Isolated Sub-Dehumidification Strategies in Large Supermarkets and Grocery Stores

Final Report

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Refrigeration Project Team
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1. Goals and Objectives

The objective of this project was to determine the potential energy savings associated with reducing the relative humidity in the vicinity of refrigerated display cases in supermarkets, as compared to the widely accepted current practice of maintaining a relatively higher and uniform humidity level throughout the entire supermarket. Existing and new strategies for maintaining lower relative humidity levels near the vicinity of refrigerated display cases were analyzed to determine their effectiveness and limits of application.

2. Motivation and Benefits of Reduced Humidity

Retail food stores and supermarkets are energy-intensive commercial buildings, with energy usage intensities ranging from 43 kW to 70 kW per square foot of floor area (460 kW/m^2 to 750 kW/m^2) (EIA 2006). Supermarkets operate their refrigeration systems continuously to maintain proper food storage conditions within their refrigerated display cases and food storage areas. The continual operation of this equipment accounts for approximately 50% of the total electrical energy consumption of a typical supermarket (Westphalen, et al. 1996). In addition, approximately 10% of the total electrical energy of a typical supermarket is consumed by the HVAC system in order to maintain thermal comfort for the occupants and suitable climatic conditions for the refrigerated display cases (Capozzoli, et al. 2006) (Kosar and Dumitrescu 2005).

There is a significant interaction between a supermarket's refrigeration system, its HVAC system and the ambient conditions within the store. Heat and moisture contained in the conditioned store air are transferred to the refrigerated display cases, contributing both a latent and sensible heat load to the supermarket refrigeration system. The moisture which infiltrates into the refrigerated display cases leads to condensation on the interior surfaces of the display cases and to frost formation on the evaporator coils. To minimize condensation, anti-sweat heaters are often used to heat door frames and other interior surfaces of refrigerated display cases. In addition, defrost heaters may be required to remove frost accumulation on evaporator coils. These heaters add additional sensible heat to the refrigerated display cases and increase the overall electrical energy consumption of the supermarket.

Dehumidification of the supermarket air with its refrigeration system, rather than its HVAC system, is an inefficient and energy intensive process. As shown in Figure 1, approximately 1.8 kWh to 2.7 kWh of energy is required to remove one pound of moisture using the low temperature display cases (Khattar 1992). On the other hand, approximately 0.25 kWh to 0.45 kWh of energy is required to remove one pound of moisture using the HVAC system (Khattar 1992). Due to the higher suction temperature at the evaporator in the HVAC system as compared to the refrigeration system, and with the resulting higher coefficient of performance (COP) of the HVAC system as compared to the refrigeration system, it is more efficient to dehumidify the air using the HVAC system.

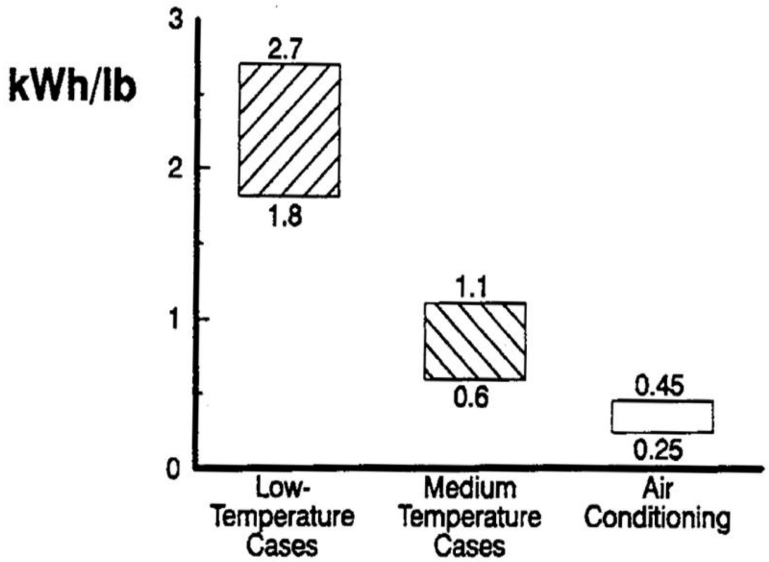

Figure 1. Energy required to remove move moisture from air for low-temperature refrigeration, medium temperature refrigeration and HVAC (Khattar 1992).

Reducing the ambient relative humidity within the supermarket by shifting the dehumidification to the more efficient HVAC system reduces the latent load on the refrigeration system, thereby reducing the compressor power requirements of the refrigeration system. It has been estimated that for every 0.6°F (1°C) reduction in dew point temperature, a 2% reduction in refrigeration system energy use can be achieved (Capozzoli, et al. 2006). In addition, due to less frost formation, fewer and/or shorter defrost cycles will be required, resulting in lower energy consumption for electric defrost systems. As an added benefit, fewer defrost cycles reduces the temperature fluctuation of the stored food items within the display cases, thereby minimizing detrimental impacts on product shelf life and quality. Furthermore, anti-sweat heater power requirements will be reduced due to less condensation on display case surfaces.

Other impacts of reduced humidity levels in supermarkets include improved product appearance due to lack of frost formation on frozen product; the creation of a more comfortable shopping environment; the reduction in the growth of mold, mildew and fungi; and the increased life of furniture, fixtures and electronics (Khattar 1992) (Khattar 2004).

The area near the refrigerated display cases requires humidity control in order to obtain efficient performance from refrigeration system while the dry goods sales area generally requires no humidity control. Since there are no partitions between the refrigerated display case area and the dry goods area in the supermarket, moisture will diffuse through the air and migrate to the refrigerated display case area if the humidity in the dry goods area is higher than in the refrigerated display case area (Khattar

2004). Therefore, several researchers have attempted to quantify the impact of reduced humidity level on the performance of supermarket refrigeration systems as well as to develop new and efficient strategies to reduce the overall store humidity or the humidity near the refrigerated display cases.

Outline of Report

Through this project, an attempt has been made to determine the impact of supermarket relative humidity on the energy consumption of supermarket refrigeration systems and HVAC systems. The estimated savings associated with the refrigeration and HVAC systems is then used to determine the total potential energy savings associated with reducing the relative humidity in the vicinity of refrigerated display cases in supermarkets.

The following outline summarizes the content of this report.

- Section 3, *Effects of Humidity on Supermarket Refrigeration Systems*

  A summary of research is given regarding the effects of humidity level on the performance of supermarket refrigeration systems.

- Section 4, *Control of Supermarket Humidity using HVAC Systems*

  A description is given of various HVAC systems which can be used to control the humidity level within supermarkets. In addition, a summary of simulation studies and field studies of supermarket HVAC system performance is given.

- Section 5, *Whole Store Dehumidification vs. Isolated Sub-Dehumidification*

  Results of studies regarding the effectiveness of isolated sub-dehumidification versus whole store dehumidification are discussed.

- Section 6, *Estimated Energy and Cost Savings of Isolated Sub-Dehumidification*

  Estimated energy and cost savings of implementing isolated sub-dehumidification in supermarkets are given.

- Section 7, Implementation

  Methods for implementing isolated sub-dehumidification in supermarkets are given, including new construction as well as retrofit applications.

- Section 8, *Conclusions*

  Conclusions and recommendations regarding isolated sub-dehumidification in supermarkets are given.

3. Effects of Humidity on Supermarket Refrigeration Systems

Several researchers have attempted to determine the impact of supermarket relative humidity on the performance of supermarket refrigeration systems. Studies have included both analytical and experimental investigations of refrigeration system energy consumption.

The performance of refrigerated display cases and refrigeration systems as a function of supermarket relative humidity are reported by several researchers (Faramarzi, Sarhadian and Sweetser 2000) (Sweetser 2000) (Kosar and Dumitrescu 2005) (Getu and Bansal 2006). Faramarzi et al. (2000) and Sweetser (2000) reported on laboratory testing that was performed at Southern California Edison's Refrigeration Technology and Test Center. The performance of four types of refrigerated display cases was investigated in a controlled laboratory setting, including:

- Medium temperature open five shelf dairy case
- Medium temperature open two shelf meat case
- Low temperature coffin case
- Low temperature reach-in five shelf case

The controlled environment in which the refrigerated display cases were tested was maintained at 75°F (23.9°C), and the relative humidity ranged from 35% to 55% in 5% increments during testing. The suction and discharge temperatures were fixed while the refrigerant flow rate was allowed to vary according to the load. In addition, the test protocol included simulated product and simulated shopper traffic.

The effects of relative humidity on the evaporator load of the four types of refrigerated display cases are summarized as follows (Faramarzi, Sarhadian and Sweetser 2000) (Sweetser 2000):

- Dairy case: The evaporator load was reduced 21% when the relative humidity decreased from 55% to 35% (refrigerant flow rate dropped from 264 lb/hr to 218 lb/hr). The latent heat load represented 28% of the total load at 55% RH and 9.3% of the total load at 35% RH.
- Meat case: The evaporator load was reduced 21.6% when the relative humidity decreased from 55% to 35% (refrigerant flow rate dropped from 218 lb/hr to 171 lb/hr). The latent heat load represented 32.2% of the total load at 55% RH and 14.8% of the total load at 35% RH.
- Coffin case: The evaporator load was reduced 12.6% when the relative humidity decreased from 55% to 35% (refrigerant flow rate dropped from 64.5 lb/hr to 56.4 lb/hr). The latent heat load represented 9.6% of the total load at 55% RH and 5.2% of the total load at 35% RH.
- Reach-in case: There was little impact on energy consumption as the relative humidity decreased from 55% to 35%. However, it was noted that if the anti-sweat heaters were deactivated at 35% RH, the fog recovery time would be the same as that which was observed at 47.5%RH with the anti-sweat heaters active.

Based on a review of analytical and experimental data reported by display case manufactures, energy consultants, utility company researchers and academic researchers, Kosar and Dumitrescu (2005) reported that by decreasing the relative humidity within a supermarket from 55% to 35%, compressor energy savings ranging from 5.9% to 17.3% could be realized for low-temperature open coffin-type display cases. Since low temperature display case compressors account for approximately 40% to 60% of the total compressor energy consumption, Kosar and Dumitrescu (2005) estimated that a savings of 2.5 to 3.0 kWh/day per linear foot of open low-temperature display case could be achieved by reducing the humidity from 55% to 35%.

For medium-temperature open display cases, Kosar and Dumitrescu (2005) reported that compressor energy savings ranged from 14.2% to 21.4% when the relative humidity within the supermarket was reduced from 55% to 35%. A 20% energy savings was estimated to yield compressor energy reductions of 0.55, 0.77 and 0.88 kWh/day per foot for open multi-deck produce, diary and meat display cases, respectively.

Getu and Bansal (2006) reported on the impact of store relative humidity on the performance of a low-temperature refrigeration system for a supermarket located in Auckland, New Zealand. The low temperature refrigeration systems consisted of an air-cooled, parallel compressor rack with subcooling. Coffin-type frozen food cases, glass-doored reach-in frozen food cases, fish/meat cases and walk-in freezer rooms were serviced by the low temperature refrigeration system. The electrical energy consumption of the low temperature refrigeration system as well as display case and store temperature and relative humidity were monitored for a period of 15 days.

It was found that the electrical power required by the low temperature compressors increased by approximately 225 W for every 1% rise in store relative humidity, from approximately 18 kW at 35% RH to approximately 22 kW at 50% RH (Getu and Bansal 2006). This represents approximately a 1.1% increase in compressor power for every 1% increase in relative humidity.

Getu and Bansal (2006) found that the electrical power of the accessories, including the defrost heaters, anti-sweat heaters, lights and evaporator fans, increased by approximately 130 W for every 1% rise in store relative humidity. This represents approximately a 0.67% increase in accessory power for every 1% increase in relative humidity.

Finally, Getu and Bansal (2006) reported that the total electrical power required by the low temperature refrigeration system, including the compressors and accessories, increased by approximately 355 W for every 1% rise in store relative humidity. This is approximately a 0.91% increase in total power for every 1% increase in relative humidity.

Several studies also report the energy savings associated with defrost heaters and anti-sweat heaters as a function of supermarket relative humidity (Tassou and Datta 1999) (Henderson and Khattar 1999) (Kosar and Dumitrescu 2005). It should be noted that the energy savings of these components is strongly dependent upon their control strategy.

