EMERGING, COST-EFFECTIVE APPLICATIONS FOR DESICCANT DEHUMIDIFICATION IN THE U.S.

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ABSTRACT

Gas-fired desiccant dehumidification systems are now being specified for many supermarkets, ice arenas and cold warehouses, with installations now numbering in the thousands. Their cost effectiveness is due to the economic benefits of improved refrigeration operations resulting from the introduction of drier air. New application niches in the commercial sector are emerging due to the increased outside air quantities required by Indoor Air Quality codes and standards such as ASHRAE Standard 62-1989. In this paper, a competitive economic analysis of desiccant and other HVAC equipment, generated using a new desiccant screening tool driven by DOE-2.1E simulations, will be presented for several representative buildings in selected U.S. cities

INTRODUCTION

Gas-fired desiccant dehumidification systems are now being specified into many supermarkets, ice arenas and cold warehouses, with installations now numbering in the thousands. Their cost effectiveness is due to the economic benefits from improved refrigeration operations resulting from the introduction of drier air. The benefits are similar to the improvements realized in the industrial sector where dry air increases productivity in moisture sensitive manufacturing processes or reduces humidity related losses in storage applications. The commercial sector though has few, if any, other niches which can generate these "process improvement" economics. So what is driving desiccants from these niches into the commercial market mainstream? Well the answer has been anticipated for years -- it's Indoor Air Quality (IAQ).

Outside Air Requirements

In the commercial sector today, IAQ equates directly to the quantity of fresh air brought into a building, as prescribed by the 1989 version of ASHRAE Standard 62, "Ventilation for Acceptable Indoor Air Quality". This 1989 edition increased outside air requirements by 2 to 4 times over the 1981 version in response to the appearance of sick building syndrome of the 1980's. During the early 1990's, these ventilation rates were adopted by all three major model building codes (BOCA 1996, SBC 1994, ICBO 1994). In turn, those revised model codes were accepted into many state and local codes by the mid 1990's. This results in air handling units processing large percentages of outside air (%OA) in many commercial building types. It is these moisture laden outside air streams that are now requiring new air handling equipment solutions to properly control humidity.

Humidity Control Requirements

Increased outside air volumes can result in periods of increased indoor humidity levels in non-arid climates. Examples of the harmful effects of elevated humidity levels on humans and buildings have been documented (Sterling 1986, AHMA 1993). As manufacturers and specifiers react to the magnitudes of moisture present in outside air and to the threat posed to IAQ by uncontrolled indoor humidity, alternative HVAC equipment solutions to precondition outside air will be increasingly utilized to isolate and solve this problem.

Emerging Desiccant Applications

In many applications, gas-fired desiccant systems can be a cost effective choice in commercial building systems which treat larger fractions of outside air. These emerging applications include:

- Hospital Surgical Suites
- Hotels
- Theaters
- Schools
- Restaurants
- Retail Stores
- Nursing Homes

Figure 1 shows the design sensible heat ratios (SHR) for these applications in Atlanta for both the 2.5% dry-bulb and 2.5% dewpoint design conditions

