FLORIDA SOLAR ENERGY CENTER®

FINAL REPORT

Energy and Peak Power Reduction from Air Misting Products Applied to Commercial Condensing Units in Hot and Humid Climates

Final Report

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Executive Summary

Project Description

Commercial misting systems are designed to save energy consumed by refrigeration equipment by reducing the air temperature of air drawn into condensing coils that exchange heat to outdoor air. Water is released through very small openings which results in a fine mist. The mist is introduced to the air stream that is drawn across the coils. Evaporation of the fine water particles removes heat from the air and lowers its temperature. The drop in air temperature created by the mist is called T_{drop} in this report. The potential energy reduction depends in large part upon the evaporation rate of the mist, which in turn depends on outdoor relative humidity (RH). A low RH has more capacity to take on added moisture and results in a greater temperature drop than when high outdoor RH is present. The potential energy reduction also depends upon condenser coil capture effectiveness which is related to what portion of the evaporative cooled air goes through the coil and what portion escapes. Effective capture of mist cooled air can be significantly affected by wind speed and direction.

The objective of this project was to provide performance assessments of the following characteristics of a commercially available air misting product when applied to commercial refrigeration and air conditioning equipment located in a climate dominated by hot and humid weather:

- Seasonal cooling energy and peak demand reductions of cooling/refrigeration units.
- Electrical energy and peak demand usage of components of the installed misting system (such as water pumps).
- Water consumption amounts and time-of-day patterns of the misting system.
- Any potential degradation such as mineral scaling or oxidation on equipment and other surfaces exposed to the water spray, to the extent practicable.

Evaporation of fine water particles can lower the (dry bulb) temperature of air to a level approximating the wet bulb temperature. If we consider an average hot and humid summer cooling condition of 84°F and 75°F dew point temperature (RH = 74%), the wet bulb temperature is 77.5°F. Therefore, if the misting system is operating effectively to distribute and evaporate the mist fully across the stream of air impacting the condenser coils, then the entering temperature would be about 77.5°F (6.5°F cooler) than it otherwise would be. Based on data collected from other experiments, a reduction of 6.5°F in entering air temperature to the condenser of an air conditioning system with EER of about 10 could yield an improvement in air conditioner operating efficiency of about 15%.

Compared to drier climates, evaporative cooling has lower energy savings potential in Florida where the wet bulb temperature is generally higher than in dryer portions of the country. In Phoenix, Arizona, for example, common cooling conditions are 100°F dry bulb and 35°F dew point temperature, with a wet bulb temperature of 64°F. There is, therefore, a potential 36°F air temperature depression, or nearly 6 times as great as that on a typical Florida summer day.

So while the misting system is limited to producing 77.5°F entering conditions for 84°F and 75°F dew point ambient conditions in Florida, the same misting system can produce 64°F entering conditions for 100°F and 35°F dew point ambient conditions in Phoenix.

A supermarket was selected for these experiments in Melbourne, Florida. This site is less than 1 mile west of the Intracoastal Waterway and 3.5 miles from the Atlantic Ocean. Space conditioning is provided by a two-stage 50-ton roof top unit (RTU) and two smaller RTUs. Food refrigeration is provided through four separate systems. Systems 1 and 2 are low temperature refrigeration systems each served by Witt condenser rack model # RCS041VE. Systems 3 and 4 are medium temperature systems using Witt condenser rack model # RCS081VE. Each of the four food refrigeration systems is served by four compressors, with all compressors located inside a well-ventilated rooftop mechanical room. Figure ES-1 shows a rooftop image with the location of space conditioning and food refrigeration systems. The systems at this site are relatively old and showed signs of considerable weathering at the start of the project, compared to other supermarkets. However, they were considered by supermarket management to be in good working order with more years of life expected. All the refrigeration and space cooling systems use R22 refrigerant and are considered to be typical for supermarkets in Florida.