Tassou and Datta (1999) performed both field and laboratory investigations to quantify the effect of supermarket humidity on the frost accumulation on the evaporators of medium temperature vertical multi-deck display cases. For a supermarket located in Airdrie, UK, where the summertime store relative humidity ranged from 45% to 55% and the wintertime store relative humidity ranged from 22% to 25%, it was found that twice as much defrost condensate was collected from a multi-deck dairy display case in the summer as compared to the winter. Thus, the diary case studied at the supermarket required 50% fewer defrost cycles in the winter as compared to the summer.

In laboratory testing, Tassou and Datta (1999) found that for a medium temperature open display case, frost accumulation increased exponentially with increasing relative humidity. After the display case being studied had operated for a six hour period, 2.2 gallons (8.4 liters) of condensate were collected after defrosting when the relative humidity was 65% while 1.7 gallons (6.3 liters) of condensate were collected at 45% relative humidity. This represents a 23% reduction in condensate collection when the relative humidity is reduced from 65% to 45%. However, condensate data presented by Faramarzi et al. (2000) exhibits a linear trend with relative humidity. They noted a 61.7% reduction in condensate collection from an open, medium temperature meat case when the relative humidity was reduced from 55% to 35%, while a 73.2% reduction in condensate collection occurred from an open, medium temperature dairy case when the relative humidity was reduced from 55% to 35%.

Henderson and Khattar (1999) performed field investigations at two supermarkets to determine the impact of store relative humidity on anti-sweat heater power consumption. One supermarket was located in Minneapolis, MN while the other was located in Indianapolis, IN. In each supermarket, the power consumption of the dew point temperature controlled anti-sweat heaters was measured as a function of store humidity. It was found that anti-sweat heater energy consumption decreased 1% to 2% for each 1% reduction in relative humidity.

From the database review performed by Kosar and Dumitrescu (2005), it was found that if proactive measures were taken, then defrost electrical energy savings anywhere from 0% to 7.4% could be achieved by using temperature terminated electric defrost rather than time-terminated defrost in low-temperature display cases, if the store relative humidity was maintained at 35% rather than 55%. In addition, if dew point temperature control rather than 100% continuous operation were implemented with anti-sweat heaters, then anti-sweat heater electrical energy savings of 67% to 100% could be achieved if the store relative humidity was maintained at 35% rather than 55%.

Summary

In summary, reducing the relative humidity within a supermarket can reduce the energy consumption of the supermarket's refrigeration system, including the energy consumption of compressors, defrost heaters and anti-sweat heaters. The refrigeration system energy savings is dependent upon refrigerated case type and refrigeration system temperature level. Lower humidity levels significantly improve the performance of open display cases while the benefit for doored display cases is minor. However, for doored display cases, a substantial reduction in anti-sweat heater energy requirements can be realized with lower humidity levels.

For medium temperature refrigeration systems, refrigeration system energy use decreases anywhere from 15% to 22% when the relative humidity is reduced from 55% to 35%. For low temperature refrigeration systems, refrigeration system energy use decreases anywhere from 0% to 17% when the relative humidity is reduced from 55% to 35%.

For medium temperature open display cases, defrost cycles can be reduced by 25% to 50% by reducing the relative humidity from 55% to 35%. Also, for low temperature cases, anti-sweat heaters can be deactivated at a relative humidity of 35%.

For the low temperature refrigeration system, an increase of 1% in power consumption results from every 1% increase in relative humidity from 35% RH to 50% RH. Furthermore, for the entire supermarket refrigeration system (medium temperature and low temperature), a 2% reduction in refrigeration system energy consumption can be achieved for every 0.56°F (1°C) reduction in the dew point temperature.

4. Control of Supermarket Humidity using HVAC Systems

Compared to typical commercial buildings, supermarkets have a relatively high latent cooling load due to the sensible cooling which is provided by the refrigerated display cases. This high latent load can be further intensified in humid climates where humid air enters the supermarket through ventilation and/or from infiltration through doors and other openings.

The ratio between the sensible and latent cooling can be quantified with the Sensible Heat Ratio (SHR), which is defined to be the ratio of the sensible cooling load to the sum of the sensible cooling load and the latent cooling load. The SHR approaches one for high sensible loads and decreases towards zero for increasing latent loads. For a typical commercial building, the SHR ranges from 0.75 to 0.90, indicating that the cooling load is mainly sensible. Conventional HVAC systems can easily handle SHRs in this range, and therefore, humidity problems are rarely seen in typical commercial buildings. However, the SHR for a supermarket ranges from 0.50 to 0.75, indicating more latent load. Since most conventional HVAC systems are not capable of handling such low SHRs, humidity control in supermarkets is difficult (Khattar 1992).

Moisture can be removed from the air through mechanical or chemical means. In a mechanical process, the moisture is removed from the air by condensing the moisture on a cold surface, thereby cooling the air. In a chemical process, moist air comes into contact with a chemical desiccant which either absorbs or adsorbs the moisture. During the absorption or adsorption process, the air is heated (Khattar 1992).

HVAC/Dehumidification System Types

Several methods for controlling the temperature and humidity with a supermarket are discussed below, including the following:

- Conventional, Single-Path HVAC Systems
- Improved Single-Path Systems
- Dual-Path Systems
- Desiccant-Based Systems
- Heat Pipe Heat Exchanger Enhancements

Conventional, Single-Path HVAC Systems

In many supermarket applications, both the temperature and the humidity within the store are maintained at or below desired levels by using a constant-air-volume HVAC system. As shown in Figure 2, this type of system consists of a direct expansion cooling coil, a heating coil and a supply fan. As the supply air passes over the cooling coil, the air is cooled below its dew point, thereby causing moisture to condense out of the air. The cool, dehumidified supply air is then reheated, via the heating coil, to achieve the required comfort conditions before being discharged into the store. In this humidity control strategy, the cooling system must operate at a much lower evaporating temperature in order to cool the air below its dew point for dehumidification purposes as opposed to simply cooling the air for comfort conditions. This leads to lower coefficients of performance for the cooling system. This inefficient process of overcooling and reheating of the store air can be made more efficient by utilizing the waste heat from the supermarket refrigeration system to partially or fully reheat the store air.

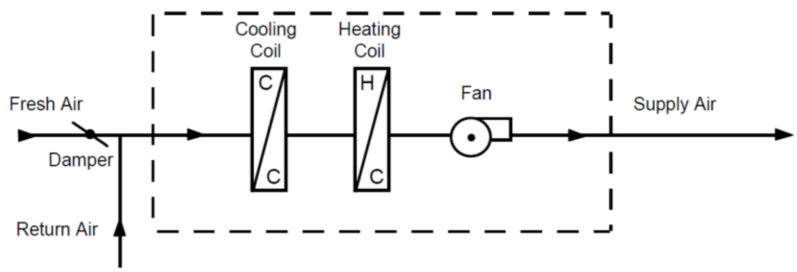

Figure 2. Conventional, single-path HVAC system (Khattar 2004).

Improved Single-Path Systems

The improved single-path HVAC system is very similar to the conventional HVAC system except that the cooling coil suction pressure is lower than that of the conventional system, and thus, the air exiting the cooling coil in the improved single-path system is around 40°F (4.4°C), rather than 50°F (10°C) as is typical in the conventional system. The lower cooling coil temperature allows the air to be cooled to a lower dew point temperature, thereby removing more moisture from the air. The air flow rate in an improved single-path system is typically less than that of a conventional system, in order to increase the air's contact time with the cooling coil and increase the cooling coil's moisture removal capacity. Finally, the improved single-path system typically incorporates a bypass damper which allows a portion of the air to be routed around the cooling coil. The effect of the bypass damper is to reduce the air flow through the cooling coil, thereby reducing the leaving air humidity. When the bypass air stream mixes with the coil leaving air stream, the mixed air temperature and humidity results in a lower SHR and improved dehumidification performance (Khattar and Brandemuehl 1996).

Dual-Path Systems

In a dual-path HVAC system, shown in Figure 3, outside ventilation air is conditioned separately from the recirculation air. Typically, the outside air is cooled and dehumidified to a low dew point temperature, then mixed with the return air, and finally supplied to the space. Compared to the conventional, single-path system, the dual path system has improved dehumidification performance at part-load conditions.

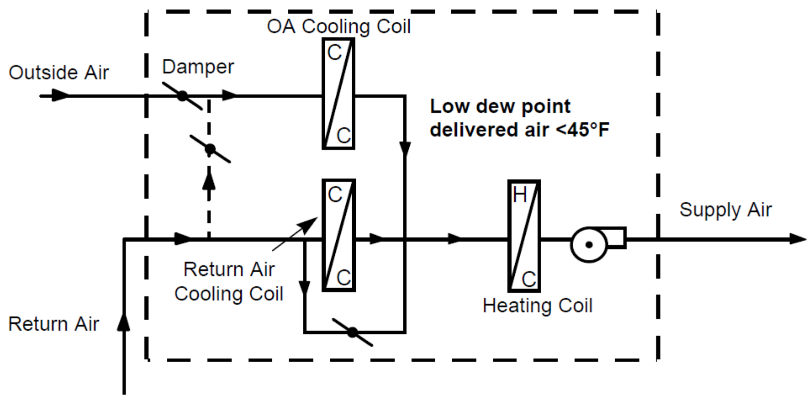

Figure 3. Dual-path HVAC system (Khattar 2004)

Desiccant-Based Systems

The use of hybrid HVAC systems that incorporate desiccant dehumidification have been proposed for supermarket applications (GRI 1984) (Mitchell, et al. 1992) (Capozzoli, et al. 2006) (Lazzarin and Castellotti 2007). In these desiccant dehumidification HVAC systems, shown in Figure 4, outdoor air passes over a solid or liquid desiccant that removes the moisture from the air, either by adsorption or absorption. The dehumidified air then passes over cooling and heating coils and through a humidifier for final conditioning prior to being discharged into the conditioned space. In order to continually remove moisture from the supply air, the moisture rich desiccant must be frequently regenerated. This is done by passing warm air over the desiccant. The warm air absorbs the moisture from the desiccant, and the desiccant is thereby regenerated and able to remove moisture from the supply air stream. The warm regeneration air stream is often produced via a gas-fired heater. To increase system efficiency, waste heat from the supermarket refrigeration system could be used to preheat the regeneration air stream or to heat the supply air.

Desiccant dehumidification systems have several advantages over that of conventional single-path and dual-path HVAC systems, including separate control of the latent and sensible heat loads and smaller cooling system requirements (Capozzoli, et al. 2006). In addition, the smaller cooling requirements of desiccant systems can be met with cooling systems having a higher COP than that compared to conventional and dual-path systems. However, desiccant-based systems require a source of heat to regenerate the desiccant material, and often times this is provided via gas-fired heaters. Thus, while cooling system capacity may be reduced with a corresponding reduction in electrical energy consumption, an additional natural gas consumption may be required for desiccant systems.

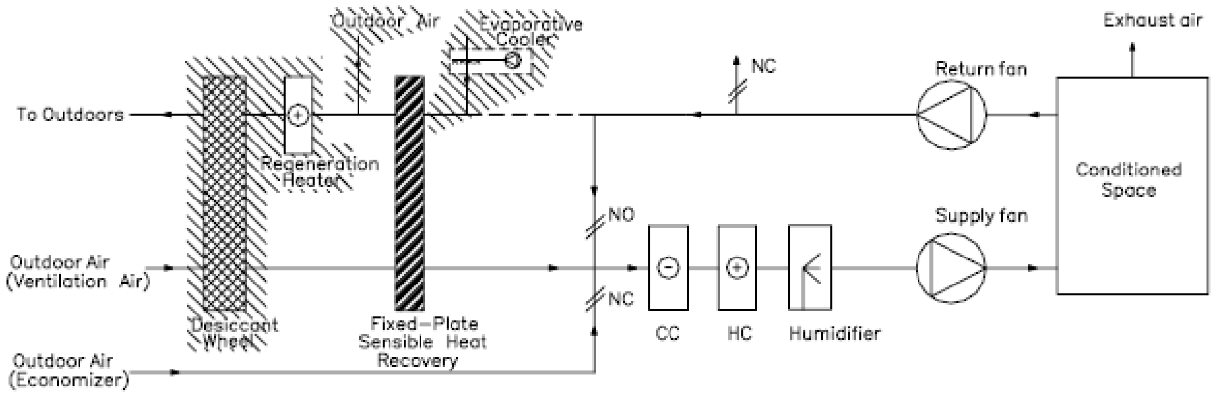

Figure 4. Desiccant-based dehumidification system (Capozzoli, et al. 2006).