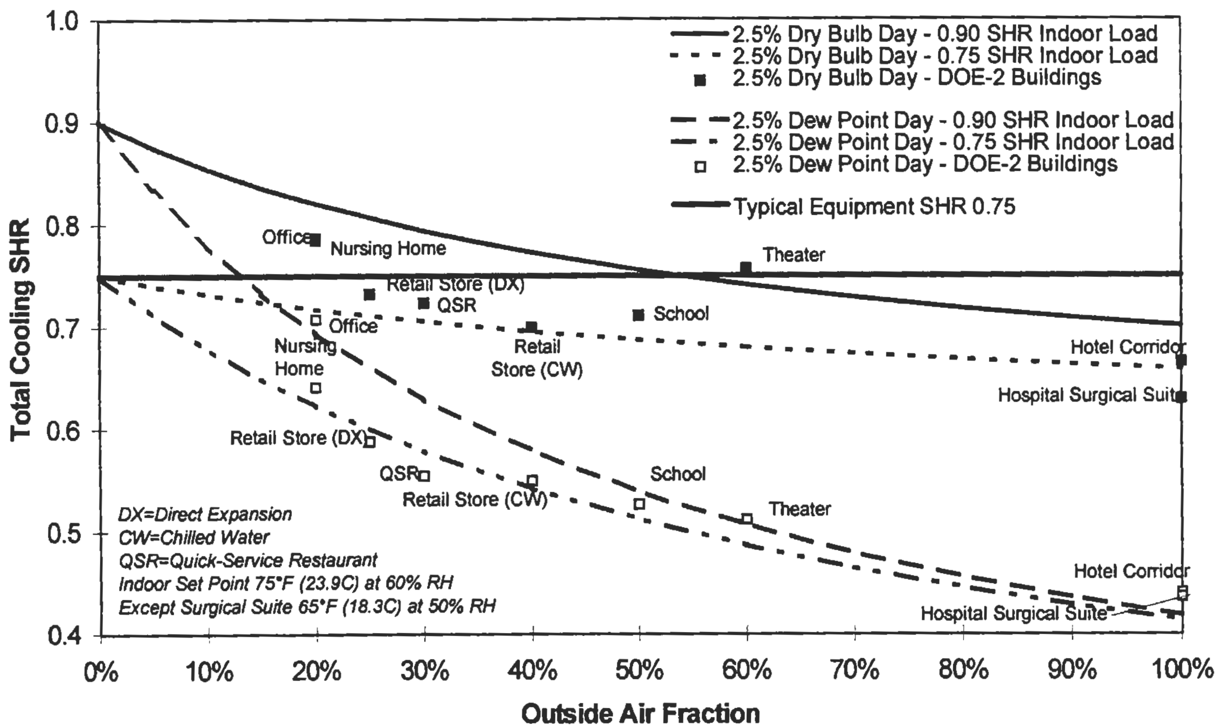

Figure 1 Total Cooling SHR at Design Conditions for Atlanta. *As the outside air fraction increases, the design cooling load SHR decreases at a modest slope for design dry bulb conditions; and at a steep slope for design dew point. At the design dewpoint condition, even outside air fractions as low as 15% will push the SHR below the 0.75 typically provided by conventional equipment.*

(ASHRAE 1997). This relationship between %OA and the resulting SHR of the cooling load (for the entire building or specific air handling units in that building) forms the premise for preconditioning OA. Conventional cooling equipment typically matches up well with cooling loads with SHR's of 0.75 or higher. At 0% OA, commercial buildings have cooling loads with SHR's typically approaching 0.90 but possibly as low as 0.75 (with high occupancy activities per square foot). Figure 1 shows that as %OA increases from 0 to 100%, the design cooling load SHR decreases: at a modest slope for design dry bulb conditions; and at a steep slope for design dew point. As Figure 1 illustrates for Atlanta, it is imperative that designers consider the design dew point when evaluating cooling equipment sizing and selection, otherwise traditional design dry bulb analysis will not reveal the significant mismatches between conventional equipment performance and cooling load SHR's.

These mismatches occur whenever the generic design load curves fall below the 0.75 SHR line. Under these circumstances one can anticipate excess moisture loads that cannot be wholly removed by conventional equipment and which will allow relative humidity levels to climb above 60% in the building space.

The emerging desiccant applications were modeled in the DOE 2.1E analysis program at the design dry bulb and dew point conditions. The individual building points plotted in Figure 1 identify modeled building cooling load SHR at the ASHRAE standard 62-1989 prescribed %OA for that building. Applications with the highest outside air fractions, such as hospital operating rooms, hotel corridor make-up air systems, theaters, and schools, will be the most difficult to satisfy with conventional HVAC. As a result, these will be the first to see the use of alternative ventilation air treatment technologies. This is already being borne out in the marketplace. For example, manufacturers are successfully specifying alternative systems, using desiccant dehumidifiers or enthalpy exchangers, into a number of schools and hospitals throughout the southern United States.

A series of monitoring activities has been underway to document the application economics of desiccant systems in these promising commercial settings. The series began with hotel corridor makeup air systems for guest room fresh air handling. Monitoring of large retail store outside air handlers was the second activity to be completed in the series. Now the series is being continued with monitoring activities in various stages for: theaters, restaurants, hospitals, nursing homes and schools. Results to date from these monitoring activities in the U.S. will be presented at this conference (Yborra 1998).

In this paper, a competitive economic analysis of desiccant and other HVAC equipment, generated by the desiccant screening tool, will be presented for several representative buildings in selected U.S. cities. In addition, software driven by DOE-2.1E was recently introduced in the U.S. to screen desiccant system applications. This work is also being presented at this conference, (Czachorski and Worek 1998).

