Preconditioning
Market Analysis

Phase 1

DESICCANT-BASED PRECONDITIONING MARKET ANALYSIS

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1. INTRODUCTION: TASK 1 MARKET STUDY

1.1 PURPOSE OF THIS OUTDOOR AIR PRECONDITIONING MARKET EVALUATION

The primary purpose of this phase 1 marketing evaluation is to define the markets and applications that would best benefit from a design approach that uses a specialized outdoor air-handling unit (SOAHU) to "*decouple*" (or separate) the latent load associated with the outdoor air (or outdoor and space loads) from the sensible (cooling) load. This approach would allow down-sized conventional cooling equipment to handle the indoor sensible load. It can result in better humidity control and reduced energy consumption without significantly increasing the project first cost.

Key objectives include establishing the following:

*   Which markets and applications will embrace an SOAHU approach? Which of the preconditioning system configurations identified in this study would most likely be used in each market?

*   Which SOAHU benefits are most important to the individual markets, and are these benefits quantifiable and important enough to justify a first-cost premium to the facility owner?

*   Which markets are best served by desiccant systems, both heat-regenerated and non-regenerated (total energy recovery) and why (Task 1)? Which desiccant system configurations need to be offered to effectively serve these markets (i.e., component arrangement, sizes, performance, construction, size, and price) (Task 2)?

*   What is the estimated annual sales volume potential that could be targeted with a line of desiccant-based preconditioning products, both regenerated and non-regenerated?

*   What are the current sales estimates of various SOAHU designs?

*   Which market drivers (ASHRAE 62-89, new weather data, re-evaporation from direct expansion (DX) cooling coils, moisture storage within structures, demand reduction, dry ductwork, improved humidity control, etc.) are most important to the justification of the SOAHU approach? What training tools are most required to convey this to the field?

*   Which desiccant-based cooling (DBC) system configuration shows the most promise to current heating, ventilation, and air-conditioning (HVAC) equipment manufacturers? The involvement of HVAC manufacturers may be crucial in construction of a laboratory prototype that may incorporate conventionally available hardware components. [Such a laboratory prototype is to be built as part of this total program (Phase 2)].

1.2 PRELIMINARY POSITIONS AND ASSUMPTIONS CONCERNING THIS STUDY

Only outdoor air preconditioning systems will be considered in this evaluation. The mass market required to justify a product introduction by one of the major HVAC equipment manufacturers was considered to be an important consideration in assessing the market potential of desiccant systems to handle outdoor air latent loads.

All available methods of handling this latent load must be evaluated against the merits of the regenerated desiccant system approach (DBC), because such an analysis will surely be done by the marketplace on a job-by-job-basis. Only the markets that clearly benefit from the DBC approach and are justifiable without utility rebates will be considered viable markets in this study.

Based on previous modeling experience, the desiccant cooling system performance used for this analysis is optimized for $/ton and not for coefficient of performance (COP) or efficiency. This is because the marketplace will demand a "cost-competitive" and compact product, and most energy savings will come from demand savings and not reduced kWh consumption.

The modeling results for an active desiccant system show the annual Btu consumption can actually be greater for the active desiccant system than it is for the conventional approach. Even in areas where the cost of electricity is high and the cost of gas is low, the energy cost savings are relatively small, until the demand charge is factored into the analysis (most utility structures utilize a relatively low kWh-consumption charge and a sizable demand-charge element). An active desiccant system optimized for $/ton of latent cooling is significantly less expensive than a similar system optimized for energy efficiency (the higher efficiency comes at the expense of lower wheel face velocities and thereby a much larger system size processing for the same airflow). However, because the two systems may provide a similar reduction in outdoor air latent load at design conditions, the peak electrical demand savings will be similar. Once again, because the modeling confirmed that most of the energy-cost savings are associated with the peak-demand savings and not kWh reduction, the active desiccant system optimized for $/ton of latent cooling will clearly be the choice of the marketplace.

The weather data for most Phase I analyses used ASHRAE dry-bulb frequency groupings, or BINS, with corresponding wet bulbs because it was available when this work was performed. Phase 2 analyses will be based on ASHRAE outdoor air wet bulb BINs with a mean coincident dry-bulb temperature, because that more accurately reflects the latent load in outdoor air. Phase 1 will compare these two methods to quantify the resulting difference.

The cost comparisons for all of the system approaches are based on typical central-station air-handler selling prices so that a fair assessment can be made. These are mass-produced, pre-engineered, air-handling modules such as the current Trane Climate Changer® offering.

Reduced chiller capacity credits (for example) used are from component pricing provided by data obtained from literature available from the major HVAC equipment manufacturers.

Many markets will not require optimum-space humidity control and may be effectively served by straight non-regenerated desiccant energy recovery (no supplemental cooling and reheat) feeding conventional cooling systems. However, all performance comparisons in this study will assume that dry air at a "space-neutral" temperature (an air temperature range of 65–80°) is desired, (thereby providing a more favorable comparison to the DBC approach). The preconditioning systems are designed to provide dry air at a space-neutral temperature (i.e., close to that of the space) so that this air can be provided directly to the occupied space without overcooling. There are many cases where a conventional ventilation air-preconditioning approach needs to cool the air to remove the humidity, but sensible cooling is not required (i.e., 68° and raining outdoors). As a result, outdoor air-preconditioning systems sold today are typically controlled to provide space-neutral air.

The air that leaves an active desiccant system is hot, so it needs to be cooled before it is introduced to a space. If hot air is provided, it will overheat the room, or, if it is provided to another more conventional air handling system, that system will need to expend energy to post-cool this pretreated (dehumidified) outdoor air stream.

To satisfy the "space neutral" requirements specified above, the modeling assumes that the conventional cooling approaches would reheat to 65° and the active desiccant systems would be post-cooled to 80°. This energy is factored into the analyses completed as part of this investigation. These considerations are shown in the range of air outlet conditions for the seven SOAHU options illustrated and discussed in the exhibit on the following pages.

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The projected size of the overall market and percentage allocation to SOAHU equipment and the various approaches come from market information provided by surveys of the major HVAC manufacturers and input from the marketing questionnaires provided to select consulting engineers, owners, sales offices, and SEMCO in-house and field sales staff.

All analyses assume that the DBC systems will be reliable and will perform as modeled, to eliminate the barrier of considering an unknown technology.

2. ANALYSIS

As stated previously, the primary purpose of this marketing evaluation is to define the markets and applications that would best benefit from a design approach that uses an SOAHU to decouple the latent load associated with the outdoor air (or outdoor and space loads). This could allow down-sized conventional cooling equipment to handle the indoor sensible load. A further objective is to determine which of the available SOAHU approaches would most often be used in each market segment. Finally, based on the needs of the market segments showing the most promise for regenerated DBC systems, this analysis attempts to determine what the product offering needs to be and how much business potential ($/year) exists for the technology.

2.1 PERFORMANCE ANALYSES

This investigation studied seven SOAHU approaches for decoupling outdoor air loads. These approaches are described briefly, including simple schematics, in the exhibit "Preconditioning Approaches Analyzed by this Study." They include conventional cooling with reheat, conventional cooling with sensible recovery to provide free reheat, two total recovery approaches using non-regenerated desiccant wheels, and three approaches using regenerated desiccant wheels. Table 1 summarizes the nine markets investigated for each approach. It also lists the six cities chosen for analysis and the reasons they were selected. Table 2 lists the energy costs used for each city selected; these were obtained from a report sponsored by the Gas Research Institute that lists 1995 electric and gas rates.

Models were prepared for each SOAHU approach to allow for a comparison at different operating conditions, in different climates, and for different energy costs. Appendix A gives sample energy BIN analysis sheets for each SOAHU approach; for 2500-cfm, 7500-cfm, and 20,000-cfm systems; and for each of these city locations. Appendix B presents more detailed illustrations of each of the modeling applications for the Atlanta area. Also included in Appendix B are modeling data for the DBC approach for locations in Houston, Minneapolis, New York, and Orlando; these data highlight the impact that weather and energy cost have on performance of this option.

Results of this modeling for differing levels of pre-conditioning dehumidification are summarized as Cases 1 through 4 in Appendix C. Case 1 assumes preconditioning outdoor air to 50 grains, Case 2 to 65 grains, and Case 3 to 45 grains. Each of these cases assumes a consistent cost of energy. Case 4 was prepared based on delivering 50-grain air but uses the actual energy cost for each individual city.

In the second phase of this study, the new dew point weather data will be used to complete the final energy analyses for the most promising markets. For this study, dry bulb BINs with mean coincident wet bulb data were generally used. This provides a conservative estimate because it understates both the humidity content and enthalpy of the outdoor air volumes. In Appendix B, there are two examples of analyses for the conventional cooling/reheat, dual total energy wheel, and DBC approaches that use the new dew point weather data for Atlanta (see Appendix B). The impacts of using these design dew point data are summarized in Table 3.

To keep a consistent comparison in Table 3, all regeneration and reheat that is not available from a source of waste heat is provided by a boiler burning gas at an efficiency level of 78%. The second conventional system showing a run-around heat exchanger system is the most efficient way of using conventional cooling without desiccants. An example utilizing DX dehumidification and condenser reheat was not considered because the performance of the cooling system would have to be compromised to provide the amount of heat necessary for an all-outdoor-air application. This would make comparisons of energy efficiency impossible, because it could not provide the same supply conditions chosen for the analysis. More important, the run-around example provides a more energy-efficient option, and therefore a good complement to the conventional approach, using vapor-compression DX for dehumidification with hot-water reheat (still a very common approach).

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Exhibit: Preconditioning Approaches Analyzed by this Study

Options 1 and 2: Cooling and Reheat

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This approach offers the lowest first cost in most cases and requires the highest energy consumption (Fig. 1). It is also the technology most familiar to the major HVAC manufacturers, and thereby the easiest to integrate into their manufacturing and marketing organizations. It is compact and does not require an exhaust air stream to operate. Adding a run-around coil or heat pipe improves performance. This option is limited in its ability to deliver low dew points, does not handle fluctuations in outdoor air moisture loading very well (when using DX), and offers no recovery in the heating or cooling modes.

Option 3: Desiccant-Based Total Recovery with Conventional Cooling and Reheat

This approach offers a more energy-efficient version of Option 1 (Fig. 2). It also improves the control of the DX cooling approach because it stabilizes the conditions introduced to the cooling coil. It provides for efficient heating mode energy recovery, which dramatically expedites the payback of the increased cost over Option 1. This approach significantly reduces the condensate on the cooling coil when compared to Option This option is also limited in its ability to deliver low dew points and requires an exhaust air stream to operate. Because it combines a supply and exhaust system, it is larger than Option 1.

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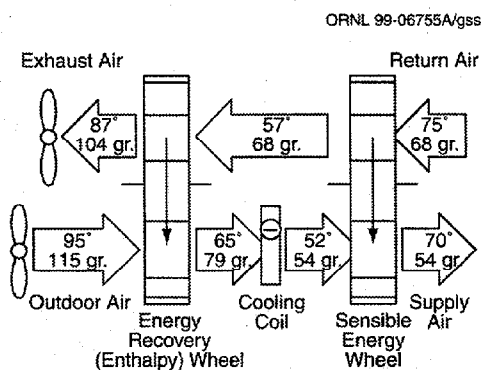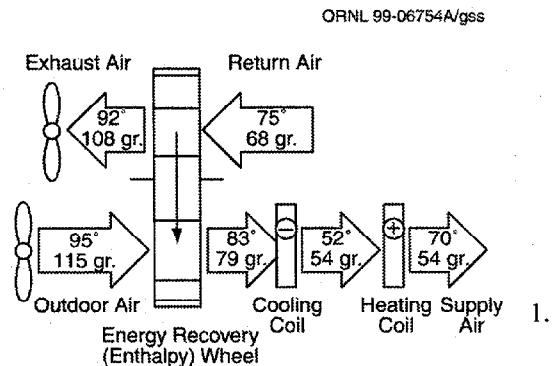
Option 4: Desiccant-based Total Recovery and Free Reheat with Dual-wheel Approach

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This approach offers a very energy-efficient version of Option 2. It further improves the control of the DX cooling approach, making it an ideal application for DX despite the 100% outdoor air. It provides for free reheat via the sensible recovery wheel. It offers heating mode energy recovery. This approach minimizes the mechanical cooling tons required to dehumidify the outdoor air. This option is also limited in its ability to deliver low dew points and requires an exhaust air stream to operate. Because it combines a supply and exhaust system and a sensible wheel, it is larger than Option 2.

Option 5: Heat-Regenerated Desiccant-Based Cooling Approach

This approach uses gas, steam, or waste heat to regenerate a desiccant dehumidification wheel, which eliminates the need for conventional cooling to remove moisture from outdoor air streams. A sensible energy wheel removes most of the heat added by the regeneration process and uses this heat to minimize the amount of regeneration energy required. This option can provide, in some climates, drier air than possible with conventional cooling approaches. Other advantages, such as dry cooling coils, airborne pollutant removal, improved economizer operation due to indirect evaporative cooling and reduced energy consumption over Option 1 are common. An exhaust air stream is beneficial but not required for this approach, which is a key advantage over Options 2 and 3.
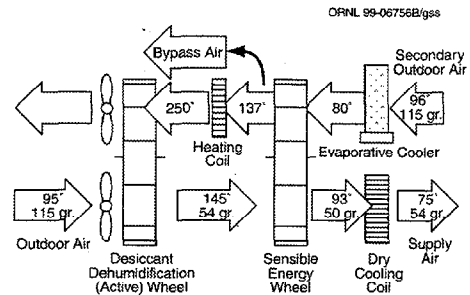
This option requires a more complex control system for optimization than do the previous approaches. Because the face velocities through the desiccant wheel need to be moderate to recognize the desired performance, the units are very large compared to Option 1. Because raw outdoor air is introduced to the dehumidification wheel, the heat generated is significant, making the second wheel essential.
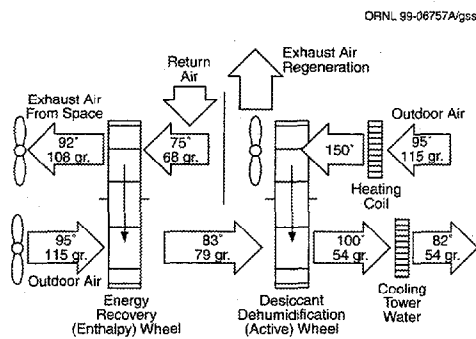
Option 6: Heat-Regenerated Desiccant-Based Cooling and Desiccant-Based Recovery Wheel Hybrid Approach 1

This approach uses gas, steam, or waste heat to regenerate a desiccant dehumidification wheel, which eliminates the need for conventional cooling to remove moisture from outdoor air streams. Because much drier air is introduced to the desiccant wheel than in the case of Option 4, much less heat is added to the supply air stream, making conventional post-cooling (e.g., from cooling tower water) feasible. Much lower dew points are possible, and this system is more energy efficient than Option 4. Advantages such as dry cooling coils, airborne pollutant removal, improved economizer operation as a result of indirect evaporative cooling, and reduced energy consumption over Option 1 are common. Full benefits of heating-mode total energy recovery are recognized. The secondary evaporative cooler, which is very important for Option 4, is eliminated with this approach. From the standpoint of maintenance and IAQ concerns, this is an advantage. An exhaust air stream is required for this approach. The same challenges listed for Option 4 above apply.

Option 7: Heat-Regenerated Desiccant-Based Cooling and Desiccant-Based Recovery Wheel Hybrid Approach 2

This approach combines the best of Options 4 and 5 by integrating an additional sensible wheel after the dehumidification wheel to reduce the sensible load added by adsorption and, at the same time, reduce the regeneration energy requirement. This system provides a very high COP and low operating cost. This approach will be evaluated primarily for this reason.
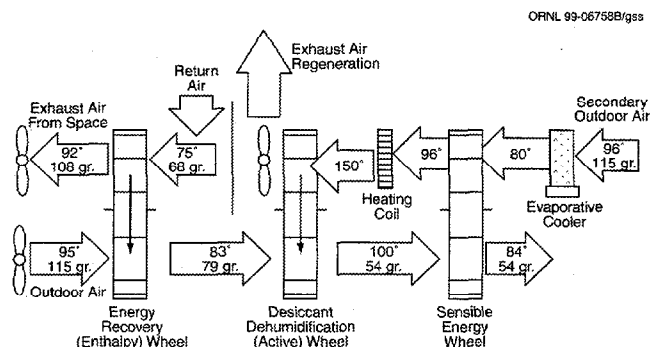
Once again, advantages such as dry cooling coils, airborne pollutant removal, improved economizer operation due to indirect evaporative cooling and reduced energy consumption over Option 1 are common. Full benefits of heating mode total energy recovery are recognized. An exhaust air stream is required for this approach. The same challenges listed for Option 4 above apply. Size and first cost are the big obstacles with this approach; however, it is very appropriate for designs where energy efficiency is most important and for combining with engine-driven chillers, for example, where waste heat is available and dry coils are desired.