Heat Pipe Heat Exchanger Enhancements

In the improved single-path and dual-path HVAC systems described above, the cooling coil temperature is lower than that which is typically used for traditional commercial applications. By reducing the coil temperature, the dehumidification capacity of the coil is increased since the dew point temperature of the air passing through the coil is reduced. However, there is a practical lower limit on the coil temperature. If the surface temperature of the coil is below 32°F (0°C), then frost will form on the coil. Frost accumulation will eventually degrade the heat transfer performance of the coil and defrost cycles are required to remove the frost. Thus, coil temperatures must be maintained above 32°F (0°C) to avoid frost formation.

A heat pipe is a simple device which can be used to increase the dehumidification performance of air cooling coils while still maintaining the coil temperature above 32°F (0°C). A heat pipe is a sealed tube that is partially filled with a volatile working fluid such as a refrigerant. Warm air is passed over the lower end of the heat pipe which causes the refrigerant in the tube to boil and absorb heat from the warm air stream, thereby cooling the warm air. The vapor then travels to the higher end of the heat pipe where it is in contact with a cold air stream. The vapor now condenses, transferring heat from the refrigerant to the cold air stream, thereby heating the cold air. The condensed refrigerant now flows via gravity to the lower end of the heat pipe and the cycle repeats. The net effect is to transfer heat from the warm air stream to the cold air stream.

The two ends of a heat pipe can be placed upstream and downstream of an air cooling coil to precool the incoming air and reheat the exiting air. By doing so, the total amount of cooling required by the coil is reduced and thus, a portion of the coil's sensible capacity is exchanged for additional latent capacity. Therefore, heat pipes can increase the dehumidification capacity of air cooling coils.

Heat pipes may be used in a side-by-side configuration, as shown in Figure 5. In this configuration, the lower ends of the heat pipes are located in the inlet air flow path. The higher ends of the heat pipes are located in the outlet air flow path. The air must change direction 180 degrees in order to pass through the low and high sections of the heat pipes.

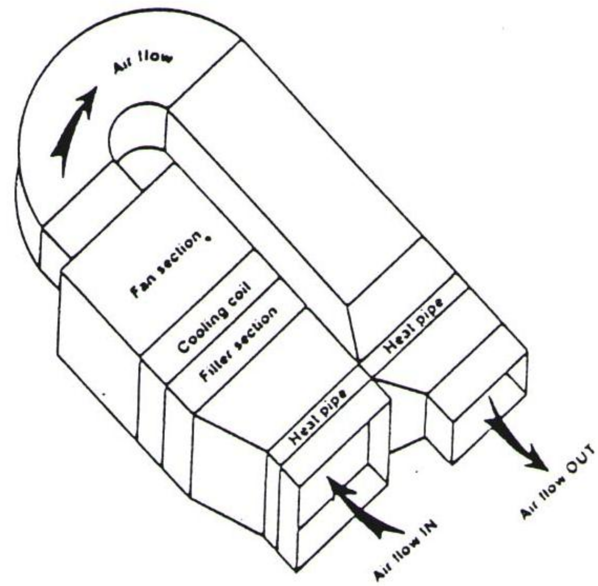

Figure 5. Side-by-side heat pipe heat exchanger configuration (Khattar 2004).

Another configuration makes use of U-shaped heat pipes, as shown in Figure 6. In this configuration, U-shaped heat pipes are placed around a cooling coil in such a way that the lower ends of the heat pipe are in the inlet air flow stream and the higher ends of the heat pipes are in the outlet air flow stream. Since the air flow does not need to change directions, this configuration is more compact than the side-by-side configuration.

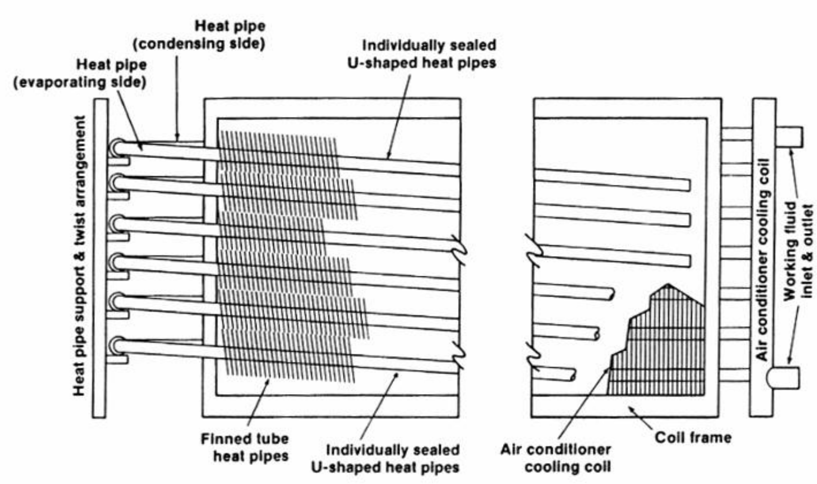

Figure 6. U-shaped heat pipe heat exchanger configuration (Khattar 2004).

Evaluation of HVAC/Dehumidification Systems

Several numerical simulations and field studies have been performed to determine the performance of various HVAC/dehumidification systems for supermarket applications. These studies are summarized below.

Heat Pipe Heat Exchanger Enhancements

Flannick (1992) discussed energy and cost savings associated with heat pipe retrofits at several supermarkets. The use of heat pipes allowed for substantially smaller system capacities as compared to conventional systems, as well as reduced airflow rates, thereby resulting in significant annual energy savings. For the supermarkets studied, a 14% to 33% reduction in HVAC system capacity was realized with the addition of heat pipe heat exchangers, with an average capacity reduction of 26.8%. Fan power requirements were found to decrease anywhere from 2 kW to 6 kW after adding heat pipe heat exchangers. It was noted that fan power savings increased with increasing change in HVAC system capacity due to the addition of the heat pipes. On average, the change in fan power requirements per change in HVAC capacity was found to be 0.267 kW/ton.

Keebaugh et al. (1992) present energy savings data for a heat pipe system installed in a 35,000 ft^2 (3,250 m^2) supermarket in Lithonia, GA. A 40-ton rooftop AC unit was modified to include a heat pipe run-around coil. It was noted that a 1°F (1.8°C) reduction in interior dew point resulted in a 1 kW reduction in refrigerated case compressor demand. It was also noted that the heat pipe significantly increased the moisture removal capacity of the air conditioner.

Dual-Path Systems with Heat Pipe Heat Exchanger Enhancements

Khattar and Brandemuehl (1996) attempted to quantify changes in supermarket energy consumption due to various HVAC operating strategies. Field monitoring and testing was performed at a supermarket with a sales area of 34,585 ft^2 (3,213 m^2), located in Gulf Breeze, FL. The original HVAC system in the store consisted of a DX system with a remote air handling unit (AHU). The AHU was customized to test the following operating strategies:

- Heat pipe heat exchanger around cooling coil
- Variable speed blow for adjustable system airflow
- Face-split cooling coil with six separately controllable refrigerant circuits
- Operable damper for air bypass around cooling coil
- Operable damper for separation of return and outdoor air streams (dual path)

The modified AHU is shown in Figure 7. The return air enters the AHU in the upper left chamber while the outside ventilation air enters the AHU in the lower left chamber. These two chambers are separated with a dual-path damper. Single-path operation occurs when the dual-path damper is open while dual-path operation occurs when the dual-path damper is closed. In addition, the return air chamber includes a bypass damper so that a portion of the return air can be routed around the DX cooling coil. Heat pipe heat exchangers were installed around the DX cooling coils. These heat pipes were initially uncharged to simulate operation without heat pipes. The heat pipes were then charged to determine the effects of heat pipes on system performance. Finally, the coiling coils were "face split" into three refrigerant circuits which were served by three separate compressors. This allows for part-load operation of the HVAC unit.

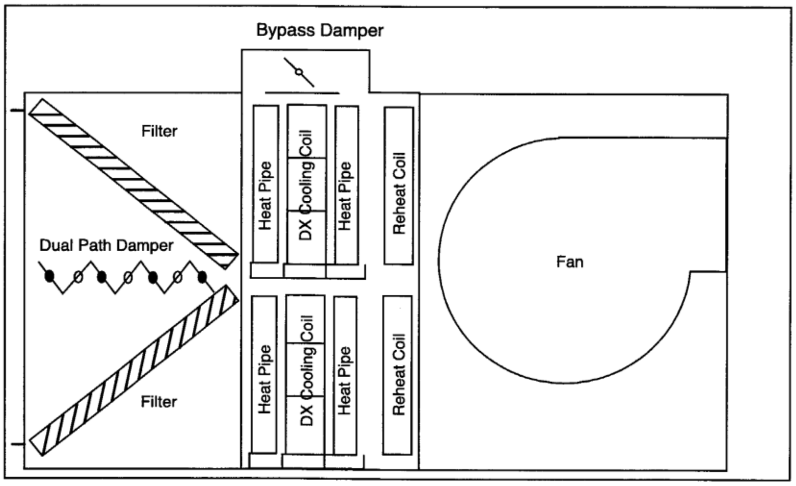

Figure 7. Modified AHU used in Khattar and Brandemuehl (1996) field study.

The operating strategies investigated by Khattar and Brandemuehl (1996) are summarized in Table 1.

Table 1. Operating Strategies Investigated by Khattar and Brandemuehl (1996).

Variable	Action to Improve Dehumidification	Base System	Improved System
HPHX	Charge the heat pipe	Uncharged	Charged
Supply airflow rate	Reduce airflow	1.0 cfm/ft^2	0.7 cfm/ft^2
Bypass damper	Open bypass damper	Closed	Open
Coil temperature	Lower suction pressure	48°F	40°F
Dual-path damper	Close damper	Open	Closed

Khattar and Brandemuehl (1996) noted that the HVAC system at the selected field site was undersized. However, important trends were observed. The heat pipe heat exchangers were found to provide a 28% improvement in dehumidification efficiency under typical summer operating conditions. Also, a 20% reduction in airflow produced a 17% improvement in dehumidification efficiency. Furthermore, the combination of the heat pipe heat exchanger and reduced airflow increased the dehumidification efficiency of the system by 42%.

In addition, Khattar and Brandemuehl (1996) found that the sensible loads were small compared to latent loads, which were driven by infiltration. It was estimated that the infiltration into this particular supermarket ranged from 0.11 cfm/ft^2 to 0.17 cfm/ft^2 (0.56 $L/s{\cdot}m^2$ to 0.86 $L/s{\cdot}m^2$). Khattar and Brandemuehl noted that effective humidity control requires that infiltration and ventilation be controlled to minimum acceptable levels. They recommend slight pressurization of the space.

Furthermore, Khattar and Brandemuehl (1996) also found that the sensible cooling provided by the refrigerated display cases was significant such that this particular supermarket did not require any net cooling until the average daily outdoor temperature exceeded 74°F (23°C).

Khattar (2004) investigated the performance of a dual-path water-source heat pump system in a supercenter located in Moore, OK. A schematic of the dual path system used in the field demonstration is shown in Figure 8. This system was used to provide variable ventilation rate through the ventilation air path, conditioned ventilation air below a set point of 45°F (7°C) at all airflow rates, and space heating through recovery of refrigeration waste heat from a circulating water loop. This system used a water-cooled DX coiling coil in the ventilation air path and a water-source heat pump in the recirculation air path. A central circulating water loop was used in which the waste heat from the refrigeration racks was transferred to the rooftop HVAC units. Cooling towers were used to reject the excess heat in the water loop which was not used for space heating.

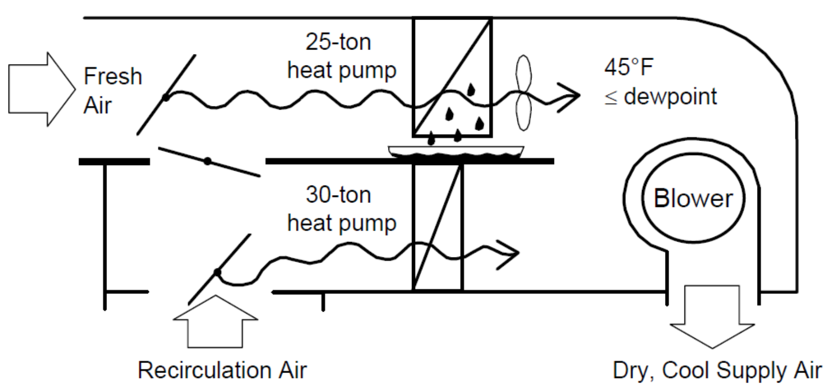

Figure 8. Dual path utilized in Wal-Mart field study (Khattar 2004).

