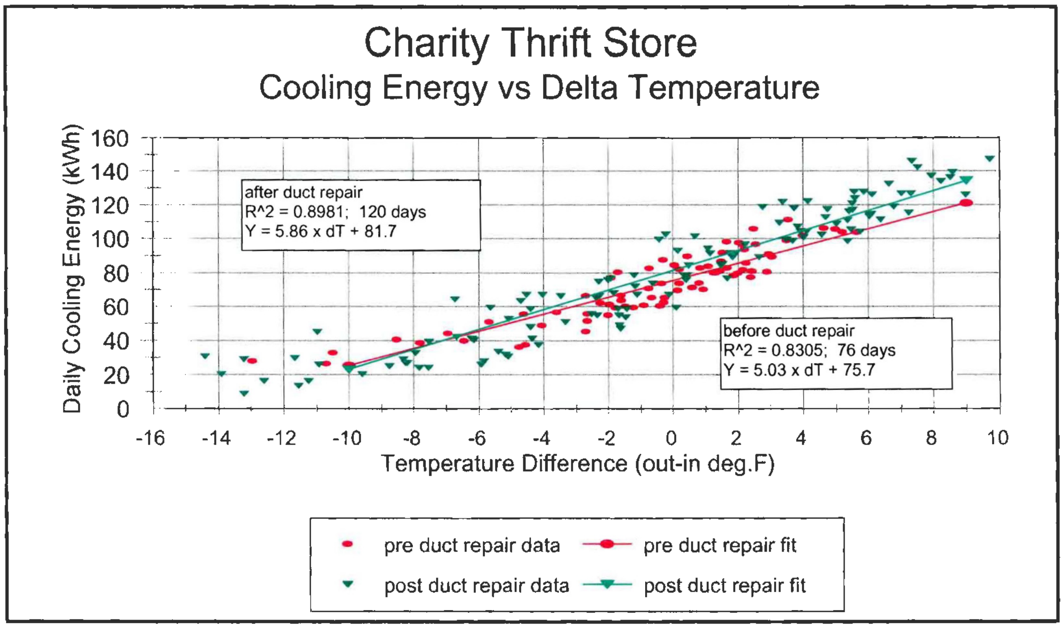

Figure 8 Cooling energy used before and after duct repairs.

The fact that energy use actually increased by 8.5% after duct repair is certainly a surprising finding considering that some duct leakage was repaired. At worst one would expect no increase in energy use. There are two factors, however, which may account for the increased energy use. First, the temperature of the roof deck and ceiling space is about 1 degree Fahrenheit higher after repair (Table 16). Second, and more importantly, outdoor dew-point temperature (as measured by our outdoor temperature/relative humidity monitor) increased substantially from 61.5 F to 69 F from before to after duct repair. Because of the much higher outdoor dew-point temperature, the latent load associated with air infiltration would be much higher during the post-repair period.

Table 16
Average business hours conditions before and after duct repair.

	Outside			Inside				
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	Roof deck T
Pre duct fix	86.0	44.4	5838	75.9	51.7	85.9	46.6	89.4
Post duct fix	86.8	55.5	5332	75.1	53.6	86.7	51.6	90.1

Zone environmental conditions

Indoor temperature declined by 0.8 F after duct repair. Two factors may have caused this temperature decline. There are indications that staff were lowering the thermostat on some days after repair. Also, during some peak load portions of the day, it appears that the cooling system could not meet cooling load prior to duct repair. This is suggested by the fact that peak indoor temperature declined by 1.4 degrees Fahrenheit during the 3 PM to 5 PM period (Table 17).

Table 17
Average business peak hours conditions before and after duct repair.

	Outside			Inside				
	Out T	Out RH	Solar	In T	In RH	Ceiling T	Ceiling RH	Roof deck T
Pre duct fix	90.0	29.8	940	77.9	50.3	91.0	35.9	93.0
Post duct fix	89.7	46.5	991	76.5	51.8	91.5	43.2	95.0

Summary of Retrofit Impacts

Three retrofits were implemented to examine cooling energy savings. Note that there were no heating energy savings because no heat was used during the entire monitoring period. Note also that there were no cooling season peak demand savings because total cooling capacity was sufficiently small so that even after all retrofits, the cooling systems ran at essentially 100% capacity during the 3 PM to 5 PM period.

Retrofit one. Covering the roof and exterior walls with a white coating yielded no energy savings because the two air-conditioning systems could not meet cooling load during any hours of the 7:30 AM to 6 PM work-day, either before or after the white coating. Based on estimates of 8% decline in cooling energy use per degree Fahrenheit increase in thermostat setpoint, the 3.2 degree Fahrenheit decrease in indoor temperature indicates that the white coating retrofit would have yielded cooling energy savings of 28% or 6050 kWh/year if air conditioners had been sufficiently large to maintain the indoor setpoint temperature both before and after the retrofit. At $0.08/kWh, the savings of $484 per year yields a 7.5 year simple payback period.

Retrofit two. Balancing return air for the 2.5-ton air-conditioning system yielded measured energy savings of 2450 kWh over an 8 month season which is 14.8% of the total air conditioner energy used before the return retrofit. Indoor temperature also declined by 1.8 degrees Fahrenheit. As a result of this decline in indoor temperature, we project that an additional 14.4% cooling energy savings would have occurred if the air conditioning systems had been able to meet load both before and after this retrofit. Based on this, we can project additional savings of approximately 2384 kWh. Combining measured energy savings and projected energy savings from indoor temperature reduction, total savings would be 4834 kWh or 29% of total cooling energy use. Projected annual savings of approximately $387 would mean a simple payback of less than 3 months.

Retrofit three. Duct repair produced no cooling energy savings. In fact, the monitored data finds an 8.5% increase in cooling energy use after duct repair. It would be reasonable to expect almost no energy savings since only 30% of the duct leakage was repaired. However, to find an increase of 8.5% seems unreasonable. One variable may account for this unexpected result. Outdoor dew-point temperature was measured to be 69F post-repair compared to 61.5F pre-repair. The higher latent load associated with air infiltration may account for some or all of the increased cooling use.

Measured energy savings were 0.0% for the building/roof white coating, 14.8% for balancing of return air, and -8.5% for duct repair. Combined, this building is only seeing a 7.5% reduction in cooling energy use.

Potential savings were much greater. If this building had been changed in only one way; that is, if it had much larger air-conditioning systems that could meet the indoor setpoint temperature both before and after the retrofit, then we project that energy savings from the three retrofits would have totaled 45%.

