ORNL/SUB/94-SV044/2 REPORT

Active Desiccant-Based Preconditioning Market Analysis & Product Development

Phase 2

ORNL/SUB/94-SV044/2

ACTIVE DESICCANT-BASED PRECONDITIONING MARKET ANALYSIS AND PRODUCT DEVELOPMENT

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1. INTRODUCTION: MARKET ANALYSES DRIVE ACTIVE DESICCANT SYSTEMS DEVELOPMENT

The Phase 1 report (ORNL/Sub/94-SV044/1), completed earlier in this program, involved a comprehensive field survey and market analysis comparing various specialized outdoor air handling units. This initial investigation included conventional cooling and reheat, conventional cooling with sensible recovery, total energy recovery systems (passive desiccant technology) and various active desiccant systems. The report concluded that several markets do promise a significant sales opportunity for a *Climate Changer*–based active desiccant system offering. (Climate Changer is a registered trademark of Trane Company.) This initial market analysis defined the wants and needs of the end customers (design engineers and building owners), which, along with subsequent information included in this report, have been used to guide the determination of the most promising active desiccant system configurations.

This Phase 2 report begins with a summary of a more thorough investigation of those specific markets identified as most promising for active desiccant systems. Table 1 estimates the annual sales potential for a cost-effective product line of active desiccant systems, such as that built from Climate Changer modules.

The Product Development Strategy section describes the active desiccant system configurations chosen to best fit the needs of the marketplace while minimizing system options. Key design objectives based on market research are listed in this report for these active desiccant systems. Corresponding performance goals for the dehumidification wheel required to meet the overall system design objectives are also defined.

Table 1. Projected annual sales potential for active desiccant systems in targeted markets

Application category	DBC approach ($M)	Total recovery desiccant DH hybrid ($M)	Dehumidification stand-alone module ($M)
Nursing homes/assisted living (1)	4.75	4.04	
Hospitals and operating rooms		8.65	4.14
Research laboratories	6.24		0.98
Retail stores (1)	6.24	3.75	2.5
Hotels, dormitories (1)	6.24	6.24	3.12
School and university classrooms		25.65	
Subtotal by product	23.47	48.32	10.73

Note 1: Significant penetration into this market is contingent upon a low-cost, commercial product to augment packaged HVAC equipment.

Note 2: This table estimates sales potential on an annual basis and assumes compliance with ASHRAE 62-89.

Note 3: Sales potential dollars assume a Climate Changer offering priced as per Table 2.

The Performance Modeling section describes the strategy used by SEMCO to design the dehumidification wheels integrated into the prototype systems currently being tested as part of the U.S. Department of Energy's Advanced Desiccant Technology Program. Actual performance data from wheel testing was used to revise the system performance and energy analysis modeling initially presented in the Phase 1 report. This section also provides a revised payback analysis comparing the active desiccant systems selected with the other, more conventional approaches to specialized outdoor air handling units listed in the Phase 1 report.

This report concludes with a look at the importance of analyzing active desiccant systems on the basis of "latent air conditioning" and not the more traditional "total cooling capacity". Since handling latent load will always be the primary reason for using an active desiccant system, it is concluded that the technology must be positioned as a "latent air conditioner" in the marketplace. In past analyses, active desiccant systems have been too often compared with conventional cooling systems based on total cooling output. The resulting $/ton ratios that have resulted almost always made the active approach appear as if a very significant first-cost premium were required.

The tables in Sect. 5 show that when the most promising active desiccant systems are compared with a conventional cooling approach based on latent capacity, both the first cost and operating efficiency are found to be similar. The active desiccant systems have the advantage of lower operating cost because they use gas rather than electricity. This and other performance advantages make the active desiccant approach an attractive design alternative in the targeted markets.

2. SUMMARY OF MARKET ANALYSES

2.1 SUMMARY OF PROMISING MARKETS FOR ACTIVE DESICCANT SYSTEMS: NEED DRIVEN

The Phase 1 market evaluation for active desiccant systems concluded that several markets have a high level of need for the dehumidification capabilities offered by active desiccant systems. These markets include hospitals, nursing homes, and research facilities. The need is driven by a quantifiable benefit and/or importance associated with controlling space relative humidity at (or sometimes below) 50% during the cooling season, combined with a desire or need to provide high percentages of outdoor air on a continuous basis.

A secondary level of need is associated with reducing the cost of operation of these facilities. Given that these facility types are generally constructed to be owner-occupied and to have a long life cycle, and that they typically have waste or low-cost energy available during the cooling season (excess steam or low-cost gas), the humidity control desired will often be achieved most economically using some type of active desiccant approach. Owner-occupied facilities are often more interested in the long-term operating costs than first cost, one of the most common impediments to integrating active desiccant systems.

In an attempt to determine which of the many active desiccant system approaches best fit the specific needs of these need-driven markets, each was analyzed to determine which approach was most easily integrated into current design schemes. The specific drivers behind increased ventilation and humidity control were also determined for each market. This market-by-market analysis provided the basis for deciding which active desiccant approach(s) would best fit each market and what performance, cost, size and other parameters would be most important. The following information provides a summary of these analyses.

2.1.1 Hospitals

Hospitals are bound by charter to do no harm to their patients. Humidity control is very important in hospital facilities because it has been well established that there exists a strong relationship between space relative humidity and airborne infectious diseases. There are many sources for information on this topic; one of the best sources is a paper by Arundel et al. (1986). (This paper is the source of the Optimum Relative Humidity Range diagram referenced in the ASHRAE 62-89 IAQ Standard).

One body of evidence demonstrates that mid-range relative humidities (defined as between 40 and 60%) are more lethal to airborne bacteria than lower or higher relative humidities. Bacteria cause pneumonia, tuberculosis, anthrax and Legionnaire's disease, all of which are air-transmitted. Nonpathogenic bacterial infections such as streptococcus and staphylococcus have also been shown to be less viable in space conditions controlled at 50% relative humidity.

Research conducted to determine the space relative humidity required to minimize the risk of viral infections (such as influenza) suggests an optimum range between approximately 40 and 70% relative humidity.

It is also well established that the greater the ventilation rate, the lower the colony count (concentration) of the airborne pathogens. There is strong correlation between the colony count and the incidence of infection. In fact, it is suggested by Arundel et al. (1986) that where high air change rates exist, the humidity control impact is very small, and therefore, less important.

This basic research provides a quantitative, compelling reason to provide high quantities of outdoor air to hospitals while maintaining the space relative humidity between approximately 40 and 60%.

Two additional, important factors favor the use of active desiccant systems in hospital facilities. First, they allow operation of the facility (specifically the operating rooms) with dry cooling coils. This is a significant advantage since the most common design approach used today minimizes the outdoor air quantity by employing high-efficiency filters (handling increased recirculated air rates) downstream of the cooling coils. The justification is that bacteria and viruses are large enough to be filtered out, so dilution ventilation can be greatly reduced.

The serious flaw in this approach is that engineers almost never recognize the need for a sizable reheating coil after the cooling coil to avoid saturated air, which wets the final filter bank. Wet filters can exacerbate the very problem that they are designed to eliminate by serving as an amplification site for microbial growth. This is a common and serious problem in hospitals. Active desiccant systems can dehumidify the recirculated air sufficiently that the air leaving the cooling coils does not wet the final filters.

The second key advantage is that the active desiccant system can provide the optimum relative humidity (40 to 50%) in operating rooms even when the space temperature is controlled to as low as 62 to 65°F. This low space temperature has become very common because of the high density of electronics and lights now common in operating rooms, linked with the added protective gear worn by personnel because of the concern about exposure to AIDS and about other liability issues. It is also common during certain surgical procedures, such as heart surgery, to maintain very low space temperatures to improve the success rate of the procedure. Many surgeons fight with the hospital facilities group because they are given the choice of either comfortable conditions (cool space) or optimum space humidity. They need both. Current practice is to cool to 55°F and control the space temperature at 62°F, resulting in a space relative humidity of 75%. This market is searching for a cost-effective, compact solution.

As a result, the opportunity for active desiccant systems built from Climate Changer modules is significant for the hospital market. As is discussed later, the best solutions for this market will range from dehumidification-only units to dehumidification/total energy recovery hybrid systems.

2.1.2 Nursing Homes/Assisted Living

Critical care nursing homes (facilities for the very elderly or those in need of critical care) are designed similarly to a hospital facility's patient care area. In some ways, humidity control and indoor air quality are even more critical since all of the patients are elderly and therefore more susceptible to discomfort and infection. As a result, these types of facilities have the same ventilation and humidity control requirements as hospital patient care facilities.

Since the elderly are generally more comfortable at higher space temperatures (say 78°F), the level of dehumidification required to maintain the relative humidity at 50% (56°F dew point) is significantly less than for a hospital operating room operated at 65°F (45°F dew point). Therefore, it can be reached with conventional cooling equipment quite easily. Based on market research and project history, the ventilation requirements for these facilities are driven more by a desire to keep odor levels low (no one wants to place a loved one in a smelly environment); as a result, these facilities are typically designed to have a high percentage of outdoor air.

According to the building statistics and market observations, the number of critical care type facilities constructed each year is declining as a result of the significant rise in the construction of assisted living facilities. These assisted living facilities, marketed by huge organizations such as Marriott and Service

Master, are being built at a very high rate. Demographics suggest that this rate should remain strong for some time. Based on preliminary discussions with the designers of these facilities and on review of current design practices, they are more like condominiums than hospitals. As a result, they are typically built using packaged equipment (i.e., heat pumps in each room) with corridor delivery of outdoor air to compensate for kitchen and/or bathroom exhaust.

In both cases there are compelling reasons to control humidity and ventilate. The opportunity for active desiccant systems based upon a Climate Changer product will most likely be limited to the critical care type of facilities that are currently in decline. The assisted living facilities would be served very well (in humid climates) by a product designed to augment conventional packaged rooftop units to provide 100% outdoor air to the corridors. If an exhaust air path is available, such a product would likely be a dual wheel passive system. If an exhaust air path is not available, an active desiccant module linked with a packaged rooftop air conditioner, such as the Voyager product line manufactured by Trane, would best fit the needs of this market.

2.1.3 Research Facilities

Research facilities offer an exceptional opportunity for applying regenerated desiccant systems. They require high percentages of outdoor air to replace the air exhausted from fume hoods, animal areas, and so on. Most research facilities require humidity control—not necessarily low humidity, but steady humidity typically in the range of 50%. These facilities currently demand mechanical equipment of at least the complexity and quality offered by the Climate Changer. It is generally understood by those designing laboratory facilities that the cost of operation is very high, so these projects are not often first-cost driven (making the initial first-cost premium associated with a desiccant system less an issue than in other markets). Research facilities also typically overcool and reheat to maintain space humidity, so pay-back periods for active desiccant systems can be short. Payback cycles are significantly improved by the availability of low-cost steam for regeneration during the cooling season, which is often available.

An added benefit that some active desiccant systems can provide to a research facility is the ability to remove airborne pollutants from the outdoor air along with the adsorbed water vapor. Initial testing in this area looks very promising. More follow-up research is planned to verify this benefit, including measurements at actual installation sites.

Active desiccant systems based upon a Climate Changer format will meet the needs of this market segment in most cases. Research facilities often have numerous exhaust fans that cannot be easily connected together and deal with chemical contaminant concentrations that may not be suitable for recovery (in the opinion of the owner or designer). Therefore, the conventional DBC approach (discussed later) provides the most attractive active desiccant system solution because it does not require access to an exhaust air path.

2.2 SUMMARY OF PROMISING MARKETS FOR ACTIVE DESICCANT SYSTEMS: SALES-POTENTIAL DRIVEN

The SEMCO Phase 1 report also concluded that several markets have only a moderate level of need but still offer a high level of sales potential for a cost-effective active desiccant system. These markets include hotels/dormitories, retail stores, and school facilities. The sales potential is driven by the fact that many of these types of facilities are constructed each year compared with the number of hospitals, nursing homes, and research facilities. If only a small percentage of these facilities chose to utilize active desiccant approaches, the potential sales volume increase for an active desiccant system, such as that delivered in Climate Changer modules, would be significant.

In these market segments, benefits such as low operating cost (associated with low-cost fuel or waste heat) or dry cooling coils associated with the active desiccant approach must offset the familiarity factor, compact size, and lower first cost typically associated with more conventional approaches.

2.2.1 Hotel/Dormitory

This market segment is currently quite active because of renovations and acquisitions of various hotel facilities, as well as a healthy expansion in new construction. The mechanical design approach used for these facilities is similar to that described for assisted living facilities. Hotels are not driven by the same health-related issues associated with hospitals and nursing homes. They are affected by wide humidity fluctuations associated with the room layout (showers in each room) and the inability of the room air conditioning units to handle these high latent conditions.

As a result, a direct cost can be attributed to humidity problems (e.g., mold, mildew) in hotel and dormitory rooms. A study available from the Gas Research Institute (Collier 1988) and conducted by Cargocaire Engineering quantifies the benefits of improved humidity control within hotel facilities. The need and justification for improved indoor air quality and, specifically, improved humidity control is relatively clear for hotel facilities. This market has not evolved, however, since the successful test site demonstration completed by Cargocaire and GRI more than 5 years ago. The reason is likely a perception of high first cost and the lack of effective sales distribution to the large national accounts that purchase most of the HVAC equipment sold to the hotel construction market.

An active desiccant system offering by Trane would overcome the distribution obstacle because Trane is best positioned, because of its access to national accounts, to change the way hotels are designed and operated. A fully commercialized active desiccant system offering by Trane would be far more cost-effective than options currently available that would also accelerate market acceptance.

A recent survey of several Trane sales offices supports these conclusions. The salespeople in these offices were aware of the need for an outdoor air preconditioner for hotel/dormitory and assisted living facilities and recognized these as growing markets. They also acknowledged that Trane did not currently have an effective product to meet the needs of these markets. As a result, most were selling non-Trane products to meet the needs of such projects.

Several 5000-cfm, packaged overcooling/reheating units originally designed for swimming pool dehumidification have been sold by Trane for $39,000, or approximately $8/cfm. Based on this pricing level, an active desiccant system offering by Trane would be both energy efficient and cost competitive ($5/cfm is a good estimate of the selling price of a 5000-cfm Climate Changer–based DBC system, according to Table 2).