Table 1. Markets and cities investigated for outdoor air preconditioning

Applications–markets	Cities investigated for energy analysis
1. Hospital operating rooms	1. Hot/humid, high-electric, moderate-gas costs (b)
2. Nursing homes and hospital areas other than operating rooms	2. Hot/humid, moderate-electric, moderate-gas costs (c, d)
3. Research laboratories	3. Moderate humidity, high-electric, moderate-gas costs (b, e)
4. Quick service restaurants	4. Moderate humidity, moderate-electric, moderate-gas costs (a)
5. Retail stores	5. Cities representative of different parts of the country:
6. Hotels, high-rise apartments, dormitories	(a) Minneapolis, MN
7. School and university classrooms	(b) Orlando, FL
8. Movie theaters and auditoriums	(b) New York, NY
9. Office buildings	(c) Atlanta, GA
	(d) Houston, TX
	(e) Los Angeles, CA

Table 2. Utility costs (1994) of electricity and gas for the regions analyzed in this study
(electric rates based on a commercial customer using more than 500 kW demand)

City	Electrical cost $/kWh	Monthly demand charge ($/kW)		Gas cost $/million Btu
		Summer	Winter	
Atlanta	0.060	8.00	8.00	4.80/3.50[a]
Houston	0.035	8.00	8.00	3.40
Los Angeles	0.046	7.20	0	5.33
Minneapolis	0.034	8.47	6.20	4.10
New York	0.050	23.50	18.98	6.26
Orlando	0.035	8.10	8.10	4.20

[a] Represents a gas-cooling rate.

In the Conclusions (Sect. 4), energy savings for each approach are combined with the projected market potential to provide an estimate of the overall energy reduction for the United States if SOAHU systems were to be applied to only a portion of new construction and renovation projects annually.

Preconditioning system approach	Annual estimate based on current dry bulb BIN method ($/year)	Annual energy cost estimate based on revised ASHRAE data (new wet bulb BIN method) ($/year)	Percent change
Conventional cooling w/reheat	58,180	61,597	6
Conventional cooling w/run-around recovery	44,020	46,205	5
Total energy wheel w/cooling and reheat	43,046	43,390	0.8
Dual-wheel total energy recovery	26,928	27,060	0.5
Dual-wheel, desiccant-based (DBC) w/post-cooling [a]	35,463	32,625	-8
Desiccant dehumidification— Total recovery hybrid [a]	34,113	32,066	-6
Dual-wheel DBC— Total recovery hybrid [a]	25,264	24,001	-5

Table 3. Impact of new ASHRAE weather data on a 20,000-cfm system in Atlanta, GA (wet bulb BINS, corresponding to dry bulb)

[a] Reflects the use of a favorable gas cooling rate of $0.35/therm.
Wet bulb BIN data is from ASHRAE.
Units are assumed to operate continuously.
Electricity is $0.06/kWh and $8.00/kW demand.
Gas at $0.48/therm for all but DBC summertime use. A 78% boiler efficiency is assumed.
Location is Atlanta.

2.2 COST ANALYSES

To compare the various SOAHU approaches, the relative cost differences must be known. Given that the conventional cooling equipment is already mass-produced, while DBC systems are still custom built, the current cost to the market for these technologies does not allow for a meaningful comparison. To do so, Table 4 was created using sales prices for commercially produced, central-station air-handling units provided by one of the major HVAC manufacturers to project what the selling price to the market would be if all products were mass-produced. The cost of the chiller and cooling tower required for each preconditioning approach is shown enclosed in parentheses, and the total system cost (air handler + chiller + cooling tower) is shown in bold. Table 4 shows the cost for each approach at three airflow capacities and provides the approximate physical dimensions of the air handling unit in inches for each system.

Table 5 combines the energy analyses completed with the cost information provided by Table 4 to show the simple payback period if systems with various sizes were applied in Atlanta.

Table 4. Selling price and physical dimensions of various preconditioning approaches at different airflow capacities

Preconditioning system approach	2,500 cfm		7,500 cfm		20,000 cfm	
	Sales price[a] in $[b]	Dimensions[c] H×W×L (in.)	Sales price[a] in $[b]	Dimensions[c] H×W×L (in.)	Sales price[a] in $[b]	Dimensions[c] H×W×L (in.)
Conventional cooling w/reheat (comparable total)	2,200 (9,500) **11,700**	27.5×44×154	7,100 (28,500) **35,600**	44×74×177	16,700 (76,000) **92,700**	74×120×251
Conventional cooling w/run-around recovery	4,000 (7,750) **11,750**	55×44×141	12,150 (23,250) **35,400**	88×74×193	31,000 (62,000) **93,000**	148×120×260
Total energy wheel w/cooling & reheat	7,000 (5,500)	55×44×171	16,750 (17,000)	88×74×194	37,300 (45,000)	148×120×268
Dual-wheel total energy recovery	11,100 (3,500) **14,600**	55×44×188	24,750 (10,500) **35,250**	88×74×211	53,650 (28,000) **81,650**	148×120×285
Dual-wheel, desiccant-based (DBC) w/post cooling	14,700 (2,000) **16,700**	55×44×188	30,200 (6,000) **36,200**	88×74×211	64,800 (16,000) **80,800**	148×120×285
Desiccant dehumidification—total recovery hybrid	15,750 (3,000) **18,750**	55×44×203	31,350 (8,500) **39,850**	88×74×226	67,850 (22,500) **90,350**	148×120×300
Dual-wheel DBC—total recovery hybrid	20,500 (1,000) **21,500**	55×44×235	39,300 (3,250) **42,550**	88×74×258	84,200 (8,500) **92,700**	148×120×332

[a] All sales prices are estimates of the market price using Climate Changer Modules.
[b] Parentheses denote cost of chiller and cooling tower capacity required at $500/ton. Bold denotes total system cost.
[c] All dimensions assume the use of Trane Climate Changer Modules.

Table 5. Simple payback of various preconditioning approaches at different airflow capacities

Preconditioning system approach	2,500 cfm (simple payback in months)[*]	7,500 cfm (simple payback in months)[*]	20,000 cfm (simple payback in months)[*]
Conventional cooling w/reheat	N/A	N/A	N/A
Conventional cooling w/run-around recovery	0.3 months ($2,009/yr in energy savings)	($200 first cost savings) Immediate ($6,028/yr in energy savings)	0.2 months ($16,074/yr in energy savings)
Total energy wheel w/cooling and reheat	5 months ($1,892/yr in energy savings)	($1,850 first cost savings) Immediate ($5,676/yr in energy savings)	($10,400 first cost savings) Immediate ($15,136/yr in energy savings)
Dual-wheel total energy recovery	9 months ($3,907/yr in energy savings)	($350 first cost savings) Immediate ($11,720/yr in energy savings)	($11,050 first cost savings) Immediate ($31,253/yr in energy savings)
Dual-wheel, desiccant-based (DBC) w/post cooling	21 months ($2,840/yr in energy savings)	0.8 months ($8,520/yr in energy savings)	($11,900 first cost savings) Immediate ($22,720/yr in energy savings)
Desiccant dehumidification—total recovery hybrid	28 months ($3,008/yr in energy savings)	5.6 months ($9,024/yr in energy savings)	($2,350 first cost savings) Immediate ($24,064/yr in energy savings)
Dual-wheel DBC—total recovery hybrid	29 months ($4,115/yr in energy savings)	7 months ($12,345/yr in energy savings)	($0 first cost savings) Immediate ($32,920/yr in energy savings)

Notes:
[*] The simple payback compares the various preconditioning approaches with the conventional cooling with reheat approach
[*] All dimensions assume the use of Trane Climate Changer Modules and SEMCO wheel modules
[*] All sales prices are estimates of the market price using Climate Changer Modules
[*] The Atlanta weather data and 1994 local energy costs are used for this comparison
[*] The energy savings are based on continuous operation
[*] The conventional cooling with run around uses the plate heat exchanger currently offered by Trane.

3. RESULTS

This investigation has provided a substantial amount of valuable information. The combination of the numerous energy analyses completed for the various SOAHU approaches, the equipment cost projections, the market size analyses, and the market survey conducted by SEMCO provide a fairly clear picture of where the opportunities for regenerated DBC systems exist.

3.1 MODELING RESULTS

The results of the modeling are summarized in Tables 3–5 (Sect. 2), by the Summary Sheets for Preconditioning Approaches in Appendix A, and by Cases 1–4 in Appendix C. The more important findings with respect to the market for regenerated desiccant systems include the following:

1. In general, if dry, space-neutral temperature air is desired (i.e., the basis of this investigation) and if an exhaust air path is available, the dual-wheel total-energy recovery (non-regenerated desiccant wheel) approach is by far the most desirable because it provides the highest energy savings. It is very cost-effective in that it also provides wintertime humidification.

2. If first cost is most important, and if maintaining humidity below 70% and above 20% is an acceptable design parameter, then desiccant preconditioning by single total-energy wheel (non-regenerated) is by far the most economical solution, provided there is access to an exhaust air path.

3. Where an exhaust air path is not accessible, the DBC approach may provide an attractive solution, depending on the hours of operation, demand charge, availability of waste energy or gas cost, and the importance of humidity control.

4. If a DBC system is to be applied to process humid outdoor air (120–140 grains) and supply it at 50–60 grains, the latent load is so significant that an oversized DBC system is required (very low face velocities across the wheels), making it a high-cost alternative. In such applications, combining total energy recovery with desiccant dehumidification makes sense.

5. If the DBC approach is used without access to a return air path, the energy savings projected by the sample modeling are significantly reduced because (1) the winter heat recovery is lost, and (2) more energy is required for regeneration because the inlet air contains a higher humidity content, and (3) the temperature entering the sensible wheel on the secondary side is much warmer, resulting in hotter air leaving the supply side of the system. Without access to the exhaust air path, the savings listed in this investigation for the DBC approach are reduced by approximately 50%.

6. If weather data that more accurately reflect the humidity content of outdoor air (wet bulb BINs with corresponding mean coincident dry bulb temperatures) are used to complete energy analyses, compared with the dry bulb BINs used for most of this investigation, the energy consumption estimates for the conventional approach increase by approximately 6%, the DBC energy consumption estimates decrease by approximately 10%, and the total energy recovery estimates remain approximately the same (see Table 3).

7. The demand charges are a very significant portion of the cooling season energy savings for the DBC approach. The winter humidification savings are significant for the total energy recovery approach, especially in the cool, less-humid climates.

3.2 PAYBACK AND COST ANALYSES

Table 4 (Sect. 2) summarizes the projected selling price for the various SOAHUs assuming they were built using mass-produced, pre-engineered air-handling units for commercial buildings.

Table 5 (Sect. 2) uses these costs and the energy cost analyses to project a simple payback based on chiller and cooling tower costs of $500/ton. Some of the more important results from these analyses are these:

1. The small systems (2500-cfm) had payback periods ranging from 0.5 to 2.5 years, while the large systems had an immediate payback. This highlights one of the most important aspects of comparing air-side approaches, which is that the cost of the chillers required to solve ventilation air pretreatment problems is very high compared with the air-side solutions.

2. Size of the SOAHU is a very important factor in the design process. All SOAHU systems were similar in size, except for the conventional cooling and reheat systems, which were significantly smaller (proving a significant advantage).

The $500/ton figure was selected as a very "conservative" number. This number includes the chiller, the cooling tower, piping, insulation, and incremental installation costs. Clearly, increasing it slightly would decrease the payback period resulting from the use of active desiccant systems. Decreasing it would extend the payback periods of the active desiccant systems. The $500/ton value was chosen in collaboration with Trane as a figure that would result in a fair analysis.

3.3 SIZE OF POTENTIAL MARKET FOR SOAHUs

Table 6 was prepared from information obtained from F. W. Dodge marketing forecasts (http://www.mag.fwdodge.com) for the year 1994. This table combines the information provided by ASHRAE standard 62-1989 with annual construction and renovation starts for the markets studied. The average size of each facility was taken from the DOE 1992 survey of Building Characteristics. Table 6 shows the potential market size for SOAHU systems by market and in total. The Final Market Analysis sheets provide these numbers based on evaluations made for each SOAHU approach for a given market. The percentage of the overall market potential listed for a given SOAHU approach is a subjective determination based on feedback from users and specifiers.

Appendix D provides a sample of the survey sheet used to weight the various market drivers. (The responses to the questions asked by the survey are summarized in Sect. 4 and tabulated in Tables 9 and 10 at the end of this section.) Factors such as importance of humidity control, first cost, energy efficiency, size of equipment, and required dew points from the SOAHU system helped in completing the subjective evaluation.

Appendix E includes nine "Final Market Segment Analysis" sheets. These sheets are used in the next section to summarize the opportunity for regenerated DBC approaches, as well as other approaches investigated for the various market segments. These sheets provide a brief summary of the benefits offered to the market by preconditioning, as well as a brief conclusion regarding the desiccant-based approaches investigated. Table 7 summarizes the information provided by the individual Final Market Segment Analysis sheets, showing the projected sales potential for each approach presented for each market.

The more important results from these analyses include the following:

Table 6. Size of potential outdoor air preconditioning market, segmented by application (all of the United States)

Application category	Outdoor air cfm/person ASHRAE 62-89	Occupancy people/ft^2 ASHRAE 62-89	Outdoor air cfm/ft^2 of facility	Number of Projects per year	Total market size estimated new construction and renovation (million ft^2/year)	Total market size ($) based on average cost of each OA preconditioning system (final market segment analyses)
Office buildings	20	0.01	0.2	28,000[a]	448	127,755,000
Nursing homes/hospital (2)	25	0.01	0.25	5,500[a]	154	71,802,500
Hospital operating rooms	30	0.02	0.6	400[a]	8	10,344,000
Research laboratories	20	0.03	0.6	1300[a]	13	16,142,100
Fast-food restaurants	20	0.1	2	6600[a]	38	110,960,000
Retail stores	20	0.01	0.2	15,600[a]	156	58,266,000
Hotels, dormitories (3)	15	0.02	0.3	13,000[a]	130	76,927,500
School and university classrooms	15	0.05	0.75	1,660[b]	166	222,979,500
Movie theatres and auditoriums	15	0.1	1.5	680[a]	14	30,082,500
Total estimated annual potential						*725,259,100*

[a]From Dodge reports for 1994 (avg. size of each facility from DOE Commercial Buildings Characteristics 1992)
[b]From figures published by the American School and University Magazine Survey.
OA=outdoor air

Table 7. Business Potential for Various Preconditioning Products, Segmented by Market

Application category	Conventional cooling or cooling reheat[a]	Conventional cooling w/ heat pipe	Conventional cooling w/ total recovery[b]	Total recovery dual wheel	DBC approach	Total recovery desiccant DH hybrid
Office buildings	$55,575,000	$10,395,000	$50,625,000	-	$3,600,000	$7,560,000
Nursing homes/hospital (2)	$16,458,750	$9,528,750	$17,325,000	$12,705,000	$7,700,000	$8,085,000
Hospital operating rooms	$2,280,000	-	$2,160,000	-	$2,880,000	$3,024,000
Research laboratories	$2,964,000	$1,930,500	$3,510,000	$514,800	$6,240,000	$982,800
Fast-food restaurants	$50,540,000	$25,080,000	$8,550,000	-	$15,200,000[c]	-
Retail stores	$11,856,000	$7,722,000	$21,060,000	$5,148,000	$12,480,000	-
Hotels, dormitories (3)	$14,820,000	$9,652,500	$17,550,000	$19,305,000	$15,600,000	-
School and university classrooms	$65,051,250	$10,271,250	$56,025,000	$61,627,500	$9,960,000	$15,687,000
Movie theatres and auditoriums	$11,970,000	$5,197,500	$9,450,000	$3,465,000	-	-
Subtotal by product	$231,515,000	$79,777,500	$186,255,000	$102,765,300	$73,660,000	$35,338,800

[a]The conventional approach includes both over-cooling reheat preconditioning of outdoor air treated by conventional HVAC units (no preconditioning)
[b]This section includes both total recovery with cooling/reheat and simple total energy recovery preconditioning to conventional systems
[c]DBC approach for this market is likely contingent upon low cost, commercial product to augment packaged HVAC equipment

1. The overall market potential for preconditioning outdoor air with SOAHUs is substantial ($725,000,000 annually).

2. The largest market potential by far is the school market ($223,000,000/year, 31% of total market potential).

3. Table 7 projects the potential sales volume by SOAHU approach and projects that the majority of the sales dollars will continue to go with conventional cooling approaches (with or without reheat). The next largest potentials are projected for single-wheel total energy recovery, dual-wheel total energy recovery, conventional cooling with heat pipe, and DBC, respectively.

4. The combined DBC and DBC hybrid market potential is estimated at approximately $109,000,000 annually (15% of total market opportunity).