ALTERNATIVE VENTILATION AIR TREATMENT TECHNOLOGIES

A wide variety of equipment configurations are available for controlling humidity. Four of the more commonly used system configurations were modeled to compare the humidity control performance and energy costs for different types of ventilation air treatment (Witte et. al. 1997):

- **Baseline**, a standard constant volume system sized to meet the design dry bulb temperature requirement in the space with no special consideration for humidity control;
- **Enhanced Reheat**, a reheat system with increased cooling capacity to meet the additional cooling load required to overcool the supply air for humidity control. Reheat energy cost is tracked separately to model applications which use a "free" source of reheat, such as a heat pipe, hot gas reheat, or condenser heat (this represents an ideal lower limit on operating costs, neglecting additional operating costs due to increased fan power, etc.);
- **Enthalpy Wheel**, the enhanced reheat system (with oversized cooling capacity and "free" reheat option) with a 70% effective sensible and latent heat exchanger between the relief air and outdoor air; and
- **Desiccant Dehumidifier**, the baseline system with a desiccant wheel to remove latent heat from the outdoor air, producing hot and dry air which is then partially cooled using a sensible heat exchanger (to the relief air) with the remaining cooling provided by a DX or CW coil (Figure 2).

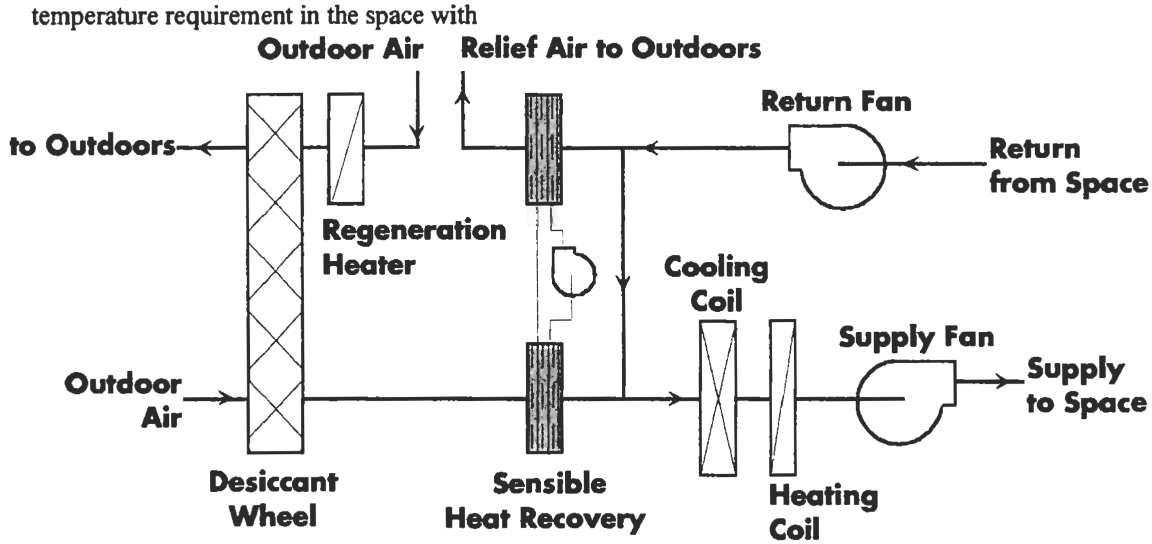

Figure 2 Desiccant Dehumidifier System Alternative. *The desiccant dehumidifier system is the baseline system with a desiccant wheel to treat the outside air stream. The desiccant wheel removes latent heat from the outdoor air, producing hot and dry air. The excess sensible heat is rejected using a sensible heat exchanger, shown here as a run-around heat recovery loop. In most applications, the sensible heat exchange is with the relief air. The desiccant wheel model uses manufacturer's data which is representative of state-of-the-art silica gel wheel performance. The sensible heat exchanger performance is based on manufacturer's data with the typical effectiveness averaging 70%.*

These four cooling systems were compared for nine applications using the DOE 2.1E-based screening tool to determine humidity control performance and the resulting energy costs of achieving improved humidity control. Table 1 summarizes the basic characteristics and key assumptions for each application.