Table 18
Measured cooling energy savings and projected cooling energy savings (from reduction in indoor temperature) from three retrofits.

RETROFIT	MEASURED SAVINGS	PROJECTED SAVINGS
WHITE COATINGS	0.0%	28%
RETURN BALANCE	14.8%	29%
DUCT REPAIR	-8.5%	-8.5%
COMBINED	7.6%	45%

If we assume that the 8.5% increase in cooling energy use is an anomaly, and treat the duct repair retrofit as if it produced neither an increase nor a decrease in energy use, then measured cooling energy savings would have been 14.8% and projected savings would have been approximately 50%.

In addition to energy savings, a large improvement in indoor temperature conditions was experienced. 3 PM to 5 PM indoor temperatures declined from about 85F to about 76F. Staff expressed appreciation for the substantial improvement in comfort of the occupants.

References

Parker, Danny S., Maria D. Mazzara, and John R. Sherwin, *Monitored Energy Use Patterns in Low-Income Housing*, Florida Solar Energy Center contract report, FSEC-PF-300, May 1996.

Fairey, Philip, A. Kerestecioglu, R. Vieira, M. Swami, and S. Chandra, *Latent and Sensible Load Distribution in Conventional and Energy Efficient Residences*, *Final Report*, prepared for Gas Research Institute, FSEC_CR-153-86, Florida Solar Energy Center, October 1986.

Polk County Classroom Buildings

The Polk County Florida school district has recently constructed a number of classroom buildings that are designed to be easy to construct, easy to maintain, and energy efficient. Two of these units were selected for testing and monitoring in this project.

The new classroom buildings at Floral Ave. Elementary School (Bartow, FL) and Eagle Lake Elementary School (Eagle Lake, FL) are of similar construction, the only significant difference is in the color of the roof. A white roof covers Floral Ave. (Figure 9) and a blue roof covers Eagle Lake. The buildings are constructed with metal framed wall studs and a metal roof truss system. The exterior wall finish is brick. The standing seam metal roof deck has approximately 1 inch of vinyl coated fiberglass insulation directly beneath. The ceiling is a typical t-bar suspended assembly located about 2 feet below the metal truss rafter system. A layer of radiant barrier foil is attached to the bottom of the trusses and R19 fiberglass batt insulation lies on top of this foil.

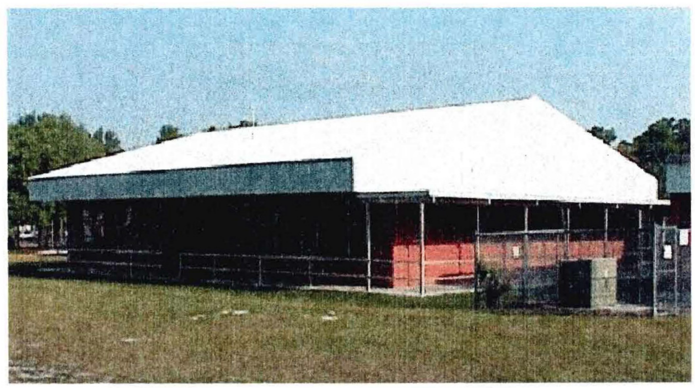

Figure 9 New building at Floral Avenue

Each new classroom building has about 5000 ft^2 of floor area, and the two schools (Floral Ave. and Eagle Lake) are located about 20 miles apart. Each building consists of four classrooms, four bathrooms, and a central teacher's preparation area (Figure 10). This design option allows the school board the flexibility to link the "quads" together in a row to form as many classrooms as needed. Each classroom is equipped with a package terminal heat pump (PTHP) with a thermostat located near the teacher's preparation area. The thermostat is wired in series with a 4-hour "crank" timer that limits operation time of the PTHP unit to four hours without human intervention. Ventilation air is introduced into the classroom space via the PTHP. The manufacturer's specification sheet

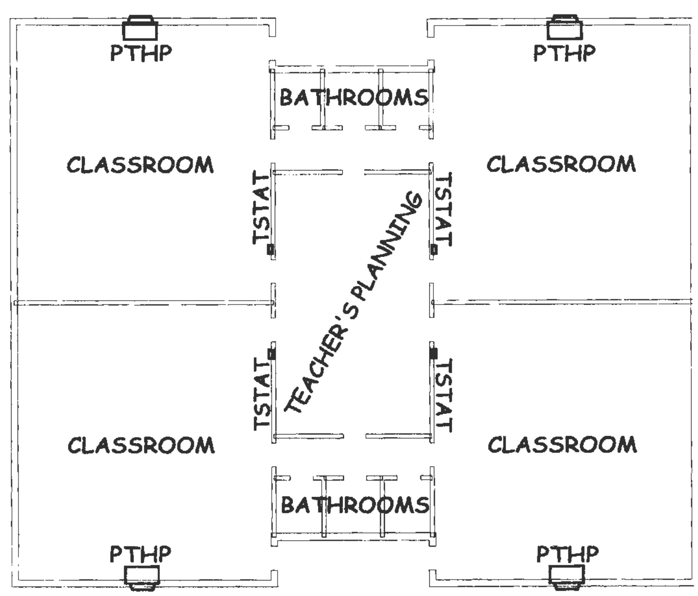

Figure 10 Typical layout of a "quad". The building contains 4 classrooms, centralized teacher planning area and bathrooms. Each classroom has its own heating, cooling and ventilation unit (package terminal heat pump) with teacher controlled thermostat.

indicates that up to 400 cfm of outside air may be introduced to the space when the control damper is in the full open position. Figure 11 shows the inside of one of the classrooms where the PTHP unit can be seen at the back as well as the suspended tile ceiling.

Figure 11 Typical classroom with PTHP unit

Building airtightness test results

A multi-point blower door test was completed on each of the buildings. The testing was completed with the exterior doors and windows closed and PTHP units in the off mode. All of the interior doors that connect the classrooms to the centralized teacher's planning area were open during the test. Test results are shown in Table 19.

Table 19
Building airtightness results

School	CFM50	ACH50	CFM50 / Ft^2	C	n	Comments
Floral Ave	16644	19.9	3.3	1641.9	0.59	As found[1]
	13692	16.4	2.7	1409.8	0.58	PTHP grills sealed[2]
Difference	2952	3.5	0.6	17.7%	reduction	
Eagle Lake	18509	22.1	3.7	1649.5	0.62	As found
	16771	20.1	3.3	1513.8	0.61	PTHP grills sealed
Difference	1738	2	0.4	9.4%	reduction	

1- The exterior doors were closed, all interior doors open, all exhaust fans off and the HVAC units were off.
2- The exterior grill of the PTHP units were sealed off temporarily with an air impermeable cover. This eliminates any air passing through or around the unit from the outside.