2.2.2 Retail Stores

The market potential for regenerated desiccant systems in retail stores is primarily limited to the new trend toward super-stores that include a grocery section. Retail stores with refrigerated food cases (i.e., supermarkets) are one of the few markets that have already accepted regenerated desiccant systems. The

Table 2. Selling price and dimensions of various preconditioning approaches at different airflow capacities

Preconditioning system approach	2,500 cfm		7,500 cfm		20,000 cfm	
	Sales price[a]	Dimensions[b] (H × W × L)	Sales price	Dimensions (H × W × L)	Sales price	Dimensions (H × W × L)
Conventional cooling w/reheat	$2,200 ($9,500)	27.5 × 44 × 154	$7,100 ($28,500)	44 × 74 × 177	$16,700 ($76,000)	74 × 120 × 251
Conventional cooling w/run around recovery	$4,000 ($7,750)	55 × 44 × 141	$12,150 ($23,250)	88 × 74 × 193	$31,000 ($62,000)	148 × 120 × 260
Total energy wheel w/cooling and 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)	55 × 44 × 188	$24,750 ($10,500)	88 × 74 × 211	$53,650 ($28,000)	148 × 120 × 285
Dual wheel desiccant based (DBC) w/post cooling	$14,700 ($2,000)	55 × 44 × 188	$30,200 ($6,000)	88 × 74 × 211	$64,800 ($16,000)	148 × 120 × 285
Desiccant dehumidification–total recovery hybrid	$15,750 ($3,000)	55 × 44 × 203	$31,350 ($8,500)	88 × 74 × 226	$67,850 ($22,500)	148 × 120 × 300
Dual wheel DBC—total recovery hybrid	$20,500 ($1,000)	55 × 44 × 235	$39,300 ($3,250)	88 × 74 × 258	$84,200 ($8,500)	148 × 120 × 332

[a]All sales prices are estimates of the market price using Climate Changer modules.

[b]All dimensions assume the use of Trane Climate Changer modules and SEMCO wheel modules.

[c]Numbers in parentheses denote cost of chiller and cooling tower capacity required to take outdoor air at 95°F/115 grains to 50 grains assuming a cost of $500/ton.

justification for the desiccant system is its ability to keep the humidity level low enough in areas around the frozen food cases that frosting on the product and within the casings is minimized.

Avoiding frost formation translates into reduced operating costs (from limiting expensive defrost cycles of the cases) and increased sales due to better presentation of the product and customer comfort (warmer aisles). If customers are comfortable, they spend more time in the stores and therefore purchase more products.

One obstacle to penetration into this market segment by Trane is that the Climate Changer is an indoor product and most of the current designs for super-stores use outdoor, compressorized equipment. As a result, this market segment may be better suited to a hybrid desiccant system based on a unitary rooftop air conditioner such as those Trane makes in Clarksville, rather than to a Climate Changer system.

In humid climates, retail stores that do not include food sections with refrigerated cases may still present a limited market opportunity for regenerated desiccant systems if the outdoor air volume equals that required for building pressurization (air leaking through doors). This results in a design where exhaust air is not available for use with desiccant-based energy recovery (passive wheels). A rooftop/active desiccant hybrid system could reduce the overall number of rooftop units installed on a typical retail store and better handle the part load conditions while reducing the cost of operation in humid climates where the gas cost/electrical cost ratio is low.

If the outdoor air volume required by indoor air quality regulations is greater than that required for pressurization, then a passive desiccant approach would be chosen most often, since it is less costly and will usually save more energy because it is a year-round technology (i.e., it provides significant heating season energy savings).

2.2.3 School Facilities

Given that school facilities often have an exhaust air stream available for recovery using passive desiccant wheels (or other recovery approaches), the opportunity for active desiccant systems continues to be either hybrid systems where the energy rejected from an engine-driven chiller is used for regeneration or where the benefit of maintaining "dry" cooling coils is considered very significant. A more thorough investigation of this market completed since the initial Phase 1 report further supports this conclusion.

An active desiccant offering made available in the Climate Changer product line would be limited primarily to school facilities designed with central chilled water systems located in humid climates, where waste heat is available or where the electrical cost/gas cost ratio is extremely high.

Considering these limitations, the percentage of schools constructed that would be viable candidates for an active desiccant system based upon a Climate Changer approach would likely be small. Once again, however, the number of schools constructed each year is very large, and even a small percentage may provide a large sales potential for Trane if its distributors market aggressively.

One potential market driver for active desiccant systems in schools is the increased awareness of the rise in the incidence of asthma, specifically among children. Numerous papers on this topic were presented at the worldwide indoor air quality conference held in Washington, D.C., in the fall of 1997. Two papers presented at the conference (Hodgson 1997; Bascom 1997) confirmed a rise in asthma cases—as well as other lung diseases and allergic conditions—among children and linked the increase in part to high humidity and/or mold and microbial infestation (which can be directly attributed to lack of adequate humidity control in humid climates).

(These papers and others presented at IAQ 1997 are collected in a proceedings, which can be obtained from the American Society of Heating, Refrigerating and Air-conditioning Engineers).

As more research is completed in this area, especially in the school environment where children spend a large portion of their time during the developmental years (when most asthma occurs), and if it is shown that humidity is a major contributor to the asthma "epidemic," then the argument for dry cooling coils and careful, year-round humidity control is likely to become widely accepted.

This scenario would significantly increase the market opportunity for active desiccant systems in school facilities as well as in the hotel/dormitory markets.

SEMCO has already observed first-hand the positive impact that effective humidity control has had on avoiding (in new construction) and eliminating (in retrofits) microbial activity in school facilities. We are hopeful that the school investigations being conducted by the Georgia Tech Research Institute and Georgia State University as part of Phase 2 of this program will provide quantitative evidence to support these observations.

2.2.4 Positive Trends Impacting Active Desiccant System Solutions

In addition to the market drivers already mentioned (e.g., conformance with ASHRAE 62-89, the relationship of high humidity to asthma), current trends in the design community and utility deregulation favor an increase in future market potential for active desiccant systems.

The 1997 ASHRAE *Fundamentals* has provided easy access to the true humidity design criteria for the entire country. As discussed in the Phase 1 report and shown in Table 6, the dew point design condition typically has a higher overall enthalpy value than the value obtained from the previously utilized dry bulb/wet bulb BIN data. More important, this table shows that the grain levels are much higher than previously thought by most engineers and that the sensible temperature that corresponds with this humidity design condition is likewise more moderate.

In short, outdoor air has a much higher latent-to-sensible load ratio than previously considered by most designers; this favors active desiccant systems and presents performance challenges to conventional compressor-driven, vapor compression equipment.

A growing number of design engineers are beginning to understand the benefits associated with providing outdoor air preconditioners to manage the increased latent loads resulting from accommodation to the ASHRAE 62 standard. The major HVAC equipment suppliers, such as Trane, are also beginning to promote conventional cooling methods of preconditioning such as dual path systems, over-cooling with compressor reheating units, and passive energy recovery. Acceptance of this fundamental change in the way systems are designed should proliferate as the impact of increased latent loads is better understood by more designers and end customers. This trend will greatly simplify the marketing of active desiccant systems.

GRI has been working to provide comprehensive weather data on an hour-by-hour basis that would allow completion of energy analyses that can accurately reflect the impact of latent outdoor air loads. According to Lew Harriman of Mason Grant, who has worked with GRI on this project, this comprehensive data base provides improved energy savings results over those provided by the current ASHRAE weather database. As these data are integrated into tools for engineers to evaluate the cost of conditioning outdoor air, the projected payback periods for active desiccant systems will shorten.

The analysis tool developed by SEMCO and Kirk Collier as part of this program is nearing completion (a beta copy is available for evaluation). These types of tool are badly needed to educate designers and owners. SEMCO will investigate the possibility of integrating the hour-by-hour weather data now available from GRI to further improve the accuracy of the comparative systems modeling.

Utility deregulation has started a trend toward mega-utility companies that provide both gas and electric energy. According to the experts, this trend will likely result in higher electrical demand charges during peak use periods, more time of day rates reflecting the true cost to the electrical service provider, and more energy service companies that will be looking for ways to use gas during peak electrical periods.

3. PRODUCT DEVELOPMENT STRATEGY

3.1 LIMITING ACTIVE DESICCANT SYSTEMS OPTIONS FOR SUCCESSFUL COMMERCIALIZATION

As discussed at length in the Phase 1 report, there are numerous ways to configure an active desiccant system. The benefit of the Climate Changer approach is that the modules can be arranged in many different configurations, offering the flexibility to tailor the system to the needs of each individual market. On the other hand, this relatively new technology needs to be presented to sales distributors and end customers in a pre-designed, easy-to-use format, or the benefits will be lost because complex engineering requirements increase the likelihood of misapplication.

Based on the market analysis portion of this program, it has been concluded that the needs of the target markets can be met effectively with a limited number of system configurations. Schematics of these configurations are shown as Figs. 1 through 6. A summary of the function and strength of each approach is provided below.

3.1.1 Traditional Desiccant-Based Cooling Approach

The desiccant-based cooling (DBC) system uses a desiccant dehumidification wheel and a sensible-only energy recovery wheel, plate, or pipe to provide dehumidified outdoor air at a temperature that is similar to the outdoor air condition (see Figs. 1 and 2) prior to post-cooling. At moderate humidity conditions, this system approach can also provide indirect evaporative cooling to assist sensible cooling.

This approach is best suited for applications where an exhaust air path is not available for passive energy recovery. If it is available, the desiccant dehumidification–total energy recovery approach or other passive energy recovery options would likely be far more efficient, and often less costly. For this reason,

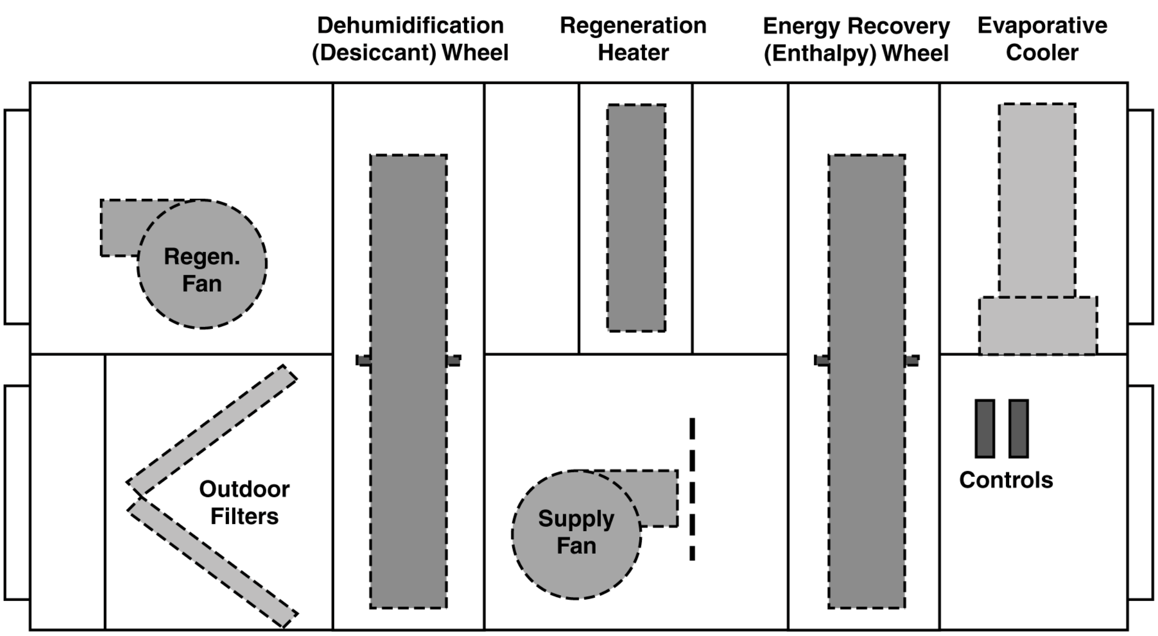

Fig. 1. Desiccant-based cooling (DBC) approach (from Climate Changer modules).

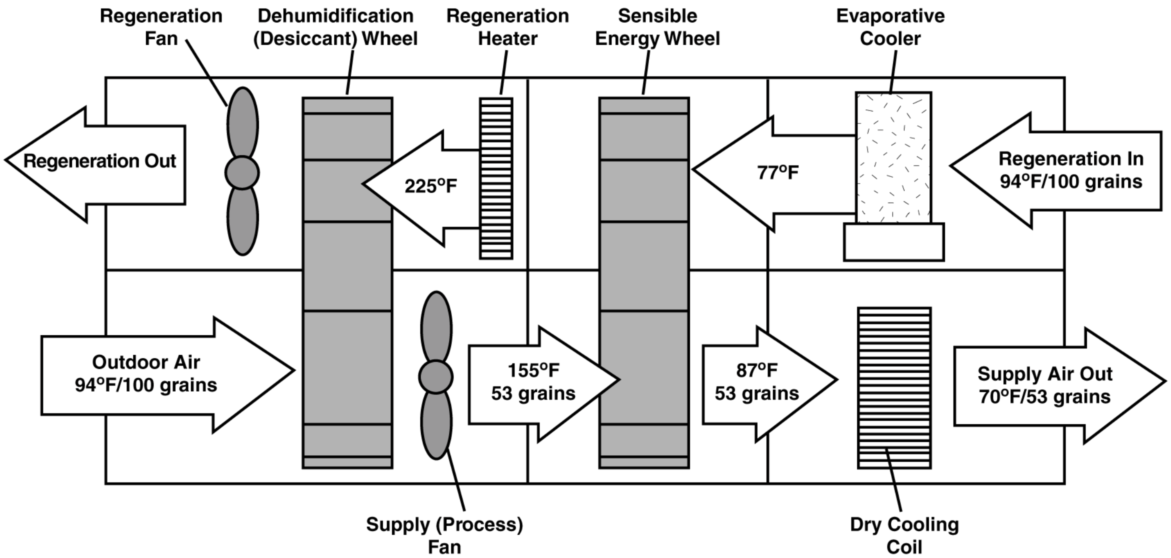

Fig. 2. Desiccant-based cooling approach: typical configuration.

a DBC unit is generally deactivated during the heating season, although sensible recovery is an option if an exhaust air path is used for regeneration.

3.1.2 Desiccant Dehumidification–Total Energy Recovery Hybrid

The desiccant dehumidification–total energy recovery system uses a desiccant dehumidification wheel and a passive total energy recovery wheel to provide much lower humidity levels more efficiently than is possible with the traditional DBC approach (Figs. 3 and 4). This approach requires a return air path for the total energy recovery wheel; the traditional DBC approach is typically applied without the need for a return air path.

This dehumidification–total recovery hybrid approach provides total energy recovery during the heating season, significantly reducing the payback period. It also eliminates the evaporative cooler used in the DBC approach, a fact often considered a plus because of the maintenance costs and the concerns about microbial activity and odors that are sometimes associated with evaporative coolers.