Khattar (2004) found that this dual path system was capable of maintaining the indoor dew point between 45°F to 48°F (7°C to 9°C) regardless of outdoor dew point temperature (unless the outdoor dew point fell below 45°F (7°C), in which case, the indoor dew point also dropped). This corresponds to approximately 40% to 45% RH at 70°F (21°C). Furthermore, no additional heat was required to maintain space conditions in the winter other than the heat which was recovered from the water loop. Thus, it was demonstrated that the refrigeration waste heat could be practically and efficiently used, without the need for additional heat sources.

Desiccant Systems

Banks (1992) noted several advantages that desiccant systems have to offer as compared to single-path and dual-path systems. Based on the information from over 200 operating supermarkets, Banks claims that desiccant dehumidification has lowered operating costs. In addition, the cost effectiveness of the desiccant systems can be further enhanced by utilizing the waste heat from the supermarket refrigeration system to preheat the regeneration airflow. It was noted that desiccant systems can maintain the relative humidity as low as 30% at 75°F, with airflow rates as low as 0.5 cfm/ft^2 (2.5 L/s per m^2), as compared to dew points of 45°F to 50°F (7°C to 10°C) with airflow rates of 1 cfm/ft^2 (5.1 L/s per m^2) for conventional systems. Finally, Banks noted that with desiccant dehumidification, the central air conditioning is used for sensible cooling only. Thus, central air conditioning capacity can be reduced by 20% to 30%. In addition, central AC evaporator coils can operate at a higher suction pressure, thus increasing the COP.

On the other hand, Malone (1992) describes experiences with desiccant-based systems which are not quite as optimistic as those given by Banks (1992). Malone notes that, while desiccant-based systems can maintain humidity levels at 40%-45%, consideration of first cost and the ongoing maintenance requirements must be considered. Even though his organization was an early adopter of desiccant technology, they are now using more cost effective technologies such as heat pipes, dual coil and sub-cooled reheat concepts. Malone has found that the simple payback for a desiccant-based system can be 10 years or more. In addition, significantly more maintenance is required for desiccant systems as compared to conventional packaged rooftop units. In terms of first cost and operating costs for desiccant-based systems, Malone provides the following information:

- Compared to a custom, dual-path type supermarket HVAC system designed to maintain 45% RH, the first cost for a similarly sized desiccant system designed to maintain 40% RH is 37% more (cost includes system first cost, delivery, installation and ductwork).
- Compared to a custom, dual-path type supermarket HVAC system designed to maintain 45% RH, the annual operating costs (utilities and maintenance) of a similarly sized desiccant system designed to maintain 40% RH is 14% more.
- Compared to a conventional packaged RTU system designed to maintain 45% RH, the first cost for a similarly sized desiccant system designed to maintain 40% RH is 39% more (includes system first cost, delivery, installation and ductwork).
- Compared to conventional packaged RTU system designed to maintain 45% RH, the annual operating costs (utilities and maintenance) of a desiccant system designed to maintain 40% RH is 3% less.

Brandemuehl and Khattar (1997) provide data from field monitoring of a desiccant-based system installed in a 38,427 ft^2 New Jersey supermarket. The original HVAC system at this supermarket was a conventional rooftop DX cooling system which provided cooling and dehumidification. The system provided 75 tons (265 kW) of cooling and dehumidification capacity with an supply airflow of 28,000 cfm (13,200 L/s). The system was then retrofitted with an all-electric desiccant dehumidifying system. The modified system consisted of the original rooftop unit, with reduced airflow (18,000 cfm or 8,494 L/s) and reduced capacity (45 tons or 158 kW). The desiccant system had a supply air flow rating of 7,000 cfm (3,300 L/s), a regeneration air flow rating of 4,400 cfm (2,080 L/s) and a total fan power of 8.8 kW. Waste heat from the refrigeration system was used to regenerate the desiccant material. The desiccant material that was used in the system was newly developed, and could be effectively regenerated at temperatures as low as 130°F (54°C).

Two zones were created with the air distribution system: the desiccant system served the freezer aisles while a conventional DX system served the remainder of the store. Thus, the desiccant system could focus dehumidification in the freezer aisles where the greatest benefit from dehumidification could be realized.

Brandemuehl and Khattar (1997) found that the new desiccant-based system was able to maintain a substantial humidity gradient across the supermarket, where the freezer aisles had a lower humidity than the main grocery aisles. The humidity gradient was found to disappear as the overall humidity levels dropped. In addition, the desiccant system heats as well as dehumidifies the supply air. It was found that the additional heat added to the store did not appear to increase the cooling energy use of the HVAC system. The heat generally provided beneficial warming to the freezer aisles, improving occupant comfort. The heat added to the supermarket by the desiccant system was less than the need for reheat with a conventional HVAC system.

Capozzoli et al. (2006) modeled the variation in energy consumption versus humidity level for a supermarket that utilized a desiccant dehumidification HVAC system. A schematic of the hybrid system is shown in Figure 4. The 39,800 ft^2 (3,700 m^2) prototype supermarket was divided into four zones, including three peripheral zones (bakery, store and offices) and one core zone (sales center and check-out area). The three peripheral zones were each serviced by their own individual, traditional HVAC systems, while the core zone was serviced by a desiccant dehumidification HVAC system. Both a desiccant wheel hybrid HVAC system and a liquid desiccant hybrid HVAC system were modeled in the core zone. Furthermore, the energy modeling of the prototype supermarket was performed for three Italian cities (Bolzano, Rome and Trapani).

For the simulations performed for Rome, Capozzoli et al. (2006) found that, excluding refrigeration, the electrical energy use of the hybrid systems was 15% to 17% lower than that of the traditional HVAC system. In addition, natural gas use of the hybrid systems was 1% to 6% lower than that of the traditional HVAC system. There was a substantial summer reheating and winter dehumidification gas use by the traditional HVAC system which the hybrid systems did not have, while the hybrid systems had a regeneration gas usage which the traditional system did not have. Finally, Capozzoli et al. (2006) found that the overall energy costs (including both electric and gas) for the entire supermarket (including both HVAC and refrigeration) were 10% to 12% lower for the hybrid systems as compared to the traditional HVAC system.

Analysis of Several Dehumidification Alternatives

Using computer modeling, Mitchell et al. (1992) investigated the energy impacts of several dehumidification strategies for supermarket applications. Four HVAC systems were considered, including:

- Conventional (single path) system
- Improved single path system

- Dual path system
- Hybrid gas-fired desiccant/vapor compression cycle cooling system

These systems were evaluated for a typical supermarket located in Miami, FL. Annual energy simulations were performed and operating costs and first costs of the systems were analyzed.

The key findings from the energy simulation studies performed by Mitchell et al. (1992) include the following:

- In Miami, the supermarket HVAC system accounts for approximately 7% to 25% of the total store energy use.
- Lowering the store humidity reduced the supermarket's total electrical energy use for all but the conventional HVAC system. Shifting the dehumidification to the HVAC system rather than the refrigeration system was found to be more efficient since the HVAC compressors have a higher COP than the refrigeration compressors.
- Lowering the store relative humidity from 55% to 40% reduced the refrigeration system energy use by approximately 20%. This savings included the reduction in frost and anti-sweat heater operation as well as improved refrigeration system efficiency. The HVAC system energy use increased as much as 36% to 89% to maintain the lower humidity, but, for the high efficiency HVAC systems, the overall energy use of the store was reduced by about 6%. However, maintaining a reduced humidity with the conventional HVAC system increased the total energy use of the supermarket by 3%.
- The dual path and desiccant systems consumed about 10% to 15% less energy as compared to the conventional system. While the desiccant system used about the same electrical energy as the dual path system, the desiccant system also required natural gas to reactivate the desiccant.
- Compared to the improved single path and dual path systems, desiccant systems had similar operating costs but significantly higher initial costs.
- The heat pipe heat exchanger added little benefit to the single and dual path systems when the relative humidity was maintained at 55%. However, at 40% RH, the heat pipe heat exchanger produced a 7% energy benefit for the single path system but little improvement for the dual path system.
- At 40% RH, the energy costs associated with operating the high efficiency HVAC systems were about 9% to 14% less than that of the conventional HVAC system at 55% RH, and about 12% to 17% less than that of the conventional HVAC system at 40% RH.
- The improved single and dual path systems were estimated to cost less than the conventional system due to the reduced capacity requirements for the air conditioner. Desiccant systems were estimated to cost 20% to 24% more than the conventional HVAC system. These estimated installed costs included equipment purchase, installation, wiring, piping and ductwork costs for each system.

Walker (2001) presents the results of a study in which the energy consumption of a prototypical supermarket was modeled using various refrigeration and HVAC system configurations. Refrigeration and HVAC systems studied included conventional rooftop units, refrigeration heat reclaim and water-source heat pumps, as well as multiplex refrigeration with air-cooled condensing and mechanical subcooling and advanced, low-charge refrigeration systems. The simulations were performed for four cities in the U.S., including Worcester, MA, Washington, D.C., Memphis, TN and Los Angeles, CA.

Walker (2001) found that a heat reclaim system which incorporated a water-source heat pump HVAC system had the lowest annual operating costs as compared to the conventional HVAC and heat reclaim systems. Savings of the water-source heat pump system ranged from 2.2% to 30.7% as compared to the conventional HVAC system and from 3.8% to 22.1% as compared to the conventional HVAC system with heat reclaim for space heating. Walker noted that the results were extremely varied and were highly dependent on the local rates for electricity and natural gas.

Summary

Dual path and desiccant HVAC systems which maintain the indoor relative humidity of a supermarket at 40% were estimated to produce electrical energy savings of 10% to 15% as compared to conventional HVAC systems which maintain the indoor relative humidity at 55%. Retrofitting conventional HVAC systems with heat pipe heat exchangers was found to reduce the required air conditioning capacity of the supermarket HVAC units by 14% to 33% while improving the dehumidification efficiency by up to 28%. In addition, dehumidification efficiency can be increased by reducing the air flow rate through HVAC systems. A 20% reduction in air flow rate was reported to result in a 17% increase in dehumidification efficiency for a dual path HVAC system with heat pipe heat exchangers (Khattar and Brandemuehl 1996).

Desiccant systems were reported to maintain low humidity levels within supermarkets, on the order of 30% to 35%, with low air flow rates of approximately 0.5 cfm/ft^2 (2.5 L/s per m^2). In addition, supermarket air conditioning capacity requirements are reduced by 20% to 40% with desiccant systems. However, desiccant systems require a heat source in order to regenerate the desiccant material. This regeneration heat may be partially or fully supplied by waste heat from the supermarket's refrigeration system. Nonetheless, one study suggested that the natural gas use of a desiccant-based system can be 1% to 6% less than that of a conventional HVAC system (Capozzoli, et al. 2006).

Advanced HVAC systems that incorporate water-source heat pumps have been investigated (Walker 2001) (Khattar 2004). These systems use water loops to recover waste heat from the supermarket refrigeration system and transfer that energy to heat pump which can supply heating and cooling to the supermarket. No additional heat sources are required by these systems, and compared to conventional HVAC systems, energy savings of 2% to 31% may be achieved.

Operating costs of these various HVAC options have been found to vary widely with local rates for electricity and natural gas (Walker 2001). Thus, local utility rates must be considered when selecting cost-effective HVAC equipment for supermarket applications.

5. Whole Store Dehumidification vs. Isolated Sub-Dehumidification

Based on the assumption that the area near refrigerated display cases requires humidity control in order to obtain efficient performance from the refrigeration system while the dry goods sales area generally requires no humidity control, it has been proposed to maintain the refrigerated display case area at a lower humidity than the dry goods sales area. The effectiveness of this technique depends upon how fast moisture will diffuse through the air and migrate to the refrigerated display case area if the humidity in the dry goods area is higher than that in the refrigerated display case area. A few analytical and field studies have been performed to determine the feasibility of isolated sub-dehumidification strategies versus whole-store dehumidification strategies.

Early Studies

A US patent (no. 5,749,230) has been issued for the concept of isolated sub-dehumidification using a combination of conventional HVAC and desiccant dehumidification units to supply varying levels of humidity to different regions within a single-zone space, such as that typically found in supermarkets (Coellner and Calton 1998).