Airtightness tests found 16,430 CFM50 (cubic feet per minute at 50 pascals) at Floral Avenue and 17,756 CFM50 at Eagle Lake. This airtightness can also be expressed as 21.9 ACH50 (air changes per hour at 50 pascals) and 23.7 ACH50, respectively. Given that we often recommend that building tightness fall in the range of 4 to 8 ACH50, these buildings are very leaky.

Repeating the airtightness test with the PTHP grills sealed allowed us to examine how much of the building leakage occurs through the PTHP units.

Air and thermal barrier conditions

The fact that the buildings were very leaky prompted the search for the source of the leak. Leak paths were identified at two locations.

- Ceiling plane - consisting of a t-bar assembly, an air space of approximately 18 inches, a metal grid covered with an attached radiant barrier material, and an R-19 fiberglass batted insulation. The perimeter vertical wall section between the t-bar ceiling and the radiant barrier covered grid had a generally poor air barrier, especially over the bathrooms.

- PTHP assembly - the unit is placed into the wall without any seal between the wall and unit. This gap varied in size depending on the rough opening dimensions of the wall and closeness of the unit to the wall, and often represented a cumulative two square foot hole.

Photos in Figures 12 - 14 show some of the failures. The radiant barrier plane (on which the insulation batts were located) was very leaky, most often due to numerous areas where the radiant barrier material had fallen or was not properly installed (Figure 12). Additionally, the vertical exterior wall section above the t-bar ceiling (but below the radiant barrier) utilized the paper backing of the fiberglass insulation batts as the air barrier. The material appeared to be installed fairly well, however, there were numerous gaps that exposed the space between the t-bar ceiling and the radiant barrier to the soffit and attic air spaces as it is shown in Figure 13. The insulation material on top of the radiant barrier failed to provide complete coverage; in fact, there appeared to be as much as 25% to 30% insulation void illustrated in Figure 14.

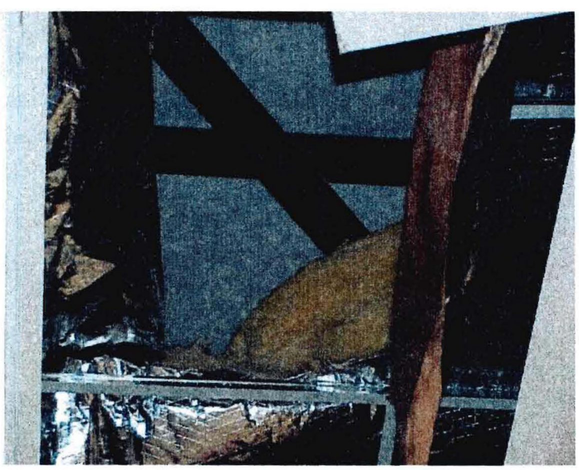

Figure 12 Missing air barrier & insulation that had been moved to gain access to attic.

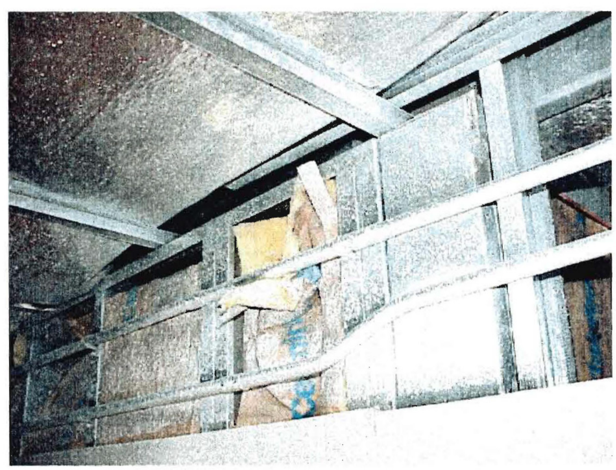

Figure 13 Incomplete air barrier over t-bar ceiling.

Figure 14 Typical insulation void approximately 25% to 30%.

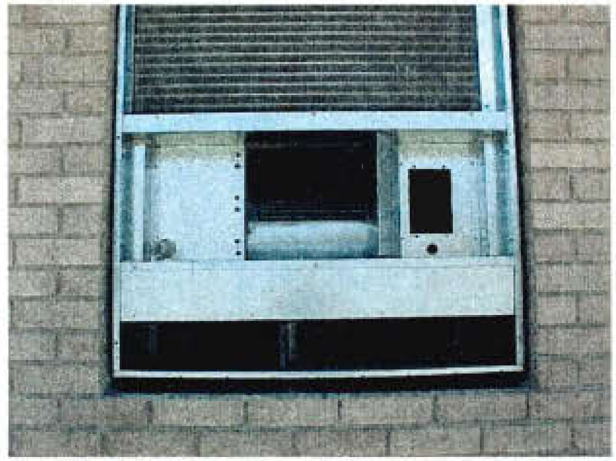

Figure 15 Exterior view of PTHP unit with grille assembly removed. Approximately 1 inch of clearance exists between the unit and the wall.

Retrofit description and schedule

The schools needed a retrofit measure that would provided a better air and thermal barrier above the suspended t-bar ceiling assembly and improve the overall tightness of the buildings. During the third week of July 1999, retrofits began and included tightening the ceiling space between the t-bar suspended ceiling and the insulation level approximately 24 inches higher, eliminating voids in insulation to ensure that a uniform R19 coverage existed, and sealing leakage around each PTHP. The PTHP gap is shown in Figure 15 from the outside. The ceiling space tightening was done with a spray foam insulation called Icynene™. This material provides an effective air seal and supplements the existing insulation. A sample of pictures showing the retrofits are shown using Figures 16-19.

Detailed monitoring of these two classroom buildings was performed for a period of approximately one year (April 1999 - May 2000) to compare the energy consumption, equipment operation, and interior conditions. This effort occurred in three stages. The first two stages were organized to compare energy savings from a retrofit, and the third was designed to compare the two retrofit schools to two other schools of a different type of construction.

- Interior temperature of the two buildings was set at 75F and the 4 hour crank timers were disconnected. This set-up allowed the cooling equipment to operate as needed to maintain temperature 24 hours per day. The adjustment damper of the ventilation air in the PTHP units were closed to minimize outside air entry. No other modifications were done. (May'99 - Jun'99)

- Retrofits were performed on both buildings to air tighten the building envelope and correct deficiencies in the ceiling insulation. Thermostat control continued as in stage 1 for the period Jun'99 - Aug'99.