3.4 OVERALL ENERGY IMPACT ANALYSIS FOR SOAHUs

Assuming that the Final Market Segment Analysis Sheets (Appendix E) accurately reflect the breakdown of SOAHU utilization by market; and assuming that ASHRAE 62-1989 is followed in the future; and, for simplicity, assuming that all SOAHUs are operated continuously, their use, compared with conventional over-cooling and reheat approaches, could reduce overall energy usage by 0.032 quads (32×10^{12} Btu) annually (Table 8). This figure is only for new construction and renovation projects on an annual basis. It also reflects the use of a SOAHU approach other than conventional cooling in approximately 48% of these buildings.

3.5 SURVEY RESULTS

A significant amount of data was collected as part of the survey. The results are summarized in Tables 9 and 10. The more important findings with respect to the markets for regenerated desiccant systems include the following:

1. The vast majority of the respondents listed first cost as most important for almost all markets. Energy efficiency was always listed last. (This would agree with the percentage of high-efficiency equipment sold by the major HVAC suppliers).

2. Only one market, hospital operating rooms, needed supply air dew points that were lower than could be accomplished with a conventional cooling coil.

3. DX equipment is the most often used in all markets except nursing homes, hospitals, and research laboratories. Most respondents are aware that maintaining the desired indoor humidity levels with conventional DX systems processing a significant amount of outdoor air is not possible in many markets and locations.

4. In most cases, the respondents thought that space humidity levels needed to be controlled to a maximum of 70% and a minimum of 20%, and that there are benefits to be recognized by controlling the humidity in the occupied spaces. Most felt that the conditions would preferably range between 60% and 30%, but few felt that the market would pay a significant premium to maintain this range.

5. In all cases but fast-food restaurants and research laboratories, access to air to be used for recovery or the secondary side of a DBC system was available at least 50% of the time.

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Table 8. Annual estimated energy savings projected for the preconditioning approaches analyzed (new construction and renovations, assumes continuous operation, reflects results of final market segment analysis, uses Atlanta weather data and energy costs)

Preconditioning system approach	Outdoor air CFM preconditioned, estimate of total market	Energy savings/CFM, excluding demand[a]	Energy savings, all markets combined ($/year)	Energy savings, all markets combined (Million BTU/year)
Conventional cooling w/ run around recovery[c]	48,350,000	$0.73	$35,295,500	7,059,100
Total energy wheel w/ cooling & reheat[b]	82,780,000	$0.67	$55,462,600	11,092,520
Dual wheel total energy recovery[c]	31,141,000	$1.43	$44,531,630	8,906,326
Dual wheel desiccant based (DBC)[c]	18,415,000	$0.95	$17,494,250	3,498,850
Desiccant dehumidification—total recovery hybrid[c]	8,413,800	$0.74	$6,226,212	1,245,242
			$159,010,192	31,802,038
		Annual total	(with demand savings: $190,000,000)	**0.032 quadrillion BTU/year**

[a]Savings estimate use Atlanta DB bin data, assume continuous supply of outdoor air, and only new construction/renovation projects for 1995 projected by "Final Market Segment Analysis" are considered.
[b]The cost of operating this system is compared against a conventional over-cooling/reheat system conditioning outdoor air to 65 degrees/50 grains during the cooling season and 65 degrees and 35 grains during the heating season.
[c]The cost of operating this system is compared against a conventional cooling system without reheat conditioning outdoor air to 51 degrees/50 grains during the cooling season and 65 degrees and 35 grains during the heating season.
[d]Had the new dewpoint weather data been used, savings would have been approximately 10% greater than shown

Table 9. Survey responses for office buildings, nursing homes/hospitals, operating rooms, research labs, and fast-food restaurants					
Questions	Office buildings	Nursing homes/ hospitals	Hospital operating rooms	Research labs	Fast-food restaurants
How important is humidity control for the application?	Not very	Somewhat	Very	Very	Not very
What minimum space relative humidity is acceptable?	No minimum	30%	50%	30%	No minimum
Is humidity control required during unoccupied periods?	No	Yes	Yes	Yes	No
What percentage of total system airflow is outdoor air?	0-20%	30–50%	More than 50%	More than 50%	30–50%
What percentage of the time will ASHRAE 62-89 be followed?	<25%[a]	>75%	>75%	>75%	25–50%
What type of AC equipment is most often used?	DX	Chilled water	Chilled water	Chilled water	DX
Of the following, which is most important?					
First cost	✓				✓
Energy efficiency					
Environmental control		✓	✓	✓	
Project % with access to exhaust/return relief air?	25–50	50–75	>75	25–50	<25
Is low-cost steam or waste heat typically available?	No	Yes	Yes	Yes	No
Are significant benefits gained by controlling humidity?	Somewhat	Yes	Yes	Yes	Somewhat
Are you aware of the new ASHRAE dew-point data?					
Is removing pollutants from the outdoor air important?	No	Somewhat	Somewhat	Yes	No
Rate the opportunity based on ASHRAE 62-89 compliance for the following:					
Desiccant dehumidification or DBC	1	3	3	3	2[c]
Total energy recovery	2	3	3	2	1
Over-cool reheat	2	3	3	3	2

[a] 75% of the offices designed by engineers were reported to be designed in accordance with ASHRAE 62-89, few built by "design build contractors" meet the standard.
[b] 1=poor, 2=fair, 3=good
[c] Provided low-cost commercial product available to this cost-conscious market.

Table 10. Survey responses for retail stores, hotels/dormitories, classrooms, and theaters/auditoriums				
Questions	Retail stores	Hotels/ dormitories	School/university classrooms	Movie theaters/ Auditoriums
How important is humidity control for the application?	Not Very	Somewhat	Somewhat	Not Very
What maximum space relative humidity is acceptable?	70%	60%	60%	70%
What minimum space relative humidity is acceptable?	No minimum	No minimum	20%	No minimum
Is humidity control required during unoccupied periods?	No	Yes	Yes	No
What percentage of total system airflow is outdoor air?	20–30%	20–30%	30–50%	30–50%
What percentage of the time will ASHRAE 62-89 be followed	25–50%	50–75%	50–75%	25–50%
What type of AC equipment is most often used?	DX	50/50	DX	DX
Of the following, which is most important?				
First cost	✓	✓	✓	✓
Energy efficiency				
Environmental control		✓	✓	
Project % that has access to exhaust/return-relief air?	50–75%	50–75%	>75%	>75%
Is low cost steam or waste heat typically available?	No	No	No	No
Are significant benefits gained by controlling humidity?	Somewhat	Yes	Yes	Somewhat
Are you aware of the new ASHRAE dew-point data?				
Is removing pollutants from the outdoor air important?	No	No	No	No
Rate[a] the opportunity based on ASHRAE 62-89 compliance:				
Desiccant dehumidification or DBC	2[b]	2	1	1
Rate the opportunity for total energy recovery	3	3	3	2
Rate the opportunity for over-cool reheat	1	2	2	3

[a] 1=poor, 2=fair, 3=good
[b] Provided low-cost commercial product available to this cost-conscious market.

4. CONCLUSIONS

A number of important conclusions can be drawn as a result of this broad, first-phase market evaluation. The more important conclusions include the following:

1. A very significant market opportunity will exist for SOAHUs as more construction and renovation projects are designed to incorporate the recommendations made by the ASHRAE 62-1989 standard. Based on this investigation, the total potential market is currently $725,000,000 annually (see Table 6, Sect. 3). Based on the market evaluations completed, it is estimated that approximately $398,000,000 (55%) of this total market could be served by DBC systems if they were made cost-effective through mass production.

   Approximately $306,000,000 (42%) of the total can be served by a non-regenerated, desiccant-based total recovery approach, based on the information provided by this investigation. Approximately $92,000,000 (13%) can be served by a regenerated desiccant-based cooling approach (see Table 7, Sect. 3).

2. A projection of the market selling price of various desiccant-based SOAHU systems was prepared using prices provided by Trane for central-station, air-handling modules currently manufactured. The wheel-component pricing was added to these components by SEMCO. This resulted in projected pricing for these systems that is significantly less than that currently offered by custom suppliers (see Table 4, Sect. 2). Estimated payback periods for all SOAHU approaches were quite short when compared with conventional over-cooling and reheat systems. Actual paybacks may vary significantly depending on site-specific considerations.

3. In comparing cost vs benefit of each SOAHU approach, it is critical that the total system design be evaluated. For example, the cost premium of a DBC system is very significant when compared to a conventional air handling system, yet the reduced chiller, boiler, cooling tower, and other expense often equals or exceeds this premium, resulting in a rapid payback period while providing significant energy savings (see Table 5, Sect. 2).

4. The survey conducted as part of this investigation provided several key positions that impact the likely use of one SOAHU approach vs another. The more important ones included these:

   • Most projects are designed using DX systems and the agreement by most interviewed that such systems cannot control humidity while processing the continuous outdoor air quantities required by the ASHRAE IAQ standard provides the single most significant market driver for using a SOAHU approach.

   • First cost was almost always viewed as more important that energy efficiency and environmental conditions by those surveyed.

   • There is a wide difference in opinion regarding the need to control humidity in spaces when they are unoccupied, and regarding the level at which the space humidity needs to be controlled during both occupied and unoccupied modes.

   • Currently, little value is placed on the ability to maintain a dry cooling coil.

   • Wintertime humidification, provided by the desiccant-based recovery wheel, is considered a significant benefit.

   • With the exception of fast-food restaurants and research laboratories, an exhaust air path can be made available for recovery or for the secondary side of a DBC system.

5. The most promising opportunities for regenerated desiccant approaches based on **market need** were determined to be hospital operating rooms, nursing homes and hospitals, and research laboratories. These markets offer approximately $29,000,000 annually in sales potential for DBC and DBC/total energy recovery hybrid systems.

   The hospital operating room market was determined to be the only market investigated that required a dew point below that obtainable with conventional chilled water cooling coils. Nursing homes, hospital bed areas, and research laboratories share the advantage of being less sensitive to project first cost, requiring continuous operation of the outdoor air systems, having a need for humidity control, showing interest in operating dry cooling coils, often having access to low-cost steam for regeneration, and needing features offered only by non-unitary equipment.

   To effectively serve these markets, an institutional grade SOAHU needs to be offered. It would include dual wall construction, high-efficiency filtration, backward-curve airfoil fans, good access for maintenance, options for a variety of coils and humidifiers, and controls designed to work with industry standards.

6. The most promising opportunities for regenerated desiccant approaches based on annual **sales potential** were determined to be hotels and dormitories, retail stores, and school facilities. These markets offer approximately $53,727,000 annually in sales potential for DBC and DBC/total energy recovery hybrid systems.

   The potential for regenerated desiccant systems in retail stores reflects the conditioning of outdoor air to stores that combine food sales and retail, in addition to a portion of large retail facilities using gas-engine cooling in high-demand cost areas. Such facilities as supermarkets and ice rinks have proven offer a significant market potential to desiccant dehumidification systems, yet since they process recirculated air and benefit from the refrigeration loads (ice rinks or freezer loads) and not loads associated with outdoor air or the interior space conditions, these markets were not included in this investigation. (If they were, it is projected that they could add an additional $15 to $25 million annually to the regenerated desiccant-based SOAHU approaches).

   The potential for regenerated desiccant systems in hotels/dormitories reflects the conditioning of outdoor air to these facilities, most likely the corridors, to control humidity in order to avoid moisture problems. The hotel industry has established that humidity problems result in very significant replacement and maintenance costs each year. The regenerated desiccant systems would likely be used where access to an exhaust air path is limited.

   The comparison of outdoor air preconditioning systems shows that passive desiccant systems clearly provide the highest energy savings and lowest first cost. This is a result of the consistent availability of a cool, dry exhaust air path that is perfect for the "total recovery" option. Also, schools need to process large quantities of ventilation air at very high outdoor-air humidity levels (130 grains), dehumidify it, typically to 50 to 55 grains, and provide it to the conditioned space at 65 to 70°. This perfectly fits the capability of the passive-desiccant, dual-wheel system. The active system cannot provide this 75- to 80-grain reduction in a single pass across an active wheel. In addition, the air supplied would be very warm and would require significant post-cooling.

   Therefore, the market opportunity for active desiccant systems in schools is limited to "advanced system designs" using waste heat, engine chillers, ice systems, or areas where electrical demand charges are extremely high. In these applications, where designers feel that there is a benefit provided by operating a dry coil, the active systems may also find acceptance despite the significant first-cost premium.

These markets are far more cost-sensitive than the hospital, nursing home, and research facility markets mentioned earlier. Although the benefits of the technology would allow a percentage of these markets to be served by the same institutional-grade products required by facilities such as hospitals and nursing homes, the construction of the equipment for these markets would have to be more of a unitary, rooftop design in order to gain maximum market penetration.

7. Fast-food restaurants offer a significant potential for regenerated desiccant systems. The kitchen exhaust creates a need for makeup air. The loads within the space are variable and often result in high latent-to-sensible ratios. The hours of operation are high, and gas is always available. However, since these markets are very much driven by first cost, market acceptance will likely depend on a low-cost, commercial-grade product that could be easily coupled with the packaged HVAC units currently used.

8. The investigation concluded that if an exhaust air path is available, the non-regenerated desiccant total recovery approach will almost always be the SOAHU of choice. It typically provides more energy savings at a lower cost than regenerated desiccant systems. It is often more capable of handling the extreme humidity conditions encountered by some environments (120 to 150 grains). It provides winter humidification that makes it a year-round device applicable to most climates. It allows customers and engineers to continue using familiar conventional cooling technologies. It also eliminates the need for evaporative coolers, viewed as a maintenance item and a potential contributor to poor indoor air quality.

    For applications where careful humidity control is not a requirement (most markets), unitary systems can usually be coupled with a single-wheel, non-regenerated, desiccant-based total energy recovery preconditioning systems to meet the needs of the application.

9. Energy analyses completed using weather data that more accurately reflect outdoor air humidity content show the cost of operating conventional cooling systems to be approximately 6% higher than estimates using the traditional dry bulb BIN information. Total energy recovery system operating costs remained approximately the same, while those of regenerated desiccant systems are reduced by approximately 10% (assuming that there is an exhaust air path available for the secondary side of the system) (Table 5, Sect. 2).

10. A very significant amount of energy would be saved in the United States annually should the SOAHU projections made by this investigation occur. Assuming that the systems are operated 50% of the time on average for all markets, a reduction of 0.016 quads per year would be expected (0.032 quadrillion Btu/year if operated continuously). This is projected energy consumption savings compared with using conventional cooling and conventional cooling/reheat systems, using steam heat for the heating mode.

Based on the information resulting from this investigation, the most attractive markets are the following:

*   Nursing homes
*   Hospitals (patient areas and operating rooms)
*   Hotels/dormitories
*   School and university classrooms

The estimated sales opportunity for active desiccant systems at full market penetration, based upon the market investigation, amounts to approximately $63,000,000 annually for the four market segments identified. This business potential appears large enough to merit a serious consideration of technology integration by the major HVAC manufacturers.

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APPENDIX A

SAMPLE ENERGY BIN ANALYSIS SHEETS
FOR EACH SOAHU APPROACH

Summary Sheet for Preconditioning Approach 1: Conventional Cooling with Reheat

ORNL 99-06759/gss
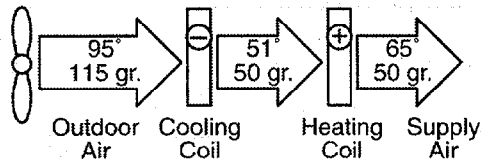
Approach 1. Conventional cooling with reheat.

This approach uses conventional cooling, either chilled water or DX to overcool the outdoor air to remove humidity. Reheat is applied via hot water, heat of rejection from a condenser coil, hot gas with fan heat, steam or electric reheat (if allowed by the local energy code).

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	**7500**	**20000**
Conventional cooling tons at design*	19 tons	57 tons	152 tons
Electrical demand at design* (KW)	25	75	200
Reheat energy at design* (BTU)	48462	145384	387692
Annual energy consumption**			
Atlanta (DB BIN/New WB BIN)	$7,272/$8,000	$21,818/$23,099	$58,180/$61,597
Houston	$6150	$18450	$49199
Los Angeles	$3169	$9507	$25353
Minneapolis	$5640	$16920	$45120
New York	$8988	$26963	$71901
Orlando	$6610	$19830	$52881
Dimensions	27.5"Hx44"Wx154"L	44"x74"x177"	74"x120"x251"
Approximate sales price to market	$2200	$7100	$16700

Key Benefits and Weaknesses:

Benefits include compact design and low first cost. This approach does not require an exhaust air path.