3.1.3 Dehumidification Only

The dehumidification-only module is used with conventional cooling approaches to provide the required dehumidification without the use of the sensible-only or total energy recovery wheel (Figs. 5 and 6). The advantage of this approach is compact size and, in some cases, a lower first cost. In most cases the operating cost will be high compared with the first two approaches, and no winter time recovery is provided. It will be common to have space limitations drive projects toward this approach despite the operational cost advantages of the others, since active desiccant systems will always be much larger than their conventional counterparts.

This approach would typically involve precooling to reduce the grain level of the air entering the dehumidification wheel while simultaneously raising its relative humidity, a process that increases the grain depression of the dehumidification wheel. The heat of adsorption and carryover associated with the dehumidification process would be handled by post-cooling or simply mixed with a cool airstream to provide the desired mixed air temperature.

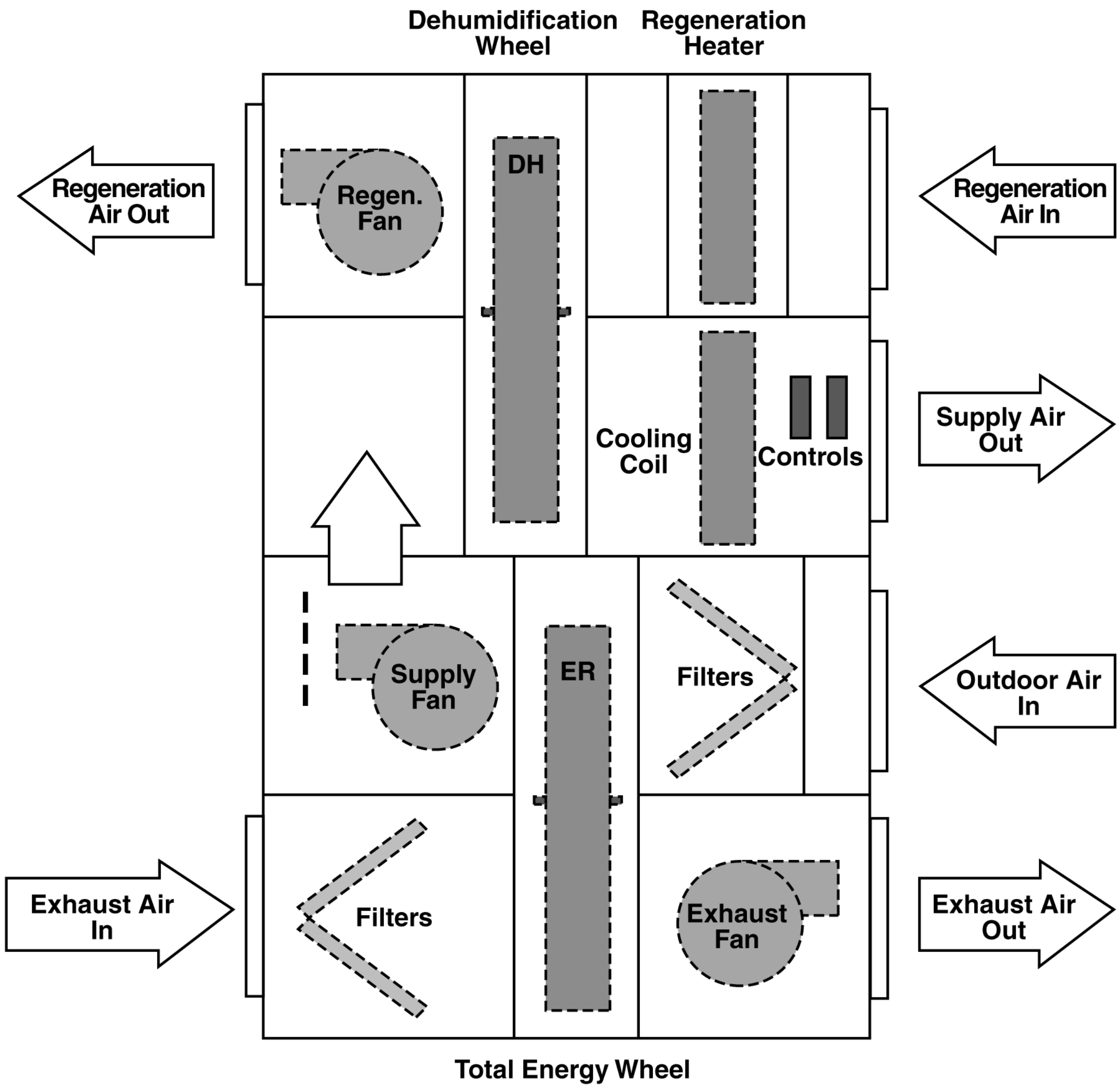

Fig. 3. Dehumidification–total recovery hybrid approach (from Climate Changer modules).

Given the flexibility of the Climate Changer offering, these three system configurations represent only two new configurations because the dehumidification–total energy recovery hybrid (Fig. 3) is simply a combination of Trane's existing total energy recovery wheel system with the dehumidification-only module shown in Fig. 5. Likewise, the dehumidification-only module shown in Fig. 5 is simply the dehumidification half of the traditional DBC approach shown in Fig. 1.

As a result, the goal of being able to meet the needs of the most attractive markets for an active desiccant system approach is easily met with only a few customized Climate Changer modules, and the system configurations required can be limited to three straightforward solutions.

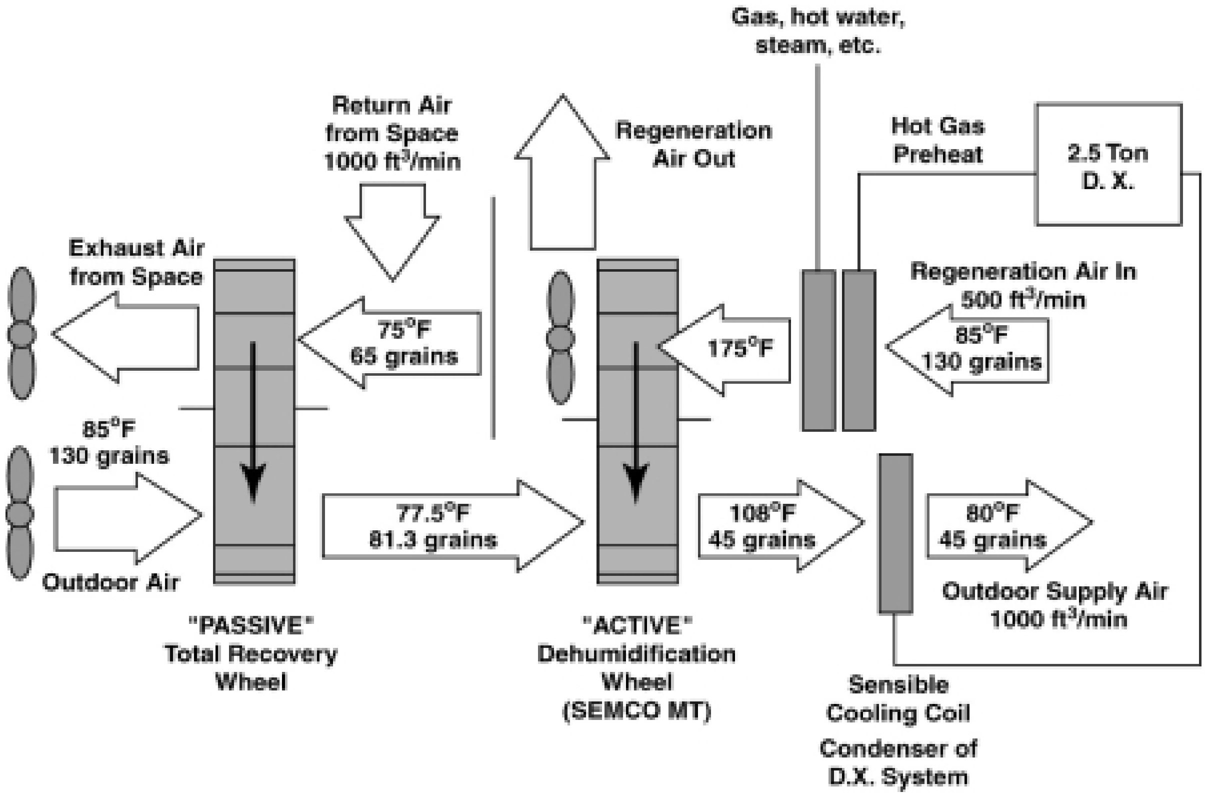

Fig. 4. Dehumidification–total energy recovery hybrid system.

3.2 LIMITED ACTIVE DESICCANT SYSTEM OPTIONS MEET THE MARKET-DRIVEN NEEDS AND OPPORTUNITIES

3.2.1 Hospitals

The dominant market driver for the use of active desiccant–based systems in hospitals is the need for humidity levels below those obtainable with conventional cooling approaches. This market would like a grain reduction of as much as 85 grains (from 120 grains down to 35) in order to maintain conditions in an operating room of 65°F and 50% RH, assuming a moderate internal latent load. For such applications, the systems shown in Figs. 4 and 6 are the most appropriate since they can easily supply the 35-grain air desired. The DBC approach shown schematically in Fig. 2 has a practical grain reduction limit of 60–65 grains and requires the use of evaporative coolers in the regeneration side, which is a serious concern for many hospital designers.

The dehumidification-only approach (Fig. 6) provides the most compact solution. The dehumidification–total energy recovery hybrid approach is the most energy-efficient while providing for significant heating season energy savings.

3.2.2 Nursing Homes/Assisted Living

The grain level required for nursing homes and assisted living facilities is obtainable with conventional cooling. Where a return air path is not available (i.e., corridor pressurization), the DBC approach is the most viable option (Fig. 1). It can provide the grain levels desired by these markets. When a return air

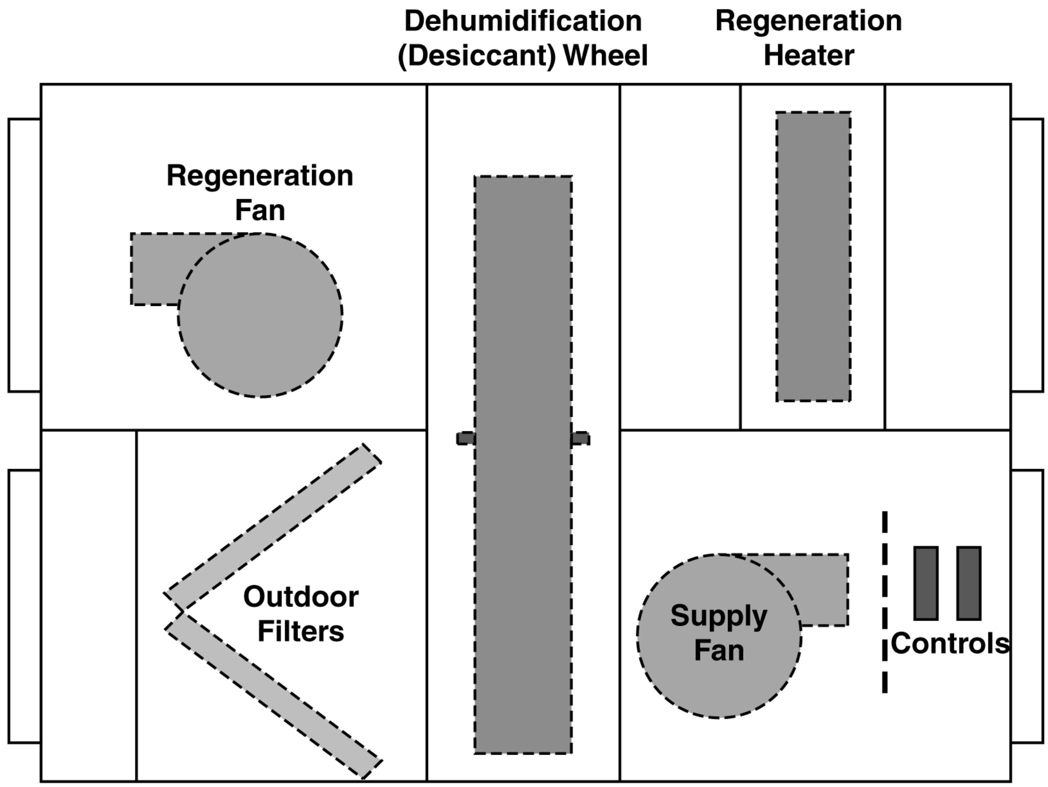

Fig. 5. Dehumidification-only approach (from Climate Changer modules).

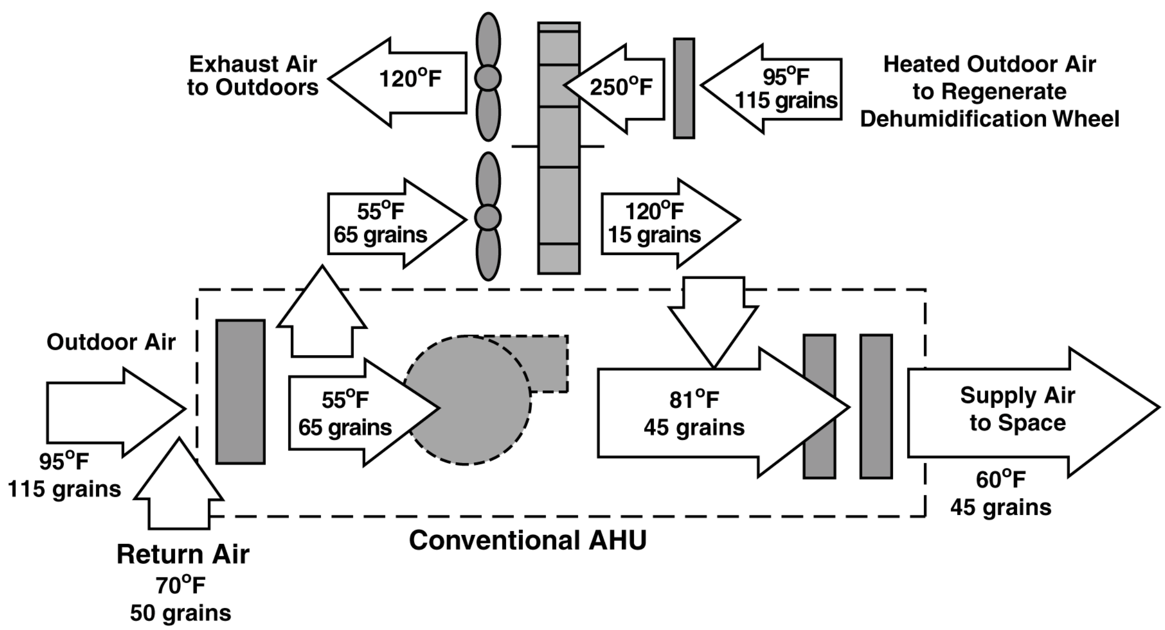

Fig. 6. Dehumidification-only assist to conventional cooling approach (from Climate Changer modules).

path is available, the dehumidification–total energy recovery hybrid will likely provide the best solution because it offers winter time heat and humidity recovery, reduced system complexity, and elimination of the evaporative cooler.