- The crank timers were reconnected and control of HVAC equipment was returned to the teachers and maintenance personal. Normal school operations resumed. (Aug'99-May'00)

Figure 16 Finished air sealing above the t-bar ceiling.

Figure 17 Attic space after foam was installed It can be seen along the bottom of the truss.

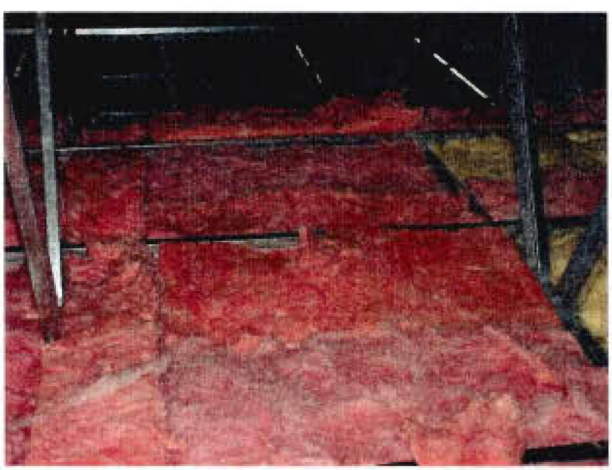

Figure 18 Attic space insulation after all repairs are finished.

Figure 19 Gasketing material placed between the PTHP and the wall to create a seal.

Building airtightness changes in metal building retrofit

Building airtightness testing was repeated after the retrofits were completed. The airtightness of Floral Avenue was reduced from a CFM50 of 16430 to 11091 (32.5 % reduction) and Eagle Lake was reduced from CFM50 of 17756 to 11091 (37.5 % reduction). Figure 20 shows the results of various air tightness tests at the two monitored school buildings.

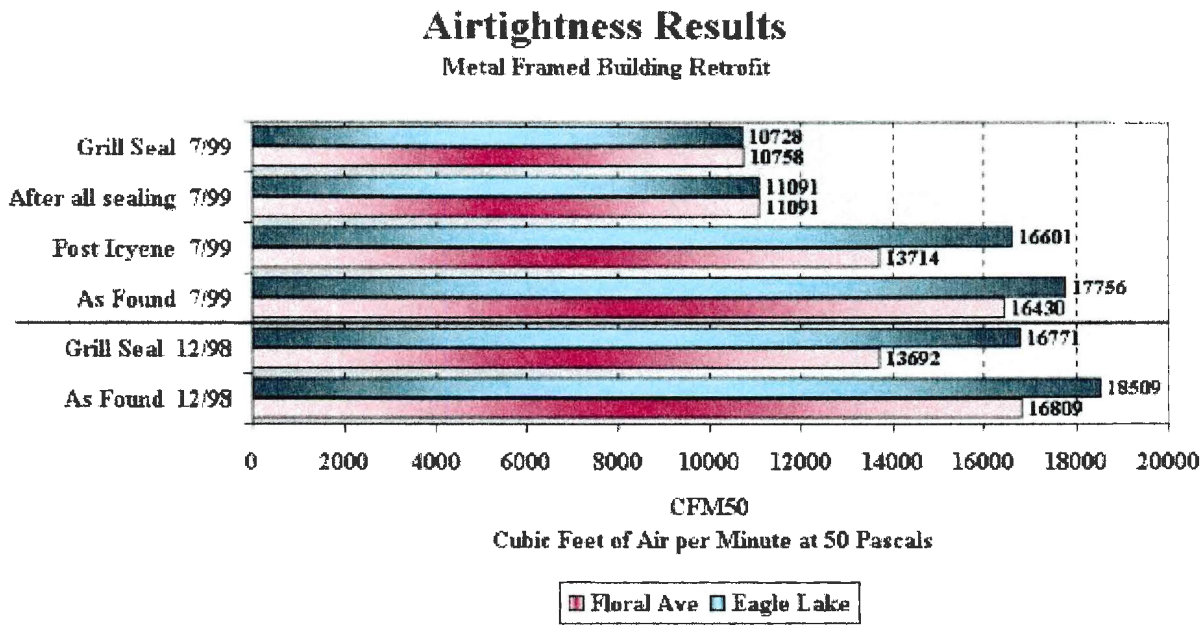

Figure 20 Airtightness results through the stages of retrofit of the metal framed buildings.

Ventilation impacts

Ventilation air is provided by the PTHP units. It is brought into the unit, cooled and dehumidified during the cooling season, then distributed into the classroom. The amount of ventilation brought into the space can be controlled by a damper. Using tracer gas decay methodology (ASTM E 741, "Standard Test Method for Determining Air Leakage Rate by Tracer Dilution"), the building infiltration/ventilation rate was measured with the HVAC equipment operating. This test standard uses a tracer gas distributed evenly into the conditioned space of the building and a specific gas analyzer that measures the gas concentration over time. As time passes, the gas is diluted by outside air which has 0 ppm concentration of the tracer gas in it. The dilution rate of the gas indicates the infiltration rate of outside air, but only for the specific period during the test. Ventilation test results from measurements taken before the retrofit are shown in Table 20.

. The school board determines the amount of ventilation required based on the average occupancy of a classroom for the entire school day. The assumption for the elementary school is that the room will be occupied 77.7% of the time. If the classroom holds 25 students, then, for ventilation purposes, the outside air requirements would be for 19.4 students (25 x 77.7% = 19.4). The total cfm of outside air needed per classroom can then be determined to be 291 cfm (15 cfm/student x 25 students x 77.7% occupancy loading = 291 cfm). The average airflow passing through the building during the test period was less than the recommended air change rate. It averaged 21.7% low or about 63 cfm.

Table 20
Building infiltration rates with HVAC operational

School	ach	cfm[1]	cfm/classroom[2]	difference from 291 cfm	percentage difference
Floral Ave	1.00	839	210	81	27.9%
Eagle Lake	1.33	1114	278	13	4.3%
average	??	??	??	??	??

Notes:
1- Hourly average airflow passing through the building during the test period with outside dampers as found.
2- Total cfm in previous column divided equally amongs the 4 classrooms.
The ventilation air damper was left in the "as found" position, return air and outdoor air filters were clean and the PTHP were ON.

Unfortunately the ventilation test was not repeated after the retrofit so the actual ventilation rate is not known. However, since ventilation is dominated by the mechanical system, it is not expected to be much different after the retrofit. Based on initial ventilation measurements and assumed occupancy, the classroom buildings should have more ventilation air according to ASHRAE Standard 62-99.