Weaknesses include very high cost of operation and very poor operation with DX approach. No winter time recovery benefit. Concern for freezing the cooling coil or hot water reheat coil during winter operation. Very large chiller/DX condenser required in addition to large capacity for reheat, which must be factored into first cost evaluation.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 2: Conventional Cooling with Run-around Recovery

ORNL 99-06760/gss
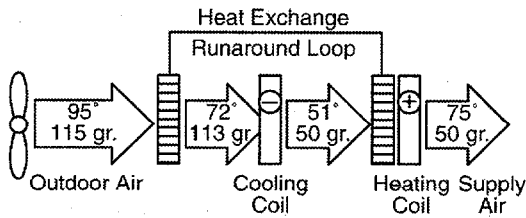
Approach 2. Conventional cooling with runaround recovery reheat.

This approach uses conventional cooling, either chilled water, or DX to overcool the outdoor air to remove humidity. Reheat is applied via the integration of a run-around coil, heat pipe, or plate exchanger and assisted, in some cases, with additional hot water, steam, or electric heat.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	7500	20000
Conventional cooling tons at design*	15.5 tons	46.5 tons	124 tons
Electrical demand at design* (KW)	21	63	169
Reheat energy at design* (BTU)	0	0	0
Annual energy consumption**			
Atlanta	$5263	$15790	$42106
Houston	$4615	$13844	$36918
Los Angeles	$2518	$7555	$20147
Minneapolis	$5097	$15291	$40777
New York	$6992	$20976	$55937
Orlando	$3826	$11479	$30611
Dimensions	55"Hx44"Wx141"L	88"x74"x193"	148"x120"x260"
Approximate sales price to market	$4000	$12150	$31000

Key Benefits and Weaknesses:

Benefits include compact design (at lower airflows) and relatively low first cost. This approach does not require an exhaust air path. It is less energy consuming that a conventional over cooling with reheat system and it reduces the required cooling capacity (modestly) and reheating source (significantly).

Weaknesses include high pressure losses associated with two wet components. No winter time benefit to this approach despite the horsepower penalty year round. This often offsets much of the cooling season savings. The temperature leaving the unit is too cool on many overcast, humid days (ie: 72 degress and raining) so additional reheat is often required. This approach does not stabilize coil entering conditions well enough over various outdoor air conditions to allow DX to be utilized effectively.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 3: Dessicant Total Energy Recovery with Cooling/Reheat

ORNL 99-06761/gss

Approach 3. Total recovery and conventional cooling with reheat.
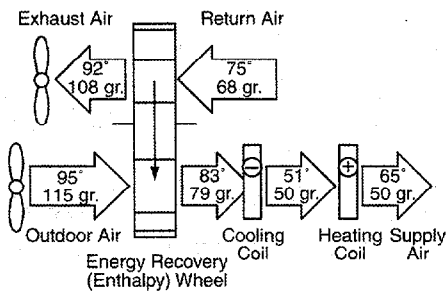
This approach uses a total energy wheel to precool and predehumidify the outdoor air delivered to a conventional cooling coil, either chilled water, or DX, to overcool the outdoor air to remove humidity. Reheat is applied via hot water, steam, or electric.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	7500	20000
Conventional cooling tons at design*	11 tons	34 tons	90 tons
Electrical demand at design* (KW)	16	48	128
Reheat energy at design* (BTU)	48462	145384	387692
Annual energy consumption**			
Atlanta	$5381	$16142	$43046
Houston	$4891	$14673	$39127
Los Angeles	$4226	$12678	$33808
Minneapolis	$3151	$9452	$25205
New York	$5217	$15651	$41735
Orlando	$5202	$15605	$41614
Dimensions	55"Hx44"Wx171"L	88"x74"x194"	148"x120"x268"
Approximate sales price to market	$7015	$16783	$37275

Key Benefits and Weaknesses

Benefits include a significant reduction in operating cost over approach 1, as well as reduction in refrigeration tons required. The total energy wheel provides more consistent outdoor air conditions to the cooling coil making the use of DX more functional. The energy recovery wheel also provides very significant wintertime energy savings as well as free humidification. It protects the coils during winter operation. Peak demand is cut significantly.

Weaknesses include higher cost than option 1 and a larger size since it is an exhaust as well as supply unit. Biggest weakness is that it needs an exhaust air volume to function.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 4: Dual-Wheel Total Energy Recovery

ORNL 99-06762/gss

Approach 4. Dual wheel total recovery configuration.
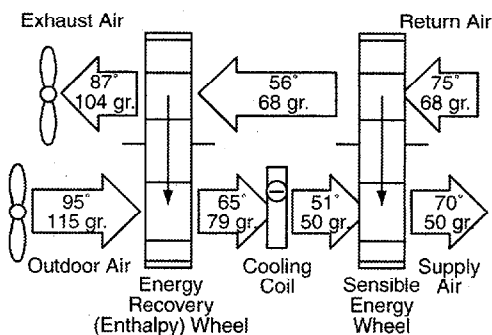
This approach uses two energy recovery wheels, one sensible and one latent, and a cooling coil to provide the same conditions as approach 3, except the sensible wheel provides free reheat and provides precooled air to to the return side of the total wheel, significantly reducing the required conventional cooling capacity.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	**7500**	**20000**
Conventional cooling tons at design*	7 tons	21 tons	56 tons
Electrical demand at design* (KW)	11	33	88
Reheat energy at design* (BTU)	0	0	0
Annual energy consumption**			
Atlanta (DB BIN/New WB BIN)	$3,366/$3,383	$10,098/$10,147	$26,928/$27,060
Houston	$2587	$7762	$20698
Los Angeles	$1906	$5717	$15246
Minneapolis	$1981	$5942	$15845
New York	$3367	$10100	$26933
Orlando	$2776	$8328	$22209
Dimensions	55"Hx44"Wx188"L	88"x74"x211"	148"x120"x285"
Approximate sales price to market	$11100	$24750	$53650

Key Benefits and Weaknesses:

Benefits include a very significant reduction in operating cost over options 1, 2 and 3, as well as the greatest reduction in refrigeration tons required. The sensible wheel provides the required reheat without allocating any energy source for this benefit since the heat comes from the space sensible load. This approach provides the most consistent outdoor air conditions to the cooling coil making it ideally suited for cost effective DX cooling. The energy recovery wheel also provides very significant wintertime energy savings as well as free humidification. It protects the coils during winter operation. Peak demand is cut significantly.

Weaknesses include slightly higher cost than options 1-3 for the system (although reduced cooling and heating plant often offset this difference) and a larger size since it incorporates two wheels. Biggest weakness is that it needs an exhaust air volume to operate.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 5: Dual-wheel Dessicant-based Cooling and Post Cooling

ORNL 99-06763A/gss

Approach 5. Dual wheel desiccant-based cooling.
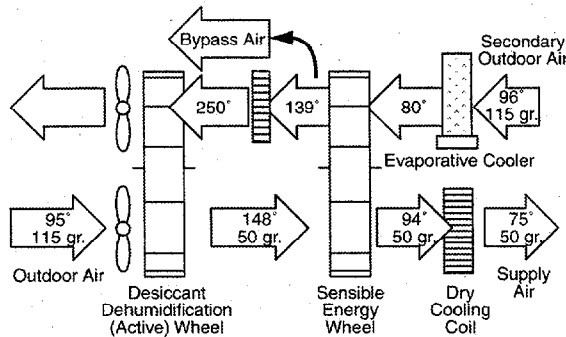
This approach uses two wheels, a dehumidification, and sensible recovery wheel to provide dry air at a moderate temperature. Heat is used to regenerate the dehumidification wheel once it has adsorbed moisture from the outdoor airstream. An evaporative cooler is usually required to enhance the cooling effect associated with the sensible wheel. Modest post cooling is often required during peak conditions.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	7500	20000
Conventional cooling tons at design*	4 tons	12 tons	32 tons
Electrical demand at design* (KW)	7	21	57
Reheat energy at design* (BTU)	159875	479625	1279000
Annual energy consumption**			
Atlanta (DB BIN/New WB BIN)	$4,433/$4072	$13,298/$12,216	$35,463/$32,573
Houston	$4718	$14154	$37744
Los Angeles	$3925	$11774	$31397
Minneapolis	$3542	$10626	$28336
New York	$4948	$14843	$39582
Orlando	$6308	$18925	$50466
Dimensions	55"Hx44"Wx188"L	88"x74"x211"	148"x120"x285"
Approximate sales price to market	$14700	$30215	$64838

Key Benefits and Weaknesses:

Benefits include the ability to shift the cooling load from high cost electricity to low cost gas. To provide air that is drier (in some cases) than obtainable with conventional cooling systems. Key advantage over desiccant based recovery wheels is that a return airstream is not necessary, although helpful. No wet cooling coils in this system.

Weaknesses include higher cost than previous options. The size of the system is quite large due to the two wheel arrangement operating at moderate face velocities. Often requires the evaporative cooler for economics, yet it can be a high maintenance item and is avoided in some applications.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 6: Dessicant Dehumidification—Total Recovery Hybrid

Approach 6. Desiccant dehumidification—total recovery hybrid.
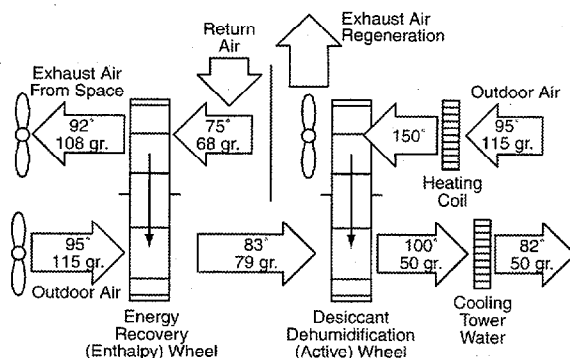
This approach combines the best of total recovery with the best of dehumidification wheel performance. The total energy wheel removes much of the moisture and precools the outdoor air, which is then introduced to a dehumidification wheel to remove only the remaining moisture desired.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	7500	20000
Conventional cooling tons at design*	6 tons	17 tons	45 tons
Electrical demand at design* (KW)	9	28	73
Reheat energy at design* (BTU)	78125	234375	625000
Annual energy consumption**			
Atlanta	$4264	$12792	$34113
Houston	$3455	$10365	$27640
Los Angeles	$2857	$8571	$22856
Minneapolis	$2661	$7983	$21288
New York	$4442	$13327	$35538
Orlando	$4592	$13777	$36738
Dimensions	55"Hx44"Wx203"L	88"x74"x226"	148"x120"x300"
Approximate sales price to market	$15750	$31350	$67850

Key Benefits and Weaknesses:

Benefits include the ability to shift the cooling load from high cost electricity to low cost gas. To provide air that is much drier than obtainable with conventional cooling systems. No wet cooling coils in this system. Since the DH wheel does less work than in option 5, the temperature leaving the desiccant is far more moderate, allowing the sensible recovery wheel to be replaced with a tower or conventional cooling coil. The energy efficiency (COP) when compared to option 5 is much improved if free post cooling is available or not required. Air much drier than possible with option 5 is produced using this approach. This approach also provides for latent recovery in the winter as well as summer.

Weaknesses include those listed for option 5 plus the fact that a return air stream is required for the total energy recovery wheel portion of the design.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

Summary Sheet for Preconditioning Approach 7: Dual-wheel Dessicant-based Cooling—Total Recovery Hybrid

Approach 7. Dual wheel desiccant-based cooling—total recovery hybrid.
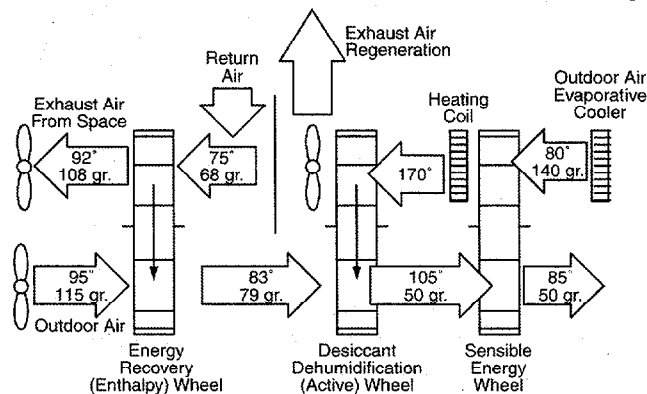
This approach combines the best of total recovery with the best of dessicant-based cooling performance. The total energy wheel removes much of the moisture and precools the outdoor air, which is then introduced to a dehumidification wheel to remove only the remaining moisture desired. The added sensible wheel eliminates the need for post cooling at most conditions while reducing the regeneration energy consumption.

Preconditioning Approach Analysis Summary	System CFM (Nominal)		
	2500	7500	20000
Conventional cooling tons at design*	2 tons	6.5 tons	17 tons
Electrical demand at design* (KW)	6	17	45
Reheat energy at design* (BTU)	70000	210000	560000
Annual energy consumption**			
Atlanta	$3158	$9474	$25264
Houston	$3030	$9091	$24242
Los Angeles	$2506	$7517	$20045
Minneapolis	$2334	$7001	$18669
New York	$3573	$10719	$28584
Orlando	$3924	$11772	$31392
Dimensions	55"Hx44"Wx235"L	88"x74"x258"	148"x120"x332"
Approximate sales price to market	$20500	$39300	$84200

Key Benefits and Weaknesses:

Benefits include the ability to shift the cooling load from high cost electricity to low cost gas. To provide air that is much drier than obtainable with conventional cooling systems. No wet cooling coils in this system. Since the DH wheel does less work than in option 5, the temperature leaving the desiccant is far more moderate, allowing the sensible recovery wheel to replace the tower water or conventional cooling coil. The energy efficiency (COP) when compared to option 5&6 is much improved.

Weaknesses include those listed for option 5 plus the fact that a return air stream is required for the total energy recovery wheel portion of the design. Due to the three wheels involved, space is very often a problem and the cost is at a premium of over other approaches.

* Design Based on outdoor air at 95 deg/115 grains, supply outdoor air at 50 grains, 65-75 degree dry bulb
** Energy consumption based on continuous operation and local 1994 utility costs for commercial facility and adjusts for cooling tower, pumps & parasitic fan energy assuming 1.5" of external static pressure, and includes cost of heating and humidifying supply air to 65 degrees and 35 grains during the heating season

APPENDIX B

DETAILED SPECIALIZED OUTDOOR AIR HANDLING UNIT (SOAHU)
MODELING INPUTS AND RESULTS FOR ATLANTA
and
DESSICANT-BASED COOLING MODELING DATA FOR HOUSTON,
MINNEAPOLIS, NEW YORK, AND ORLANDO

Annual Operating Cost Estimate for "Conventional Over-cooling and Reheat Approach"

APPROACH: Cooling/Reheat

LOCATION: Atlanta, Ga.

Given Information		
Supply/Exhaust CFM	20000	
Cooling coil temperature/enthalpy	51	20.5 Btu/lb
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb
Summer return air temperature/enthalpy/grains	75	28 Btu/lb
Winter desired supply air temperature/enthalpy	65	21 Btu/lb
Winter return air temperature/enthalpy	72	22.5 Btu/lb
Total recovery effectiveness	n/a	
Sensible recovery effectiveness	0.6	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output
Electrical Demand Charges ($/KW)	$8.00	
Cost of heating fuel	$4.80	$/million BTU of heating fuel
Boiler efficiency	78%	
% time of operation	100	Average KW/ton 1

Conventional Cooling/Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)					(A) BTU Required to dehumidify with Cooling Coil	(B) BTU Required to Reheat (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.				
102	73	1	35.9	72.2	1385426	387692		$3
97	74	20	36.8	85.9	29397738	7753846		$57
92	74	135	36.9	94.0	199025632	52338462		$387
87	72	367	35.2	90.9	484297069	142283077		$967
82	70	612	33.5	88.3	717068969	237267692		$1,477
77	69	839	32.7	91.2	924615943	325273846		$1,937
72	67	1201	31.2	89.2	1156997361	465618462		$2,523
67	60	986	26.2	65.0	506776650	382264615		$1,407
62	57	845	24.3	60.6	289163559	327600000		$988
57	52	773	21.3	49.4				$507
52	48	709	19.2	43.2			149078575	$465
47	43	665	16.7	34.9			331817717	$436
42	39	608	14.8	30.8			433252807	$399
37	34	471	12.7	24.6			452807222	$309
32	30	303	11.1	22.0			347683123	$199
27	25	134	9.2	17.5			183064295	$88
22	21	51	7.7	16.0			78113544	$33
17	16	23	6.0	12.7			39739816	$15
12	11	9	4.4	10.1			17226474	$6
7	6	1	2.9	7.8			2091896	$1
2	2	1	1.7	7.9			2229795	$1
-3	-3	0	0.2	6.3				

Subtotal Energy Cost	$21,544	$9,316	$9,778	$12,206

Cooling Season Energy Cost	$40,607	
Heating Season Energy Cost	$12,237	Total Annual Energy Cost Estimate for Conventional Approach $58,180
Demand Charges (Cooling Season Only)	$5,337	

Annual Operating Cost Estimate for "Conventional Over-cooling and Reheat Approach"

APPROACH: Cooling/Reheat

LOCATION: Atlanta, Ga.