The inventors suggest that a desiccant dehumidification unit should supply dehumidified air near the refrigerated display case area of the supermarket while a standard HVAC system supplies higher humidity air to the main sales and produce areas. The suggested conventional/desiccant HVAC system layout for a supermarket is shown below in Figure 9. The entire supermarket (labeled Zone 20) is divided into two spaces: the “refrigerated space”, shown on the left in light blue (24), and the “non-refrigerated space”, shown on the right in light red (25). The produce cases, shown in dark red (21), are located in the non-refrigerated space while the refrigerated display cases, shown in dark blue (22), are located in the refrigerated space. A desiccant dehumidification unit (26) supplies air to the refrigerated space via supply ducts (26b) and the air is returned to the dehumidification unit via return ducts (26a). A conventional air conditioning unit (27) supplies air to the non-refrigerated space via supply ducts (27b) and the air is returned to the air conditioning unit via return ducts (27a). The temperature and humidity in the refrigerated zone are controlled by a thermostat (26c) and a humidistat (26d) while the temperature and humidity in the non-refrigerated space are controlled by a thermostat (27c) and a humidistat (27d).

The inventors claim that the arrangement shown in Figure 9 has been tested in a supermarket with a sales area of 20,000 ft^2 (1,860 m^2). A desiccant dehumidification unit rated to remove 150 lb/hr (68 kg/hr) of moisture while delivering air at 8,000 cfm (3,780 L/s) was used. In addition, a 40-ton (141 kW) air conditioning unit capable of delivering air at 24,000 cfm (11,300 L/s) was used. The arrangement was capable of maintaining the temperature inside the supermarket at 75°F (24°F) while the relative humidity gradient ranged from 45% in the refrigerated display case area to 55% in the main sales area. Furthermore, the inventors claimed that the energy needed for air circulation was substantially reduced. In addition to the layout described in Figure 9, two alternate sub-dehumidification HVAC system layouts are also specified in the patent.

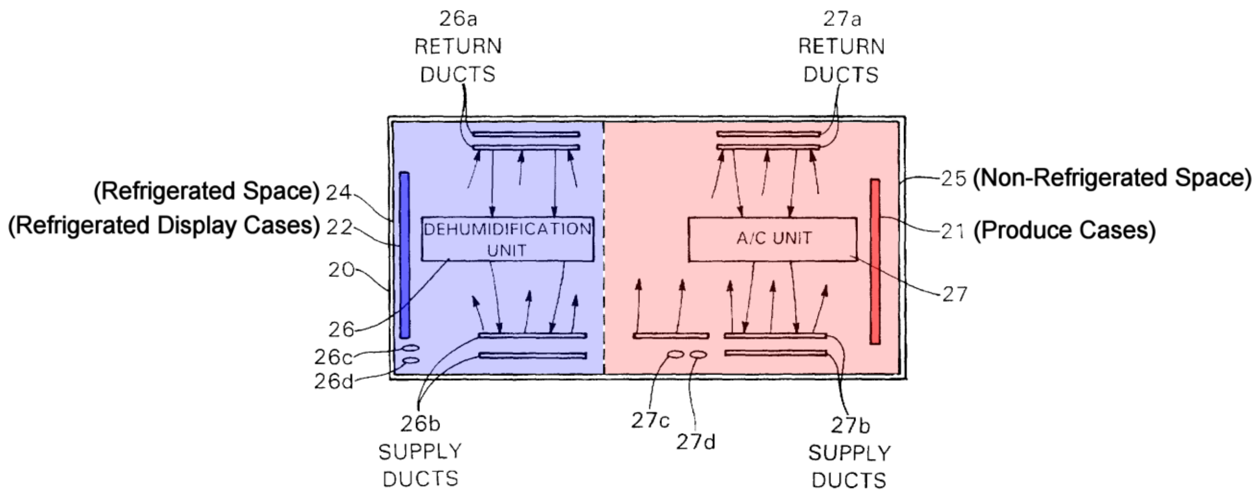

Figure 9. Conventional/desiccant HVAC system layout suggested by Coellner and Calton (1998).

As discussed previously in Section 4, *Control of Supermarket Humidity using HVAC Systems*, Brandemuehl and Khattar (1997) conducted field monitoring at New Jersey supermarket of a combined system, such as that described in the patent by Coellner and Calton (1998), where a desiccant-based dehumidification system was used in the refrigerated display case area and a conventional HVAC system was used in the main sales area. A newly developed desiccant material was used that could be effectively regenerated at temperatures as low as 130°F (54°C). Thus, waste heat from the refrigeration system was used to regenerate the desiccant material. Two zones were created for air distribution: the desiccant system delivered warm, dehumidified air to the freezer aisles while a conventional DX system delivered air-conditioned air to the remainder of the store. Thus, the desiccant system provided focused dehumidification in the freezer aisles where the greatest benefit from dehumidification could be realized.

The results from the study showed that a substantial humidity gradient existed across the supermarket, where the freezer aisles had a lower humidity than the main grocery aisles. During the summer, the difference in the dew point temperature between the grocery aisles and the freezer aisles was approximately 10°F, while during the winter, the difference in the dew point temperature between the grocery aisles and the freezer aisles was approximately 5°F. Brandemuehl and Khattar found that the humidity gradient disappeared as the overall humidity levels dropped.

In the system tested by Brandemuehl and Khattar, the desiccant dehumidification unit delivered warm, dry air to the display case area. The additional heat added to the supermarket by the dehumidification unit did not appear to increase the cooling energy use of the HVAC system. In fact, it was found that the heat provided beneficial warming to the freezer aisles, thereby improving occupant comfort. Before the desiccant system was operational, the temperature in the freezer aisles was 3°F to 4°F (1.7°C to 2.2°C) lower than that in the grocery aisles. However, after the desiccant system was operational, the temperature in the freezer aisles was only 1°F to 2°F (0.6°C to 1.1°C) lower than that in the grocery aisles. Furthermore, the heat added to the supermarket by the desiccant dehumidification unit was less than that which would have been required for reheat with a conventional HVAC system.

Since the desiccant dehumidification system used waste heat from the refrigeration system to regenerate the desiccant material, the only purchased energy required to operate the system was associated with the fans. Compared to a well-designed conventional HVAC system, Brandemuehl and Khattar found that the desiccant dehumidification system was approximately three to four times more efficient at removing moisture. Furthermore, Brandemuehl and Khattar reported that a slight reduction in refrigeration system energy consumption was noted when the desiccant dehumidification system was in operation.

Recent Studies

Recently, several modeling studies have been performed to determine the effect of humidity level on supermarket energy consumption.

It has been found that dehumidifying an entire supermarket with a desiccant-based HVAC system may not be cost effective (Walker 2001). Thus, based upon this experience, some supermarket chains limit the use of desiccant systems, or other low humidity HVAC systems, to the refrigerated display case areas of the supermarket where lower humidity levels are beneficial. It has been reported that desiccant-based systems work well in this application because they deliver warm, dry air directly into the refrigerated food aisles, which helps offset the overcooling produced by the display cases. The drying of the air is limited to the vicinity of the low temperature refrigerated cases, where maximum benefit to the refrigerated cases is obtained (Walker 2001).

As discussed previously in Section 4, *Control of Supermarket Humidity using HVAC Systems*, Capozzoli et al. (2006) modeled the variation in energy consumption versus humidity level for a supermarket that utilized a desiccant dehumidification HVAC system. The 39,800 ft^2 (3,700 m^2) prototype supermarket was divided into four zones, including three peripheral zones (bakery, store and offices) and one core zone (sales center and check-out area). The three peripheral zones were each serviced by their own individual, traditional HVAC systems, while the core zone was serviced by a desiccant dehumidification HVAC system. Both a desiccant wheel hybrid HVAC system and a liquid desiccant hybrid HVAC system were modeled in the core zone. Furthermore, the energy modeling of the prototype supermarket was performed for three Italian cities (Bolzano, Rome and Trapani).

The simulation results indicated that a total operating cost savings of between 5% to 13%, depending upon the climatic conditions, could be achieved by using a hybrid system rather than a traditional HVAC system in the central zone. These cost savings estimates include both electric and gas usage. In addition, for the city of Rome, it was found that the electrical energy consumption of the refrigeration system could be reduced by 11% to 12% by using a hybrid HVAC system. Similar results were noted for the other cities studied (Bolzano and Trapani). A simple payback of approximately 1 year was estimated for the hybrid HVAC system.

While the simulations performed by Capozzoli et al. (2006) modeled the supermarket as four zones, with traditional HVAC systems in the peripheral zones and a hybrid system in the central zone, this configuration is not necessarily the same as that in which the refrigerated display cases are serviced by a dehumidification unit while the remainder of the store is serviced by traditional HVAC. However, the results show that it could be beneficial to dehumidify only certain portions of the supermarket.

An on-going project funded by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE), Research Project 1467-RP, "Balancing Latent Heat Load between Display Cases and Store Comfort Cooling," is being conducted to optimize the design and operation of the combined HVAC and refrigeration systems of supermarkets. The researchers have reported preliminary results regarding the effect of refrigerated display case spatial distribution on supermarket humidity profiles (M. Brandemuehl 2010).

In this project, a prototypical 48,400 ft^2 (4,500 m^2) supermarket was developed and the effect of refrigerated display case distribution on the supermarket humidity profile was determined using computational fluid dynamics (CFD). The basic layout of the prototype supermarket is shown in Figure 10 (M. Brandemuehl 2010). The front of the store is at the top of Figure 10 and the refrigerated display cases, shown as grey rectangles, are generally distributed around the perimeter of the store.

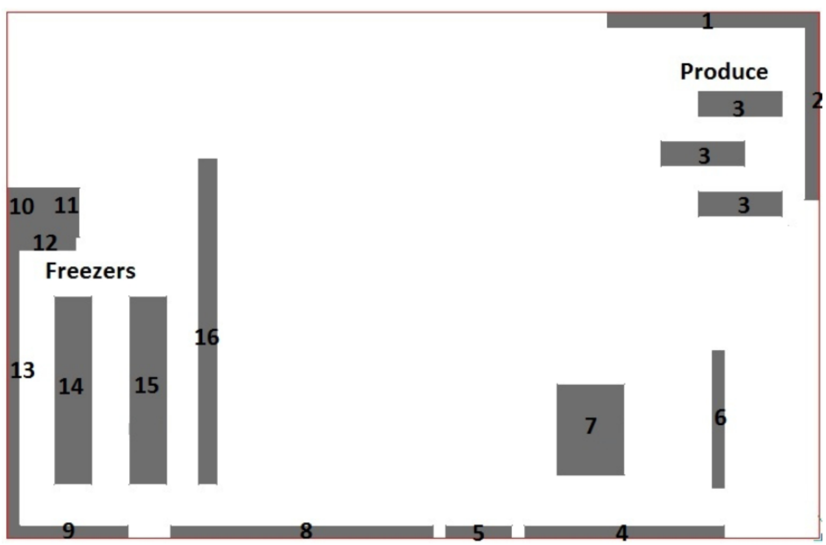

Figure 10. Distribution of refrigerated display cases in base store layout (M. Brandemuehl 2010)

The following four scenarios were modeled:

1. Base Case: A single air handler delivered supply air near the front of the store (top of Figure 10) and removed return air at the rear of the store.
2. Side Location: Same as the Base Case, but the refrigerated display cases were redistributed to be on one side of the store.
3. Scattered Location of Cases: Same as the Base Case, but the refrigerated display cases were scattered throughout the store.

4. Separate AHUs: Base Case layout with two separate air handling units that delivered different supply streams to the freezer area and the main sales area. The air handler in the freezer area supplied air at a lower humidity.

Figure 11 shows the humidity ratio profile within the supermarket for the base case. It can be seen that there is a significant variation in the humidity ratio throughout the store, with lower values of humidity ratio near the display cases and higher values within the dry goods area.

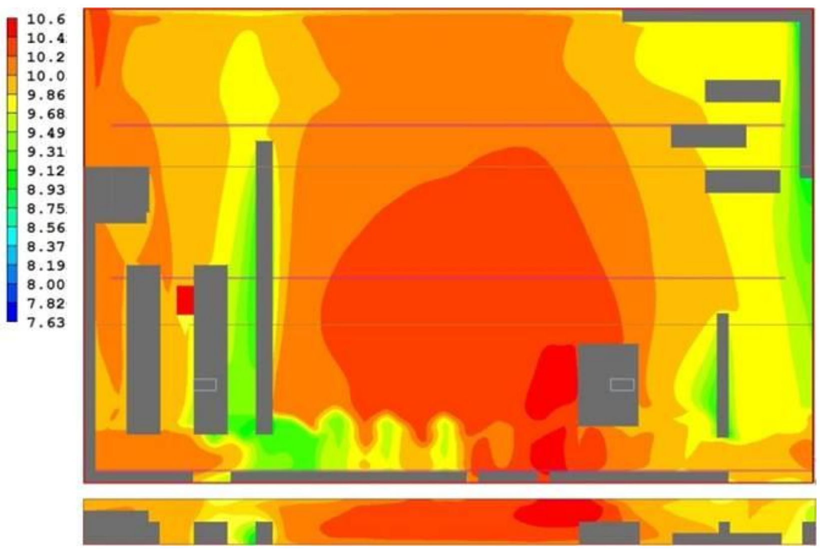

Figure 11. Humidity ratio (g/kg) profile for the base case (M. Brandemuehl 2010).