A Campbell data logger, temperature and relative humidity sensors, outdoor weather station, and energy meters were installed on site to measure indoor and outdoor conditions and system energy at 15 minute intervals.

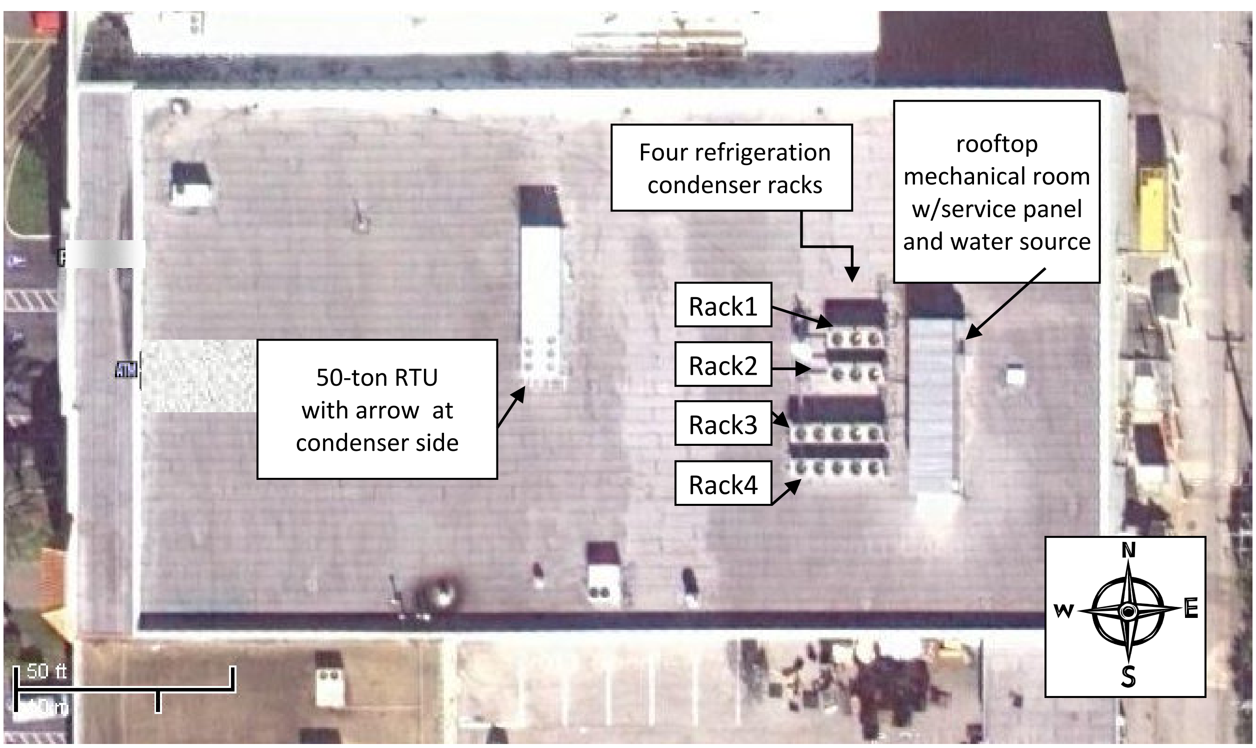

Figure ES-1. Rooftop view shows RTU#1 at roof center and four refrigeration condenser racks near mechanical room to the right. An approximate scale is shown in the lower left corner of illustration. **(Photo credit to Google Maps)**

Experimental Setup and Method

A search for commercially available mist systems appropriate for supermarket refrigeration condensing units found only two companies. Of these two, only one had the capability to turn mist off when condensing unit fans were not operating. This company, Cloudburst, was selected for this research. The system can cycle mist on and off based upon two variables: 1) outdoor temperature and 2) condenser unit fan on/off status. Temperature control cycles the mist off if outdoor conditions fall below a user-selectable temperature. The secondary control cycles the mist off if none of the condenser fans are operating for a given refrigeration system, even if outdoor temperature conditions are above the control point. A third-party humidistat was later added to the control strategy in early November in an attempt to further optimize energy savings and water conservation. The humidistat was designed to turn off the mist system whenever outdoor RH exceeded the set point of 75% (opposite from a dehumidifier control). When humidistat control was activated, the mist control thermostat set points were lowered to 50^{o}F with humidistat RH control set to 75%. This allowed evaluation of system performance at lower dry bulb temperatures but without permitting it to operate during most overnight hours or during rainy periods.