Given Information		
Supply/Exhaust CFM	20000	
Cooling coil temperature/enthalpy	51	20.5 Btu/lb
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb
Summer return air temperature/enthalpy/grains	75	28 Btu/lb
Winter desired supply air temperature/enthalpy	65	21 Btu/lb
Winter return air temperature/enthalpy	72	22.5 Btu/lb
Total recovery effectiveness	n/a	
Sensible recovery effectiveness	0.6	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output
Electrical Demand Charges ($/KW)	$8.00	
Cost of heating fuel	$4.80	$/million BTU of heating fuel
Boiler efficiency	78%	
% time of operation	100	Average KW/ton 1

Conventional Cooling/Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with Cooling Coil	(B) BTU Required to Reheat (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
87	78.5	26	41.2	129.5	48488819	10080000		$90
86	76.5	111	39.3	118.6	187493839	43033846		$354
83	74.5	405	37.4	111.6	616882294	157015385		$1,191
78	72.5	634	35.7	108.4	866254198	245796923		$1,715
75	70.5	776	34.0	102.3	942611896	300849231		$1,923
74	68.5	573	32.4	93.5	612049656	222147692		$1,294
72	66.5	363	30.8	86.7	337120369	140732308		$744
69	64.5	374	29.3	82.0	297660839	144996923		$692
67	62.5	378	27.9	76.0	252484173	146547692		$627
66	60.5	311	26.6	68.8	169402929	120572308		$458
64	58.5	327	25.2	63.5	139653311	126775385		$424
62	56.5	405	24.0	58.6	127226101	157015385		$457
61	54.5	376	22.8	52.4	77141272	145772308		$362
59	52.5	347	21.6	48.0				$228
57	50.5	376	20.5	44.0			21461432	$247
55	48.5	326	19.4	40.2			59016638	$214
52	46.5	344	18.4	38.2			103250842	$226
50	44.5	346	17.4	34.9			143762055	$227
48	42.5	278	16.4	31.8			146477396	$182
45	40.5	230	15.5	30.4			145903007	$151
43	38.5	215	14.6	27.7			158800129	$141
42	36.5	181	13.7	23.6			151982627	$119
37	32.5	320	12.1	20.6			330374886	$210
32	29	303	11.4	24.1			336006440	$199
27	24	184	9.5	19.8			243820712	$121
22	20	138	7.9	16.9			209267273	$91
17	15	78	6.2	13.7			133431286	$51
12	12	35	4.7	12.1			65731216	$23
		8760			Subtotal Energy Cost $23,372	$9,414	$10,797	$12,065

Cooling Season Energy Cost	$41,416	Total Annual	$61,597
Heating Season Energy Cost	$14,113	Energy Cost	
Demand Charges (Cooling Season Only)	$6,068	Estimate for Conventional Approach	

Annual Operating Cost Estimate for "Conventional Over-cooling, Run-Around Recovery, Reheat Approach"

APPROACH: Cooling/Reheat
with Run-Around.
Sensible Recovery

LOCATION: Atlanta, Ga.

Given Information		
Supply/Exhaust CFM	20000	
Cooling coil leaving air temperature/enthalpy	51	20.5 Btu/lb
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb
Summer return air temperature/enthalpy/grains	75	28 Btu/lb
Winter desired supply air temperature/enthalpy	65	21 Btu/lb
Winter return air temperature/enthalpy	72	22.5 Btu/lb
Sensible recovery effectiveness	0.6	
Sensible recovery effectiveness	0.6	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output
Electrical Demand Charges ($/KW)	$8.00	
Cost of heating fuel	$4.80	$/million BTU of heating fuel
Boiler efficiency	78%	
% time of operation	100	Average KW/ton 1

Conventional Cooling/Run-Around Sensible Recovery/Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with Cooling Coil	(B) BTU Required to Reheat (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	73	1	35.9	72.2	867026	0		$2
97	74	20	36.8	85.9	19029738	0		$47
92	74	135	36.9	94.0	129041632	0		$320
87	72	367	35.2	90.9	313069549	0		$814
82	70	612	33.5	88.3	471191849	0		$1,280
77	69	839	32.7	91.2	641906503	0		$1,749
72	67	1201	31.2	89.2	830133201	46561846		$2,371
67	60	986	26.2	65.0	302319690	120140308		$1,378
62	57	845	24.3	60.6	168700359	173160000		$1,045
57	52	773	21.3	49.4				$724
52	48	709	19.2	43.2			149078575	$664
47	43	665	16.7	34.9			331817717	$623
42	39	608	14.8	30.8			433252807	$570
37	34	471	12.7	24.6			452807222	$441
32	30	303	11.1	22.0			347683123	$284
27	25	134	9.2	17.5			183064295	$126
22	21	51	7.7	16.0			78113544	$48
17	16	23	6.0	12.7			39739816	$22
12	11	9	4.4	10.1			17226474	$8
7	6	1	2.9	7.8			2091896	$1
2	2	1	1.7	7.9			2229795	$1
-3	-3	0	0.2	6.3				

		(A)	(B)	(C)	(D)
Subtotal Energy Cost		$14,381	$1,631	$9,778	$12,519

Cooling Season Energy Cost	$25,019	Total Annual
Heating Season Energy Cost	$13,291	Energy Cost
Demand Charges (Cooling Season Only)	$3,796	Estimate for
		Conventional Approach
		with Run-around Sensible Recovery

Total Annual Energy Cost Estimate $42,106

(E) Supply Air Temp Entering Reheat Coil	(F) Outdoor Air Enthalpy Entering Cooling Coil
75	30.13
75	31.07
75	31.12
72.6	29.98
69.6	29.05
66.6	29.00
63.6	28.18
60.6	23.91
57.6	22.72

Note 1: assumes parasitic loss of all components in the system as well as an average .3 KW/ton for operating the chilled water pumps and cooling tower
Note 2: assumes that a control strategy is in place to limit sensible recovery to avoid over-heating

Annual Operating Cost Estimate for "Total Recovery with Conventional Cooling and Reheat"

APPROACH: Cooling/Reheat with Total Energy Recovery

LOCATION: Atlanta, Ga.

Given Information		
Supply/Exhaust CFM	20000	
Cooling coil leaving air temperature	51	20.5 Btu/lb
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb
Summer return air temperature/enthalpy/grains	75	28 Btu/lb
Winter desired supply air temperature/enthalpy	65	21 Btu/lb
Winter return air temperature/enthalpy	72	22.5 Btu/lb
Total recovery effectiveness	0.76	0.75 Pressure Loss/side
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output
Electrical Demand Charges ($/KW)	$8.00	
Cost of heating fuel	$4.80	$/million BTU of heating fuel
Boiler efficiency	78%	
% time of operation	100	Average KW/ton 1

Total Recovery/Conventional Cooling/Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidification wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	73	1	35.9	72.2	845502	387692		$2
97	74	20	36.8	85.9	17315457	7753846		$45
92	74	135	36.9	94.0	117021152	52338462		$302
87	72	367	35.2	90.9	304502297	142283077		$801
82	70	612	33.5	88.3	486052553	237267692		$1,303
77	69	839	32.7	91.2	652314826	325273846		$1,765
72	67	1201	31.2	89.2	893792367	465618462		$2,466
67	60	986	26.2	65.0	505818000	382264615		$1,683
62	57	845	24.3	60.6	288990000	327600000		$1,225
57	52	773	21.3	49.4				$724
52	48	709	19.2	43.2				$664
47	43	665	16.7	34.9				$623
42	39	608	14.8	30.8			24005289	$570
37	34	471	12.7	24.6			46719118	$441
32	30	303	11.1	22.0			43587796	$284
27	25	134	9.2	17.5			26309277	$126
22	21	51	7.7	16.0			12038789	$48
17	16	23	6.0	12.7			6512171	$22
12	11	9	4.4	10.1			2950508	$8
7	6	1	2.9	7.8			370517	$1
2	2	1	1.7	7.9			403612	$1
-3	-3	0	0.2	6.3				
Subtotal Energy Cost					$16,333	$9,316	$782	$13,104

Cooling Season Energy Cost	$35,241	
Heating Season Energy Cost	$4,295	Total Annual $43,046
Demand Charges (Cooling Season Only)	$3,511	Energy Cost Estimate for Total Recovery/Reheat

(E) Supply air temperature Entering Reheat Coil	(F) Outdoor Air Enthalpy Entering Cooling Coil
51	29.89
51	30.12
51	30.13
51	29.72
51	29.32
51	29.14
51	28.77
51	26.20
51	24.30
	22.22
	21.70
	21.10
	20.66
	20.14
	19.75
	19.30
	18.95
	18.55
	18.16
	17.79
	17.50

Note 1: assumes parasitic loss of all components in the system as well as an average .3 KW/ton for operating the chilled water pumps and cooling tower

Annual Operating Cost Estimate for "Dual wheel Total Recovery/Conventional Cooling/Sensible Recovery Reheat Approach"

APPROACH: Cooling/Dual Wheel Total Rec. w/ Sensible Rec. Reheat

LOCATION: Atlanta, Ga.

Given Information				
Supply/Exhaust CFM	20000			
Cooling coil leaving temperature/enthalpy	51	20.5 Btu/lb		
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb		
Summer return air temperature/enthalpy/grains	75	28 Btu/lb	65 Grains	
Winter desired supply air temperature/enthalpy	65	21 Btu/lb		
Winter return air temperature/enthalpy	72	22.5 Btu/lb		
Sensible recovery effectiveness	0.78	0.7 Pressure Loss/side		
Total recovery effectiveness	0.76	0.75 Pressure Loss/side		
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output		
Electrical Demand Charges ($/KW)	$8.00			
Cost of heating fuel	$4.80	$/million BTU of heating fuel		
Boiler efficiency	78%			
% time of operation	100	Average KW/ton	1	

Dual Wheel Total Recovery/Conventional Cooling/Sensible Recovery Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with Cooling Coil	(B) BTU Required to Reheat (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	73	1	35.9	72.2	545876	0		$2
97	74	20	36.8	85.9	11322933	0		$39
92	74	135	36.9	94.0	76571617	0		$267
87	72	367	35.2	90.9	194539486	0		$705
82	70	612	33.5	88.3	302681327	0		$1,142
77	69	839	32.7	91.2	400928457	0		$1,545
72	67	1201	31.2	89.2	533941318	0		$2,152
67	60	986	26.2	65.0	332012977	0		$1,607
62	57	845	24.3	60.6	249700127	0		$1,325
57	52	773	21.3	49.4				$869
52	48	709	19.2	43.2				$797
47	43	665	16.7	34.9				$748
42	39	608	14.8	30.8			24005289	$684
37	34	471	12.7	24.6			46719118	$530
32	30	303	11.1	22.0			43587796	$341
27	25	134	9.2	17.5			26309277	$151
22	21	51	7.7	16.0			12038789	$57
17	16	23	6.0	12.7			6512171	$26
12	11	9	4.4	10.1			2950508	$10
7	6	1	2.9	7.8			370517	$1
2	2	1	1.7	7.9			403612	$1
-3	-3	0	0.2	6.3				

	Subtotal Energy Cost	$10,511	$0	$782	$12,998

Cooling Season Energy Cost	$19,294	
Heating Season Energy Cost	$4,997	
Demand Charges (Cooling Season Only)	$2,637	

Total Annual Energy Cost Estimate for Dual Wheel Total Recovery/Sensible Rec. Reheat $26,928

(E) Supply Air Temp Entering Reheat Coil	(F) Outdoor Air Enthalpy Entering Cooling Coil	(G) Exhaust Air Temp Entering Total Energy Wheel	(H) Exhaust Air Enthalpy Entering Total Energy Wheel
69.6	26.57	56.4	23.6
69.6	26.79	56.4	23.6
69.6	26.80	56.4	23.6
69.6	26.39	56.4	23.6
69.6	26.00	56.4	23.6
69.6	25.81	56.4	23.6
69.6	25.44	56.4	23.6
69.6	24.24	56.4	23.6
69.6	23.78	56.4	23.6
	20.66		
	20.14		
	19.75		
	19.30		
	18.95		
	18.55		
	18.16		
	17.79		
	17.50		

Note 1: assumes parasitic loss of all components in the system as well as an average .3 KW/ton for operating the chilled water pumps and cooling tower

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant Based Cooling Typical Configuration

LOCATION: Atlanta, Ga.

Given Information				
Supply/Regeneration CFM	20000 cfm		10000 cfm	
Summer desired supply grain level	50			
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb		
Summer return air temperature/enthalpy/wet bulb	75	28 Btu/lb	65	62.5
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	Grains	Wet Bulb
Winter return air temperature/enthalpy	72	22.5 Btu/lb		
Dehumidification wheel pressure loss	1	1.25 Regen side		
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side		
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output		
Electrical Demand Charges ($/KW)	$8.00			
Cost of heating fuel	$3.50	$/million BTU of heating fuel		
Boiler efficiency	78%			
% time of operation	100	Average KW/ton	1	
		0.65 Pressure Loss/side		

Desiccant Based Cooling Approach

WEATHER DATA (TWELVE YEAR AVERAGE)					(A) BTU Required to dehumidify with dehumidification wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.				
102	73	1	35.9	72.2	818994	29253		$1
97	74	20	36.8	85.9	17646139	1031930		$30
92	74	135	36.9	94.0	118512628	7264936		$202
87	72	367	35.2	90.9	306375129	7833685		$550
82	70	612	33.5	88.3	542299688	0		$918
77	69	839	32.7	91.2	762351575	0		$1,258
72	67	1201	31.2	89.2	974532660	0		$1,801
67	60	986	26.2	65.0	912927148	0		$1,479
62	57	845	24.3	60.6	751744120	0		$1,267
57	52	773	21.3	49.4				$1,159
52	48	709	19.2	43.2				$1,063
47	43	665	16.7	34.9				$997
42	39	608	14.8	30.8			25881688	$912
37	34	471	12.7	24.6			85225639	$706
32	30	303	11.1	22.0			77619173	$454
27	25	134	9.2	17.5			48859324	$201
22	21	51	7.7	16.0			21382517	$76
17	16	23	6.0	12.7			11620802	$34
12	11	9	4.4	10.1			5231610	$13
7	6	1	2.9	7.8			648916	$1
2	2	1	1.7	7.9			675797	$1
-3	-3	0	0.2	6.3				

	(A)	(B)	(C)	(D)
Subtotal Energy Cost	$19,686	$81	$970	$13,127

Cooling Season Energy Cost	$27,274	Total Annual	$35,463
Heating Season Energy Cost	$6,590	Energy Cost	
Demand Charges (Cooling Season Only)	$1,599	Traditional Desiccant Based Cooling Approach	

(E) Humidity Grains Entering DH Wheel	(F) Temperature Entering DH Wheel	(G) Enthalpy Entering DH Wheel	(H) Temperature Leaving the Dehumidification Wheel	(I) Humidity Leaving the Dehumidification Wheel	(J) Regeneration Temperature Required	(K) Temperature Leaving Regen Evaporative Cooler	(L) Temperature to Regeneration Coil	(M) Temperature Leaving Sensible Wheel To Cooling Coil	(N) Enthalpy Leaving Sensible Wheel
72.2	102.0	35.9	126.8	50	190	63.8	114.2	76	
85.9	97.0	36.8	131.9	50	200	63.8	118.3	77	
94.0	92.0	36.9	132.5	50	200	63.8	118.7	77	
90.9	87.0	35.2	124.9	50	190	63.8	112.7	76	
88.3	82.0	33.5	119.0	50	190	63.8	108.0	75	
91.2	77.0	32.7	116.4	50	190	63.8	105.9	74	
89.2	72.0	31.2	108.9	50	175	63.8	99.9	73	
65.0	67.0	26.2	89.4	50	170	63.8	84.3	69	
60.6	62.0	24.3	81.1	50	160	63.8	77.6	67	
30.8	42.0	14.8						66	20.6
24.6	37.0	12.7						65	19.4
22.0	32.0	11.1						64	18.8
17.5	27.0	9.2						63	17.8
16.0	22.0	7.7						62	17.4
12.7	17.0	6.0						61	16.6
10.1	12.0	4.4						60	16.0
7.8	7.0	2.9						59	15.4
7.9	2.0	1.7						58	15.1

Note1: assumes parasitic loss for all components in the system as well as an average .3KW/ton for operating the chilled water pumps and operating the cooling tower

Annual Operating Cost Estimate for "Dehumidification/Total Recovery Hybrid Approach"

| APPROACH: | Dehumidification with Total Energy Recovery Hybrid |
| LOCATION: | Atlanta, Ga. |

Given Information

Supply/Regeneration CFM	20000 cfm	7000 cfm
Summer desired supply grain level	50	
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb
Summer return air temperature/enthalpy/grains	75	28 Btu/lb	65 Grains
Winter desired supply air temperature/enthalpy	65	21 Btu/lb
Winter return air temperature/enthalpy/grains	72	22.5 Btu/lb	34 Grains
Dehumidification wheel pressure loss	1	1.25 Regen side
Total recovery effectiveness	0.76	0.75 Pressure Loss/side
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output
Electrical Demand Charges ($/KW)	$8.00	
Cost of heating fuel/Summer cooling rate	$3.50	$/million BTU of heating fuel
Boiler efficiency	78%	
% time of operation	100	Average KW/ton 1