Table 1 Basic Characteristics and Key Assumptions

See Notes Below	Supermarket	Hosp. Surg. Suite	Large Hotel	Small Hotel	Movie Theater	School	Quick-Service Rest.	Retail Store	Nursing Home
Area (ft^2)									
Total Building	39,000	500,000	300,000	40,000	40,000	165,000	2,000	60,000	45,000
Dehumidified	26,000	22,000	210,000	36,000	40,000	120,000	2,000	60,000	27,000
HVAC Schedule						*Schl. Yr.*			
M-F	24 hr	5a-8p	24 hr	24 hr	12n-11p	5a-6p	4a-12m	4a-12m	24 hr
Saturday	24 hr	5a-8p	24 hr	24 hr	9a-1a	8a-1p	4a-12m	4a-12m	24 hr
Sun & Hol	24 hr	5a-8p	24 hr	24 hr	9a-1a	Unocc.	4a-12m	6a-10p	24 hr
Occupied Setpoints									
Heating (°F)	70	65	72	72	70	72	70	72	74
Cooling (°F)	75	65	75	75	75	75	75	75	75
Humidity (%RH)	55%	50%	60%	60%	60%	60%	60%	60%	60%
Desiccant Setpoint (%RH or gr/lb)	35%	50%	65 gr	65 gr	60%	65 gr	60%	60%	65 gr
Cooling Coil Type	DX	CW	CW	DX	DX	DX	DX	DX	DX
Outside Air									
(%)	15%	100%	100%	100%	40%	100%	20%	30%	100%
(CFM/person, /ft^2, or /room)	15/per		65/rm	65/rm	15/per	15/per	1.5/ft^2	0.3/ft^2	25/per
Economizer	Sens.	n/a	n/a	n/a	Sens.	n/a	Sens.	Sens.	n/a

<u>Assumptions</u>

Restaurant OA volume based on balancing kitchen exhaust air volume.

School operates on a full schedule during the school year (September - May) and a shortened schedule during the summer. Summer occupancy is 50% from 9 a.m. to 2 p.m. Monday - Friday.

Large Hotel, Small Hotel, School, and Nursing Home are configured with separate makeup air systems for treating outside air.

DX=direct expansion rooftop unit or PTAC
CW=chilled water system

Supermarket uses refrigeration condenser waste heat as much as possible for heating and reheat.

Figures 3-6 and Table 2 compare the energy costs and occupied hours over the desired RH setpoint for each application in eight southern U.S. cities:

- Atlanta, Georgia
- Charleston, South Carolina
- Dallas/Ft.Worth, Texas
- Houston, Texas
- Jackson, Missippi
- Miami, Florida
- New Orleans, Louisiana, and
- Tampa/St. Petersburg, Florida.

A representative pair of electric and gas rates was used in each city. While the size range of these applications would cause different applications to be on different rates, the purpose of this analysis was to show the trends among the applications more than to give specific examples of savings. The rates used were actual utility rates for medium-sized buildings with minor adjustments to prevent excess charges due to contract minimums. (See Appendix for details.)

The desiccant systems generally provide equal or better humidity control with lower utility costs for most of the applications. Given the diversity of OA requirements, utility rates, and different process requirements, it is difficult to generalize the results. However, certain trends are apparent:

- Higher OA requirements give greater savings.
- Savings are greater in DX applications than in chilled water applications.
- As shown by the movie theater with modified setpoints, additional savings may be obtained by using a higher drybulb setpoint combined with a lower relative humidity setpoint, providing equal or superior comfort.
- Applications such as supermarkets and surgical suites offer advantages of product presentation, productivity, and sanitation over and above energy cost savings.

Table 2 Desiccant vs. Reheat Energy Cost Savings.

Average energy cost savings are stated as percentage of whole building energy costs, averaged across the eight cities.

Application	Average Energy Cost Savings
Supermarket (DX)	7%
Hospital (Surgical Suite CW)	3%
Large Hotel (CW)	8%
Small Hotel (DX)	30%
Movie Theater - Base (DX)	29%
Movie Theater - Modified (DX)	30%
School (DX)	24%
Quick-Service Restaurant (DX)	7%
Retail Store (DX)	16%
Nursing Home (DX)	12%