If a hybrid combination of an active desiccant dehumidification module and a vapor-compression air-conditioning rooftop product were available, it would likely be the preferred active desiccant choice for this market given the growth in the nursing home/assisted living industry and the price sensitivity and outdoor mounting requirements typical of assisted living facilities.

3.2.3 Research Facilities

Research facilities likely to be designed to incorporate an active desiccant approach are those where a significant quantity of exhaust air exists that is considered unavailable for recovery. As a result, such facilities will be a solid market for the traditional DBC approach (Fig. 1).

3.2.4 Hotel/Dormitory

The hotel/dormitory market fits criteria similar to those mentioned for nursing homes and assisted living facilities. The likely system choices are the same, although there also appears to be a market for the straight dehumidification approach to solve humidity problems in existing hotels located in hot and humid climates and in situations where the DBC system is too large or too costly. This market would also be well served by a Clarksville offering.

3.2.5 Retail Facilities

Retail facilities could benefit from each of the three active desiccant approaches discussed. If the facility is designed to conform to the ASHRAE standard recommendations, then there should be exhaust air available that can be used for the dehumidification–total recovery hybrid approach. If the outdoor air quantities are minimized, then the DBC approach located over the food area or a dehumidification module combined with a conventional packaged HVAC system is the best option.

3.2.6 Schools

If the benefits associated with dry cooling coils and year-round humidity control are embraced in the future by school designers, then the dehumidification–total energy recovery hybrid approach will provide an excellent solution for schools, as a return air path is almost always available. The ability to provide very dry air with this approach can complement a number of advanced system designs such as ice storage, super-cold systems, and gas engine–driven chiller designs. The sensible and latent heat recovery provided during the heating season operation is a big plus and therefore favors this hybrid approach.

Table 3 summarizes the active desiccant approach most likely to be used by the identified markets.

Table 1 is a revision to Table 7 from the Phase 1 report. The Table 1 summary applies potential sales dollars to the information in Table 3 to provide a breakdown of estimated sales potential for each active desiccant approach by target market. The annual sales potential for active desiccant systems produced from Climate Changer modules and SEMCO dehumidification wheel modules is estimated at $82.5 million, presenting a significant opportunity, though some part of this increased business would be at the expense of lost chiller sales in a separate business unit.

Table 3. Active desiccant approach most likely to be used in targeted markets

	Traditional-desiccant based cooling approach	Dehumidification—total recovery hybrid	Dehumidification only
Hospitals		X	X
Nursing homes	X	X	
Research facilities	X		X
Hotels/dormitories	X	X	X
Retail facilities	X	X	X
Schools		X	

3.3 KEY DESIGN OBJECTIVES DRIVING ACTIVE DESICCANT SYSTEM DEVELOPMENT (CONCLUSIONS OF MARKET RESEARCH)

Both Phase 1 and subsequent marketing research have provided firm design direction to the active desiccant–based system choices presented in this report. SEMCO believes that following the market feedback will greatly increase the likelihood of widespread market acceptance of this technology. Some of the more important conclusions based on market research are as follows.

1. Unit first cost continues to be more important than optimizing energy efficiency. As with other mainstream HVAC equipment, the active desiccant product needs to be designed to provide the most "bang for the buck" in order to gain market acceptance. This translates into providing the most latent cooling output (pounds of moisture removed) for a given system first cost. This position is also supported by performance modeling, since most of the energy savings dollars associated with an active desiccant system come from reduced demand charges. As a result, optimizing COP is secondary to the ratio of $/ton of latent cooling capacity.

2. As with passive desiccant systems, market acceptance will be based more on the equipment's ability to improve the quality of the indoor air (i.e., minimize pollutants and prohibit microbial activity), maintain the comfort of the building occupants, or meet an established code requirement than on a projected reduction in utility costs.

3. The primary function of active desiccant systems will be preconditioning the outdoor air delivered to buildings, linked with a design shift toward decoupling the outdoor air latent load with desiccant systems while using downsized conventional HVAC equipment to handle the internal sensible loads.

4. Size of the equipment will always be a major impediment to the use of active desiccant-based systems. The equipment will have to be designed to be as compact as possible while providing the required performance and serviceability.

5. Most opportunities for active desiccant systems involve systems sized for 10,000 cfm or less (as is the case for other HVAC units). All desiccant systems are less competitive, from a first-cost perspective, the smaller they become (see Table 2). This conclusion reinforces conclusion 1 that a cost-effective design approach is critical for successful commercialization.

6. The fact that low-cost hot water or steam is often available in the targeted markets significantly helps in justifying active desiccant systems. Therefore, it is desirable that the dehumidification wheel be capable of providing adequate performance at regeneration temperatures between 180 and 225°F.

7. Minimizing the pressure loss through the active desiccant system is important because it has a significant impact on the cost of operation, as well as the installed fan horsepower, electrical components, and so on.

8. Evaporative coolers are not currently well received in the targeted markets. They are viewed as potential sources of microbial activity and odors and as being maintenance-intensive. Avoiding the use of these devices would be a plus for market acceptance.

Based on these market driven conclusions, the following "key design objective" was derived for the active desiccant system approaches:

> *Design the most compact active desiccant system(s) possible, maximizing the latent cooling (grain reduction) vs. cost ratio, utilizing a dehumidification wheel capable of removing approximately 60 grains of moisture from outdoor air having a relative humidity of 80–90% (dew point design data), and using a regeneration temperature in the range of 180 to 225 °F.*

3.4 SYSTEM DESIGN OBJECTIVES SET THE PERFORMANCE GOALS FOR THE ACTIVE DESICCANT WHEEL

The performance, size and cost of the dehumidification wheel used for the active desiccant system approaches have the most significant impact on whether or not the stated "key design objective" is achieved. To reach the stated design objective for the active desiccant system, the desiccant dehumidification wheel must be designed to meet the following criteria:

- Operates at relatively high face velocities (600 to 700 ft/min) to minimize the overall system size
- Limits pressure loss to 1 in. wg for both the supply and regeneration air
- Provides the stated grain reduction at the stated regeneration temperature range (60 grain reduction at the dew point design condition of 84°F and 130 grains)
- Operates in a flow configuration that matches the current Climate Changer product offering dimensional aspect ratios
- Is cost-effective to produce when commercialized in large quantities

Additional desirable but less critical performance objectives include

- Requires minimal maintenance, resists plugging and performance degradation
- Optimizes latent cooling output/ regeneration energy input (latent COP)
- Co-sorbs airborne contaminants from the outdoor or recirculated airstream

These performance objectives parallel the criteria used by SEMCO in developing its current 1M composite active desiccant dehumidification wheel. The major exception is that the current product's desiccant adsorption parameters are optimized for a regeneration temperature range between approximately 250 and 300°F. The SEMCO MT (moderate regeneration temperature) wheel tested as part of this program is being developed to fill the performance gap for the targeted (more moderate) regeneration temperature range (see Fig. 7).

The current SEMCO 1M wheel uses a desiccant surface that has an isotherm optimized for high regeneration temperatures. Based on SEMCO's in-house wheel testing thus far, the goal of removing 60 grains of moisture from outdoor air at the latent design condition (85°F and 130 grains) should be obtainable with a regeneration temperature in the range of 250°F once the casing, seals, purge, and spoke designs are optimized. This moisture removal target reflects the need to provide outdoor air at 70 grains (57°F dew point) during the 1% dew point design extreme.

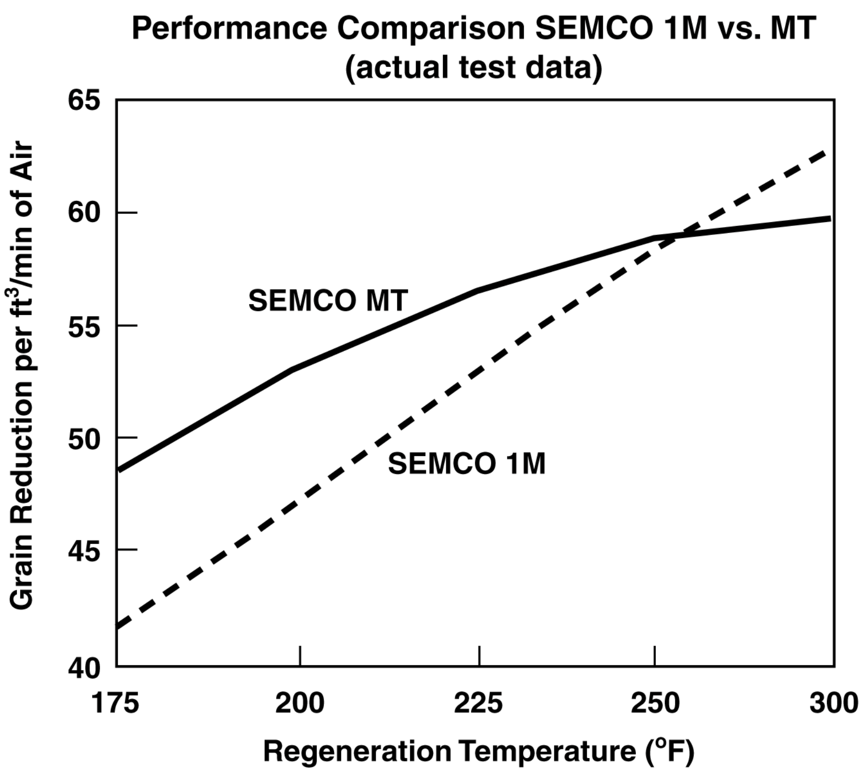

Fig. 7. Sample performance comparison: SEMCO 1M and SEMCO MT wheels with all conditions and operating parameters being equal.

In addition to providing the targeted dehumidification capacity, this wheel has been specifically designed to co-sorb common indoor and outdoor airborne pollutants. Based on market feedback, this feature is considered very beneficial by some of the markets targeted for active desiccant systems.

SEMCO has also developed and tested a second-generation wheel, type MT. The purpose for this new wheel is to respond to the need identified in the market analysis for an active desiccant wheel that can be regenerated more effectively at moderate regeneration temperatures (optimized for 200 to 225°F). This wheel would be used where hot water, low-temperature steam, or waste heat is available.

It appears that this wheel may be slightly more costly to produce than the current SEMCO 1M wheel and may not have the same ability to remove targeted indoor and outdoor air pollutants as the 1M wheel. The initial results do look very promising and clearly justify further optimization of this product, which is under way.

As shown by Fig. 7, the performance advantage of the SEMCO MT wheel is clear compared with the 1M wheel at moderate regeneration temperatures. A significant improvement in moisture removal is recognized between approximately 150 and 225°F regeneration temperatures, with all other design parameters being equal.

Further testing of this new MT wheel is under way at SEMCO as part of the total system testing. A prototype wheel and cassette has been installed in the dehumidification–total energy recovery hybrid system built from Climate Changer modules (shown in Fig. 3, as well as in actual equipment photographs attached to select copies of this report).

Since the MT wheel is still under development, the modeling and optimization presented in the next section are based on the current 1M technology. The performance data presented for the DBC approach are based upon the SEMCO 1M test data. The performance presented for the dehumidification–total energy hybrid approach uses the test data obtained thus far for the MT wheel. The performance shown for both of these wheels can and will be improved by further optimization.

4. PERFORMANCE MODELING

4.1 ACTIVE DESICCANT WHEEL OPTIMIZATION TO MEET THE MARKET-DRIVEN PERFORMANCE CRITERIA AND STATED KEY DESIGN OBJECTIVE

4.1.1 Wheel Size and Process/Regeneration Area Percentages

Although optimum performance may be provided with a process area to regeneration area split that is 75/25, for example, fitting such a product into the Climate Changer modules (or any other system for that matter) would be difficult because of the airflow pattern that would result. The Climate Changer modules must be stacked to provide the two separate airstreams required for an active desiccant system (see Fig. 1 and the attached photographs).

Based on modeling, performance testing, and airflow patterns through the Climate Changer modules, an approximate 60 to 40% allocation of process to regeneration area appears to be the optimum selection. This configuration provides increased area to process the maximum quantity of outdoor (or recirculated) air while matching well with the size, aspect ratio, and flow pattern that exists within the Climate Changer product.

Table 4 clearly shows the importance of optimizing the dehumidification wheel for relatively high process face velocities. Note that at 600 ft/min, the process flow volume through the dehumidification wheel represents on average 82% of the current design capacity of the Climate Changer modules. At 650 ft/min, the resulting flow would be 90%.

In contrast, operating at a face velocity of 400 ft/min, even for a wheel with a 60/40 split, would only use

Table 4. Comparison of Climate Changer design airflow rates with dehumidification wheel geometry and face velocity
(Using SEMCO SMCC wheel diameters)

	Trane Climate Changer model number					
	CC-6	CC-10	CC-14	CC-21	CC-30	CC-40
Design airflow for Climate changer (cfm)	2,930	4,820	7,110	10,390	14,505	19,650
Design flow SMCC DH[a] at 600 ft/min (cfm)	2,520	3,744	6,048	8,424	12,096	16,128
Percent optimum capacity used	80%	78%	85%	81%	83%	82%
Design flow SMCC DH[a] at 500 ft/min (cfm)	2,100	3,120	5,040	7,020	10,080	13,440
Percent optimum capacity used	72%	65%	71%	68%	69%	68%
Design flow SMCC DH[a] at 400 ft/min (cfm)	1,680	2,496	4,032	5,616	8,064	10,752
Percent optimum capacity used	57%	52%	57%	54%	56%	55%

[a]SMCC DH wheel is a modification to the standard SEMCO SMCC total energy wheel that has a 50/50 split between the exhaust and the outdoor air halves of the wheel. THe SMCC DH wheel has 60% of the area in the outdoor air portion and 40% of the wheel area in the regeneration air portion.

approximately 55% of the capacity of the Climate Change modules. Since the cost of the final system is a function of the size of the Climate Changer modules used, and not of the airflow quantity processed, operating the system at reduced capacity would significantly increase the cost/ton of latent cooling ratio, one of the primary optimization goals based on market feedback.

For example, a commercialized DBC system produced with conventional Climate Changer modules, using a desiccant wheel with a 60/40 split operated at 600 ft/min and processing 2500 cfm, would have a projected selling price of approximately $5.60/cfm. If the wheel were operated at 400 ft/min in lieu of 600, the selling price would increase by 61% to approximately $9/cfm. Higher face velocities result in larger pressure drops across components and correspondingly more fan power.

As a result, optimizing for a high face velocity through the desiccant dehumidification wheel becomes one of the most important criteria impacting successful commercialization. Process face velocity significantly impacts both the overall system cost and size, two of the most important optimization parameters based on market feedback.