Desiccant Dehumidification/Total Recovery/Hybrid

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidificaiton wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidity (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	73	1	35.9	72.2	514080	548026		$1
97	74	20	36.8	85.9	11037600	10960522		$24
92	74	135	36.9	94.0	79606800	73983520		$164
87	72	367	35.2	90.9	202539960	201125570		$447
82	70	612	33.5	88.3	337750560	335391958		$746
77	69	839	32.7	91.2	494741520	459793877		$1,022
72	67	1201	31.2	89.2	708205680	658179316		$1,463
67	60	986	26.2	65.0	581424480	540353710		$1,201
62	57	845	24.3	60.6	530220600	463082034		$1,030
57	52	773	21.3	49.4				$942
52	48	709	19.2	43.2				$864
47	43	665	16.7	34.9				$810
42	39	608	14.8	30.8			19354883	$741
37	34	471	12.7	24.6			43262080	$574
32	30	303	11.1	22.0			41425492	$369
27	25	134	9.2	17.5			25399911	$163
22	21	51	7.7	16.0			11703398	$62
17	16	23	6.0	12.7			6369159	$28
12	11	9	4.4	10.1			2897763	$11
7	6	1	2.9	7.8			365009	$1
2	2	1	1.7	7.9			398275	$1
-3	-3	0	0.2	6.3				
Subtotal Energy Cost					**$13,219**	**$13,717**	**$529**	**$10,666**

Cooling Season Energy Cost	$33,036	Total Annual $39,187
Heating Season Energy Cost	$5,096	Energy Cost
Demand Charges (Cooling Season Only)	$1,056	Estimate for Desiccant
		Dehumidification/Total Recovery Hybrid

(E) Humidity Grains Entering DH Wheel	(F) Temperature Entering DH Wheel	(G) Enthalpy Entering DH Wheel	(H) Temperature Leaving the Dehumidification Wheel	(I) Humidity Leaving the Dehumidification Wheel	(J) Regeneration Temperature Required
66.7	81.5	30.0	100.4	50	170
70.0	80.3	30.2	101.3	50	170
72.0	79.1	30.2	101.4	50	170
71.2	77.9	29.8	98.8	50	160
70.6	76.7	29.4	96.9	50	155
71.3	75.5	29.3	96.2	50	155
70.8	74.3	28.9	94.3	50	150
65.0	73.1	27.7	89.3	50	145
63.9	71.9	27.2	87.6	50	145
33.2	64.8	20.7			
31.7	63.6	20.2			
31.1	62.4	19.8			
30.0	61.2	19.4			
29.7	60.0	19.0			
28.9	58.8	18.6			
28.3	57.6	18.2			
27.7	56.4	17.8			
27.7	55.2	17.5			

Note1: assumes parasitic loss for all components in the system as well as an average .3KW/ton for operating the chilled water pumps and operating the cooling tower

Annual Operating Cost Estimate for "Total Recovery with Traditional DBC Approach"

APPROACH: Total Recovery with Traditional DBC Approach

LOCATION: Atlanta, Ga.

Given Information			
Supply/Regeneration CFM	20000 cfm		7000 cfm
Summer desired supply grain level	50		
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb	
Summer return air temperature/enthalpy/grains	75	28 Btu/lb	65 Grains
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	
Winter return air temperature/enthalpy/grains	72	22.5 Btu/lb	34 Grains
Dehumidification wheel pressure loss	1	1.25 Regen side	
Total recovery effectiveness	0.76	0.75 Pressure Loss/side	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output	
Electrical Demand Charges ($/KW)	$8.00		
Cost of heating fuel/summer cooling rate	$3.50	$/million BTU of heating fuel	
Boiler efficiency	78%		
% time of operation	100	Average KW/ton	1
Sensible Recovery effectiveness	0.8	0.65 Pressure Loss/side	

Desiccant Dehumidification/Total & Sensible Recovery/Hybrid

					(A) BTU Required to dehumidify with dehumidificaiton wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
WEATHER DATA (TWELVE YEAR AVERAGE)								
OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.				
102	73	1	35.9	72.2	632944	129477		$2
97	74	20	36.8	85.9	13234249	2848165		$34
92	74	135	36.9	94.0	89362033	18109576		$228
87	72	367	35.2	90.9	222074259	30637429		$619
82	70	612	33.5	88.3	356578658	21582239		$1,032
77	69	839	32.7	91.2	493973715	6891272		$1,415
72	67	1201	31.2	89.2	679452117	0		$2,026
67	60	986	26.2	65.0	560855504	0		$1,663
62	57	845	24.3	60.6	493515367	0		$1,425
57	52	773	21.3	49.4				$1,304
52	48	709	19.2	43.2				$1,196
47	43	665	16.7	34.9				$1,122
42	39	608	14.8	30.8			0	$1,026
37	34	471	12.7	24.6			43262080	$795
32	30	303	11.1	22.0			41425492	$511
27	25	134	9.2	17.5			25399911	$226
22	21	51	7.7	16.0			11703398	$86
17	16	23	6.0	12.7			6369159	$39
12	11	9	4.4	10.1			2897763	$15
7	6	1	2.9	7.8			365009	$2
2	2	1	1.7	7.9			398275	$2
-3	-3	0	0.2	6.3				
Subtotal Energy Cost					$13,056	$401	$461	$14,768

Cooling Season Energy Cost	$21,902	Total Annual $30,486
Heating Season Energy Cost	$6,784	Energy Cost
Demand Charges (Cooling Season Only)	$1,799	Estimate for Total Recovery with Traditional DBC Approach

(E) Humidity Grains Entering DH Wheel	(F) Temperature Entering DH Wheel	(G) Enthalpy Entering DH Wheel	(H) Temperature Leaving the Dehumidification Wheel	(I) Humidity Leaving the Dehumidification Wheel	(J) Regeneration Temperature Required
66.7	81.5	30.0	100.4	50	170
70.0	80.3	30.2	101.3	50	170
72.0	79.1	30.2	101.4	50	170
71.2	77.9	29.8	98.8	50	160
70.6	76.7	29.4	96.9	50	155
71.3	75.5	29.3	96.2	50	155
70.8	74.3	28.9	94.3	50	150
65.0	73.1	27.7	89.3	50	145
63.9	71.9	27.2	87.6	50	145
33.2	64.8	20.7			
31.7	63.6	20.2			
31.1	62.4	19.8			
30.0	61.2	19.4			
29.7	60.0	19.0			
28.9	58.8	18.6			
28.3	57.6	18.2			
27.7	56.4	17.8			
27.7	55.2	17.5			

Note1: assumes parasitic loss for all components in the system as well as an average .3KW/ton for operating the chilled water pumps and operating the cooling tower

Annual Operating Cost Estimate for "Dual wheel Total Recovery/Conventional Cooling/Sensible Recovery Reheat Approach"

APPROACH: Cooling/Dual
    Wheel Total Rec.
    w/ Sensible Rec. Reheat

LOCATION: Atlanta, Ga.
   New ASHRAE WB Bin Data

Given Information

Supply/Exhaust CFM	20000		
Cooling coil leaving temperature/enthalpy	51	20.5 Btu/lb	
Summer desired supply temperature/enthalpy	65	23.4 Btu/lb	
Summer return air temperature/enthalpy/grains	75	28 Btu/lb	65 Grains
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	
Winter return air temperature/enthalpy	72	22.5 Btu/lb	
Sensible recovery effectiveness	0.78	0.7 Pressure Loss/side	
Total recovery effectiveness	0.76	0.75 Pressure Loss/side	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output	
Electrical Demand Charges ($/KW)	$8.00		
Cost of heating fuel	$4.80	$/million BTU of heating fuel	
Boiler efficiency	78%		
% time of operation	100	Average KW/ton	1

Dual Wheel Total Recovery/Conventional Cooling/Sensible Recovery Reheat

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with Cooling Coil	(B) BTU Required to Reheat (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
87	78.5	26	41.2	129.5	17185036	0		$55
86	76.5	111	39.3	118.6	68683015	0		$228
83	74.5	405	37.4	111.6	234468145	0		$807
78	72.5	634	35.7	108.4	343180006	0		$1,228
75	70.5	776	34.0	102.3	391804935	0		$1,460
74	68.5	573	32.4	93.5	269155113	0		$1,048
72	66.5	363	30.8	86.7	158363583	0		$646
69	64.5	374	29.3	82.0	151240408	0		$647
67	62.5	378	27.9	76.0	141251503	0		$637
66	60.5	311	26.6	68.8	107015959	0		$510
64	58.5	327	25.2	63.5	103290032	0		$523
62	56.5	405	24.0	58.6	116950659	0		$631
61	54.5	376	22.8	52.4	98742460	0		$571
59	52.5	347	21.6	48.0			0	$390
57	50.5	376	20.5	44.0			0	$423
55	48.5	326	19.4	40.2			0	$367
52	46.5	344	18.4	38.2			0	$387
50	44.5	346	17.4	34.9			0	$389
48	42.5	278	16.4	31.8			0	$313
45	40.5	230	15.5	30.4			0	$259
43	38.5	215	14.6	27.7			0	$242
42	36.5	181	13.7	23.6			0	$204
37	32.5	320	12.1	20.6			5782767	$360
32	29	303	11.4	24.1			11039410	$341
27	24	184	9.5	19.8			16250327	$207
22	20	138	7.9	16.9			18524163	$155
17	15	78	6.2	13.7			14106127	$88
12	12	35	4.7	12.1			7735641	$39
		8760						

Subtotal Energy Cost $11,007 $0 $353 $13,154

Cooling Season Energy Cost	$17,763	Total Annual	$27,060
Heating Season Energy Cost	$5,356	Energy Cost	
Demand Charges (Cooling Season Only)	$3,940	Estimate for Dual Wheel Total Recovery/Sensible Rec.Reheat	

(E) Supply Air Temp Entering Reheat Coil	(F) Outdoor Air Enthalpy Entering Cooling Coil	(G) Exhaust Air Temp Entering Total Energy Wheel	(H) Exhaust Air Enthalpy Entering Total Energy Wheel
69.6	27.84	56.4	23.6
69.6	27.38	56.4	23.6
69.6	26.93	56.4	23.6
69.6	26.51	56.4	23.6
69.6	26.11	56.4	23.6
69.6	25.72	56.4	23.6
69.6	25.35	56.4	23.6
69.6	24.99	56.4	23.6
69.6	24.65	56.4	23.6
69.6	24.32	56.4	23.6
69.6	24.01	56.4	23.6
69.6	23.71	56.4	23.6
69.6	23.42	56.4	23.6
	23.14		
	22.87		
	22.61		
	22.37		
	22.13		
	21.89		
	21.67		
	21.45		
	21.24		
	20.84		
	20.68		
	20.23		
	19.84		
	19.43		
	19.08		

Note 1: assumes parasitic loss of all components in the system as well as an average .3 KW/ton for operating the chilled water pumps and cooling tower.

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant Based Cooling Typical Configuration

LOCATION: Atlanta, Ga. New ASHRAE Weather Data

Given Information

Supply/Regeneration CFM	20000 cfm		10000 cfm
Summer desired supply grain level	50		
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb	
Summer return air temperature/enthalpy/wet bull	75	28 Btu/lb	65 Grains / 62.5 Wet Bulb
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	
Winter return air temperature/enthalpy	72	22.5 Btu/lb	
Dehumidification wheel pressure loss	1	1.25 Regen side	
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side	
Electrical energy cost ($/KWH)	$0.060	$5.00 $/million BTU of cooling output	
Electrical Demand Charges ($/KW)	$8.00		
Cost of heating fuel	$3.50	$/million BTU of heating fuel	
Boiler efficiency	78%		
% time of operation	100	Average KW/ton 1	
		0.65 Pressure Loss/side	

Desiccant Based Cooling Approach

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidification wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)	(E) Humidity Grains Entering DH Wheel
87	78.5	26	41.2	129.5	31601872	4030564		$39	129.5
86	76.5	111	39.3	118.6	141999325	13665588		$166	118.6
83	74.5	405	37.4	111.6	541984162	37921669		$607	111.6
78	72.5	634	35.7	108.4	852242865	40344067		$951	108.4
75	70.5	776	34.0	102.3	897187540	0		$1,164	102.3
74	68.5	573	32.4	93.5	526716794	0		$859	93.5
72	66.5	363	30.8	86.7	299191449	0		$544	86.7
69	64.5	374	29.3	82.0	307787355	0		$561	82.0
67	62.5	378	27.9	76.0	291893086	0		$567	76.0
66	60.5	311	26.6	68.8	208990950	0		$466	68.8
64	58.5	327	25.2	63.5	186106980	0		$490	63.5
62	56.5	405	24.0	58.6	227345219	0		$607	58.6
61	54.5	376	22.8	52.4	135058051	0		$564	52.4
59	52.5	347	21.6	48.0				$520	48.0
57	50.5	376	20.5	44.0			0	$564	44.0
55	48.5	326	19.4	40.2			0	$489	40.2
52	46.5	344	18.4	38.2			0	$516	38.2
50	44.5	346	17.4	34.9			0	$519	34.9
48	42.5	278	16.4	31.8			0	$417	31.8
45	40.5	230	15.5	30.4			9627331	$345	30.4
43	38.5	215	14.6	27.7			26211449	$322	27.7
42	36.5	181	13.7	23.6			41248490	$271	23.6
37	32.5	320	12.1	20.6			107867015	$480	20.6
32	29	303	11.4	24.1			87717589	$454	24.1
27	24	184	9.5	19.8			79235695	$276	19.8
22	20	138	7.9	16.9			74299970	$207	16.9
17	15	78	6.2	13.7			50728907	$117	13.7
12	12	35	4.7	12.1			25412436	$52	12.1
Subtotal Energy Cost					$17,458	$480	$1,758	$11,549	

Cooling Season Energy Cost	$23,396	
Heating Season Energy Cost	$7,578	Total Annual Energy Cost: $32,573
Demand Charges (Cooling Season Only)	$1,599	Traditional Desiccant Based Cooling Approach

Note 1: assumes paras

(F) Temperature Entering DH Wheel	(G) Enthalpy Entering DH Wheel	(H) Temperature Leaving the Dehumidification Wheel	(I) Humidity Leaving the Dehumidification Wheel	(J) Regeneration Temperature Required	(K) Temperature Leaving Regen Evaporative Cooler	(L) Temperature to Regeneration Coil	(M) Temperature Leaving Sensible Wheel To Cooling Coil	(N) Enthalpy Leaving Sensible Wheel
87.0	41.2	155.9	60	250	63.8	137.5	82	
86.0	39.3	148.5	55	250	63.8	131.5	81	
83.0	37.4	141.7	50	250	63.8	126.1	79	
78.0	35.7	134.7	50	245	63.8	120.5	78	
75.0	34.0	125.2	50	220	63.8	112.9	76	
74.0	32.4	115.2	50	190	63.8	104.9	74	
72.0	30.8	107.4	50	175	63.8	98.7	72	
69.0	29.3	101.3	50	170	63.8	93.8	71	
67.0	27.9	94.7	50	160	63.8	88.5	70	
66.0	26.6	87.5	50	145	63.8	82.8	69	
64.0	25.2	80.7	50	130	63.8	77.3	67	
62.0	24.0	75.3	50	125	63.8	73.0	66	
61.0	22.8	67.5	50	100	63.8	66.7	70	24.9
59.0	21.6						69	24.1
57.0	20.5						69	23.4
55.0	19.4						69	22.7
52.0	18.4						68	22.3
50.0	17.4						68	21.7
48.0	16.4						67	21.1
45.0	15.5						67	20.7
43.0	14.6						66	20.2
42.0	13.7						66	19.5
37.0	12.1						65	18.8
32.0	11.4						64	19.1
27.0	9.5						63	18.2
22.0	7.9						62	17.5
17.0	6.2						61	16.8
12.0	4.7						60	16.3

sitic loss for all components in the system as well as an average .3KW/ton for operating the chilled water pumps and operating the cooling tower

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant
Based Cooling
Typical Configuration

LOCATION: Houston, Tex.