Cooling energy savings

Energy savings from the retrofits were monitored. There were two significant problems that made analysis difficult. First, human interference was one problem. During the 4 ½ weeks before retrofit, significant custodial activities occurred such as stripping and waxing floors and other maintenance activities that normally follow a school year. Custodians would lower thermostat settings, at times, to as low as the 50 degree indicator mark despite notes on each thermostat indicating not to change settings. They would also forget to reset the thermostat to the original setting which would require monitoring staff to reset them correctly. Days with unusual thermostat settings were generally deleted from the analysis. Second, there was a limited amount of data with a good range of average daily outdoor temperature, which is critical in establishing a dependable evaluation of cooling energy use versus delta temperature.

As a consequence, data analysis was performed somewhat differently than in the other buildings of this study. Due to a lack of range in the measured average daily outdoor temperature, cooling energy was plotted against the delta temperature on an hourly basis instead of a daily basis. This explains the curvature of the best fit lines shown in Figures 21 and 22.

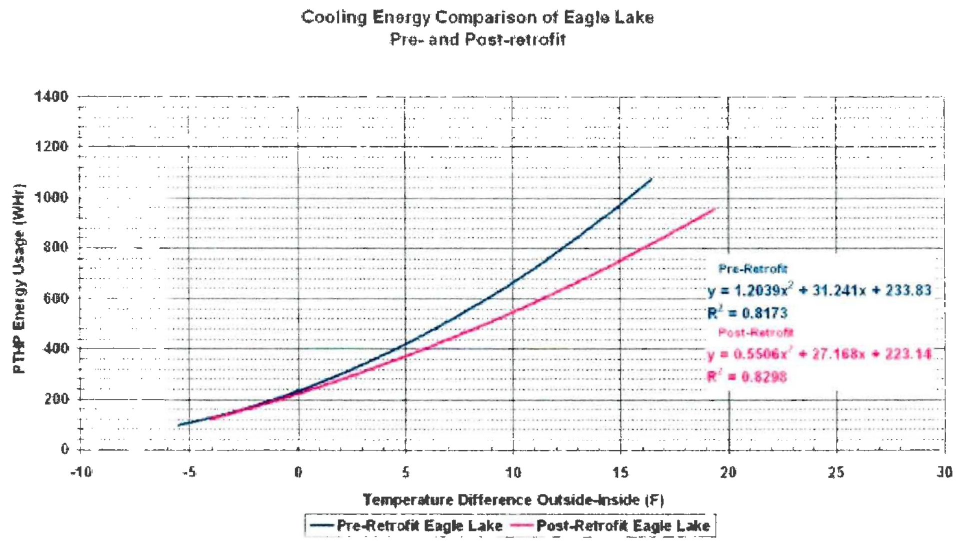

Figure 21 Eagle Lake air conditioning hourly energy use comparison

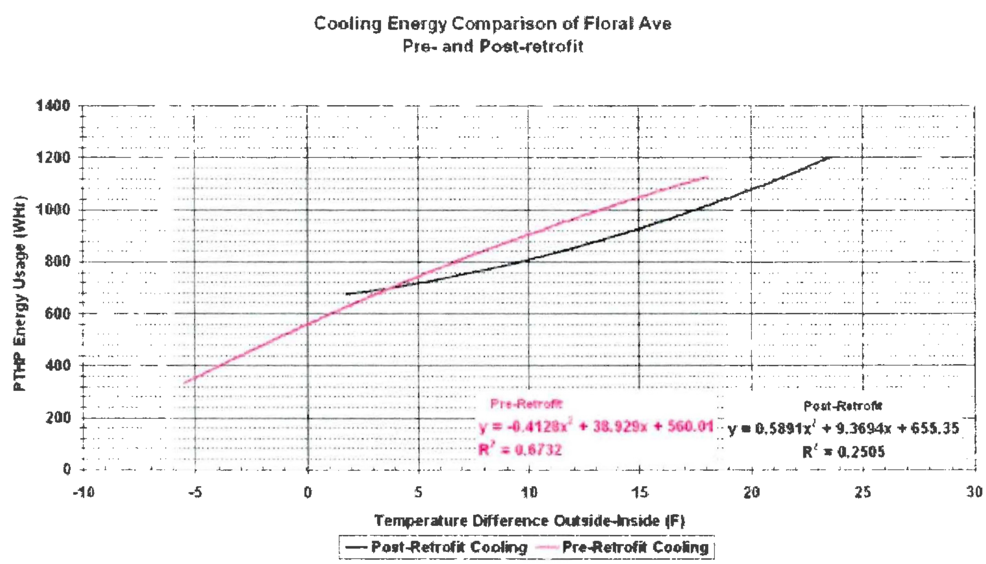

Figure 22 Floral Ave air conditioning hourly energy use consumption

When the air and thermal barriers between the conditioned space and the attic were repaired, there was an energy reduction. Since there was not enough reliable data to analyze the daily energy versus temperature difference, we did not calculate linear equations that could be used to calculate the total energy that would be used before and after retrofits. Therefore, the total seasonal energy savings has not been calculated as it was for the other buildings. Cooling energy savings have instead been estimated for these two buildings based on the best-fit curves of hourly energy use. The total energy use of the PTHP units was reduced in Floral Avenue Elementary by 11.0% assuming an average 10°F temperature difference between the conditioned space and outside ambient. Eagle Lake Elementary experienced a 17.5% energy reduction at 10°F temperature difference. Days with similar interior operation were used in the comparison. It makes sense that savings would be greater at Eagle Lake since the attic space is much hotter at this building because of the dark color of the roof (the Eagle Lake roof is royal blue metal while the Floral Avenue roof is white metal).

Zonal conditions before and after retrofit

In addition to a reduction in cooling energy use, conditions in various building spaces also changed as a result of the retrofits. Figures 23 and 24 show indoor conditions for the two school buildings before the retrofits and Figures 25 and 26 show conditions after retrofits. Hourly temperature and relative humidity profiles for the period of July 15 - Aug 11 show that the ceiling space (area between the ceiling tiles and the insulation) decreased approximately 9°F in both retrofitted buildings though the attic temperatures were almost 3°F warmer. Also note that the attic of Eagle Lake is significantly hotter (~15°F) than that at Floral Avenue.