Therefore, one of the primary design goals for the SEMCO dehumidification wheel development is to provide the 60 grain moisture reduction with a process face velocity of approximately 600 ft/min while maintaining the pressure loss through the process and regeneration airstreams at a maximum of 1 in. wg each.

4.1.2 Optimizing Wheel Speed and Managing Heat Carryover

A significant amount of testing has been completed to optimize the wheel speed to provide the maximum moisture removal (i.e., latent tons) as a function of wheel face velocity, incoming grain content, regeneration temperature, and so on. At the targeted high process face velocity, as expected, an increased wheel speed is required to obtain the desired grain depression. Naturally, the higher wheel speed results in higher carryover heat than desired if minimizing supply outlet temperature is considered a key optimization parameter. Heat carryover is a function of wheel speed, non-desiccant mass fraction, desiccant heat of adsorption, and other factors.

Heat carryover in an active desiccant system is interesting in that it has both a negative and a positive effect. The negative effect is that the overall cooling capacity of an active desiccant-based cooling system is reduced as the carryover heat is increased. A GRI report covers this issue quite well and concludes that high heat of adsorption (carryover heat) typically decreases the cooling capacity of a DBC system (Collier 1988, p. 46).

On the other hand, since the DBC approach uses a very efficient sensible-only recovery wheel to capture this carryover heat to preheat the air entering the regeneration source, high heat carryover actually increases energy efficiency, raising the system COP (Collier 1988).

Figures 8 and 9 show the relative impact of increased heat carryover in a DBC system configuration. Figure 8 shows a condition that represents the SEMCO 1M wheel with increased heat carryover. Figure 9 shows what the relative total cooling tons, latent cooling tons, and COP would be for an identical system assuming a significant reduction in carryover heat (20°F), with all other design parameters being equal.

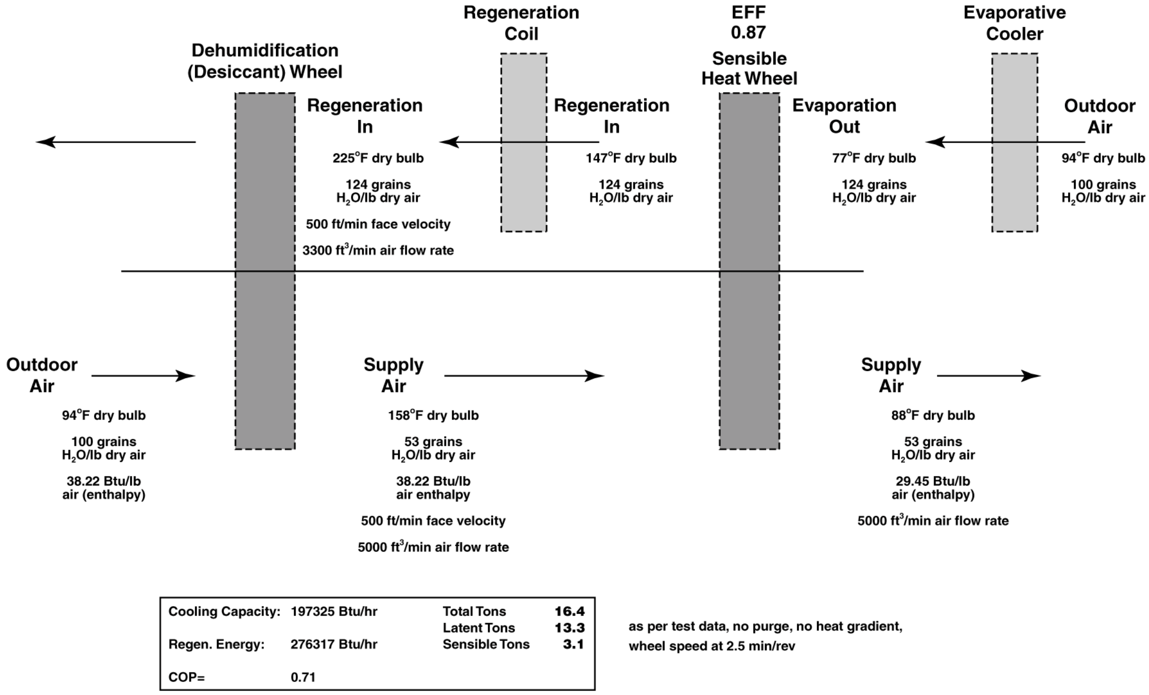

Fig. 8. High heat carry-over DBC flow diagram

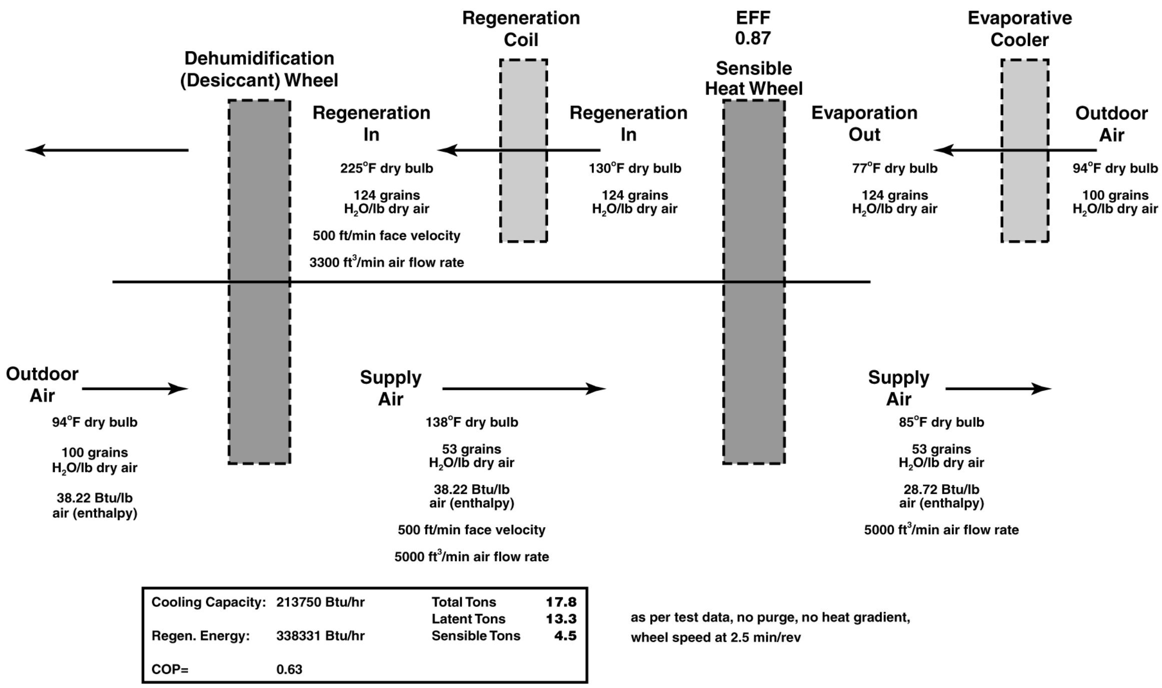

Fig. 9. Reduced heat carry-over DBC flow diagram.

Note the following:

- the most important parameter, latent cooling capacity, is identical
- the next most important factor, COP, is increased by 13% with the high carryover heat
- the least important factor, sensible cooling tons, is decreased

Note that the difference in the supply air temperature leaving the sensible wheel in Figs. 8 and 9 is only 3°F despite the significant 20°F difference in the air temperature leaving the dehumidification wheel (this is increased to only 3.8°F when the sensible wheel efficiency drops to 80%). Fan heat in a system of this type would be on the order of 2 to 5°F. In short, the net impact on sensible cooling performance of the 20°F increase in heat carryover is small and on a par with the impact of supply side fan heat.

As a result, the positive impact on COP seems to outweigh the negative impact of reduced sensible cooling load. It can therefore be concluded that the higher heat carryover associated with operating at higher process face velocities is not a significant negative and may, in fact, be a positive in justifying the active desiccant technology based on operating efficiency.

4.1.3 Purge Section Impact

Modeling has also shown that the active desiccant system benefits from the adaptation of a purge section to the dehumidification wheel cassette. This has been confirmed by SEMCO through testing and will be optimized further as part of the final system testing. A purge section uses unconditioned outdoor air to push some of the carryover heat (and associated high humidity) into the regeneration airstream, thereby increasing dehumidification performance, reducing the supply air outlet temperature, and/or reducing regeneration energy.

Figure 10 shows the same system configuration as in Fig. 8 (high heat carryover) but uses a 10°F purge angle to reduce heat carryover and regeneration energy. The performance shown in this example is based on modeling that will be confirmed when system testing is complete. Note that if the latent cooling is kept the same between examples 1 and 3, the COP is increased from 0.71 to 0.84, and the sensible cooling is also increased slightly from 3.1 to 3.6 tons. Figure 10 could have been configured to show an increase in latent cooling capacity (grain reduction), which is the most important advantage for the same energy input shown for Fig. 8.

As a result of these findings, it is expected that the use of a purge will provide an increase in latent capacity of approximately 5–10%, and it will likely be a standard feature in active desiccant systems optimized to meet the key design objective previously stated.

4.1.4 Benefit of Using the Heat Gradient of the Sensible Wheel

The improvement in COP associated with increased carryover heat becomes even more pronounced when the heat gradient leaving the sensible recovery wheel is taken advantage of. The higher temperature leaving the desiccant wheel translates into a higher average temperature entering the regeneration heater (assuming that only a fraction of the air leaving the sensible wheel on the regeneration air side is used for regeneration).

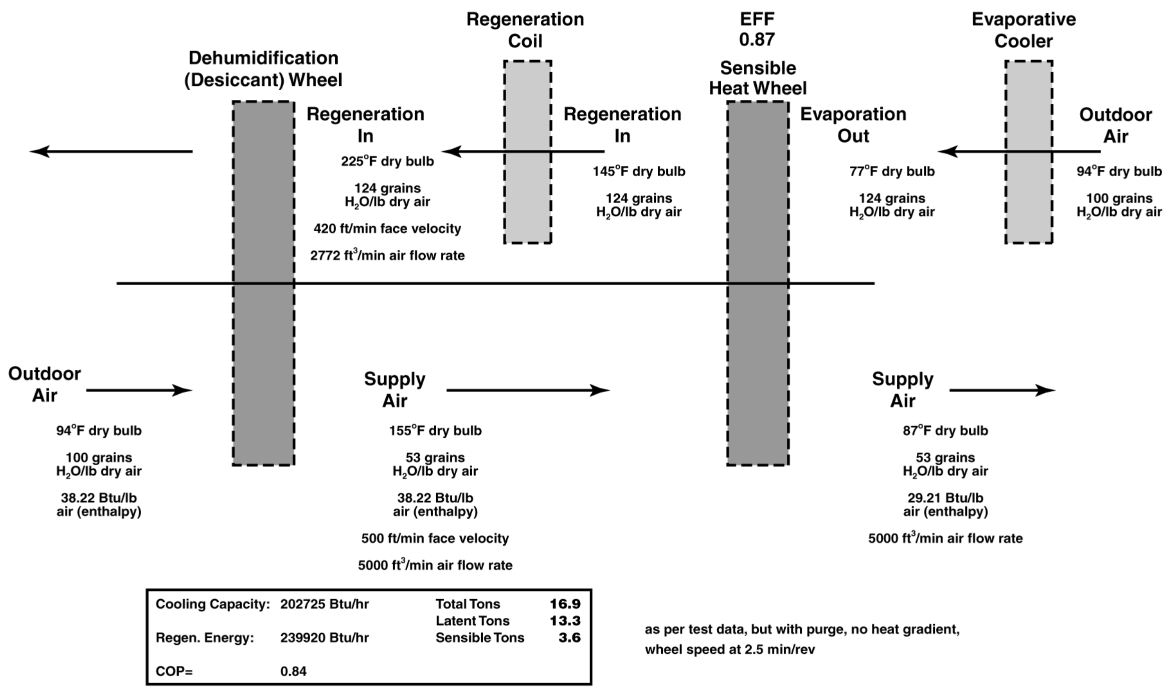

Fig. 10. High heat carry-over with purge DBC flow diagram.

As shown in Fig. 11, the air leaving the first half of the sensible wheel on the regeneration inlet side is increased from 145°F (example 3) to 150°F. This modeled gradient trend has been confirmed by testing conducted at SEMCO. The use of this heat gradient has the potential of further increasing the system COP from the 0.84 value shown in example 3 to 0.9 as shown in Fig. 11.

4.2 REVISIONS OF MODELING USING NEW DESICCANT WHEEL PERFORMANCE DATA (TESTING COMPLETED IN TASKS 2 AND 6)

4.2.1 Improved Operating Costs of DBC and DH/ER Hybrid Approaches

The SEMCO Phase 1 report included a summary of many computer-generated estimates of operating costs for the various specialized air handling units investigated. The energy analyses completed for the active desiccant systems were based on preliminary test data measured by SEMCO on its 1M wheel technology. This program has provided an opportunity to improve upon this core technology that has resulted in enhanced performance in the wheel itself.

Figures A.1 and A.2 in Appendix A show the energy analyses completed for the DBC and the dehumidification–total energy recovery hybrid configurations as part of Phase 1 work based on the previous wheel performance parameters. Figures A.3 and A.4 show the same analyses but have been modified to reflect the current performance associated with the SEMCO 1M wheel.

As a comparison of Figs. A.1 through A.4 shows, the improved performance for the dehumidification wheel reduced the operating cost of the modeled DBC system by approximately 5% and the cost of the modeled dehumidification–total energy recovery hybrid by approximately 4%.