Misting is controlled independently for each of the four refrigeration systems. An additional mist system was also installed on the large RTU with independent mist control for each of the two stages of cooling. The mist system was purchased by Florida Power and Light and installed by the manufacturer representative. Research staff assisted in the installation and learned how to operate and maintain the mist equipment. During the project, the mist system was inspected about every three weeks by the research principle investigator.

Results

Some manufacturers claim up to 30% cooling energy savings and peak demand reduction from lower temperature of incoming air to condensing units produced by evaporative cooling from an air misting device. Actual annual savings measured in this project are on the order of 2% to 5%, which is of course much lower than advertised. The reasons for much lower energy savings, than manufacturer claims, are covered in the next few paragraphs.

There are two major temperature differential effects which are at play and which could account for savings shortfall. The first is wet bulb temperature. In hot and humid climates, wet bulb temperatures are high so the air temperature (dry bulb temperature) depression that can be achieved by mist evaporation is much more limited. Evaporative cooling has lower energy savings potential in Florida where the wet bulb temperature is generally higher than in dryer portions of the country.

It should be kept in mind that energy saving potential will be higher further inland, away from the Florida coast, where there are higher summer afternoon temperatures (92°-94^{o}F typical), lower RH (50%-55% typical), and lighter winds (more on wind effect later). By comparison, the Melbourne supermarket average outdoor temperature at 5pm was only 86.2^{o}F with 64.1% RH during July-August. Actual measured temperature depression at the Melbourne supermarket was only 5.7°F for the full range of hours for which the system operated. This is slightly less than the 6.5°F indicated wet bulb temperature depression. A significant portion of the difference between theoretical T_{drop} (6.5°F) and actual measured T_{drop} (5.7°F) may be related to wind influence.

The second major temperature differential which affects percent savings is a second type of temperature drop; temperature drop from ambient to evaporator coil. The potential energy savings depends to a substantial degree upon the total temperature drop that each system is trying to achieve, from outdoor environment to delivered temperature at the evaporator coil. This can be thought of as the coil-to-coil drop (condenser coil to evaporator coil drop). This temperature drop is different for each of the three types of systems examined.

- The 50-ton RTU A/C system is delivering about 50°F supply air to the space so the system is delivering cooling to the building with a refrigerant heat exchange across about 30°F, from about 80°F outdoors to 50°F supply air stream.
- The medium-temperature refrigeration systems deliver air to display cases at about 20°F, so the system is delivering cooling to the refrigerated display cases while driving heat across about 60°F, from about 80°F outdoors to 20°F supply air stream.
- The low-temperature refrigeration systems deliver air to display cases at about -20°F, so the system is delivering cooling to the freezer display cases while driving heat across about 100°F, from about 80°F outdoors to -20°F supply air stream.

For each of these systems, the 6.5°F T_{drop} is a different proportion of the coil-to-coil temperature drop. For the A/C system, 6.5°F is about 22% of the coil-to-coil drop of 30°F. For the medium-temperature cases, 6.5°F is about 11%. For the low-temperature cases, 6.5°F is about 6.5%. In general, then, we would expect energy savings of no more than 22% for the RTU, 11% for the medium-temperature food cases, and 6.5% for the low-temperature food cases.