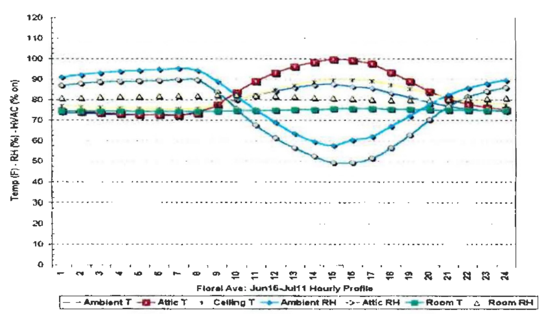

Figure 23 Floral Ave. before retrofit

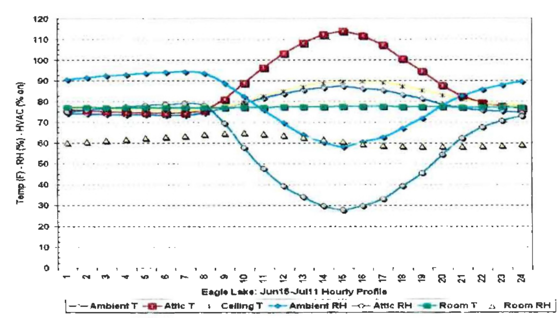

Figure 25 Eagle Lake before retrofit

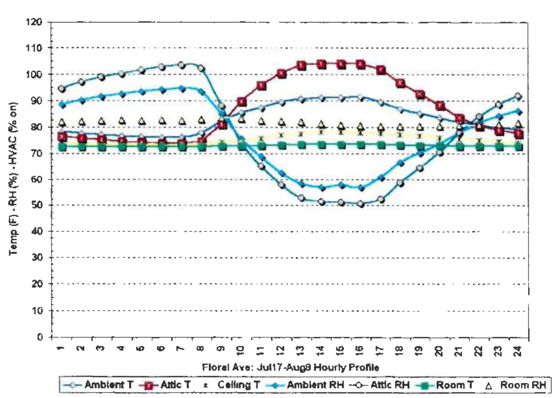

Figure 24 Floral Ave. after retrofit

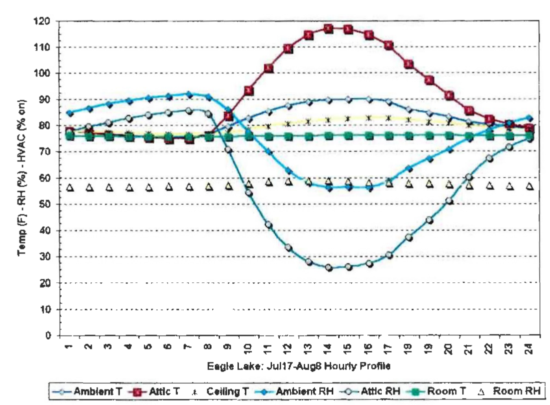

Figure 26 Eagle Lake after retrofit

Appendix D

Features and Capabilities of the Building Science Laboratory and Training Facility

Figure 1. Building Science Laboratory and Training Facility

A major part of this project was the design, construction, and instrumentation of a building science training facility in which a variety of building science and HVAC research experiments could be performed (Figure 1). Following is a description of Building Science Laboratory and Training Facility and its uses.

The first stage of the process was the development of a conceptual design for the facility. We envisioned two side-by-side small commercial buildings in which experiments could be carried out. The two buildings would be identical so that parametric testing could be performed; in other words, one variable at a time could be varied and the resulting changes could be attributed to that single variable. Funding was obtained for the "bricks and mortar" portion of the construction, but only sufficient to construct one building. Construction was begun in May 1999. At the time that the contractor turned the building over to us in November 1999, it was about 70% complete. A number of additional elements had to be installed or completed, including heating and cooling systems, water and sewer utility connections, a data acquisition system, instrumentation, dampers for building vents, intentional and controllable duct leaks, ceiling space walkways, ceiling insulation, etc. Approximately two-thirds of those additional items have been completed as of the time of this report (October 2000), and therefore the building and laboratory facilities are now about 90% complete.

Facility Capabilities

The Building Science Laboratory and Training Facility has been designed with flexibility in mind. It is intended that it can be "every building" or "any building". It has been designed, therefore, with the ability to vary many of the key parameters of the building construction and HVAC systems -- including airtightness, air flows, HVAC equipment, HVAC controls, and the location of air and thermal barriers. It will allow installation of a variety of cooling and heating systems, duct systems, and ventilation systems. This flexibility will allow experiments to evaluate a wide variety of uncontrolled air flow and HVAC problems and solutions. Building flexibility includes:

- The ability to change the leakiness of various planes of the building, including exterior walls, the ceiling, the roof deck, and exterior walls above the ceiling
- The ability to move the thermal barrier from the ceiling level to the roof level

- The ability to change the leakiness of the air distribution system

- The ability to change the type of return air, including central return with no transfers from closed rooms, central return with transfers, central return with through-the-wall transfer, central return with cross-over duct transfer, central return with mechanical closet as return plenum, ducted returns, mechanical closet as return plenum with ducted returns, and the ceiling space as a return plenum
- The ability to set and control (e.g., pre-set schedule) a variety of internal sensible and latent loads
- The ability to test a range of exhaust and intake air flows
- The ability to install a variety of ventilation systems with a variety of control strategies, make-up air flow rates and discharge patterns, and variable outdoor ventilation air
- The ability to install and test a variety of heating and cooling equipment, including fixed and variable speed air handler blowers, "cold air" systems, chilled water systems, enhanced dehumidification units, two-stage systems, roof-top package units, and heat pumps versus straight cooling units
- The ability to vary HVAC control including on/off operation, continuous blower operation, VAV (variable air volume) blower operation, simultaneous temperature and humidity control, occupancy-based outdoor air control, occupancy-based exhaust fan operation, and thermostats with various cycling behavior (deadband control).

This facility is ideally suited to performing a range of experiments on duct leakage, other forms of uncontrolled air flow, ventilation strategies and systems, dehumidification technologies, various HVAC system types, and diagnostic test methods. It is also ideally suited to building science training.

One of the unique features of this building is its ability to replicate a wide range of commercial and institutional building types, from office to retail to classroom to food preparation. Kitchen exhaust and make-up air systems are available, with variable speed fans to simulate a restaurant ventilation environment. Two separate heat pump systems (3.5 ton and 5.0 ton) serve the space and permit operation with a wide range of cooling capacities, from small capacity (3.5 tons) to medium capacity (5.0 tons) to high capacity (8.5 tons). These capacities might correspond, for example, to a small office space, a classroom space, and a restaurant space.