Given Information				
Supply/Regeneration CFM	20000 cfm		10000 cfm	
Summer desired supply grain level	50			
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb		
Summer return air temperature/enthalpy/wet bulb	75	28 Btu/lb	65	62.5
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	Grains	Wet Bulb
Winter return air temperature/enthalpy	72	22.5 Btu/lb		
Dehumidification wheel pressure loss	1	1.25 Regen side		
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side		
Electrical energy cost ($/KWH)	$0.035	$2.92 $/million BTU of cooling output		
Electrical Demand Charges ($/KW)	$8.00			
Cost of heating fuel	$3.40	$/million BTU of heating fuel		
Boiler efficiency	78%			
% time of operation	100	Average KW/ton	1	
		0.65 Pressure Loss/side		

Desiccant Based Cooling Approach

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidificaiton wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	77	1	39.6	95.6	794760	95370		$1
97	78	41	40.6	109.9	40996759	5239370		$36
92	78	304	40.7	118.1	331917159	41293421		$266
87	76	625	38.8	114.0	658799005	62944248		$547
82	74	892	37.0	110.3	947309090	0		$780
77	72	1530	35.3	107.2	1559061140	0		$1,338
72	68	1279	32.0	94.2	1252102344	0		$1,119
67	63	964	28.3	78.3	919641799	0		$843
62	58	782	24.9	64.7	679182958	0		$684
57	53	666	21.9	53.1	516096122	0		$584
52	48	580	19.2	43.2				$507
47	43	460	16.7	34.9				$402
42	39	321	14.8	30.8			13664510	$281
37	34	186	12.7	24.6			33655985	$163
32	30	87	11.1	22.0			22286693	$76
27	25	25	9.2	17.5			9115546	$22
22	20	7	7.4	13.7			3218120	$6
17	15	2	5.7	10.6			1086986	$2
12	11	9	4.4	10.1			5231610	$8
7	6	1	2.9	7.8			648916	$1
2	2	1	1.7	7.9			675797	$1
-3	-3	0	0.2	6.3				

	Subtotal Energy Cost	$27,853	$320	$305	$7,667

Cooling Season Energy Cost	$33,787	Total Annual	$37,744
Heating Season Energy Cost	$2,358	Energy Cost	
Demand Charges (Cooling Season Only)	$1,599	Traditional Desiccant Based Cooling Approach	

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant Based Cooling Typical Configuration

LOCATION: Minneapolis, Mn.

Given Information				
Supply/Regeneration CFM	20000 cfm		10000 cfm	
Summer desired supply grain level	50			
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb		
Summer return air temperature/enthalpy/wet bulb	75	28 Btu/lb	65	62.5
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	Grains	Wet Bulb
Winter return air temperature/enthalpy	72	22.5 Btu/lb		
Dehumidification wheel pressure loss	1	1.25 Regen side		
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side		
Electrical energy cost ($/KWH)	$0.034	$2.83 $/million BTU of cooling output		
Electrical Demand Charges ($/KW)	$8.47			
Cost of heating fuel	$4.10	$/million BTU of heating fuel		
Boiler efficiency	78%			
% time of operation	100	Average KW/ton	1	
		0.65 Pressure Loss/side		

Desiccant Based Cooling Approach

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidificaiton wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	80	0	42.6	114.6	0	0		$0
97	78	8	38.7	97.6	8121255	745372		$7
92	74	50	36.9	94.0	48731966	2971517		$42
87	71	136	34.3	85.5	123944741	1369629		$116
82	68	285	31.9	78.0	240182296	0		$242
77	66	442	30.4	76.2	371822314	0		$376
72	63	813	28.2	70.2	528090930	0		$521
67	60	702	26.2	65.0	616008933	0		$597
62	56	704	23.7	56.6	606805830	0		$598
57	52	614	21.3	49.4	0	0		$522
52	47	552	18.7	39.9	0			$469
47	43	478	16.7	34.9	0			$406
42	38	487	14.4	27.9			0	$414
37	34	552	12.7	24.6			0	$469
32	30	653	11.1	22.0			167278284	$555
27	25	591	9.2	17.5			215491497	$502
22	21	475	7.7	16.0			199150893	$404
17	16	379	6.0	12.7			191490600	$322
12	11	313	4.4	10.1			181943768	$266
7	6	242	2.9	7.8			157037655	$206
2	1	190	1.4	6.0			134796520	$161
-3	-8	260	-1.1	-2.6			231776920	$221
Subtotal Energy Cost					**$13,371**	**$14**	**$6,064**	**$7,415**

Cooling Season Energy Cost	$15,883	Total Annual Energy Cost Traditional Desiccant Based Cooling Approach $28,336
Heating Season Energy Cost	$10,759	
Demand Charges (Cooling Season Only)	$1,693	

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant Based Cooling Typical Configuration

LOCATION: New York, NY.

Given Information

Supply/Regeneration CFM	20000 cfm	10000 cfm	
Summer desired supply grain level	50		
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb	
Summer return air temperature/enthalpy/wet bull	75	28 Btu/lb	65 62.5
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	Grains Wet Bulb
Winter return air temperature/enthalpy	72	22.5 Btu/lb	
Dehumidification wheel pressure loss	1	1.25 Regen side	
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side	
Electrical energy cost ($/KWH)	$0.050	$4.17 $/million BTU of cooling output	
Electrical Demand Charges ($/KW)	$23.50		
Cost of heating fuel	$6.26	$/million BTU of heating fuel	
Boiler efficiency	78%		
% time of operation	100	Average KW/ton 1	
		0.65 Pressure Loss/side	

Desiccant Based Cooling Approach

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidification wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	75	0	37.7	83.6	0	0		$0
97	78	0	38.7	97.6	0	0		$0
92	74	19	36.9	94.0	16679555	1022472		$24
87	74	77	36.9	102.2	82143236	5186132		$96
82	71	196	34.4	93.6	177746585	2413707		$245
77	69	378	32.7	91.2	343467098	0		$472
72	67	608	31.2	89.2	493352088	0		$760
67	65	812	29.7	87.6	656827468	0		$1,015
62	60	809	26.2	73.1	823946901	0		$1,011
57	58	759	23.7	64.6	687884286	0		$948
52	51	705	20.8	53.7	568755571			$881
47	47	689	18.7	47.8				$861
42	42	687	16.2	39.7			0	$858
37	38	766	14.4	35.8			0	$957
32	33	838	12.3	29.8			96925887	$1,047
27	29	628	10.7	27.3			118428936	$785
22	24	365	8.8	22.9			107540179	$456
17	19	222	7.0	19.3			86078198	$277
12	15	117	5.7	18.4			50527695	$146
7	10	51	4.1	15.8			25831091	$64
2	6	17	2.8	15.6			9147060	$21
-3	1	2	1.4	13.7			1198130	$2

Subtotal Energy Cost	$20,820	$36	$3,103	$10,928

Cooling Season Energy Cost	$24,478	Total Annual	$39,582
Heating Season Energy Cost	$10,406	Energy Cost	
Demand Charges (Cooling Season Only)	$4,698	Traditional Desiccant Based Cooling Approach	

Annual Operating Cost Estimate for "Traditional Desiccant Based Cooling Approach"

APPROACH: Desiccant
                   Based Cooling
        Typical Configuration

LOCATION: Orlando, Fla.

Given Information				
Supply/Regeneration CFM	20000 cfm		10000 cfm	
Summer desired supply grain level	50			
Summer desired supply temperature/enthalpy	75	25.8 Btu/lb		
Summer return air temperature/enthalpy/wet bulb	75	28 Btu/lb	65	62.5
Winter desired supply air temperature/enthalpy	65	21 Btu/lb	Grains	Wet Bulb
Winter return air temperature/enthalpy	72	22.5 Btu/lb		
Dehumidification wheel pressure loss	1	1.25 Regen side		
Sensible recovery effectiveness	0.8	0.75 Pressure Loss/side		
Electrical energy cost ($/KWH)	$0.035	$2.92 $/million BTU of cooling output		
Electrical Demand Charges ($/KW)	$8.10			
Cost of heating fuel	$4.20	$/million BTU of heating fuel		
Boiler efficiency	78%			
% time of operation	100	Average KW/ton	1	
		0.65 Pressure Loss/side		

Desiccant Based Cooling Approach

WEATHER DATA (TWELVE YEAR AVERAGE)

OUTDOOR DRY BULB	OUTDOOR WET BULB	ANNUAL BIN HOURS	ENTHALPY BTU/LB	MOISTURE GR./LB.	(A) BTU Required to dehumidify with dehumidification wheel	(B) BTU Required to Post Cool (cooling mode)	(C) BTU Required to Heat & Humidify (heating mode)	(D) Fan Horsepower cost for the system assuming 1.5" of ESP and cooling pump and tower (note 1)
102	78	4	38.6	89.5	3691419	341290		$3
97	76	196	38.7	97.6	198970754	18261623		$171
92	74	662	36.9	94.0	645211230	39342885		$579
87	72	1004	35.2	90.9	886726391	24249805		$878
82	71	1717	34.4	93.6	1640171688	25965906		$1,502
77	68	1717	31.9	86.1	1605631032	0		$1,502
72	63	1103	28.2	70.2	1003586632	0		$965
67	58	844	24.9	56.6	777253878	0		$738
62	53	602	21.9	45.1				$527
57	48	408	19.2	35.3			988615385	$357
52	43	265	16.7	27.0			14354148	$232
47	38	148	14.4	20.0			30378428	$128
42	33	59	12.3	14.1			20175790	$52
37	29	22	10.7	11.6			9111353	$19
32	25	3	9.2	9.7			1430488	$3

Subtotal Energy Cost	$36,407	$315	$4,469	$7,656

Cooling Season Energy Cost	$43,588		Total Annual	$50,466
Heating Season Energy Cost	$5,259		Energy Cost	
Demand Charges (Cooling Season Only)	$1,619		Traditional Desiccant	
			Based Cooling Approach	

APPENDIX C

MODELED ENERGY COSTS FOR SELECTED CITIES AT DIFFERENT LEVELS OF DEHUMIDIFICATION

Estimated Energy Consumption by Preconditioning Approach for Various Cities

Conditions: Case 1
- Provide 7,500 cfm of preconditioned outdoor air at 50 grains, (a dewpoint of 48 degrees) at a temperature between 65 and 75 degrees
- Energy costs at $.06/kwh utilization, $8/KW demand and gas at $.48/therm
- Hours of operation used as shown, either continuous or 12 h/day, 5 days/week.

Preconditioning System Approach	Atlanta		Houston		Los Angeles		Minneapolis		New York		Orlando	
	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days
Conventional Cooling w/Reheat	$21,818	$11,269	$29,090	$15,127	$13,227	$6,878	$21,640	$11,253	$20,115	$14,081	$28,810	$14,981
Conventional Cooling w/Run-Around Recovery	$15,790	$8,113	$20,370	$8,759	$10,580	$4,549	$18,730	$8,054	$17,097	$8,719	$19,390	$8,338
Total Energy Wheel w/Cooling & Reheat	$16,142	$8,058	$20,390	$9,991	$9,390	$4,601	$11,980	$5,870	$12,070	$8,208	$21,210	$10,393
Dual-Wheel Total Energy Recovery	$10,098	$5,323	$12,280	$6,508	$7,150	$3,790	$8,670	$4,595	$8,290	$5,803	$13,290	$7,044
Dual-Wheel Desiccant-Based w/Post Cooling	$16,125	$5,810	$24,920	$6,095	$12,350	$3,239	$15,080	$4,825	$14,750	$8,343	$22,025	$7,813
Desiccant Dehumidification--Total Recovery Hybrid	$14,250	$5,280	$22,010	$4,425	$10,910	$3,786	$13,330	$3,574	$10,820	$5,798	$20,340	$5,660
Dual-Wheel DBC--Total Recovery Hybrid	$10,780	$4,562	$13,658	$4,205	$7,835	$3,726	$9,690	$3,520	$8,190	$6,859	$12,560	$5,238

Estimated Energy Consumption by Preconditioning Approach for Various Cities

Conditions: Case 2
- Provide 7,500 cfm of preconditioned outdoor air at 65 grains, (a 55 degree dewpoint) at a temperature between 65 and 75 degrees
- Energy costs at $.06/kwh utilization, $8/KW demand and gas at $.48/therm
- Hours of operation used as shown, either continuous or 12 h/day, 5 days/week

Preconditioning System Approach	Atlanta		Houston		Los Angeles		Minneapolis		New York		Orlando	
	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days
Conventional Cooling w/Reheat	$16,938	$11,269	$22,374	$11,634	$9,233	$4,801	$18,155	$9,441	$16,385	$11,470	$21,290	$11,071
Conventional Cooling w/Run-around Recovery	$13,527	$8,113	$16,109	$6,927	$7,202	$3,097	$16,632	$7,152	$14,747	$7,521	$15,896	$6,835
Total Energy Wheel w/Cooling & Reheat	$11,262	$8,058	$15,214	$7,455	$6,135	$3,006	$8,976	$4,398	$9,012	$6,128	$15,062	$7,380
Dual-wheel Total Energy Recovery	$7,336	$5,323	$8,430	$4,468	$4,905	$2,600	$6,812	$3,610	$6,155	$4,309	$9,161	$4,855
Dual-wheel Desiccant Based (DBC) w/Post Cooling	$14,880	$5,810	$22,988	$6,095	$11,398	$3,239	$12,642	$4,825	$13,619	$8,343	$20,324	$7,813
Desiccant Dehumidification - Total Recovery Hybrid	$10,983	$5,280	$15,595	$4,425	$6,435	$3,786	$8,250	$3,574	$7,862	$5,798	$14,125	$5,660
Dual-wheel DBC - Total Recovery Hybrid	$7,636	$4,562	$8,905	$4,205	$5,469	$3,726	$6,610	$3,520	$6,280	$6,859	$8,100	$5,238

Estimated Energy Consumption by Preconditioning Approach for Various Cities

Conditions: Case 3
- Provide 7,500 cfm of preconditioned outdoor air at 45 grains, (a 45 degree dewpoint), at a temperature between 65 and 75 degrees
- Energy costs at $.06/kwh utilization, $8/KW demand and gas at $.48/therm
- Hours of operation used as shown, either continuous or 12 h/day, 5 days/week

Preconditioning System Approach	Atlanta		Houston		Los Angeles		Minneapolis		New York		Orlando	
	24 h 7 days	12 h 5 days	24 h 7 days	24 h 7 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days
Conventional Cooling w/Reheat	$26,946	$11,269	$35,980	$18,710	$17,480	$9,090	$25,173	$13,090	$23,670	$16,569	$36,659	$19,063
Conventional Cooling w/Run-around Recovery	$19,142	$8,113	$24,826	$10,675	$12,236	$5,261	$20,411	$8,777	$18,936	$9,657	$23,079	$9,924
Total Energy Wheel w/Cooling & Reheat	$21,272	$8,058	$29,144	$14,281	$11,190	$5,483	$17,269	$8,462	$16,279	$11,070	$30,129	$14,763
Dual-wheel Total Energy Recovery	$12,686	$5,323	$18,820	$9,975	$6,915	$3,665	$10,290	$5,454	$8,995	$6,297	$17,205	$9,119
Dual-wheel Desiccant Based (DBC) w/Post Cooling	$18,750	$5,810	$28,967	$6,095	$14,363	$3,239	$17,545	$4,825	$17,160	$8,343	$25,610	$7,813
Desiccant Dehumidification - Total Recovery Hybrid	$15,579	$5,280	$24,062	$4,425	$11,790	$3,786	$14,580	$3,574	$11,830	$5,798	$22,625	$5,660
Dual-wheel DBC - Total Recovery Hybrid	$11,250	$4,562	$14,846	$4,205	$8,514	$3,726	$10,529	$3,520	$8,655	$6,859	$13,650	$5,238

Estimated Energy Consumption by Preconditioning Approach for Various Cities

Conditions: Case 4
- Provide 7,500 cfm of preconditioned outdoor air at 50 grains, at a temperature between 65 and 75 degrees
- Energy costs used are 1994 actual rates for the cities listed, electric rates based on acommercial customer with a 500+ KW demand
- See City Selection section and associated energy cost data

Preconditioning System Approach	Atlanta		Houston		Los Angeles		Minneapolis		New York		Orlando	
	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days	24 h 7 days	12 h 5 days
Conventional Cooling w/Reheat	$21,818	$11,269	$18,450	$9,594	$9,507	$4,944	$16,920	$8,798	$26,963	$18,874	$19,830	$10,312
Conventional Cooling w/Run-around Recovery	$15,790	$8,113	$13,844	$5,953	$7,555	$3,249	$15,291	$6,575	$20,976	$10,698	$11,479	$4,936
Total Energy Wheel w/Cooling & Reheat	$16,142	$8,058	$14,673	$7,190	$12,678	$6,212	$9,452	$4,631	$15,651	$10,643	$15,605	$7,646
Dual-wheel Total Energy Recovery	$10,098	$5,323	$7,762	$4,114	$5,717	$3,030	$5,942	$3,149	$10,100	$7,070	$8,328	$4,414
Dual-wheel Desiccant Based (DBC) w/Post Cooling	$13,298	$5,810	$14,154	$6,095	$11,774	$3,239	$10,626	$4,825	$14,843	$8,343	$18,925	$7,813
Desiccant Dehumidification - Total Recovery Hybrid	$12,792	$5,280	$10,365	$4,425	$8,571	$3,786	$7,983	$3,574	$13,327	$5,798	$13,777	$5,660
Dual-wheel DBC - Total Recovery Hybrid	$9,474	$4,562	$9,091	$4,205	$7,517	$3,726	$7,001	$3,520	$10,719	$6,859	$11,772	$5,238

APPENDIX D

SAMPLE CUSTOMER SURVEY SHEET

Retail Stores (Example)

<u>Sample application Evaluation Questionnaire:</u>

The purpose of this questionnaire is to help identify the market need and eventual acceptance of outdoor air, desiccant based, preconditioning approaches. Each application will be assessed using a questionnaire similar to this one, and these questionnaires will be answered internally, by field sales of both Trane and Semco, and by qualified and cooperative consulting engineers and owners.