By locating the refrigerated display cases to one side of the supermarket, it was found that the humidity ratio within the dry goods area was quite uniform and greater than that within the display case area, as shown in Figure 12. In addition, as shown in Figure 13, it was found that the local humidity within the freezer area could be maintained at a noticeably lower level than that within the dry goods area by using a separate air handling unit for the freezer area. Thus, the preliminary results from this project show that there may be opportunities to reduce the energy use of the refrigeration system by maintaining lower humidity levels near the refrigerated display cases, without the need for dehumidifying the entire supermarket (M. Brandemuehl 2010).

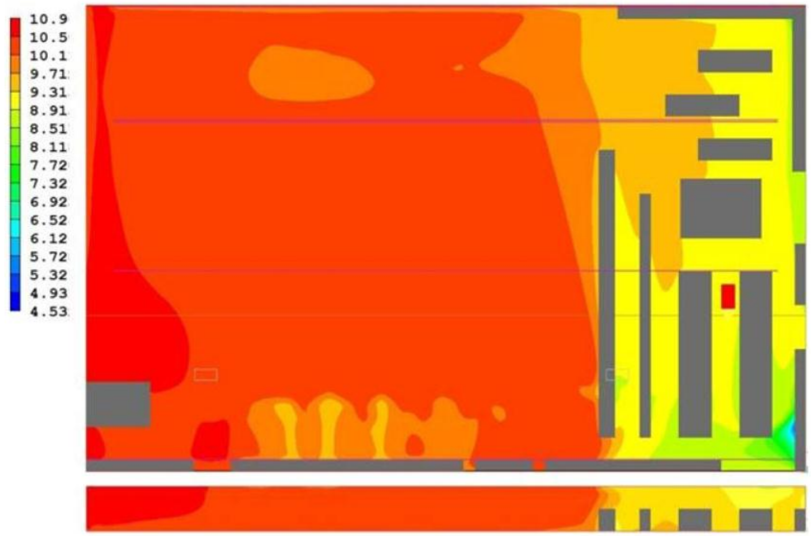

Figure 12. Humidity ratio (g/kg) profile for the side location scenario (M. Brandemuehl 2010).

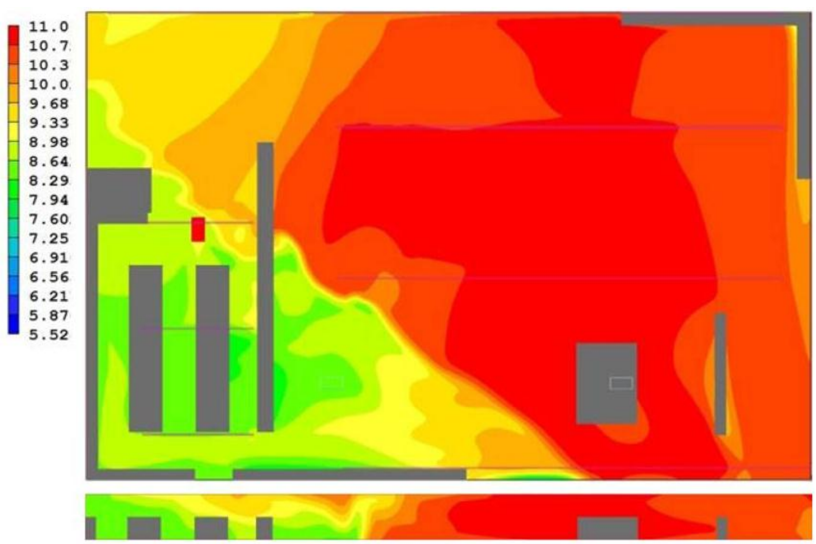

Figure 13. Humidity ratio (g/kg) profile for the separate AHU scenario (M. Brandemuehl 2010).

Summary

Several researchers have proposed the use of isolated sub-dehumidification in which dehumidified air is delivered to the refrigerated display case aisles while higher humidity air is delivered to the remainder of the supermarket area. It appears that this technique has merit and it is possible to maintain a humidity gradient throughout the store, with lower humidity levels in the refrigerated display case aisles and higher humidity levels in the remainder of the supermarket. Rather than maintaining the humidity of the entire supermarket at a lower level, it may be more cost effective to maintain only the refrigerated display case area at a lower humidity level.

6. Estimated Energy and Cost Savings of Isolated Sub-Dehumidification

As shown by Coellner and Calton (1998), Brandemuehl and Khattar (1997) and Brandemuehl (2010), it appears that the concept of isolated sub-dehumidification within is supermarket is feasible and it is possible to maintain a humidity gradient across the store. Thus, in this project, estimates have been made to determine the energy and cost savings associated with implementing isolated sub-dehumidification.

Two major systems are impacted by the implementation of isolated sub-dehumidification in a supermarket: the refrigeration system and the HVAC system. The estimated energy savings associated with each of these systems will be discussed below.

Refrigeration System Energy Savings

In order to estimate the energy savings associated with reducing the relative humidity within a supermarket, a prototypical supermarket was developed, based on the information provided by Westphalen et al. (1996). The prototypical supermarket has a plan area of 45,000 ft^2 and the refrigeration system for the supermarket consists of two medium temperature racks and two low temperature racks.

The medium temperature racks supply refrigeration to open, multi-deck cases and walk-in coolers while the low temperature racks supply refrigeration to reach-in (doored) freezer cases, coffin cases and walk-in freezers. The specifications of the refrigerated display cases and walk-in coolers/freezers used in the prototype store are given in Table 2 (Westphalen, et al. 1996). The refrigerated display cases and walk-in coolers/freezers are assumed to operate in a supermarket whose space conditions are maintained at 75°F (23.9°C), 55% relative humidity. Note that these conditions correspond to the rating conditions specified in ASHRAE Standard 72 for testing commercial refrigerators and freezers (ANSI/ASHRAE 2005). The baseline refrigeration load of the medium temperature display cases and walk-in coolers is 750,000 Btu/hr at 55% RH, while the baseline refrigeration load of the low temperature display cases and walk-in freezers is 300,000 Btu/hr at 55% RH.

Table 2. Specifications of Refrigerated Display Cases and Walk-In Coolers/Freezers (Westphalen, et al. 1996).

Temperature Range	Display Case/Walk-In	Length (ft) or Area (ft^2)	Baseline Load (Btu/hr·ft or Btu/hr·ft^2)	Baseline Total Load (Btu/hr)
Medium	Multi-deck cases, meat	120 ft	1500 Btu/hr·ft	180,000
	Multi-deck cases, other	260 ft	1500 Btu/hr·ft	390,000
	Walk-in cooler, meat	400 ft^2	60 Btu/hr·ft^2	26,000
	Walk-in cooler, other	2,600 ft^2	60 Btu/hr·ft^2	154,000
			Total, medium temperature:	750,000
Low	Reach-in cases	268 ft	560 Btu/hr·ft	150,000
	Coffin cases	128 ft	550 Btu/hr·ft	70,000
	Walk-in freezer	1,000 ft^2	80 Btu/hr·ft^2	80,000
			Total, low temperature:	300,000

The specifications of the baseline medium temperature and low temperature refrigeration racks are given in Table 3 (Westphalen, et al. 1996). Assuming a coefficient of performance (COP) of 2.5 for the medium temperature refrigeration system, a compressor power of 88 kW is required to satisfy the medium temperature loads. Similarly, assuming a COP of 1.3 for the low temperature refrigeration system, a compressor power of 68 kW is required to satisfy the low temperature loads.

Table 3. Baseline Compressor Rack Specifications (Westphalen, et al. 1996).

Compressor Rack	Saturated Suction Temperature (°F)	Saturated Discharge Temperature (°F)	System COP	Evaporator Load (Btu/hr)	Compressor Power (kW)	Heat Rejection (Btu/hr)
Medium Temperature (2 racks)	15	115	2.5	750,000	88	1,050,000
Low Temperature (2 racks)	-25	110	1.3	300,000	68	351,000
Totals				1,050,000	156	1,581,000

Based on the information provided in Section 3, *Effects of Humidity on Supermarket Refrigeration Systems*, estimates were made for the electrical energy savings of the compressors, defrost heaters and anti-sweat heaters, associated with reducing the supermarket relative humidity from 55% to 35%. These electrical energy savings estimates are given in Table 4.

Table 4. Estimated Compressor, Defrost Heater and Anti-Sweat Heater Electrical Energy Savings for Various Refrigerated Display Cases.

Case Type	Compressor Electrical Energy Savings (%)	Defrost Electrical Energy Savings (%)	Anti-Sweat Heater Electrical Energy Savings (%)
Multi-deck, meat (medium temperature)	20	25	67
Multi-deck, other (medium temperature)	20	0	0
Reach-in (low temperature)	3	5	67
Coffin (low temperature)	12	5	67

As shown in Table 5, the compressor electrical energy savings was used to estimate the reduction in the refrigeration load as a function of relative humidity. It can be seen that the medium temperature rack load decreased from 750,000 Btu/hr at 55% RH to 636,000 Btu/hr at 35 %RH. Similarly, the low temperature rack load decreased from 300,000 Btu/hr at 55% RH to 287,600 Btu/hr at 35% RH. In the analysis, it was assumed that reduced humidity levels in the supermarket did not affect the performance of the walk-in coolers and freezers.

Table 5. Reduction in Display Case Load due to Reduction in Relative Humidity.

Temperature Range	Display Case/Walk-In	Baseline Total Load, 55% RH (Btu/hr)	Total Load, 35% RH (Btu/hr)
Medium	Multi-deck cases, meat	180,000	144,000
	Multi-deck cases, other	390,000	312,000
	Walk-in cooler, meat	24,000	24,000
	Walk-in cooler, other	154,000	156,000
	Total	750,000	636,000
Low	Reach-in cases	150,000	145,600
	Coffin cases	70,000	62,000
	Walk-in freezer	80,000	80,000
	Total	300,000	287,600

The corresponding reduction in the compressor rack power requirements is given in Table 6. It can be seen that reducing the supermarket relative humidity from 55% to 35% reduced the total compressor power from 156 kW to 140 kW. In addition, the condenser heat rejection was reduced from 1,581,000 Btu/hr to 1,401,300 Btu/hr.

Table 6. Reduction in Compressor Rack Power Requirements due to Reduction in Relative Humidity.

Compressor Rack	System COP	Baseline Evaporator Load, 55% RH (Btu/hr)	Baseline Compressor Power, 55% RH (kW)	Evaporator Load, 35% RH, (Btu/hr)	Compressor Power, 35% RH (kW)
Medium temperature	2.5	750,000	88	636,000	75
Low temperature	1.3	300,000	68	287,600	65
Total		1,050,000	156	923,600	140

The reduction in electrical energy consumption of the medium temperature refrigerated display case accessories, consisting of evaporator fans, anti-sweat heaters, defrost heaters and lights, is shown in Table 7 while that for the low temperature display cases is shown in Table 8. The electrical energy consumption of the medium temperature display case accessories was reduced from 129,600 kWh/yr at 55% RH to 120,600 kWh/yr at 35% RH. For the low temperature display cases, the electrical energy consumption of the accessories was reduced from 348,200 Btu/hr at 55% RH to 221,700 Btu/hr at 35% RH.

Table 7. Reduction in Medium Temperature Display Case Electrical Energy Consumption due to Reduction in Relative Humidity (Westphalen, et al. 1996).

Case Type	Component	Baseline Power Consumption per foot (W/ft)	Baseline Total Power Consumption (kW)	Duty Cycle (%)	Baseline Energy Consumption, 55% RH (kWh/yr)	Estimated Energy Consumption, 35% RH (kWh/yr)
Multi-deck, meat (120 ft)	Evaporator fans	26.7	3.2	100	28,000	28,000
	Anti-sweat heaters	20	2.4	50	10,500	3,500
	Electric defrost heaters	135	16.2	5.6	7,900	5,900
	Lights	11.8	1.4	100	12,300	12,300
	Subtotal				58,700	49,700
Multi-deck, other (260 ft)	Evaporator fans	12.5	3.3	100	28,900	28,900
	Lights	18.3	4.8	100	42,000	42,000
	Subtotal				70,900	70,900
Total					129,600	120,600

Table 8. Reduction in Low Temperature Display Case Electrical Energy Consumption due to Reduction in Relative Humidity (Westphalen, et al. 1996).