Most unique is the ability to create three fairly typical (hot and humid weather) ceiling space environments based on the location of insulation and the degree of ceiling space ventilation.

- With the insulation at the roof and no venting, the ceiling space will be warm and dry.
- With the insulation at the ceiling and no venting, the ceiling space will be hot and dry.
- With the insulation at the ceiling and venting of the ceiling space, the ceiling space will be relatively hot and humid.

The energy, humidity, and ventilation rate impacts of duct leakage and other forms of uncontrolled air flow depend in large measure upon the conditions within the ceiling space. Knowing these impacts will provide guidance to designers of commercial buildings and to the diagnosis/retrofit decision-making process.

The energy, humidity, and ventilation rate impacts of uncontrolled air flow also depend upon the degree of communication between the occupied space and the ceiling space. Suspended T-bar ceilings are ubiquitous in commercial buildings, and they are very leaky with regard to air flow. Only a small pressure differential across the ceiling causes substantial air flow. This lab building has a suspended T-bar ceiling, but we also have the capability to make that into a tight ceiling by attaching polyethylene sheeting to the bottom surface of the suspended ceiling.

Instrumentation and Control Features

Specialized instrumentation has been installed (or in some cases will be installed) to allow monitoring of a wide variety of parameters including energy use of various systems, temperatures in various locations throughout the building and HVAC systems, humidity in occupied zones and the ceiling space, HVAC system performance, pressure differentials, air exchange rates, carbon dioxide levels, and outdoor weather conditions. Instrumentation was installed in the soil under the slab and in the slab itself to allow measurement of temperatures at 36 inches, 12 inches below the slab, the bottom of the slab itself, and the top of the slab at five separate locations (Figure 2, Figure 3, and Table 1).

Plastic tubes were located at two depths beneath the soil at these same five locations to allow characterization of the pressure field extension that emanates from the building. This will be helpful in characterizing the entry rate of radon, other soil gases, and pesticide residuals from the soil into the building. Soil moisture probes were installed in the soil under the slab at two locations. All of this instrumentation will help in the analysis of energy, humidity, and IAQ variations under various experimental conditions, and will be valuable in calibrating building computer simulation models.

Figure 2. Thermocouple wire being installed under slab.

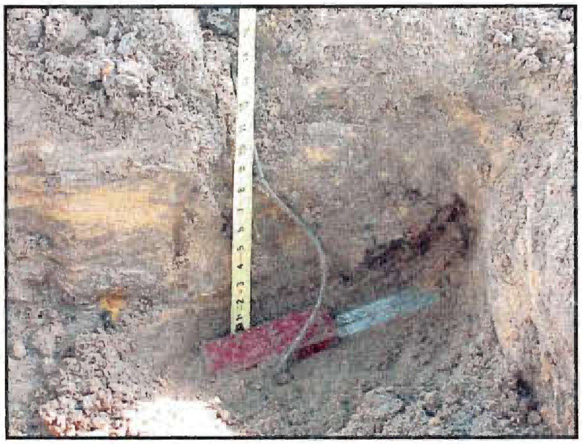

Figure 3. Soil moisture sensor being installed 12 inches below slab.

Table 1

Soil and slab instrumentation located within the building footprint.

Instrumentation Type	Vertical Location of Probe	Number of Probes
thermocouples	1/4" from top of slab	5
	top of soil	5
	12" from top of soil	5
	48" from top of soil	5
pressure tubing	1/2" from top of soil	5
	36" from top of soil	5
soil moisture probe	12" from top of soil	2

Either before or after installation, the instruments and sensors are calibrated to ensure accuracy. The 20 thermocouples, for example, that were installed in the soil and slab were calibrated in three water baths of 32°F, 77.3°F, and 107°F.

Specialized controls have been and will be installed to allow control over the principal building variables, including the airtightness of walls, ceiling, roof plane, ceiling space to outdoors, and air distribution system; the air flow rate or operating status (on or off) of exhaust, make-up, and outdoor air flows; interior heat and moisture generation rates in the building; and the cycling behavior of the heating/cooling systems. Intentional leaks will be created in the supply and return ductwork. These duct leaks will be introduced in various experiments to examine the energy, humidity, and ventilation impacts of duct leakage of different size and location.

Air flow instrumentation will be located at various points throughout the air distribution systems. Pitot tube arrays will be located in the main return and supply ducts to provide real-time measurement of system air flows. Each duct branch is being instrumented with an Iris (TM) damper which doubles as an air flow measurement device. These will allow adjustment and real-time measurement of air flows through each duct branch to various zones of the building. Calibration was performed with several types of ductwork and with a variety of configurations (Figure 4 shows calibration data for one configuration that was tested). Iris dampers yield the best measurement accuracy (±3% of flow for most damper settings) when

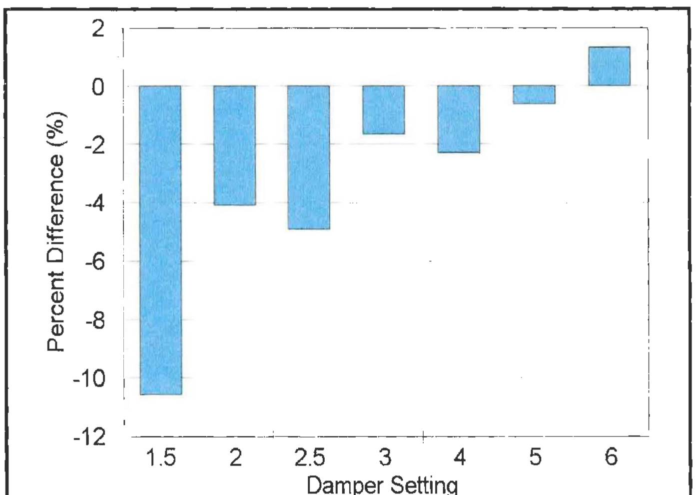

Figure 4. Iris damper calibration with metal and flex duct compared to wind tunnel. Deviations of flow measured by damper versus calibration standard.

installed downstream of a 5-foot round metal duct section and upstream of a 1-foot round metal duct section. Therefore, we have adopted this ducting protocol for our damper installation.

Uncontrolled Air Flow Research Opportunities

Several research issues related to uncontrolled air flow could be investigated with this laboratory facility, including duct leakage, unbalanced return air, and unbalanced exhaust air. Consider two examples:

1. Duct Leakage Investigations

Several different-sized supply leaks and return leaks, and combinations of return and supply leaks, can be introduced into the ductwork. Building cooling energy use would be monitored. By normalizing to outdoor weather conditions, the change in cooling energy use as a function of duct leakage type and size can be characterized. As indicated in the previous section, there are three common ceiling space configurations based on the thermal and air boundaries of the building. The duct leakage amounts and the configurations described could be repeated for each of these ceiling space configurations.