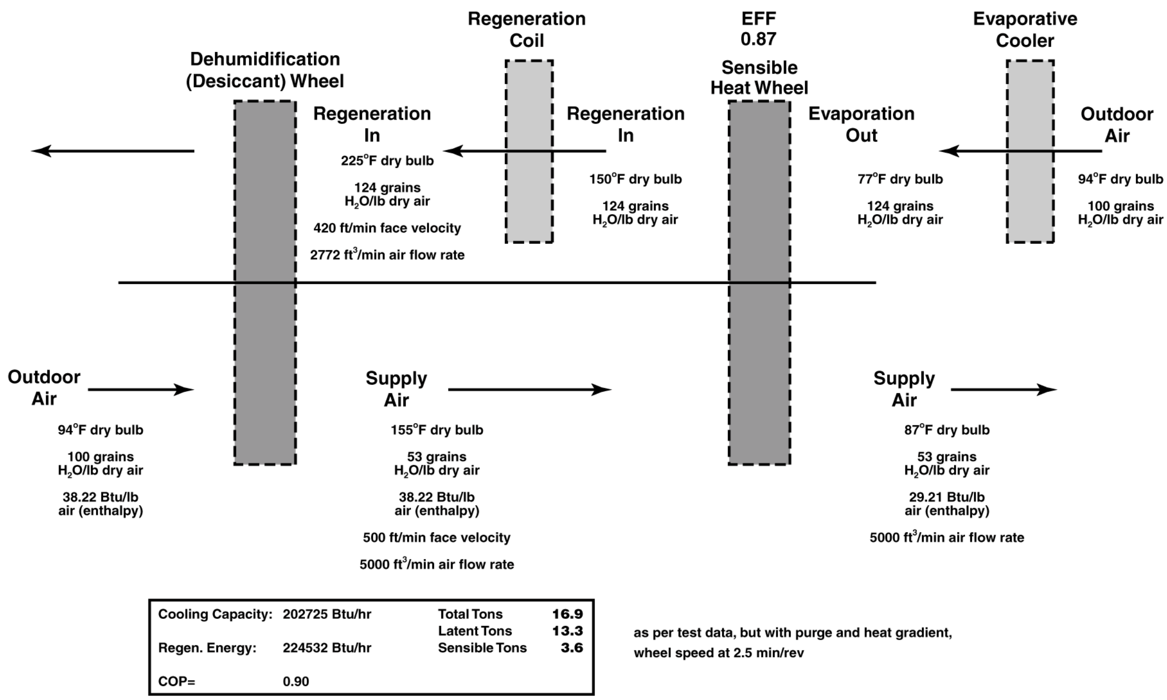

Fig. 11. High heat carry-over with purge and heat gradient.

Table 5 is a revision to Table 4 in an earlier marketing report. The payback periods for the three active desiccant-based system approaches have been modified to reflect the actual wheel performance provided by the SEMCO 1M wheel. The results of this analysis for active desiccant systems is quite favorable; a payback is very short or immediate for all systems with a capacity of 7500 cfm or greater, and approximately 2 years for systems in the 2500-cfm range. It is important to point out that these payback periods are strongly influenced by the cost of energy, as well as the cost of installed chiller and cooling tower capacity (assumed to be $500/ton for this analysis).

4.2.2 Impact of the New ASHRAE Dew Point Design Data: Good and Bad for Active Desiccant Systems

Table 6 shows the difference between dry bulb/mean coincident wet bulb design conditions and the new dew point/mean coincident dry bulb design data recently published in the 1997 *ASHRAE Fundamentals* (ASHRAE). This table, which compares the data for six cities, shows that the dew point design data result in much higher latent loads as well as total cooling loads associated with the outdoor air. This is clearly a plus for any system designed specifically to manage latent loads in outdoor air volumes. As a result, this is a plus for active desiccant systems.

Table 7 summarizes energy analyses completed for the seven specialized air handling units investigated in the earlier report. The operating costs associated with the active desiccant systems have been updated since presented in the Phase 1 report to reflect the current test data for the SEMCO dehumidification wheel. This analysis paints a positive picture for the active desiccant approach in that the cost of operation decreases by an average of 6% while the cost of the conventional approaches actually increases by the same amount. As shown, the cost of operating passive systems remains the same independent of the design conditions used.

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

Preconditioning system approach	2,500 cfm (simple payback in months[a])	7,500 cfm (simple payback in months[a])	20,000 cfm (simple payback in months[a])
Conventional cooling w/reheat	NA	NA	NA
Conventional cooling w/run around recovery	0.3 months ($2009/yr in energy savings)	($200 first cost savings) Immediate ($6028/yr in energy savings)	0.2 months ($16074/yr in energy savings)
Total energy wheel w/cooling and reheat	5 months ($1892/yr in energy savings)	($1850 first cost savings) Immediate ($5676/yr in energy savings)	($10400 first cost savings) Immediate ($15136/yr in energy savings)
Dual wheel total energy recovery	9 months ($3907/yr in energy savings)	($350 first cost savings) Immediate ($11720/yr in energy savings)	($11050 first cost savings) Immediate ($31253/yr in energy savings)
Dual wheel desiccant based (DBC) w/post cooling	20 months ($2988/yr in energy savings)	0.5 months ($8960/yr in energy savings)	($11900 first cost savings) Immediate ($23900/yr in energy savings)
Desiccant dehumidification–total recovery hybrid	27 months ($3188/yr in energy savings)	5 months ($9565/yr in energy savings)	($2350 first cost savings) Immediate ($25500/yr in energy savings)
Dual wheel DBC–total recovery hybrid	27 months ($4362/yr in energy savings)	6.5 months ($13080/yr in energy savings)	Immediate ($34895/yr in energy savings)

[a]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. Atlanta weather data and 1994 local energy costs are used for this comparison. The energy savings are based on continuous operation and chiller/cooling tower cost of $500/ton. The conventional cooling with run around uses the plate heat exchanger currently offered by Trane.

Table 6. Comparison of humidity design data: dry bulb and mean coincident wet bulb (MCWB) and mean coincident dry bulk (MCDB) data vs dew point design
(2% design condition)

City	Dry bulb, MCWB				Dew point, MCDB			
	Dry bulb (°F)	Wet bulb (°F)	Grains (H_2O/lb air)	Enthalpy (Btu/lb air)	Dew point (°F)	Dry bulb (°F)	Grains (H_2O/lb air)	Enthalpy (Btu/lb air)
Atlanta	88	73	109	37.3	72	80	123	38.5
Houston	92	77	117	40.5	76	83	137	41.4
Los Angeles	78	64	93	29.3	65	72	93	31.8
Minneapolis	85	70	90	34.5	69	79	110	36.2
New York	86	72	97	35.8	71	80	116	37.3
Orlando	92	76	110	39.4	76	81	136	40.8

Source: ASHRAE Fundamentals 1997.

The downside for active desiccant systems as a result of the new dew point weather data is reduced traditional COP values (total cooling provided/regeneration energy input) and increased post-cooling requirements (as a result of higher supply air temperatures after the dehumidification wheel).

Figures A.5 and A.6 provide schematic examples of typical active desiccant system performance for both the traditional dry bulb BIN design condition and the new dew point design condition. Note that since the dew point design condition represents a much higher absolute humidity level, more moisture needs to be removed. The greater the grain reduction required, the greater the regeneration energy required and the greater the heat of adsorption. The result is a much higher temperature leaving the dehumidification wheel than would be associated with the dry bulb design condition (Fig. A.5).

At the same time that the dew point design condition produces a much higher temperature after the dehumidification wheel, it unfortunately also provides less post-cooling capacity because the outdoor air condition is near saturation, rendering the indirect evaporative cooling side of the system ineffective (see Fig. A.6).

As shown by Fig. A.6, the performance of a DBC cycle operated at the dew point design condition can result in negative sensible cooling tons and therefore in increased post-cooling requirements. The result is also much reduced traditional COP values and total cooling output. On the other hand, the latent cooling tons (the most important criterion) increases significantly over the level obtained at the dry bulb design condition.

Figures A.7 and A.8 show a similar trend for the dehumidification–total energy recovery hybrid approach. The higher humidity content associated with the dew point design condition results in much higher latent cooling output while reducing traditional COP values and increasing the post-cooling energy required.

Table 7. Impact of new ASHRAE weather data (wet bulb BINS, corresponding dry bulb)

Preconditioning system approach	Annual energy cost estimate for a 20,000 cfm system located in Atlanta based on Air Force weather data (current dry bulb BIN method)	Annual energy cost estimate for a 20,000 cfm system located in Atlanta based on revised ASHRAE data (new wet bulb BIN method)	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	0.90
Dual wheel total energy recovery	$26,928	$27,060	0.50
Duel wheel desiccant based (DBC) w/post cooling[a]	$33,689	$30,994	–8
Desiccant dehumidification– total recovery hybrid[a]	$32,066	$30,142	–6
Dual wheel DBC–total recovery hybrid[a]	$24,000	$22,800	–5

[a]Assumptions:

- Reflects the use of a favorable gas cooling rate of $0.35/therm.
- Wet bulk BIN data are 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 summer time use. A 78% boiler efficiency is assume.
- Location is Atlanta.

What is clear from this analysis is that the dew point design conditions offer the advantage of increased latent output but also result in increased supply air temperatures off the active desiccant system, increasing the need for post-cooling. Post-cooling cannot be eliminated completely if a temperature of 70 to 80°F (room neutral temperature) is desired. The good news is, based once again on market analysis, the latent output is far more important than the traditional COP value (operating efficiency). In addition, the post-cooling energy is on a par with the energy required for reheating the conventional cooling approach.

5. CONCLUSIONS

5.1 COMPARING SYSTEMS BASED ON LATENT AIR CONDITIONING DELIVERED

The success of the introduction of any active desiccant product will depend upon the importance placed by the design community on improved humidity control, the acceptance of ASHRAE 62-89, and a paradigm shift toward decoupling the latent and sensible loads. As a result, the success of a specific active desiccant approach will be directly linked to its ability to perform as the most effective “latent air conditioning” alternative. Such a product will be applied to handle the latent load associated with the outdoor air alone or the outdoor air combined with the space latent load.

Given the intended use of this technology, to provide latent air conditioning, it appears to be inappropriate to compare it with other, more conventional approaches based on total cooling capacity. For example, a system designed to provide total cooling (sensible and latent) should be rated based on total cooling capacity, and a COP (or EER) should be based upon the ratio of total cooling to energy input. Likewise, it seems logical that a latent air conditioning system should be compared in a corresponding fashion, with a rating based on tons of latent cooling capacity and a COP measurement based on latent cooling output/energy input.

Tables 8, 9, and 10 compare a conventional overcooling–reheat approach, a DBC approach, and a dehumidification–total energy recovery hybrid approach at both the dry bulb BIN and the dew point design conditions. These figures show both the total cooling capacity and latent cooling capacity resulting from each approach.

When these systems are compared as latent air conditioning systems, the advantage of applying a commercialized active desiccant approach such as that built from Climate Changer modules becomes obvious. In short, the first-cost advantage for the conventional overcooling–reheat approach disappears when systems are compared based on latent output, the primary purpose for the equipment. Also, when the reheating required by the conventional cooling approach is compared with the post-cooling required by the active desiccant approaches, it is clear that the energy required is similar at the design condition. At part-load conditions, the active desiccant systems have the advantage since the desiccant system’s post-cooling energy decreases rapidly as the outdoor air conditions become less extreme, while the reheating energy required by the conventional system remains constant until the outdoor air humidity content falls below that delivered to the space. This seemingly contradictory result occurs because desiccated air is heated by the drying process, and additional sensible cooling is required to lower its temperature to starting conditions.

Note that even at the high-humidity condition associated with the dew point design condition (Table 8), the conventional cooling approach only provides 26 tons of latent cooling with a 47 ton total cooling input (at 7500 cfm). The DBC approach (Table 9) provides the same 26 tons of latent cooling but provides only 21 tons of total cooling. When first cost is based on $/total tons provided, the conventional approach looks far less expensive. When the systems are compared based on latent output, the appropriate method of comparison since it reflects the purpose of the system, the active DBC and the conventional cooling approach are essentially the same. Once the operating cost is factored into the equation, the active approach often provides the best return on investment.

Table 8. Conventional overcooling with reheat preconditioning approach

Outdoor air condition: 94 °F and 100 grains (typical dry bulb design condition) supplied at 53 grains and space neutral temperature

Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	16.4	NA	6.7	NA	$10,400	$1,552	Requires an additional 54,000 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F
7,500	49.2	NA	20.1	NA	$31,700	$1,577	Requires an additional 162,000 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F
20,000	131	NA	53.6	NA	$82,200	$1,534	Requires an additional 432,000 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F

Notes:
1. Total tons based upon conditioning outdoor air from 94°F and 100 grains to 51°F and 53 grains.
2. Latent tons based upon reducing the humidity content of the outdoor air from 100 grains to 53 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and required chiller tonnage at $500/ton.
4. As per the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This conventional overcooling–reheating approach requires the reheat energy shown in this column to reach the targeted 70–80°F delivery condition.

Outdoor air condition: 84 °F and 130 grains (typical dew point design condition) supplied at 70 grains and space neutral temperature

Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	15.5	NA	8.5	NA	$9,950	$1,171	Requires an additional 35,100 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F
7,500	46.5	NA	25.5	NA	$40,350	$1,190	Requires an additional 105,300 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F
20,000	124	NA	68	NA	$78,700	$1,157	Requires an additional 280,800 BTUs of reheat energy to reach the desired space neutral lower limit of 70°F

Notes:
1. Total tons based upon conditioning outdoor air from 84°F and 130 grains to 57°F and 70 grains.
2. Latent tons based upon reducing the humidity content of the outdoor air from 130 grains to 70 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and required chiller tonnage at $500/ton.
4. Based on the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This conventional overcooling–reheating approach requires the reheat energy shown in this column to reach the targeted 70–80°F delivery condition.

Table 9. Desiccant-based cooling preconditioning approach

Outdoor air condition: 94 °F and 100 grains (typical dry bulb design condition) supplied at 53 grains and space neutral temperature							
Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	8.5	0.9	6.7	0.71	$14,700	$2,194	Requires an additional 18,900 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F
7,500	25.5	0.9	20	0.71	$30,200	$1,510	Requires an additional 56,700 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F
20,000	68	0.9	53.6	0.71	$64,800	$1,209	Requires an additional 151,200 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F

Notes:
1. Total tons based upon conditioning outdoor air from 94°F and 100 grains to 87°F and 53 grains (see diagram DBC-1).
2. Latent tons based upon reducing the humidity content of the outdoor air from 100 grains to 53 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and estimates for the required SEMCO dehumidification and energy wheel modules.
4. As per the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This DBC approach requires the post-cooling energy shown in this column to reach the targeted 70–80°F delivery condition.

Outdoor air condition: 84 °F and 130 grains (typical dew point design condition) supplied at 70 grains and space neutral temperature							
Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	7.1	0.64	8.5	0.76	$14,700	$1,729	Requires an additional 31,590 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F
7,500	21.3	0.64	25.5	0.76	$30,200	$1,184	Requires an additional 94,770 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F
20,000	56.8	0.64	68	0.76	$64,800	$953	Requires an additional 252,720 BTUs of post-cooling energy to reach the desired space neutral lower limit of 80°F

Notes:
1. Total tons based upon conditioning outdoor air from 84°F and 130 grains to 91.7°F and 70 grains (see diagram DBC-2).
2. Latent tons based upon reducing the humidity content of the outdoor air from 130 grains to 70 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and estimates for the required SEMCO dehumidification and energy wheel modules.
4. As per the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This DBC approach requires the post-cooling energy shown in this column to reach the targeted 70–80°F delivery condition.