Case Type	Component	Baseline Power Consumption per foot (W/ft)	Baseline Total Power Consumption (kW)	Duty Cycle (%)	Baseline Energy Consumption, 55% RH (kWh/yr)	Estimated Energy Consumption, 35% RH (kWh/yr)
Reach-in (268 ft)	Evaporator fans	20	5.36	96	45,100	45,100
	Anti-sweat heaters	71	19	96	159,800	52,700
	Electric defrost heaters	400	107	2	18,700	17,800
	Lights	33	8.8	100	77,100	77,100
	Subtotal				300,700	192,700
Coffin (128 ft)	Evaporator fans	10	1.28	100	11,200	11,200
	Anti-sweat heaters	24	3.07	100	26,900	8,900
	Electric defrost heaters	420	53.8	2	9,400	8,900
	Lights	0	0		0	0
	Subtotal				47,500	29,000
Total					348,200	221,700

Finally, the reduction in the total refrigeration system electrical energy consumption associated with reducing the relative humidity from 55% to 35% is summarized in Table 9. It can be seen that the total refrigeration system electrical energy use was reduced from 1,571,900 kWh/yr to 1,326,000 kWh/yr, corresponding to an annual savings of 15.6%, when the relative humidity was reduced from 55% to 35%. Note that this energy savings agrees fairly well with statements made by Capozzoli et al. (2006) and Getu and Bansal (2006) in which the total refrigeration system energy consumption is reported to decrease by approximately 1% for every 1% reduction in store relative humidity.

Assuming an average cost of $0.103 per kWh for electricity, the electrical energy cost savings for the refrigeration system, associated with reducing the humidity from 55% to 35%, is $25,328 per year (EIA 2011).

Table 9. Refrigeration System Electrical Energy Savings Associated with Reducing Relative Humidity from 55% to 35%.

Component	Baseline Electrical Energy Consumption, 55% RH (kWh/yr)	Electrical Energy Consumption, 35% RH (kWh/yr)	Electrical Energy Savings (%)
Two medium temperature racks	485,700	413,900	14.8
Two low temperature racks	375,300	358,700	4.4
Condensers	99,300	77,300	22.2
Medium temperature display cases	129,600	120,600	6.9
Low temperature display cases	348,200	221,700	36.3
Medium temperature walk-ins	83,800	83,800	0
Low temperature walk-ins	50,000	50,000	0
Total	1,571,900	1,326,000	15.6

HVAC System Energy Savings

Baseline HVAC System

Based on information reported by a variety of sources, the baseline HVAC system selected for the 45,000 ft^2 prototype supermarket was a conventional rooftop unit with DX cooling and gas heat (Flannick 1992) (Mitchell, et al. 1992) (Khattar and Brandemuehl 1996) (Brandemuehl and Khattar 1997) (Henderson and Khattar 1999) (Walker 2001). The capacity was selected to be 80 tons. A schematic of the conventional rooftop unit used in the prototype supermarket is given in Figure 2.

Based on information provided by a manufacturer, the energy specifications of a conventional 80 ton rooftop unit include the following (Johnson Controls 2009):

- Maximum cooling capacity: 890,000 Btu/hr
- Maximum compressor power: 80.6 kW
- Supply fan power: 25 kW
- Condenser fan power: 9 kW
- Gas input capacity: 1,125,000 Btu/hr
- Duty cycle: 0.4 (assumed)
- Fraction of year in cooling mode: 0.6 (assumed)
- Fraction of year in heating mode: 0.4 (assumed)

From these specifications, it is assumed that the baseline HVAC system has an electrical energy consumption of 276,000 kWh/yr and a gas consumption of 1,577 MMBtu/yr.

Isolated Sub-Dehumidification System

One potential method for achieving isolated sub-dehumidification in a supermarket could include the use of a desiccant dehumidification unit to supply dry air to the refrigerated display case area of the supermarket, and a conventional rooftop unit with DX cooling and gas heat to supply conditioned air to the remainder of the sales area.

In the proposed system, which is similar to the system investigated by Brandemuehl and Khattar (1997), it was assumed that the heat required to regenerate the desiccant would be supplied by the waste heat from the refrigeration system and no additional heat would be required. In addition, it was assumed that the desiccant dehumidification unit would supply 7,000 cfm of warm, dry air to the refrigerated display case area. The specifications of the desiccant system are as follows:

- Supply air flow rate: 7,000 cfm
- Supply fan power: 1.14 kW
- Regeneration fan flow rate: 2,331 cfm
- Regeneration fan power: 0.40 kW

Assuming a duty cycle of 0.4, the annual energy consumption of the desiccant dehumidification system would be 5,388 kWh per year.

The capacity of the conventional rooftop unit will be less than that required in the baseline case. Thus, the unit capacity for the rooftop unit was selected to be 50 tons. Based on information provided by a manufacturer, the energy specifications of a conventional 50 ton rooftop unit include the following (Johnson Controls 2009):

- Maximum cooling capacity: 540,000 Btu/hr
- Maximum compressor power: 54.0 kW
- Supply fan power: 11.3 kW
- Condenser fan power: 2 kW
- Gas input capacity: 300,000 Btu/hr
- Duty cycle: 0.4 (assumed)
- Fraction of year in cooling mode: 0.6 (assumed)
- Fraction of year in heating mode: 0.4 (assumed)

From these specifications, it is assumed that the conventional rooftop HVAC system for the isolated sub-dehumidification strategy has an electrical energy consumption of 141,400 kWh/yr and a gas consumption of 420 MMBtu/yr.

Comparison of Baseline and Isolated Sub-Dehumidification HVAC Strategies

The estimated electrical energy savings associated with isolated sub-dehumidification is:

$$\begin{aligned} E_{save} &= E_{baseline} - E_{dehumid} \\ &= 276{,}000 \text{ kWh/yr} - \left(5{,}388 \text{ kWh/yr} + 141{,}400 \text{ kWh/yr}\right) \\ &= 129{,}000 \text{ kWh/yr} \end{aligned} \tag{1}$$

which represents a savings of 53%. In addition, the estimated gas energy savings associated with isolated sub-dehumidification is:

$$\begin{aligned} G_{save} &= G_{baseline} - G_{dehumid} \\ &= 1{,}577 \text{ MMBtu/yr} - 420 \text{ MMBtu/yr} \\ &= 1{,}157 \text{ MMBtu/yr} \end{aligned} \tag{2}$$

which represents a savings of 27%.

Assuming an average cost of \$0.103 per kWh for electricity and assuming 1 cubic foot of natural gas is equivalent to 1,027 Btu with an average price for natural gas of \$14.62 per thousand cubic feet, it is estimated that the annual cost of operating the baseline HVAC system and the isolated sub-dehumidification HVAC system are as follows (EIA 2011):

- Baseline HVAC system: \$50,878
- Isolated sub-dehumidification HVAC system: \$20,543

Thus, the annual HVAC system operating cost savings associated with the proposed isolated sub-dehumidification strategy is \$30,335. This represents an annual HVAC system operating cost savings of 60% as compared to the conventional HVAC system.

Summary of Refrigeration and HVAC System Energy Use and Operating Cost

In summary, for the prototypical 45,000 ft^2 supermarket, the refrigeration system electrical energy savings associated with reducing the relative humidity from 55% to 35% was found to be 245,900 kWh/year. Furthermore, by implementing an isolated sub-dehumidification HVAC system strategy to achieve a reduced relative humidity in the refrigerated display case area, it was found that the electrical energy savings of the HVAC system was 129,000 kWh/year and the gas energy savings of the HVAC system was 1,157 MMBtu/year as compared to a conventional HVAC system strategy. Thus, the total energy savings associated with isolated sub-dehumidification is:

- Total Electrical Energy Savings: 374,900 kWh/year (20% savings)
- Total Gas Energy Savings: 1,157 MMBtu/year (27% savings)

By implementing an isolated sub-dehumidification HVAC system strategy, it is estimated that a 20% savings in refrigeration and HVAC system electrical energy can be realized. Furthermore, a 27% savings in gas energy consumption can be obtained. For the prototypical 45,000 ft^2 supermarket investigated in this project, the annual operating cost of the refrigeration and HVAC system could be reduced by $55,663 through the implementation of an isolated sub-dehumidification HVAC system strategy.

7. Implementation

Isolated sub-dehumidification in supermarkets may be implemented in either new construction or in retrofit applications.

New Construction

During the design of a new supermarket, it is recommended that a “refrigerated display case zone” be created in the supermarket, in which the refrigerated display cases, and especially the low temperature display cases, are located in the same general area of the supermarket. This “refrigerated display case zone” will be supplied with lower humidity air. However, it is suggested that the refrigerated fresh produce display cases should not be located in this refrigerated display case zone since the lower humidity will tend to dehydrate and wilt the produce. For best results, the fresh produce cases should be located on the opposite side of the supermarket from the refrigerated display case zone, in the “general sales area zone” which has a higher humidity level.

Figure 14 shows the suggested floor plan of a hypothetical supermarket utilizing the isolated sub-dehumidification strategy and illustrates the distribution of low- and medium-temperature refrigerated display cases which creates a “refrigerated display case zone”. Also shown is the “general sales area zone” which contains the refrigerated fresh produce cases as well as the canned and dry goods. Note that the majority of the medium-temperature and low-temperature refrigerated display cases are located in or near the “refrigerated display case zone” while the fresh produce cases are located on the opposite side of the supermarket in the “general sales area zone”.

Figure 15 shows the suggested layout for the supermarket HVAC systems and the associated duct work. The system labeled “HVAC 1” and its associated duct work supplies lower humidity air to the “refrigerated display case zone”, which includes the frozen food display cases, the deli/diary/beverage display cases and the fresh meat display cases. The system labeled “HVAC 2” and its associated duct work supplies higher humidity air to the “general sales area zone”, which includes the fresh produce display cases as well as the canned and dry goods.

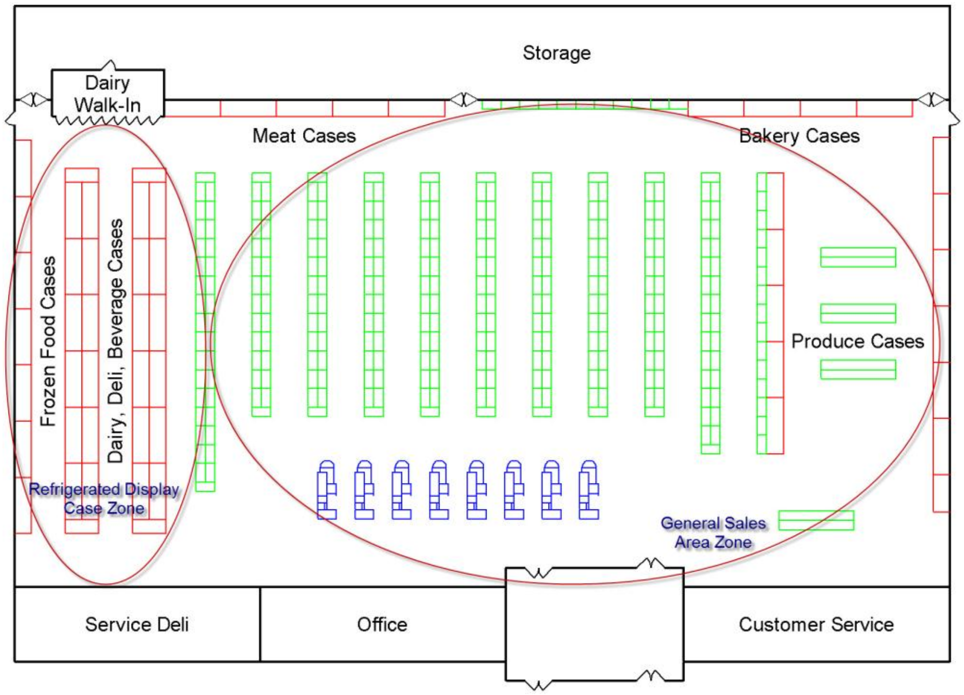

Figure 14. Suggested supermarket layout for the isolated sub-dehumidification strategy.