Based on this simple analysis, we might then expect seasonal energy savings for this supermarket of no more than 10% or so, if the system was available and operating for all hours of the year. Since the mist system only operates perhaps 40-50% of the hours of the year, it can be seen that potential savings would be even smaller. Since actual measured T_{drop} is only 5.7°F, the savings potential would be further diminished. Based on this simplified analysis, it appears that potential energy savings of no more than 5% would be reasonable for the refrigeration and A/C systems combined. In a place like Phoenix, where savings might be as much as six times greater, total annual savings might then be on the order of 30%.

Measured Results

Annual Energy Savings

Regression analysis was performed for energy versus the outdoor temperature for the four refrigeration rack systems and the large RTU. Table ES-1 summarizes the predicted annual energy consumption and savings for mist system operation from April-November. Overall the eight month combined energy savings for the five systems is predicted to be 21,126 kWh or 2.8% (21,126/743,547 = 2.8%). Almost half of the savings on site can be attributed to the RTU.

Table ES-1 Predicted annual energy use and savings with mist off and on based on TMY3 data and regression analysis of monitored data.

Test Configuration	Annual (8mo.) kWh	Annual Savings kWh	Annual Savings %
Rack 1 Mist Off	113625		
Rack 1 Mist On	110697	2928	2.6%
Rack 2 Mist Off	141444		
Rack 2 Mist On	138975	2469	1.7%
Rack 3 Mist Off	152111		
Rack 3 Mist On	148408	3703	2.4%
Rack 4 Mist Off	150975		
Rack 4 Mist On	149174	1801	1.2%
RTU1 Mist Off	185392		
RTU1 Mist On	175167	10225	5.5%
Rack and RTU Total Savings		**21126**	**2.8%**
Rack Total Savings		**10901**	**2.0%**

Summer Peak Demand Reduction

The mist system only has the potential to reduce summer utility peak demand. There is no winter utility peak demand reduction possible since the mist system should not be operated below 45°F. Measured hourly energy demand versus outdoor temperature was used in linear regression analysis. The regression equations from each system configuration were used with the utility summer peak hour temperature for each of four cities using TMY3 data. The peak values were then weighted in the same proportions as for the annual energy analysis. The results are shown in Table ES-2. The combined net peak reduction is 2.77 kW (1.5%). One out of five cases shows a small negative peak reduction of -0.23kW for rack 1. It is not expected that the mist system is actually increasing energy use in this case. Rack 3 had a peak reduction very close to zero kW. Essentially, peak demand reduction is not demonstrated for Rack 1 and 3 during the hottest hours of the hottest days of the year. These systems may be at or just under capacity during the hottest weather and instead of lowering the peak demand, the mist system may be increasing the capacity some without a measureable reduction in power. The highest reduction predicted is on Rack 4 at 3.59 kW (10.6%).

Table ES-2 Predicted utility summer peak demand and peak demand reduction with mist off and on based on TMY3 data and regression analysis of monitored data.

Test Configuration	Peak Demand kW	Peak Demand Reduction kW	Peak Demand Reduction %
Rack 1 Mist Off	23.51		
Rack 1 Mist On	23.75	-0.23	-1.0%
Rack 2 Mist Off	29.67		
Rack 2 Mist On	29.06	0.61	2.1%
Rack 3 Mist Off	35.41		
Rack 3 Mist On	35.34	0.07	0.2%
Rack 4 Mist Off	34.25		
Rack 4 Mist On	30.23	4.01	11.7%
RTU1 Mist Off	63.29		
RTU1 Mist On	61.99	1.30	2.1%
Total Mist OFF kW	**186.13**		
Total Mist ON kW	**180.37**		
Total Peak (net) Reduction		**5.76**	**3.2%**

Another method of peak demand savings analysis was also employed. Profiles of energy consumption using twenty-four hour intervals were created using 7-day periods of mist on and off operation. Data for each test configuration was reviewed to seek comparable weather periods selected from July and August. Customer peak reduction and indicated hour of the customer peak demand are shown in Table ES-3. Customer peak demand reduction is slightly greater than the utility peak demand reduction with a total reduction of 6.3 kW compared to 5.8 kW.