2. Impacts of Return Air Configurations

The lab building can be configured with a variety of return air configurations, such as the following:

- Completely "hard-ducted" return air, with return air to each room and ductwork connecting directly to the air handler
- Partially "hard-ducted" return air, with return air to each room but with the mechanical room operating as a return plenum (this means the return duct opens into the mechanical room, the mechanical room is depressurized, and the depressurization draws air from the return ductwork)
- Completely "hard-ducted"return air, with return grills located only in the central zone and <u>without</u> transfer ducts or grills from the closed rooms to the central zone
- Completely "hard-ducted"return air, with return grills located only in the central zone and <u>with</u> transfer ducts or grills from the closed rooms to the central zone
- The ceiling space used as a return plenum
- The mechanical room operating as a return plenum and drawing return air through return grills located in the central zone

Cooling energy use can be evaluated with different return air configurations in combination with different ceiling space configurations. For example, the "partially hard-ducted return air" configuration could be tested with two different ceiling space configurations; 1) the cool and dry ceiling space, and 2) the hot and humid ceiling space. The ceiling of the mechanical room can also be tested as leaky or tight; in one case as a leaky suspended T-bar ceiling and in another case as a tight gypsum board ceiling. (The mechanical room ceiling in the lab facility is suspended T-bar. We can make it airtight by application of polyethylene sheeting to the bottom of the ceiling and held in place by magnetic strips.)

The Importance of Uncontrolled Air Flow Research

This research is important because we know that the consequences of uncontrolled air flow in commercial buildings are, in many cases, large in hot and humid climates. However, the magnitude of the impacts of uncontrolled air flow are not well quantified. The consequences that need to be better understood include:

- cooling and heating load impacts
- cooling and heating energy use impacts
- peak electrical demand impacts
- indoor humidity impacts
- ventilation rate impacts

In single-family residential construction, the impacts of duct leakage, unbalanced return air, and unbalanced exhaust air are more easily identified, largely because the thermal and humidity conditions of the attic space are well known. In commercial construction, the impacts of uncontrolled air flow vary substantially because the thermal and humidity conditions of the space between the ceiling and the roof vary considerably depending upon the location of the air and thermal barriers. There is great variety regarding where these barriers are located in commercial buildings. Therefore, there is much greater complexity in commercial buildings regarding the impacts of uncontrolled air flow compared to residential construction.

It is important that a thorough understanding of these complex building interactions be developed. This information is needed to establish building design priorities (best practice) and provide guidance to those making decisions regarding retrofit measures. With this information, we can verify which design approaches work and do not work, and what the consequences will be if the wrong approach is taken. From a diagnosis and repair perspective, this research can identify which measures are cost-effective depending upon the ceiling space configuration.

Pathways for air flow between the occupied space and the ceiling space, and between the ceiling space and outdoors, are two important variables. The leakiness of suspended T-bar ceilings allows air to easily transfer between the occupied space and the ceiling space. Vent openings in the ceiling space to outdoors (if they exist) allow air to be easily transferred between the ceiling space and outdoors. The ceiling ventilation rate, in turn, is a function of wind and temperature driving forces. *Consequently, the ventilation rate of buildings with substantial uncontrolled air flow and substantial ceiling space ventilation may depend in large part upon the ventilation rate of the space between the ceiling and roof.* In this lab building, there are 23 vent openings in the ceiling space which can be closed or opened in order to change the ceiling space from tight to well-vented, and the ceiling plane itself can be leaky or tight.

Additional Research Topics

In addition to research into uncontrolled air flow, this facility is ideally suited to testing and evaluation of a variety of HVAC systems and control strategies. The building is set up with two completely separate duct systems. Space is available to add additional duct systems so that side by side experiments can be performed to evaluate HVAC or duct system efficiency. The building is configured to accept both roof top package units, split DX systems, and mini-split systems.

Dehumidification technologies can be tested and evaluated. The facility has a set-up of large exhaust and make-up air fans (with digital fan speed control) which will permit experiments which call for high indoor humidity conditions, testing of energy recovery equipment, and advanced dehumidification technologies.

HVAC control strategies can be evaluated. The dehumidification performance of various air-conditioning systems can be tested as a function of air flow rate and air handler fan cycling schedules. The dehumidification benefits of set-back thermostats can be tested (they can improve humidity control by forcing the coil to remain cold for extended periods). The humidity impacts of short-cycle thermostat operation can be documented, and this information can be transferred to controls manufacturers for improved equipment and thermostat design.

Diagnostic testing procedures can also be evaluated. These include methods for measuring duct leakage, air flow measurement techniques, and improved approaches to Test and Balance.

Training Opportunities

In addition to accommodating a wide range of air flow, moisture, ventilation, and HVAC equipment experiments, this facility will also provide an ideal environment for building science training. The building itself will act as a walk-in model. With the turn of a switch or the opening of a damper, the various characteristics of building tightness, duct leak air flows, zone pressures, etc. will change before the students eyes and the concepts of the "building is a system" will be made clear. It is likely that a "digital scoreboard" will be installed to provide continuous output of the temperatures, humidities, and pressure differentials that occur as the building systems are operated. The interior layout of the building has been designed to accommodate lecture style training, diagnostic training with hands-on testing applications, live demonstration of building operating principals, and repair training with operating models. To date, several Class 1 Energy Rater training (2-day) classes have already been held in this facility.

As part of the effort to raise funds for the bricks and mortar portion of the construction, two brochures were developed. A detailed brochure provides floor plans, construction details, and the various uses of the building. A short brochure provides an overview of the characteristics and uses of this facility. These brochures are attached to the end of Appendix D.

Dedication Ceremony

On February 17, 2000 a dedication ceremony was held for the Building Science Laboratory and Training Facility. Making presentations were Dr. David Block, FSEC Director, Mr. Phil Ardis, General Manager of Tropic-Kool Engineering Corporation, Dr. Soileau, VP of Research at UCF, Mr. Mike Ashworth, Florida Department of Community Affairs, and the Honorable Randy Ball, State Representative, 29th District, Florida House of Representatives. Presentations were also made by leaders of the Building Science and HVAC research team, Neil Moyer, Dr. Muthusamy Swami, Don Shirey, and Jim Cummings, regarding the types of research and training activities which are anticipated to be carried out in the facility. See the attached dedication ceremony flier.