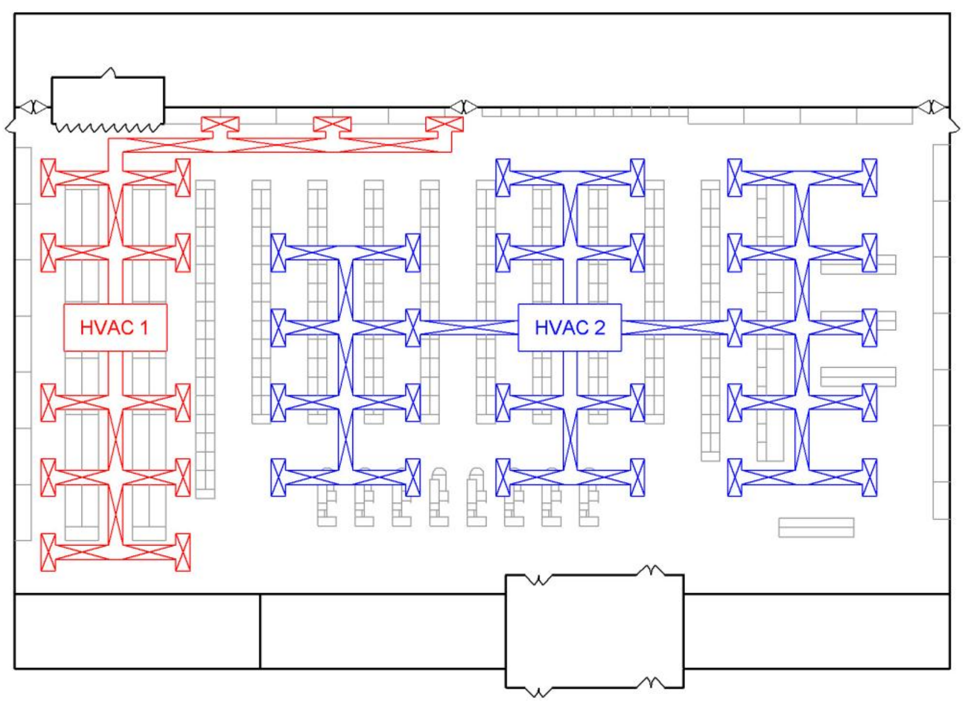

Figure 15. Location of supermarket HVAC systems and duct work for the isolated sub-dehumidification strategy.

To create the lower humidity environment within the refrigerated display case zone, several different technologies may be employed. A desiccant dehumidification unit can be used to supply less humid air directly to the refrigerated display case zone. This air may or may not be cooled prior to introducing it to the refrigerated display case zone. In fact, warm, dry air from the desiccant dehumidification unit may provide beneficial heating to the refrigerated display case aisles, thereby improving the comfort of the shoppers in the display case aisles and eliminating the need to cool the dehumidified air prior to supplying it to the refrigerated display case zone. To improve the energy efficiency of the desiccant dehumidification unit, waste heat from the refrigeration system may be used to preheat the air stream which is used to regenerate the desiccant.

Other HVAC system options for the refrigerated display case zone include the use of technologies such as advanced single path systems or dual path systems. In addition, these systems may utilize heat pipe heat exchangers to enhance dehumidification performance. Single path or dual path systems can effectively produce conditioned dry air that can be supplied to the refrigerated display case aisles. Furthermore, these systems can take advantage of the waste heat from the refrigeration system to reheat the air prior to introducing it into the refrigerated display case zone.

The air supplied to the general sales area zone can be of higher humidity, supplied by standard rooftop HVAC systems. Since higher humidity levels can be tolerated in the general sales area, and can even be beneficial for the refrigerated fresh produce cases, specialized HVAC equipment is not necessarily required in this zone. This aids in lowering the initial cost of the supermarket since traditional HVAC equipment is typically less expensive than the advanced HVAC systems for humidity control. Therefore, using standard rooftop units for the main sales area may keep initial costs lower as compared to using advanced humidity control systems for the entire supermarket.

As noted several studies, the air flow rate of the dehumidified air can be lower than that which is typically supplied by standard HVAC units. Thus an energy savings benefit can be realized by reducing the speed of the air supply fans. For example, the air flow for desiccant systems can be as low as 0.5 cfm per cubic foot of refrigerated display case zone floor area (Banks 1992) (Khattar and Brandemuehl 1996). However, others have noted that it may be beneficial to supply as much as 5 cfm of dehumidified air per cubic foot of refrigerated display case zone floor area (NYC WasteLess n.d.) (Desert Aire 2009).

Note that while Figure 15 shows only one HVAC unit in the refrigerated display case zone and in the general sales area zone, multiple units may be required to meet the supermarket's heating and cooling loads, depending upon the size of the store.

To maximize the energy savings which can be obtained by implementing the isolated sub-dehumidification strategy, it is suggested that energy efficient display cases be used in the refrigerated display case zone. This includes specifying doored display cases for both the low-temperature and medium-temperature applications. In addition, since the humidity in the refrigerated display case zone will be relatively lower, dew point controllers can be used to reduce the energy consumption of the display case anti-sweat heaters. In some instances, if the humidity level in the refrigerated display case zone is low enough, e.g., around 35%, then anti-sweat heaters may be unnecessary. Furthermore, LED display case lighting which is controlled with motion sensors will significantly reduce the electrical load and heat load of the refrigerated display cases. A high efficiency doored display case which incorporates anti-sweat heater control and LED lighting may reduce the electrical energy consumption by as much as 50% compared to a standard doored display case with continuously operating anti-sweat heaters and fluorescent lighting (Fricke and Becker 2011).

Retrofit Applications

In retrofit applications, the feasibility of implementing the isolated sub-dehumidification strategy is related to the level of retrofit which may be required. If the supermarket owner is considering a substantial whole-store remodel, then application of the isolated sub-dehumidification strategy may be attractive and cost effective. On the other hand, the pre-existing layout of the supermarket may hinder the implementation of isolated sub-dehumidification if funds are limited and/or only minor retrofits are planned.

The optimum candidate for an isolated sub-dehumidification retrofit would be a supermarket with a display case distribution similar to that shown in Figure 14. That is, the majority of the refrigerated display cases should be located along one side of the supermarket. Another factor to consider is the location of the existing HVAC units. The optimal retrofit scenario consists of one or more existing HVAC units that are located near the display case area so that they may be easily replaced with advanced dehumidification units. In addition, minimal modification of duct work would be desirable.

Note that if the refrigerated display cases, and especially the frozen food cases, are located in center of store, it would be very difficult to maintain a humidity gradient within the store. An isolated sub-dehumidification strategy would not be feasible with such a supermarket layout. Only if the display cases are moved to one side of the store would implementation of the isolated sub-dehumidification strategy be successful. Thus, in the situation where the display cases are located in the center of the store, the best approach would be to reduce the humidity of the entire store rather than to implement the isolated sub-dehumidification strategy.

As noted above for new construction applications, to maximize the energy savings which can be obtained by implementing the isolated sub-dehumidification in retrofit applications, the store owner may consider replacing older inefficient display cases with newer more energy efficient models.

8. Conclusions

The objective of this project was to determine the potential energy savings associated with reducing the relative humidity in the vicinity of refrigerated display cases in supermarkets, as compared to the widely accepted current practice of maintaining a relatively high and uniform humidity level throughout the entire supermarket. Existing and new strategies for maintaining lower relative humidity levels near the vicinity of refrigerated display cases were analyzed to determine their effectiveness.

Reducing the relative humidity within a supermarket can reduce the energy consumption of the supermarket's refrigeration system, including the energy consumption of compressors, defrost heaters and anti-sweat heaters. The refrigeration system energy savings is dependent upon refrigerated case type and refrigeration system temperature level. Lower humidity levels significantly improve the performance of open display cases while the benefit for doored display cases is minor. However, for doored display cases, a substantial reduction in anti-sweat heater energy requirements can be realized with lower humidity levels.

For medium temperature refrigeration systems, refrigeration system energy use decreases anywhere from 15% to 22% when the relative humidity is reduced from 55% to 35%. For low temperature refrigeration systems, refrigeration system energy use decreases anywhere from 0% to 17% when the relative humidity is reduced from 55% to 35%.

For medium temperature open display cases, defrost cycles can be reduced by 25% to 50% by reducing the relative humidity from 55% to 35%. Also, for low temperature cases, anti-sweat heaters can be deactivated at a relative humidity of 35%.

For the low temperature refrigeration system, an increase of 1% in power consumption results from every 1% increase in relative humidity from 35% RH to 50% RH. Furthermore, for the entire supermarket refrigeration system (medium temperature and low temperature), a 2% reduction in refrigeration system energy consumption can be achieved for every 0.56°F (1°C) reduction in the dew point temperature.

Dual path and desiccant HVAC systems which maintain the indoor relative humidity of a supermarket at 40% were estimated to produce electrical energy savings of 10% to 15% as compared to conventional HVAC systems which maintain the indoor relative humidity at 55%. Retrofitting conventional HVAC systems with heat pipe heat exchangers was found to reduce the required air conditioning capacity of the supermarket HVAC units by 14% to 33% while improving the dehumidification efficiency by up to 28%. In addition, dehumidification efficiency can be increased by reducing the air flow rate through HVAC systems. A 20% reduction in air flow rate was reported to result in a 17% increase in dehumidification efficiency for a dual path HVAC system with heat pipe heat exchangers (Khattar and Brandemuehl 1996).

Desiccant-based HVAC systems were reported to maintain low humidity levels within supermarkets, on the order of 30% to 35%, with low air flow rates of approximately 0.5 cfm/ft^2 (2.5 L/s per m^2). In addition, supermarket air conditioning capacity requirements are reduced by 20% to 40% with desiccant systems. However, desiccant systems require a heat source in order to regenerate the desiccant material. This regeneration heat may be partially or fully supplied by waste heat from the supermarket's refrigeration system. Nonetheless, one study suggested that the natural gas use of a desiccant-based system can be 1% to 6% less than that of a conventional HVAC system (Capozzoli, et al. 2006).

Advanced HVAC systems that incorporate water-source heat pumps have been investigated (Walker 2001) (Khattar 2004). These systems use water loops to recover waste heat from the supermarket refrigeration system and transfer that energy to heat pump which can supply heating and cooling to the supermarket. No additional heat sources are required by these systems, and compared to conventional HVAC systems, energy savings of 2% to 31% may be achieved.

Operating costs of these various HVAC options have been found to vary widely with local rates for electricity and natural gas (Walker 2001). Thus, local utility rates must be considered when selecting cost-effective HVAC equipment for supermarket applications.

Several researchers have proposed the use of isolated sub-dehumidification in which dehumidified air is delivered to the refrigerated display case aisles while higher humidity air is delivered to the remainder of the supermarket area. It appears that this technique has merit and it is possible to maintain a humidity gradient throughout the store, with lower humidity levels in the refrigerated display case aisles and higher humidity levels in the remainder of the supermarket. Rather than maintaining the humidity of the entire supermarket at a lower level, it may be more cost effective to maintain only the refrigerated display case area at a lower humidity level.

In this study, a prototypical 45,000 ft^2 supermarket was developed to investigate the energy use associated with implementing an isolated sub-dehumidification strategy in supermarkets. For this prototypical supermarket, the refrigeration system electrical energy savings associated with reducing the relative humidity from 55% to 35% was found to be 245,900 kWh/year. Furthermore, by implementing an isolated sub-dehumidification HVAC system strategy to achieve the reduced relative humidity level in the refrigerated display case area, it was found that the electrical energy savings of the HVAC system was 129,000 kWh/year and the gas energy savings of the HVAC system was 1,157 MMBtu/year as compared to a conventional HVAC system strategy. Thus, the total energy savings associated with isolated sub-dehumidification is:

- Total Electrical Energy Savings: 374,900 kWh/year (20% savings)
- Total Gas Energy Savings: 1,157 MMBtu/year (27% savings)

By implementing an isolated sub-dehumidification HVAC system strategy, it is estimated that a 20% savings in refrigeration and HVAC system electrical energy can be realized. Furthermore, a 27% savings in gas energy consumption can be obtained. For the prototypical 45,000 ft^2 supermarket investigated in this project, the annual operating cost of the refrigeration and HVAC system could be reduced by $55,663 through the implementation of an isolated sub-dehumidification HVAC system strategy.

The concept of isolated sub-dehumidification in supermarkets appears to be feasible and refrigeration and HVAC system energy savings can be achieved through its implementation. Therefore, strategies for implementing isolated sub-dehumidification in new store construction as well as in retrofit applications has been provided.

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