Table ES-3. Customer refrigeration peak power and peak power reduction based on composite of summer days.

System	Mist OFF kW	Mist ON kW	Savings kW	Savings %	Mist ON Peak Hour	Mist OFF Peak Hour
Rack 1	23.7	21.6	2.1	8.7%	19	19
Rack 2	29.5	28.6	0.9	3.2%	17	17
Rack 3	35.2	35.1	0.1	0.2%	17	17
Rack 4	32.8	30.8	2.0	6.0%	15	15
RTU	61.2	59.9	1.4	2.1%	16	16
Total	**182.5**	**176.1**	**6.3**	**3.5%**		
Rack Total	**121.2**	**116.2**	**5.0**	**4.1%**		

Wind Impacts Upon Air Cooling Potential

Wind has been found to be a significant cause for low energy savings. Figures ES-2 and ES-3 show T_{drop} resulting from evaporative cooling at the three fan locations of Rack 2 on two hot summer days which have different wind conditions. T_{drop} is the drop in dry bulb temperature that occurs when the mist evaporates and cools the air entering the condenser coil. Outdoor temperature and RH are also shown on the Y-axis located on the right side of figure. The first day has a dominant wind out of the west and an afternoon thunderstorm. The second day has a typical dominant east wind and no rain.

The solid green line of Fan 3 T_{drop} during the period between 12pm – 3pm is the key variable to focus on when comparing Figures ES-2 and ES-3. In Figure ES-2 (Aug.4), Fan3 T_{drop} is about 63% lower than T_{drop} from Fans 1 and 2. However in Figure ES-3 (Aug.20) Fan 3 T_{drop} is only about 1.6% lower than the average T_{drop} of Fans 1 and 2. Wind direction is the key cause for the difference between T_{drop} of Fan 3 and the other two fans in this case. More specifically, the wind is causing the most of the mist supplied to fan 3 to be blown away before it could be captured by the condenser fan. The wind speed averaged 6.5 mph out of the west on August 4, but was 10.4 mph (60% greater) out of the east on August 20. So, although the wind speed was greater on the 20th, the mechanical room provides a windscreen for easterly winds, limiting the wind from blowing away significant portions of the mist-cooled air on the racks. Another noteable item about Figure ES-3 is that the mist system ran for 24 hours on August 20 because the outdoor temperature remained above 80°F.