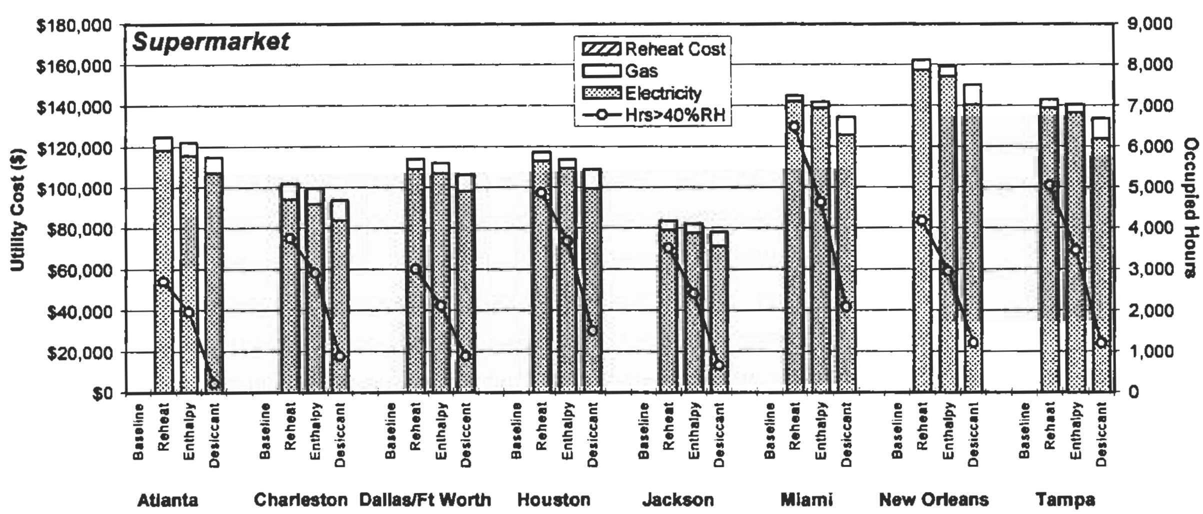

Figure 3. Energy Cost and Humidity Control Comparisons for Supermarket.