Table 10. Dehumidification–total energy recovery hybrid

Outdoor air condition: 94 °F and 100 grains (typical dry bulb design condition) supplied at 45 grains and space neutral temperature

Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	6.8	0.98	7.8	1.1	$15,750	$2,019	Requires an additional 59,400 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F
7,500	20.4	0.98	23.4	1.1	$31,350	$1,340	Requires an additional 178,200 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F
20,000	54.4	0.98	62.4	1.1	$67,850	$1,087	Requires an additional 475,200 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F

Notes:
1. Total tons based upon conditioning outdoor air from 94°F and 100 grains to 87°F and 53 grains (see diagram DH/ER Hybrid-1).
2. Latent tons based upon reducing the humidity content of the outdoor air from 100 grains to 53 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and estimates for the required SEMCO dehumidification and energy wheel modules.
4. As per the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This DH/ER hybrid approach requires the post-cooling energy shown in this column to reach the targeted 70–80°F delivery condition.

Outdoor air condition: 84 °F and 130 grains (typical dew point design condition) supplied at 45 grains and space neutral temperature

Outdoor air volume (SCFM)	Total cooling tons (1)	Total cooling COP	Latent load cooling tons (2)	Latent load COP	Estimated sales price Climate Changer modules (3)	Cost per ton latent removal	Comments (4)
2,500	7.3	0.72	12	1.2	$15,750	$1,313	Requires an additional 75,600 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F
7,500	21.9	0.72	36	1.2	$31,350	$871	Requires an additional 226,800 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F
20,000	58.4	0.72	96.3	1.2	$67,850	$705	Requires an additional 604,800 BTUs of post-cooling energy to reach the desired space neutral upper limit of 80°F

Notes:
1. Total tons based upon conditioning outdoor air from 94°F and 100 grains to 87°F and 53 grains (see diagram DH/ER Hybrid-2).
2. Latent tons based upon reducing the humidity content of the outdoor air from 130 grains to 70 grains.
3. Estimated selling price of equipment based upon Climate Changer module pricing in 1996 dollars and estimates for the required SEMCO dehumidification and energy wheel modules.
4. As per the Phase 1 work, it is assumed that the preconditioned outdoor air needs to be provided dehumidified and at a space neutral temperature. This DH/ER hybrid approach requires the post-cooling energy shown in this column to reach the targeted 70–80°F delivery condition.

When the dehumidification–total energy recovery hybrid (Table 10) is compared in a similar fashion, the advantage of this approach over conventional cooling is very clear. First, it allows for much drier air than is possible with conventional cooling approaches. Second, the cost per latent ton removed is less. And finally, the cost of operation is much lower, and the system provides for time total energy recovery in winter. It appears a clear winner.

5.2 COMPARING SYSTEMS BASED ON COST PER TON OF LATENT AIR CONDITIONING

Tables 11 and 12 show the difference in estimated cost per ton for the three approaches discussed at different design conditions and for both total and latent output. Table 11 compares the conventional overcooling and reheating approach with a DBC active desiccant system. Table 12 compares the conventional cooling approach with the dehumidification–total energy recovery hybrid approach.

Table 11 shows that the cost advantage associated with total cooling tons provided disappears for the conventional cooling approach when the systems are compared based on latent cooling output. Table 12 shows how the dehumidification–total energy recovery hybrid results in a significant cost advantage over the conventional cooling system when they are compared based on latent capacity.

Table 11. Comparison of the estimated first cost based on both total and latent cooling tonnage provided: conventional overcooling vs DBC approach
(based on a 7500 cfm outdoor air preconditioning system)

Design conditions outdoor air (grains delivered from system)	Estimated first cost selling price/ total cooling tons provided		Estimated first cost selling price/ latent cooling tons provided	
	Overcooling	DBC approach	Overcooling	DBC approach
94°F and 100 grains (typical dry bulk BIN design condition)	$644	$1884	$1577	$1510
84°F and 130 grains (typical dew point design condition)	$652	$1417	$1190	$1184

Notes: 1. Conventional cooling approach assumes the cost of chiller and cooling tower at $500/ton.
2. System costs estimates from Table 2 from Phase 1 work.
3. Based on supplying outdoor air dehumidified to 53 grains and 70 grains, respectively, as in Figs. A.6 and A.7.

Table 12. Comparison of the estimated first cost based on both total and latent cooling tonnage provided: conventional overcooling vs dehumidification–total energy recovery hybrid
(based on a 7500 cfm outdoor air preconditioning system)

Design conditions outdoor air (grains delivered from system)	Estimated first cost selling price/ total cooling tons provided		Estimated first cost selling price/ latent cooling tons provided	
	Overcooling	DH/ER hybrid	Overcooling	DH/ER hybrid
94°F and 100 grains (typical dry bulk BIN design condition)	$559	$1119	$1518	$1291
84°F and 130 grains (typical dew point design condition)	$625	$1379	$987	$839

Notes: 1. Conventional cooling approach assumes the cost of chiller and cooling tower at $500/ton.
2. System costs estimates from Table 2 from Phase 1 work.
3. Based on supplying outdoor air dehumidified to 45 grains as in Figs. A.7 and A.8.

5.3 COMPARING SYSTEM OPERATING EFFICIENCY BASED ON LATENT COOLING CAPACITY

It is clearly difficult (if not inappropriate) to attempt to compare the operating efficiency (i.e., COP) of a conventional cooling approach with that of an active desiccant system. There are many factors to consider, and the kW/ton efficiency of conventional cooling equipment varies widely based on operating conditions and equipment selected. Nevertheless, a quick, simplified comparison of the systems discussed based on latent capacity produces an interesting end result.

Figure A.2 summarizes the latent COP (latent BTU out/BTU input) for the conventional cooling, DBC and dehumidification–total energy recovery hybrid systems. This analysis is based upon the dew point design condition of 84°F and 130 grains; it assumes an electrical power plant and transmission efficiency of 45% and assumes direct gas-fired regeneration of the active desiccant systems.

The results shown are surprising in that the calculated latent COP value for the DBC approach is 70% of that calculated for the conventional overcooling approach, and the COP value calculated for the dehumidification–total energy recovery hybrid is actually higher than that of the conventional cooling approach.

5.4 KEY CONCLUSIONS BASED ON PHASE 2 WORK

The more thorough analysis conducted in Phase 2 indicates that the markets initially targeted by the Phase 1 market evaluation—including hospitals, nursing homes/assisted living centers, research facilities, hotels/dormitories, retail stores and school facilities—continue to provide a significant opportunity for a cost-competitive active desiccant–based outdoor air preconditioning system. The estimated annual sales potential for these market segments alone is in excess of $82 million.

All of this potential could be satisfied by a product offering consisting of three system configurations (Figs. 1–6) produced from current Trane Climate Changer modules combined with SEMCO active and passive desiccant wheel cassettes. If a cost-effective active desiccant module could be successfully developed to couple with the Trane Voyager packaged rooftop cooling units, approximately 30% of this identified potential market—plus an additional $15 million attributable to the restaurant market segment—could likely be served with this equipment.

The engineering community is beginning to understand the benefits provided by decoupling outdoor air latent loads from conventional HVAC equipment in their designs. Trane and other major HVAC manufacturers are beginning to market dual-path overcooling with compressorized reheating and various total energy recovery preconditioning systems. Their interest simplifies the task of gaining market acceptance for active desiccant approaches. Other trends—including revised dew point weather data, utility deregulation, growing awareness of humidity control, and the rise in childhood asthma—are all strong market drivers favoring active desiccant solutions.

The needs of the targeted markets can be effectively met by three system configurations: the DBC, dehumidification–total energy recovery hybrid, and dehumidification-only approaches. Since the dehumidification–total energy recovery hybrid configuration is a combination of the current total energy recovery preconditioner and the dehumidification-only system, only two new configurations are actually required. It is thought that minimizing the number of available options will limit confusion and the potential for misapplication and will simplify sales support, allowing the sales engineers to remain focused on the benefits of this new technology.

The feedback from the market continues to support a compact product optimized to provide the most latent cooling capacity (pounds of moisture removal) for a given system first cost. Most projects will involve systems of under 10,000 cfm, a size where the active desiccant system is least competitive with conventional alternatives, further emphasizing the need for a cost-driven design minimizing the $/ton latent cooling ratio. A significant portion of the targeted market would be best served by a product that can be regenerated at temperatures between 200 and 250°F (maximum). Based on these and other conclusions reached based on market feedback, the following key design objective was determined:

> *Design the most compact active desiccant system(s) possible, maximizing the latent cooling (grain reduction) vs. cost ratio, utilizing a dehumidification wheel capable of removing approximately 60 grains of moisture from outdoor air having a relative humidity of 80–90% (dew point design data) and using a regeneration temperature in the range of 180 to 225 °F.*

The 60-grain removal goal is based upon the new ASHRAE dew point design data of 84°F and 130 grains. A 60-grain reduction would deliver air at 70 grains, or a 57°F dew point, which is on par with the humidity level typically delivered by packaged equipment and low enough to maintain most spaces at 60% RH. At more moderate outdoor air humidity levels, or with the dehumidification–total energy recovery hybrid approach, much lower grain levels can be delivered.

It was concluded that in order to provide for the most cost-effective active desiccant solution, traditionally high wheel face velocities would be required through the process side of the dehumidification wheel. At 600 ft/min, the supply airflow through the dehumidification wheel would exceed 80% of the rated capacity of the Climate Changer modules, resulting in a cost-effective system solution, one of the key design criteria.

Testing of the SEMCO 1M wheel module suggests that the 60-grain reduction performance goal at the 600 to 650 ft/min process face velocity should be achievable with the incorporation of a purge section and other modifications to the wheel cassette and seals. Modeling has predicted very respectable COPs for the most promising active desiccant system approaches even at the high face velocities (see Figs. A.5–A.8). When actual dehumidification wheel test data are used to revise the operating cost estimates completed as part of the Phase 1 work, the active systems look more attractive than previously thought because of an approximate 5% reduction in energy consumption required by the active systems (see Figs. A.1–A.4).

It was determined that the increased latent load associated with the new ASHRAE weather data had both a positive and negative effect on active desiccant systems. On the plus side, the greater humidity levels and higher inlet relative humidities result in an increased latent capacity for a given system. On the down side, the greater the grain reduction required, the greater the regeneration energy required and the greater the heat of adsorption. The result is a much higher temperature leaving the dehumidification wheel than would be associated with the dry bulb design condition (Fig. A.5 vs Fig. A.6).

At the same time that the dew point design condition produces a much higher temperature after the dehumidification wheel, it unfortunately also provides less post-cooling capacity because the outdoor air condition is near saturation, rendering the indirect evaporative cooling side of the system ineffective (see Fig. A.6). As a result, post-cooling after an active desiccant system is unavoidable. It is therefore concluded that trading a slight increase in post-cooling requirement for a significant reduction in first cost of the overall system (by using a small dehumidification wheel, for example) is appropriate.

Finally, given the intended use of this product, to provide latent air conditioning, it appears to be inappropriate to compare it with other, more conventional approaches based on total cooling capacity. It should be rated based on tons of latent cooling capacity provided. Likewise, the COP should be based on the latent cooling output/energy input.

Tables 8–13 show that when the conventional overcooling approach is compared with the most promising active desiccant system, based on latent cooling capacity provided, the active desiccant systems are cost competitive and provide an opportunity for significant energy savings along with the advantage of delivering drier air (dew points lower than obtainable with conventional cooling systems).

The conclusion regarding the cost competitiveness of the active systems is contingent upon using relatively high wheel face velocities, matching the supply and regeneration wheel areas with the flow configuration of a traditional air handling system (i.e., a 60/40 split), and using mass-produced, pre-engineered air handling modules such as the current Trane Climate Changer offering.

The projected selling prices for the most promising active desiccant systems described by this report and built from Climate Changer modules are approximately 40% lower than those of active desiccant systems currently on the market. The projected selling prices are also 30% lower than those of compressorized packaged overcooling/reheating units currently marketed by some Trane offices (based on a 5000-cfm system).

Given the trend in the HVAC market toward improved humidity control, the sizable sales opportunities identified for active desiccant systems, and the significant reduction in cost (over equipment currently marketed) made possible by the use of a mass-produced Climate Changer type product, it appears likely that one or more of the major HVAC equipment manufacturers will choose to offer active desiccant systems in the near future.

Table 13. Comparison of system latent cooling COP based on dew point design data
(based on outdoor air conditions of 84°F and 130 grains)

Supply air humidity content (grains delivered from system)	Conventional overcooling with reheat (COP)	DBC approach (COP)	Dehumidification–total energy recovery hybrid (COP)
70 grains	1.1	0.76	—
45 grains	1	—	1.2

Notes: 1. Assumes 0.6 kW/ton for conventional cooling to 70 grains, 1 kW/ton for 45 grain delivery.
2. Assumes power generation and transmission efficiency of 35% from electrical power plant.
3. COP calculated by dividing the latent cooling output by the energy input to run chiller or regenerate the desiccant wheel.
4. Fan heat, pressure loss through the systems, pump or cooling tower energy are not considered in this analysis.

6. REFERENCES

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APPENDIX A

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) 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	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				
				Subtotal Energy Cost	$19,686	$81	$970	$13,127

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

Fig. A.1. Annual operating cost estimate for the 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

Note 1: Assumes parasitic loss for all components in the system as well as an average 0.3 KW/ton for operating the chilled water pumps and the cooling tower.

Fig. A.1 (continued)

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 & 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	514080	548026		$1
97	74	20	36.8	85.9	11037600	11346425		$24
92	74	135	36.9	94.0	79606800	76842681		$164
87	72	367	35.2	90.9	202539960	188859547		$447
82	70	612	33.5	88.3	337750560	289014233		$746
77	69	839	32.7	91.2	494741520	384218682		$1,022
72	67	1201	31.2	89.2	708205680	501543397		$1,463
67	60	986	26.2	65.0	581424480	304007147		$1,201
62	57	845	24.3	60.6	530220600	229194617		$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	$9,928	$529	$10,666

Cooling Season Energy Cost	**$29,246**	**Total Annual Energy Cost Estimate**
Heating Season Energy Cost	**$5,096**	**for Desiccant Dehumidification/**
Demand Charges (Cooling Season Only)	**$1,056**	**Total Recovery Hybrid $35,398**

Fig. A.2. Annual operating cost estimate for the dehumidification/total energy recovery hybrid 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			

Note 1: Assumes parasitic loss for all components in the system as well as an average 0.3 KW/ton for operating the chilled water pumps and the cooling tower.