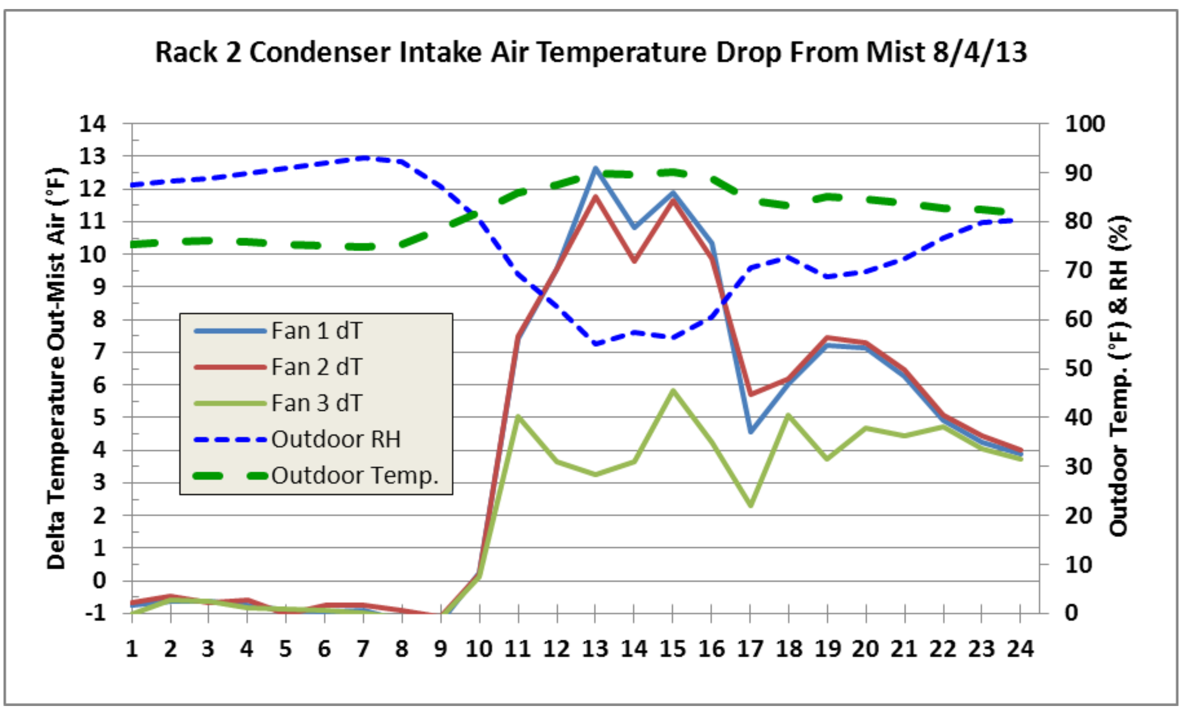

Figure ES-2. Data shows trend of temperature drop from mist over 24 hour period. Fan 3 on west side has less T_{drop} due to dominant westerly wind. Rain occurred about 3:45pm.