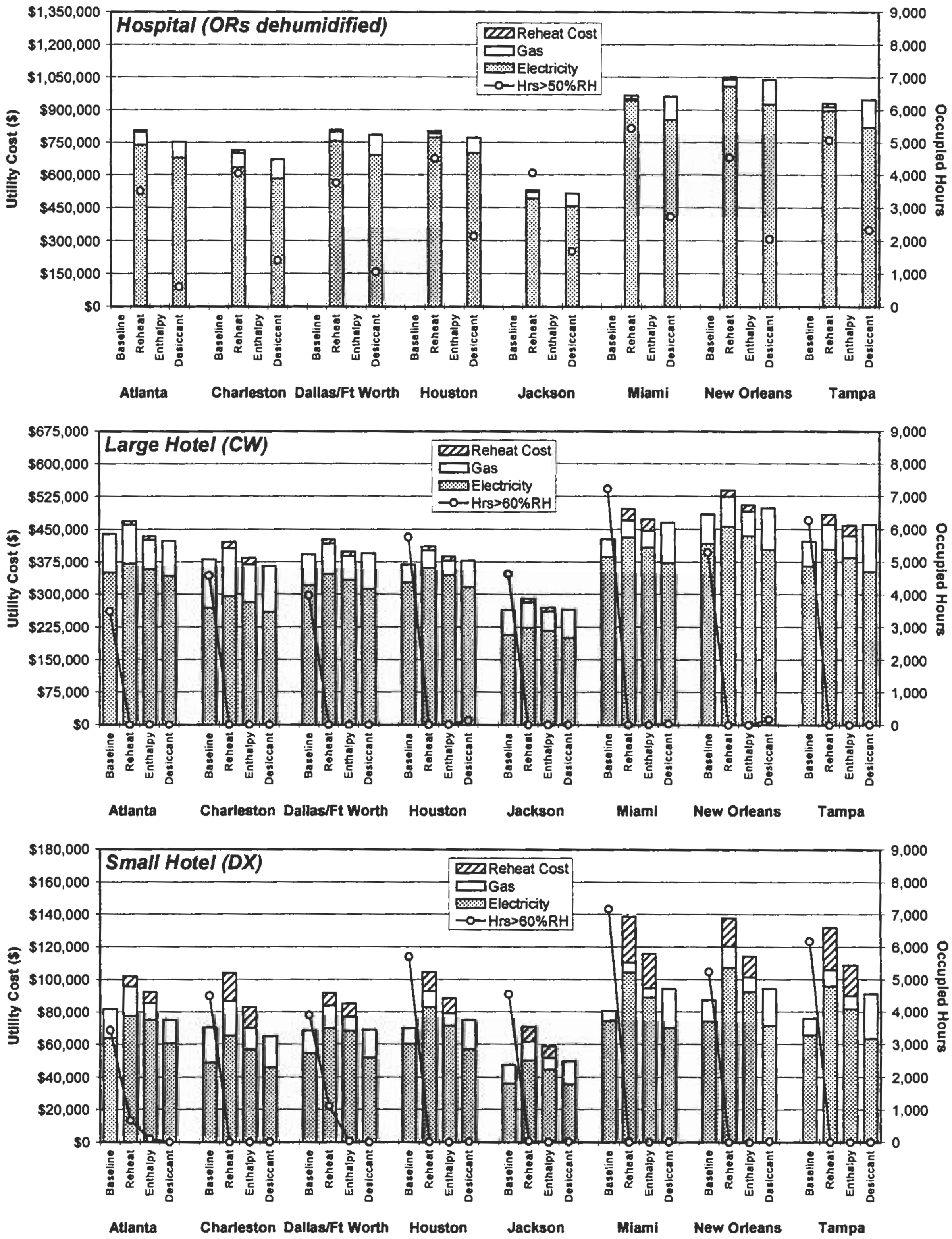

Figure 4. Energy Cost and Humidity Control Comparisons for Hospital Surgical Suite, Large Hotel (CW) and Small Hotel (DX).