Sample Questions:

1) How important is humidity control in this application?

      a. Not important   b. Somewhat important   c. Very important

2) In your opinion, it is important to maintain the space relative humidity below the following level during the cooling season for this application?

a. 50%   b. 60% c. 70%   d. No maximum required

3) In your opinion, it is important to maintain the space relative humidity at at least the following level during the heating season for this application?

a. 50%   b. 30% c. 20%   d. No minimum required

4) In your opinion, is it important to maintain these levels during evenings and weekends, as well as during the day?

a. Yes   b. No   c. Not applicable

5) What is the percentage of total system airflow that is typically outdoor air in this application?

      a. 0-20%   b. 20-30%   c. 30-50%   d. More than 50%

6) In your opinion, when designing for this application for either new construction or renovations, the ASHRAE 62-89 guidelines will be followed, (including the outdoor air

cfm/person recommendations and continuous supply of outdoor air) the following percent of the time?

    a. More than 75%   b. 50-75%   c. 25-50%   d. less than 25%

7) What type of air conditioning equipment is most often used in this application?

    a. Mostly DX b. Mostly chilled water   c. 50/50

8) Using a 1, 2 or 3, rank your impression of the importance of the following criteria as it relates to choosing the HVAC system for this application

    ( ) First cost
    ( ) Energy efficiency
    ( ) Providing environmental control (ie:temperature/humidity control, good IAQ)

9) Based on your experience, what percentage of projects in this application segment offer an exhaust air stream that can be ducted to the preconditioning system or a return air stream from which and exhaust air stream can be accessed for the same purpose?

    a. More than 75%   b. 50-75%   c. 25-50%   d. less than 25%

10) Is low cost steam, hot water or other forms of waste heat typically available for desiccant regeneration during the cooling season for this application?

    a. Yes   b. No

11) Do significant benefits exist, in your opinion, from maintaining humidity control in this application that would justify a higher first cost for the HVAC system (ie: comfort, reduced damage to furniture and wall coverings, improved research, dry cooling coils, etc.). If so list the benefit.

    a. Yes   b. No

_____

_____

_____

12) Are you aware of the new ASHRAE weather data evaluation that shows a significant increase in the outdoor air peak humidity design conditions?

    a. Yes   b. No

13) Is there a significant benefit to this application to have pollutants that may exist in the outdoor air removed from the outdoor air prior to being introduced into the facility?

     a. Yes   b. No

14) If you have had a previous experience with a desiccant wheel system (dehumidification or recovery) how would you rate it? (Only respond to personal experience please).

     a. Positive   b. Negative   c. Not applicable

APPENDIX E
FINAL MARKET SEGMENT ANALYSIS SHEETS

Final Market Segment Analysis: Office Buildings

Benefits provided by preconditioning of outdoor air loads

The increased outdoor air loads associated with ASHRAE 62-89 can increase project first cost and operating cost significantly. Preconditioning can cut demand charges significantly, reduce energy consumption, provide acceptable humidity control, and help conventional heat pumps and DX units perform with constant outdoor air loads. Removing outdoor air pollutants and providing low duct RH may be significant benefits in the future.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$127,755,000 per year[*]

Total outdoor air CFM for segment

56,000,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	65	55,575,000
Conventional w/ Heat Pipe	3	7	10,395,000
Conventional with Total Recovery	2	25	50,625,000
Total Recovery Dual Wheel	6	0	
DBC approach	5	1	3,600,000
Hybrid Total Recovery Desiccant Dehumidification approach	4	2	7,560,000
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

Of the 28,000 buildings constructed each year, only 200 are 4 stories or greater. Only these 200 would likely prove opportunities for DBC preconditioning. Gas is seldom available on rooftops of high-rise facilities. Unless dry ductwork and coils are mandated, conventional cooling/recovery will likely be used for this application. The exception will be buildings located were demand charges are high, where an exhaust air path is unobtainable, where the existing cooling capacity is fully utilized and where outdoor air flow needs to be increased (i.e. accommodate ASHRAE 62-1989). Offices designed with ice storage and super cold air distribution are viable applications for hybrid recovery/desiccant dehumidification. When ASHRAE-62-1989 is integrated into an office environment, even for small office buildings, non-regenerated desiccant total energy recovery wheels appears by far the method of choice

Final Market Segment Analysis: Hospital Operating Rooms

Benefits provided by preconditioning of outdoor air loads

Operating rooms need to have humidity controlled for antiseptic reasons. With the advent of AIDS, more protective gear is worn by the medical staff. This, in conjunction with heat generated by medical machinery, require conditions of 65-68 degrees for comfort. Relative humidity must still be controlled at approximately 50% RH which is not easily accomplished with conventional refrigeration

Potential market for this application using some type of preconditioning (Estimates for 1996)

$10,344,000 per year*

* (assumes 30% of market needs dew-points below 50 degrees)

Total outdoor air CFM for segment

4,800,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	50	$2,280,000
Conventional w/ Heat Pipe	5	0	
Conventional with Total Recovery	2	20	$2,160,000
Total Recovery Dual Wheel	7	0	
DBC approach	4	15	$2,880,000
Hybrid Total Recovery Desiccant Dehumidification approach	3	15	$3,024,000
Hybrid Total Recovery DBC approach	6	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

The driver causing the hospital to maintain humidity conditions is the surgeon, therefore significant. The surgeons insurance, liability and reputation may ride on controlling space conditions. Awareness of this appears is on the rise. Hospital facilities also typically have low cost steam available during the cooling season for desiccant regeneration. This is not a cost sensitive market and would likely pay a premium for both environmental control and energy efficiency. The requirements of the desiccant equipment would ideally accommodate high efficiency filters, heating/cooling coils, dual wall institutional construction, high pressure blowers and controls. Key benefits offered by desiccants include desired humidity control, dry ductwork, dry final filters (a big problem for hospital ORs). Hours of operation are continuous so energy savings analyses are maximized.

Final Market Segment Analysis: School Classrooms

Benefits provided by preconditioning of outdoor air loads

The increased outdoor air loads associated with ASHRAE 62-89 can increase project first cost and operating cost significantly. Preconditioning can cut demand charges and energy consumption, assure acceptable humidity control, and help conventional heat pumps and DX units perform with constant outdoor air loads. Neutral temperature, dry air is beneficial in classrooms due to the changing load associated with people and desire to use packaged equipment controlled via temperature in each classroom.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$222,979,500 per year*

* (assumes all buildings comply with ASHRAE 62-89)

Total outdoor air CFM for segment

124,500,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	2	50	$59,137,500
Conventional w/ Heat Pipe	3	10	$20,542,500
Conventional with Total Recovery	1	20	$56,025,000
Total Recovery Dual Wheel	4	15	$61,627,500
DBC approach	6	2	$9,960,000
Hybrid Total Recovery Desiccant Dehumidification approach	5	3	$15,687,000
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

Due to the high exchange rate of outdoor air in schools, there is almost always access to an exhaust air path. Since packaged cooling equipment is most often used, humidity control is critical in humid environments. The dew points required by a school project are easily obtained with conventional cooling, controlling the humidity is the problem. Regenerated desiccant systems provide dry, warm air to the school (ie: 90-95 degrees at 50 grains on a typical design day). The dual wheel energy recovery approach provides the same dehumidification yet at the desired 68 degrees, allowing the conventional HVAC system to be down-sized It also provides heating mode recovery and humidification which is important. In non-humid climates conventional single wheel desiccant based total recovery is most appropriate. Maintenance in schools is limited, evaporative coolers and gas burners are viewed as too troublesome. If schools embrace the benefits of dry coils, regenerated desiccant wheel systems will be feasible.

Final Market Segment Analysis: Nursing Homes/Hospitals

Benefits provided by preconditioning of outdoor air loads

Nursing homes and hospital bed areas need large quantities of outdoor air for health and odor maintenance. Preconditioning allows for better humidity control with the use of conventional packaged equipment. Condensate can be reduced or eliminated on the coils and down the duct. Final filters can be kept dry if all of the latent load is handled with the outdoor air. As with hospital operating rooms, the likelihood of reinfection is greatly reduced if space relative humidity is controlled.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$71,802,500 per year[*]

* (assumes all buildings comply with ASHRAE 62-89)

Total outdoor air CFM for segment   13,750,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	45	16,458,750
Conventional w/ Heat Pipe	3	15	9,528,750
Conventional with Total Recovery	2	20	17,325,000
Total Recovery Dual Wheel	4	10	12,705,000
DBC approach	5	5	7,700,000
Hybrid Total Recovery Desiccant Dehumidification approach	6	5	8,085,000
Hybrid Total Recovery DBC approach	7	0	

- Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

General hospital and nursing home areas do not have the same stringent humidity requirements as operating rooms. Odors associated with humidity problems, comfort, and dry filters and ductwork are the main drivers for regenerated desiccant systems for this application. Waste steam is often available.

Where an exhaust air path is not available and the climate is humid, a good opportunity exists for some type of regenerated desiccant approach. This market is not as cost sensitive as most and maintenance is usually not an impediment. Nursing homes are promising for preconditioning since the occupants like dry, warm air and IAQ is an important factor in deciding on one nursing home over another.

Final Market Segment Analysis: Research Laboratories

Benefits provided by preconditioning of outdoor air loads

Research laboratories often require a significant amount of outdoor air to replace that exhausted through hoods. Often humidity control is very important and the number of air changes used can meet the sensible load with moderate temperature air. As a result, over cooling and reheat is often used. Preconditioning can reduce energy consumption significantly and improve humidity control.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$16,142,100 per year*

* (assumes 50% chemical/50% biological and other labs)

Total outdoor air CFM for segment   7,800,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	40	$2,964,000
Conventional w/ Heat Pipe	4	15	$1,930,500
Conventional with Total Recovery	2	20	$3,510,000
Total Recovery Dual Wheel	6	2	$514,800
DBC approach	3	20	$6,240,000
Hybrid Total Recovery Desiccant Dehumidification approach	5	3	$982,800
Hybrid Total Recovery DBC approach	7	0	

- Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

Research laboratories that are located where demand charges are high and there are a large number of hours per year where dehumidification is needed (humid climate), then a regenerated desiccant approach becomes justifiable. Low cost steam is often available for regeneration which is a plus.
The co-sorption of pollutants is a significant benefit to many laboratory facilities.
In this market, energy efficiency and system performance usually is more important than first cost.
Some laboratories have a need for humidity conditions drier than obtainable from conventional cooling approaches. Non-regenerated desiccant total recovery is often avoided due to the contaminants in the exhaust airstream or the lack of a central exhaust location (ie: numerous exhaust fans)

Final Market Segment Analysis: Fast Food Restaurants

Benefits provided by preconditioning of outdoor air loads

The exhaust air required in the kitchen and the use of packaged equipment makes maintaining humidity within fast food restaurants very difficult in humid regions. Preconditioning outdoor air solves this problem. Increased outdoor air is also required if these facilities try to accommodate smokers. Wash down of the floors and cooking adds significant latent load to these spaces which cannot be effectively handled with packaged equipment.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$110,960,000 per year[*]

* (assumes all buildings comply with ASHRAE 62-89)

Total outdoor air CFM for segment

20,000,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	70	56,430,000
Conventional w/ Heat Pipe	2	20	28,010,000
Conventional with Total Recovery	4	5	9,550,000
Total Recovery Dual Wheel	5	0	
DBC approach	3	5	16,970,000
Hybrid Total Recovery Desiccant Dehumidification approach	6	0	
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

Since there is seldom a return air path available in a fast food restaurant, recovery is not a viable option except in larger restaurants designed to relieve cigarette smoke. In humid climates, a market for some type of DBC exists, but the product would have to be very low cost and require minimal routine maintenance. A successful product would likely be a module to augment a packaged HVAC product. The product would have to be regenerated with a gas burner or with energy released by a DX evaporator coil.

Final Market Segment Analysis: Retail Stores

Benefits provided by preconditioning of outdoor air loads

The increased outdoor air loads associated with ASHRAE 62-89 can increase project first cost and operating cost significantly. Preconditioning can cut demand charges significantly, assure acceptable humidity control in humid climates, and help conventional packaged units perform with constant outdoor air loads. Removing additional moisture to deal with humidity storage within the goods displayed in the facility may be a key benefit.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$58,266,000 per year[*]

* (assumes all buildings comply with ASHRAE 62-89)

Total outdoor air CFM for segment

31,200,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	40	11,856,000
Conventional w/ Heat Pipe	4	15	7,722,000
Conventional with Total Recovery	2	30	21,060,000
Total Recovery Dual Wheel	5	5	5,148,000
DBC approach	3	10	12,480,000
Hybrid Total Recovery Desiccant Dehumidification approach	6	0	
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

Superstores" offer the most potential for regenerated desiccant products in this market since they integrate both retail and food store functions under one roof. Regenerated desiccant systems provide a significant benefit to food stores that have freezer casings since lower dew-points result in fewer defrost cycles, lower operating costs and improved shopper comfort. Large retail stores (ie: Bloomingdale's) benefit from desiccant preconditioning since the large quantity of goods (ie. clothing) act as a moisture sink, making humidity control extremely difficult. Small shopping malls and retail stores without freezer areas would likely incorporate a non-regenerated desiccant approach combined with a unitary package AC unit. Note that for most retail stores designed to accommodate the ASHRAE 62-89 standard, the exhaust air is available from the return air path. Also note that supermarket applications process both outdoor and return air, so market number is larger than shown above for regenerated DBC in retail stores.

Final Market Segment Analysis: Hotels and Dormitories

Benefits provided by preconditioning of outdoor air loads

The exhaust air from bathrooms needs to be replaced making some of these facilities high outdoor air applications. Often the humidity associated with showers is very high for the conventional room air-conditioners to handle. Preconditioning outdoor air to handle these fluctuating loads provides benefits to comfort, energy efficiency and limits microbial and moisture damage resulting in a significant cost savings to this market segment.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$76,927,500 per year*

* (assumes all buildings comply with ASHRAE 62-89)

Total outdoor air CFM for segment

39,000,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	40	$14,820,000
Conventional w/ Heat Pipe	4	15	$9,652,500
Conventional with Total Recovery	2	20	$17,550,000
Total Recovery Dual Wheel	3	15	$19,305,000
DBC approach	5	10	$15,600,000
Hybrid Total Recovery Desiccant Dehumidification approach	6	0	
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

The market for regenerated desiccant preconditioning is limited to large facilities located in humid climates and where a return/exhaust air path is not available for non-regenerated desiccant based recovery. Humidity conditions can be controlled as desired with the total recovery dual wheel approach which also utilizes conventional cooling technology and provides heating mode recovery, important for almost all applications. Humidity control is important for this market, independent of the method used to obtain it, since the hotel industry spends a significant amount on refurbishment to rooms and corridors annually due to humidity and microbial problems.

Final Market Segment Analysis: Movie Theaters and Auditoriums

Benefits provided by preconditioning of outdoor air loads

The increased outdoor air loads associated with ASHRAE 62-89 can increase project first cost and operating cost significantly. Preconditioning can cut demand charges significantly, assure acceptable humidity control in humid climates, and help conventional packaged units perform with constant outdoor air loads.

Potential market for this application using some type of preconditioning (Estimates for 1996)

$30,082,500 per year*

Total outdoor air CFM for segment

21,000,000 CFM

Preconditioning Approach	Rank Likelihood of Utilization	% of Estimated Available Market	$ Estimated Market Potential
Conventional Cooling w/ Reheat	1	60	$11,970,000
Conventional w/ Heat Pipe	3	15	$5,197,500
Conventional with Total Recovery	2	20	$9,450,000
Total Recovery Dual Wheel	4	5	$3,465,000
DBC approach	5	0	
Hybrid Total Recovery Desiccant Dehumidification approach	6	0	
Hybrid Total Recovery DBC approach	7	0	

• Conclusions and Comments Regarding the Desiccant Based Preconditioning Market:

This is a very price sensitive market. Humidity control is not very important based on survey. No free energy available for regeneration and low hours of operation at peak occupancy, making energy efficiency a difficult sell. If designed to accommodate the ASHRAE 62-89 standard and located in an area with high demand charges, non-regenerated total energy recovery is a viable approach. There is always a return air path available to provide the exhaust required by total recovery.

This market shows little opportunity for regenerated desiccant approaches.

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