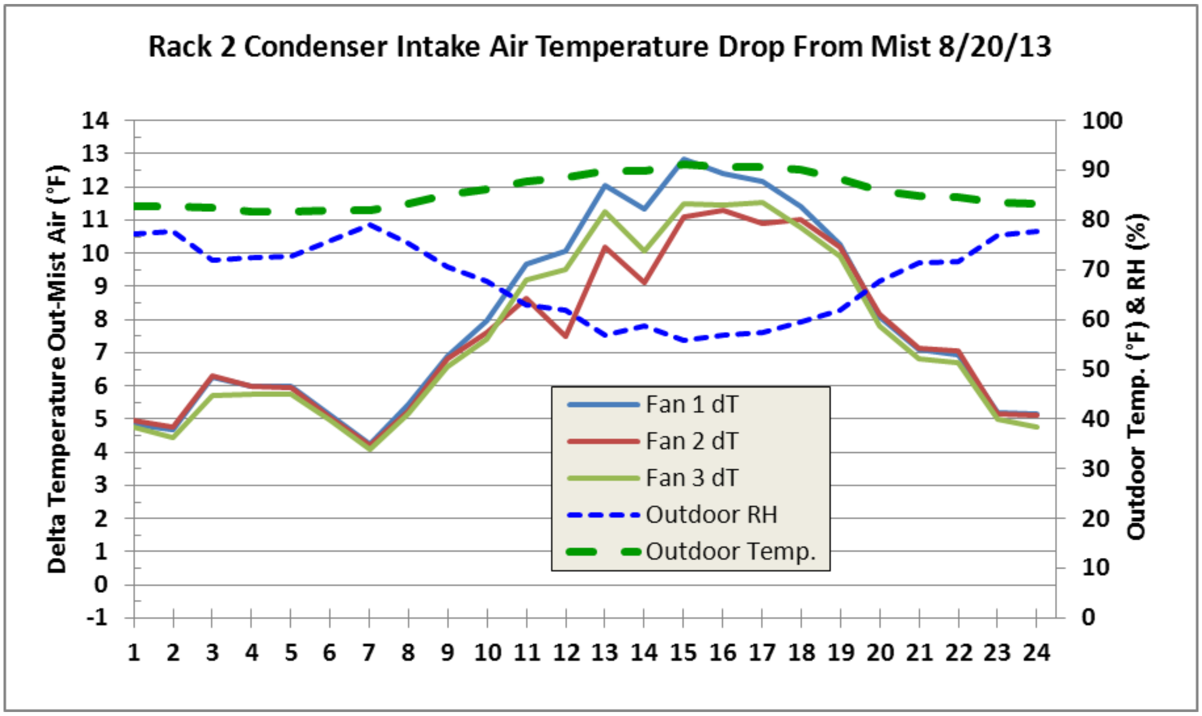

Figure ES-3. This shows temperature drop with wind out of east and no rain.

Table ES-4 provides a comparison of T_{drop} and T_{drop} potential ($T_{drop,pot}$) for 12 PM to 3 PM periods on August 4 (west wind) and August 20 (east wind). This is the time of day when the outdoor RH is at its lowest and $T_{drop,pot}$ is greatest. The first row shows data for the west wind (August 4) and the second row is for the east wind (August 20). The east fan T_{drop} most closely approaches $T_{drop,pot}$ on these two days; it is 8.8% lower than Tdrop,pot with west wind and only 4.7% lower with an east wind. The middle fan T_{drop} approaches $T_{drop,pot}$ less effectively compared to the east fan on these two days; it is 14.3% lower than $T_{drop,pot}$ with west wind and 20.0% lower with the east wind (it is unknown why the middle fan T_{drop} shows a lower T_{drop} for the east wind versus the west wind). The west fan T_{drop} is seriously impacted by the west wind. T_{drop} is 67.2% lower than T_{drop},pot with west wind and 13.7% lower with the east wind.

Table ES-4. Outdoor drybulb (Tdb) and wet bulb (Twb) temperatures and mist cooling potential compared to measured air temperature drop (T_{drop}) from ambient drybulb temperature during period 12 PM - 3 PM on August 4 and August 20, 2013, respectively.

	Tdb (°F)	Twb (°F)	T_{drop} potential (°F)	Fan1(East) T_{drop} (% difference)	Fan2 (middle) T_{drop} (% difference)	Fan3 (west) T_{drop} (% difference)
West wind Aug.4	89.9	77.0	12.91	11.77 (8.8%)	11.06 (14.3%)	4.24 (67.2%)
East wind Aug.20	90.3	77.7	12.67	12.08 (4.7%)	10.14 (20.0%)	10.93 (13.7%)

Air velocity measurements taken at the intake side of the coils found 500-600 feet per minute right at the coil. Velocity falls off quickly as you get further from the coil allowing wind to easily disturb the capture. Even with the mechanical room shielding easterly wind, air turbulence was observed wafting the mist about at times. It is clear from these results that wind speed and direction have an impact on the delivery of the evaporatively cooled air to the condenser coils. In some cases, these impacts can decrease the energy savings performance of the misting system by as much as 67% for a portion of the condenser coil system.

Conclusions

Annual energy savings over an eight month period were found to be about 2.0% for the refrigeration systems and 5.5% for the large A/C system. Peak demand savings for the hottest hours of the hottest days were found to be about 3.0% for the refrigeration systems and 1.0% for the large A/C system. Table ES-5 provides summary of information from both Tables ES-1 and ES-2. Table ES-5 shows the predicted annual and utility summer peak demand use and reductions for mist off and on. Clearly, these savings fall far short of the 30% savings threshold promoted by some manufacturers. Based on regression analysis and TMY data analysis for four Florida cities (weighted for the FPL service territory), total eight month energy savings is predicted to be 21,126 kWh ($2113 or 2.8%). These savings are about 3.5 times lower than expected for this study location.

Table ES-5 Predicted annual energy and utility summer peak demand use and savings with mist off and on based on TMY3 data and regression analysis of monitored data.