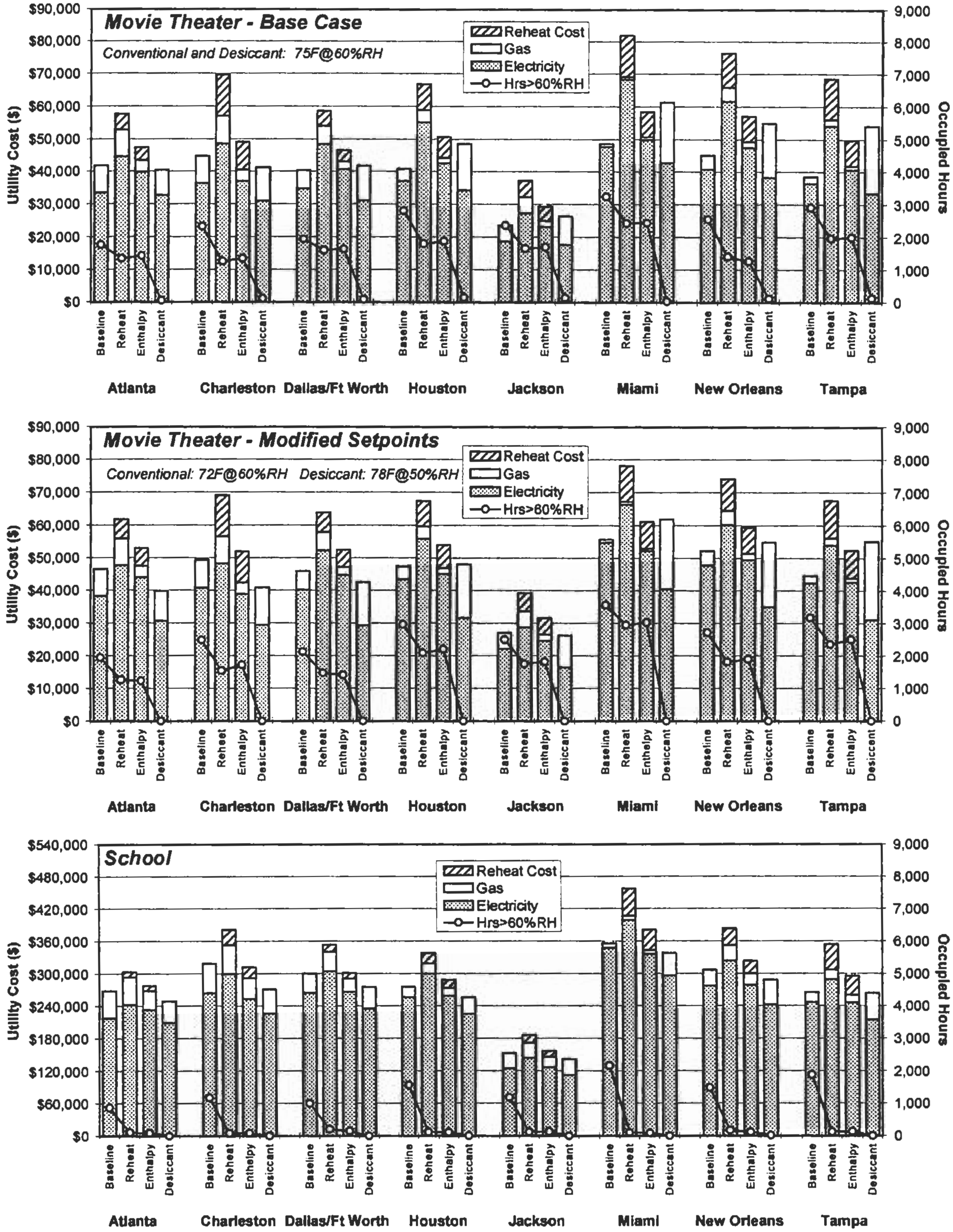

Figure 5. Energy Cost and Humidity Control Comparisons for Movie Theater (base case and with modified setpoint) and School.

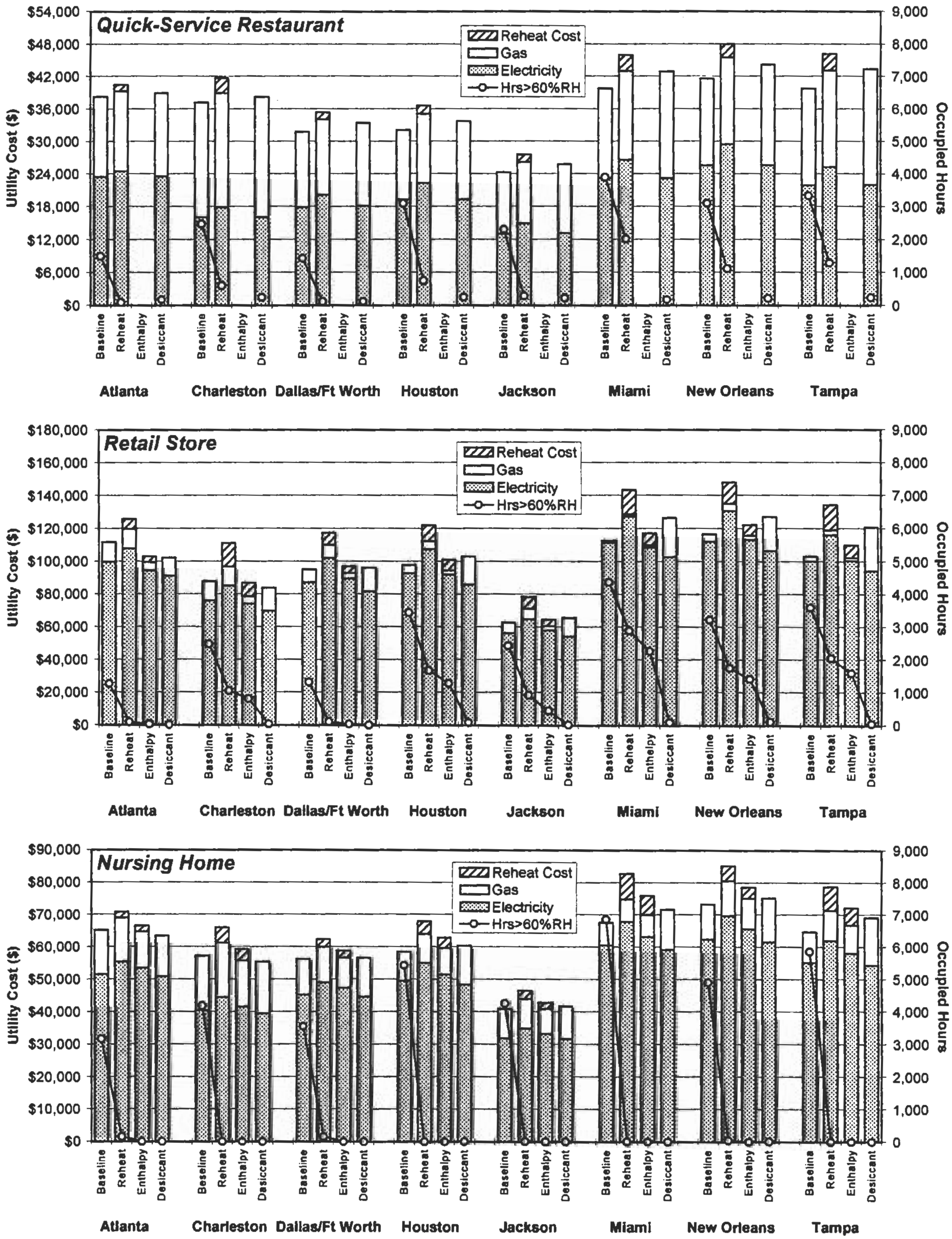

Figure 6. Energy Cost and Humidity Control Comparisons Quick-Service Restaurant, Retail Store, and Nursing Home.