Test Configuration	Annual (8mo.) kWh	Annual Savings kWh	Annual Savings %	Peak Demand kW	Peak Demand Reduction kW	Peak Demand Reduction %
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Rack and RTU Total Savings		**21126**	**2.8%**			
Rack Total Savings		**10901**	**2.0%**			
Total Mist OFF kW				**186.13**		
Total Mist ON kW				**180.37**		
Total Peak (net) Reduction					**5.76**	**3.2%**

Total system savings must also account for the cost of mist pump power, water used, and maintenance. Pump power used over 8 months of mist system operation would be about 705 kWh. Water used over the same period would be about 192,000 gallons. Based on assumed average costs of $0.10 per kWh and $4.33 per 1000 gallons, the operational cost becomes $71/year for mist pump power and $831 in water costs. Commercial users should expect ongoing maintenance of mist systems and budget at least $200 per eight months of operation to cover cost of service labor and parts. Net annual savings is then reduced to $1011 ($2113 - $902 - $200 = $1011). Based on an installed mist system cost of $10,140, the simple payback is 10 years. Based on our observations, it is unlikely that the mist system would last 10 years in a hot and humid climate, especially when located near the coast.

The energy savings produced by this system may be less than what might be expected at other Florida locations. Reasons for diminished savings:

- Because the site is close to the Atlantic Ocean, it experiences higher average RH. Review of the data found that 51% of all hours from April 1 – November 30 had hourly average outdoor RH > 70%. Out RH > 75% occurred in 38% of the time.
- It was found that wind was sweeping mist away. It is likely that some non-coastal locations will experience reduced wind speeds, especially at peak demand periods.
- The RTU at this store was not controlled solely by indoor temperature. It was also controlled by indoor dew point temperature or RH. Space has substantial cooling by food refrigeration cold air spillage, even if RTU not cooling.
- The RTU was not operating correctly and required service throughout much of the monitoring period.

It was found that controlling the mist system with a humidistat was cost-effective and avoids water accumulation and run-off problems on the roof. By deactivating the mist system when ambient RH is lower than a desired set point, it yields significant water use savings. It also provides a way to optimize system operation during the shoulder months (spring and fall). The water saved would pay for installing a humidistat in about 2 years.

Commercial users should expect that ongoing maintenance of the mist system will be required. Typical maintenance involves replacing or cleaning clogged mist heads and replacing water pre-treatment filters every 6 months or 500 hours. The mist lines should be inspected for wear and replaced if showing any signs of abrasion. Monthly inspection is essential to verifying that the system is operating as expected and that water is not being wasted. Inspection and maintenance should only require about 30 minutes per month. At the end of the season it is recommended to drain the pump module and store in an area above freezing temperature.

Concerns about advanced rates of corrosion may be founded, particularly for older equipment exposed to coastal salt-laden air. This site had equipment with pre-existing areas of surface corrosion before the mist system installation. However an advanced rate of surface corrosion was observed on portions of both the condenser rack frame members and lower supporting beam structure during the mist system operation over eight months. The proximity to the ocean three and a half miles away provided a source of salt air that is a significant factor in corrosion at this site even without a mist system. Newer equipment with adequate surface finish, not in salt air environments, would have better protection from advanced rates of corrosion during misting.

Limited energy savings and the high cost of water significantly impact the cost-effectiveness of the refrigeration misting system evaluated at this site. Net energy savings and payback could be significantly improved if better shielding from wind could be implemented. Also, installing a rainwater cistern would provide an additional cost savings. A 10,000 gallon reservoir, for example, would provide for about eleven days of peak mist use in the summer. Based on monthly average rainfall in Florida, there would be enough rain on average to provide all the water needs, although some utility water may be needed during longer than average dry spells. The installed cost of a 10,000 gallon cistern is estimated to be approximately $8500. It is a costly option with a payback of about 10 years assuming $830/year water cost. While a 10-year payback may not be particularly attractive, it would also have the advantage of reducing stress on valuable Florida water resources and limit storm water runoff into local waterways.

Acknowledgements

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