CONCLUSIONS

With ASHRAE Standard 62-1989 ventilation levels becoming applicable in more and more state and local building codes, HVAC equipment manufacturers, specifiers and end users are faced with the challenge of providing ventilation air treatment while maintaining control of indoor relative humidity. This is especially critical in hot and humid climate zones. Desiccant dehumidification is emerging as a cost-effective solution which can provide superior humidity control in many applications. While the advantages of desiccants have been recognized in refrigeration process applications such as supermarkets, ice arenas, and refrigerated warehouses, new applications with high outside air requirements are now taking hold.

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APPENDIX - UTILITY RATES

The following utility rates were used as representative of the energy prices in each city. The qualifications for the rates are indicated here, but they were neglected in the analysis. The same rates were applied to all buildings in a given city regardless of building size. This was done to allow comparison of the various building loads without complicating the results with rate changes within a city. In cities where a second gas rate is listed, this is a gas air conditioning or off-peak seasonal rate which was applied only to gas used by the desiccant dehumidifier.

Rate Name	Verified Current As of	Qualifications	Energy Charge* $/kWh or $/MMBtu Minimum	Energy Charge* $/kWh or $/MMBtu Maximum	Demand Charge* $/kW Minimum	Demand Charge* $/kW Maximum
Atlanta, GA: Georgia Power/Atlanta Gas Light						
PLM-2	Jan-96	30 - 500 kW	0.02	0.13	kWh/kW blocks**	
G-11	Jan-96	< 200 MMBtu Oct-Apr	4.83	5.35	-----	-----
GAC	Jan-96	< 200 MMBtu Oct-Apr	3.64	3.64	-----	-----
Charleston, SC: SCE&G/SCE&G						
9	Jun-97	None	0.08	0.09	2.60	2.60
31	Jun-97	< 5 MMBtu/day	12.57	13.23	-----	-----
GAC	Jun-97	< 5 MMBtu/day	8.69	8.69	-----	-----
Dallas/Ft. Worth, TX: Texas Utilities/Lone Star Gas						
GS	Jan-96	> 10 kW	0.03	0.09	9.14	9.14
951	Jan-96	None	4.91	5.30	-----	-----
Houston, TX: Houston Light & Pwr./ENTEX						
MGS	Jan-96	Single Meter	0.02	0.08	3.36	3.36
573	Jan-96	>150 MMBtu/mo.	5.65	8.04	-----	-----
Jackson, MS: MS Pwr. & Light/MS Valley Gas						
B-28	Jun-97	> 200 kW	0.03	0.04	2.51	2.63
305	Jun-97	None	4.66	5.06	-----	-----
321	Jun-97	None	3.90	3.90	-----	-----
Miami, FL: FL Pwr & Light/Peoples Gas						
GSD-1	Jan-96	20 - 500 kW	0.05	0.05	2.94	10.19
GS	Jan-96	< 2,500 MMBtu/yr.	6.57	6.57	-----	-----
LE	Jan-96	< 2,500 MMBtu/yr.	5.99	5.99	-----	-----
New Orleans, LA: NO Public Svc./NO Public Svc.						
SE-17	Jan-96	> 3 kW	0.06	0.11	4.86	5.99
SG-6, 41	Jan-96	None	5.53	6.10	-----	-----
Tampa/St. Pete, FL: Florida Power/Peoples Gas						
GSD-1	Jan-96	> 24,000 kWh/yr.	0.05	0.05	4.50	4.50
GS	Jan-96	< 2,500 MMBtu/yr.	6.84	6.84	-----	-----
GS	Jan-96	< 2,500 MMBtu/yr.	6.39	6.39	-----	-----

Notes:

* Includes any applicable energy cost adjustment, purchased gas adjustment, surcharges, and credits. Applicable taxes are included for all locations except Charleston and Jackson.

** Implicit demand charge in kWh/kW block structure; billing demand ratchet 95% of summer peak.