Fig. A.2 (continued)

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)

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	73	1	35.9	72.2	631333	15084		$1
97	74	20	36.8	85.9	19776763	1046618		$30
92	74	135	36.9	94.0	159313300	8734600		$202
87	72	367	35.2	90.9	381303024	10005738		$550
82	70	612	33.5	88.3	548010383	0		$918
77	69	839	32.7	91.2	770542900	0		$1,258
72	67	1201	31.2	89.2	1161104647	0		$1,801
67	60	986	26.2	65.0	584773727	0		$1,479
62	57	845	24.3	60.6	346403704	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				
				Subtotal Energy Cost	$17,822	$99	$970	$13,127

Cooling Season Energy Cost	**$25,428**	**Total Annual Energy Cost**		
Heating Season Energy Cost	**$6,590**	**Traditional Desiccant-Based**		
Demand Charges (Cooling Season Only)	**$1,599**	**Cooling Approach**	**$33,618**	

Fig. A.3. Annual operating cost estimate for the traditional desiccant-based cooling approach, modified to reflect the performance of the SEMCO 1M wheel.

(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	123.5	50	170	63.8	111.5	76	
85.9	97.0	36.8	132.1	50	210	63.8	118.4	77	
94.0	92.0	36.9	135.0	50	230	63.8	120.7	78	
90.9	87.0	35.2	126.3	50	210	63.8	113.8	76	
88.3	82.0	33.5	117.9	50	190	63.8	107.1	75	
91.2	77.0	32.7	115.3	50	190	63.8	105.0	74	
89.2	72.0	31.2	109.7	50	190	63.8	100.5	73	
65.0	67.0	26.2	84.2	50	135	63.8	80.1	68	
60.6	62.0	24.3	74.1	50	110	63.8	72.0	66	
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

Note 1: Assumes parasitic loss for all components in the system as well as an average 0.3 KW/ton for operating the chilled water pumps and the cooling tower.

Fig. A.3 (continued)

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 & 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	362880	490026		$1
97	74	20	36.8	85.9	8769600	10375635		$24
92	74	135	36.9	94.0	69400800	71567054		$164
87	72	367	35.2	90.9	188667360	178782490		$447
82	70	612	33.5	88.3	314616960	272712316		$746
77	69	839	32.7	91.2	463027320	361652670		$1,022
72	67	1201	31.2	89.2	662807880	470226698		$1,463
67	60	986	26.2	65.0	544153680	279105993		$1,201
62	57	845	24.3	60.6	498279600	207635354		$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	$12,340	$9,263	$529	$10,666

Cooling Season Energy Cost	**$27,702**	**Total Annual Energy Cost Estimate**	
Heating Season Energy Cost	**$5,096**	**Estimate for Desiccant Dehumidification/**	
Demand Charges (Cooling Season Only)	**$1,056**	**Total Recovery Hybrid**	**$33,853**

Fig. A.4. Annual operating cost estimate for the dehumidification/total energy recovery hybrid approach, modified to reflect the performance of the SEMCO 1M wheel.

(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	97.7	50	150
70.0	80.3	30.2	99.0	50	155
72.0	79.1	30.2	99.5	50	160
71.2	77.9	29.8	97.6	50	155
70.6	76.7	29.4	95.6	50	150
71.3	75.5	29.3	95.0	50	150
70.8	74.3	28.9	93.1	50	145
65.0	73.1	27.7	88.1	50	140
63.9	71.9	27.2	86.4	50	140
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			

Note 1: Assumes parasitic loss for all components in the system as well as an average 0.3 KW/ton for operating the chilled water pumps and the cooling tower.

Fig. A.4 (continued)

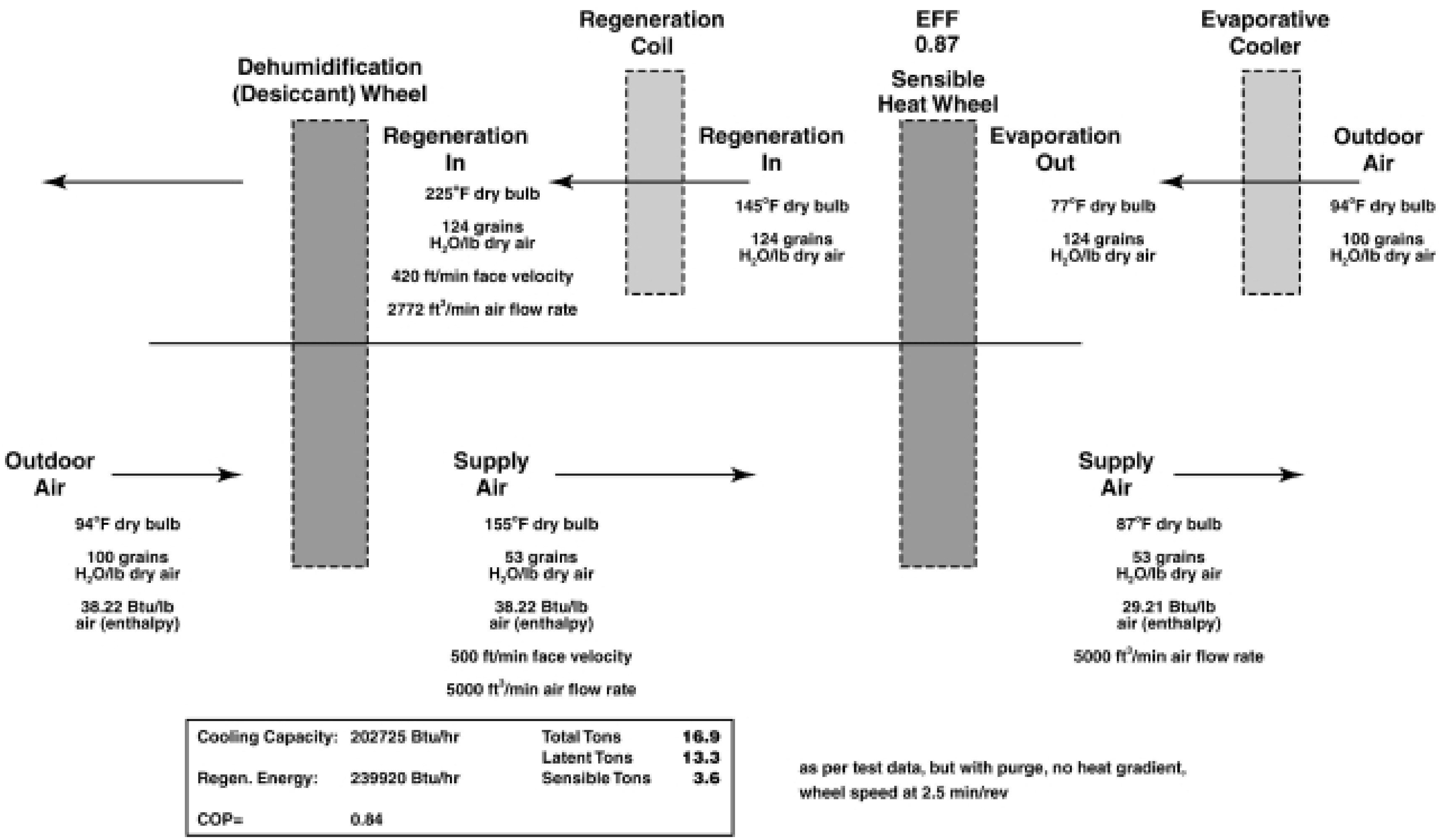


Fig. A.5. Flow schematic of typical active desiccant system performance for dry bulb BIN design condition.

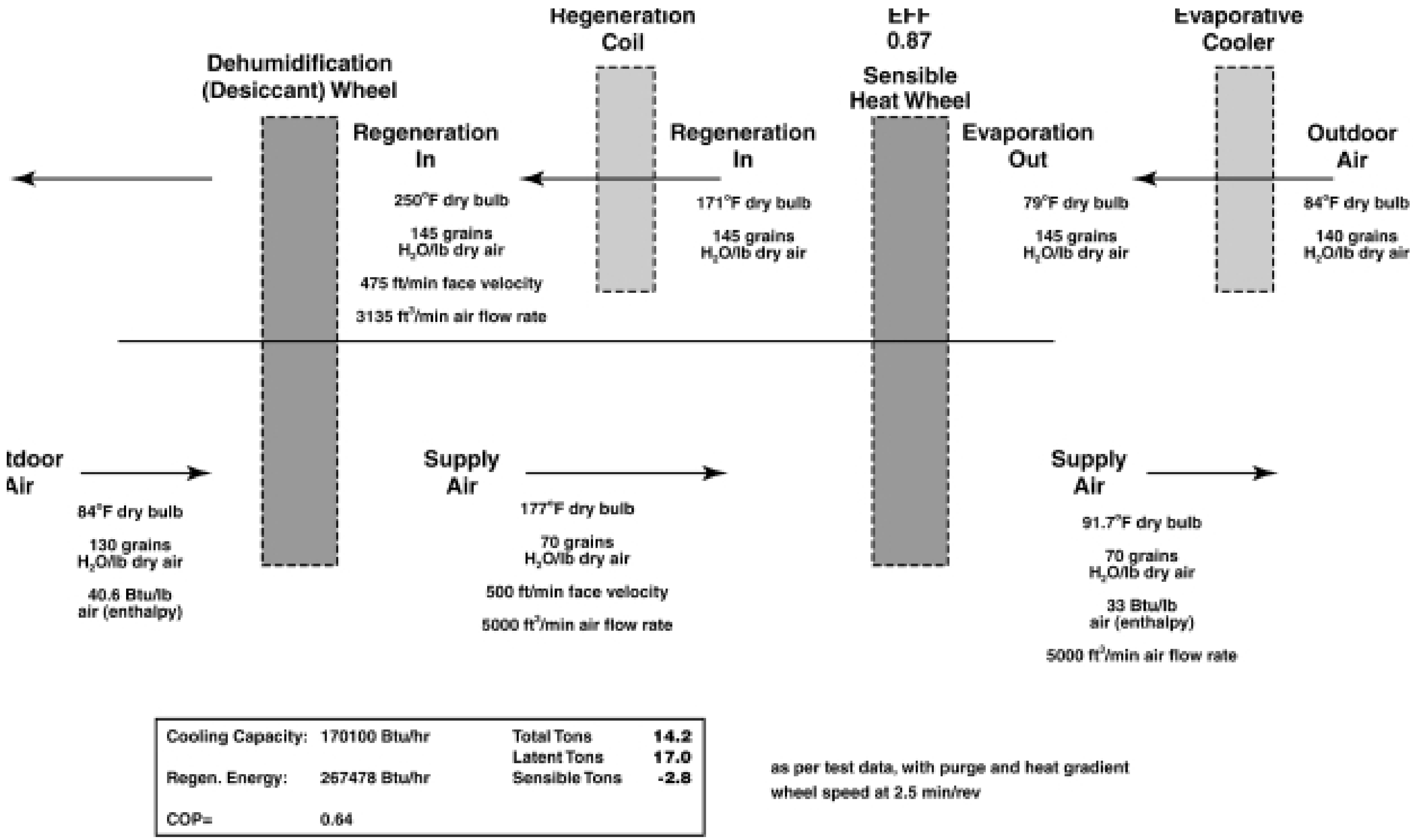
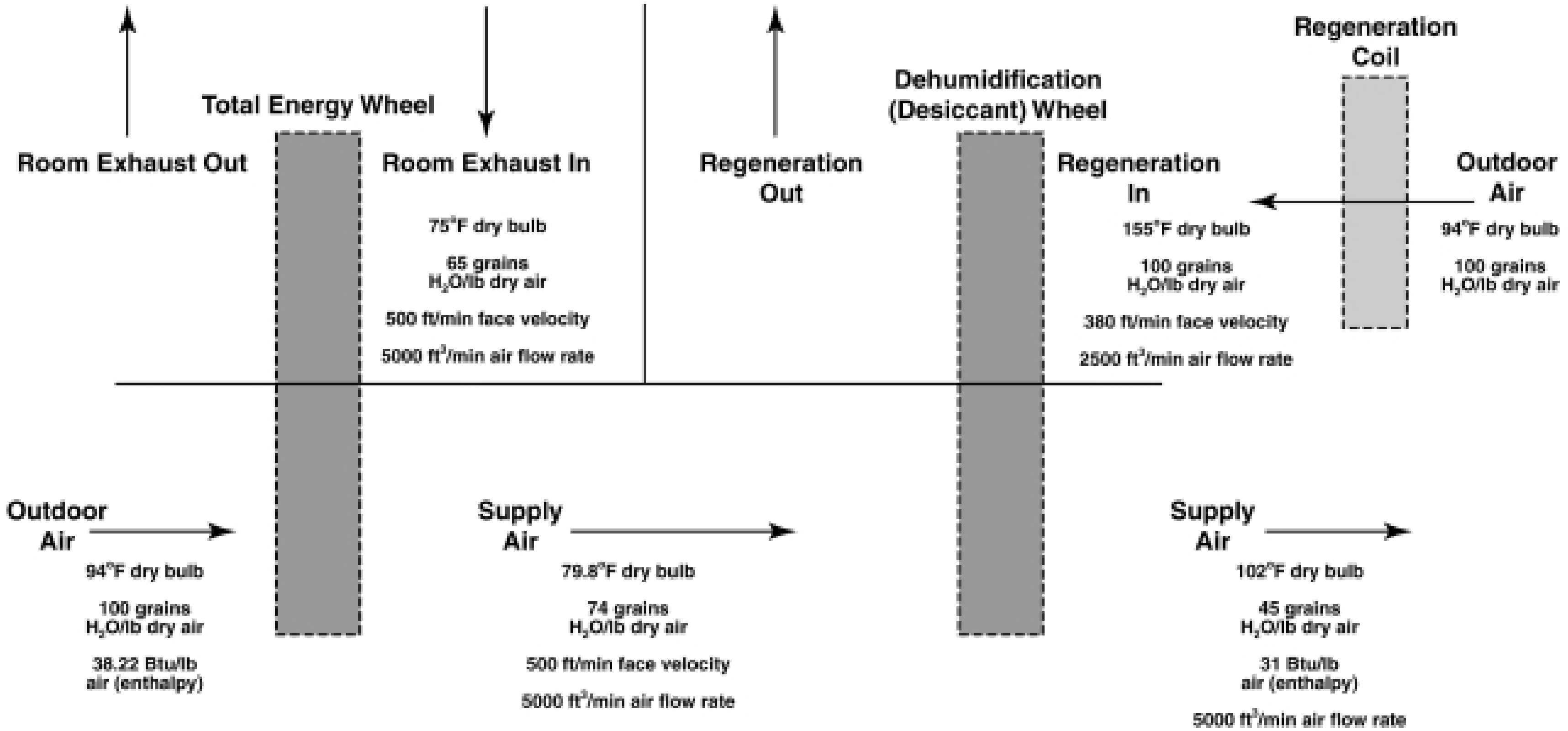
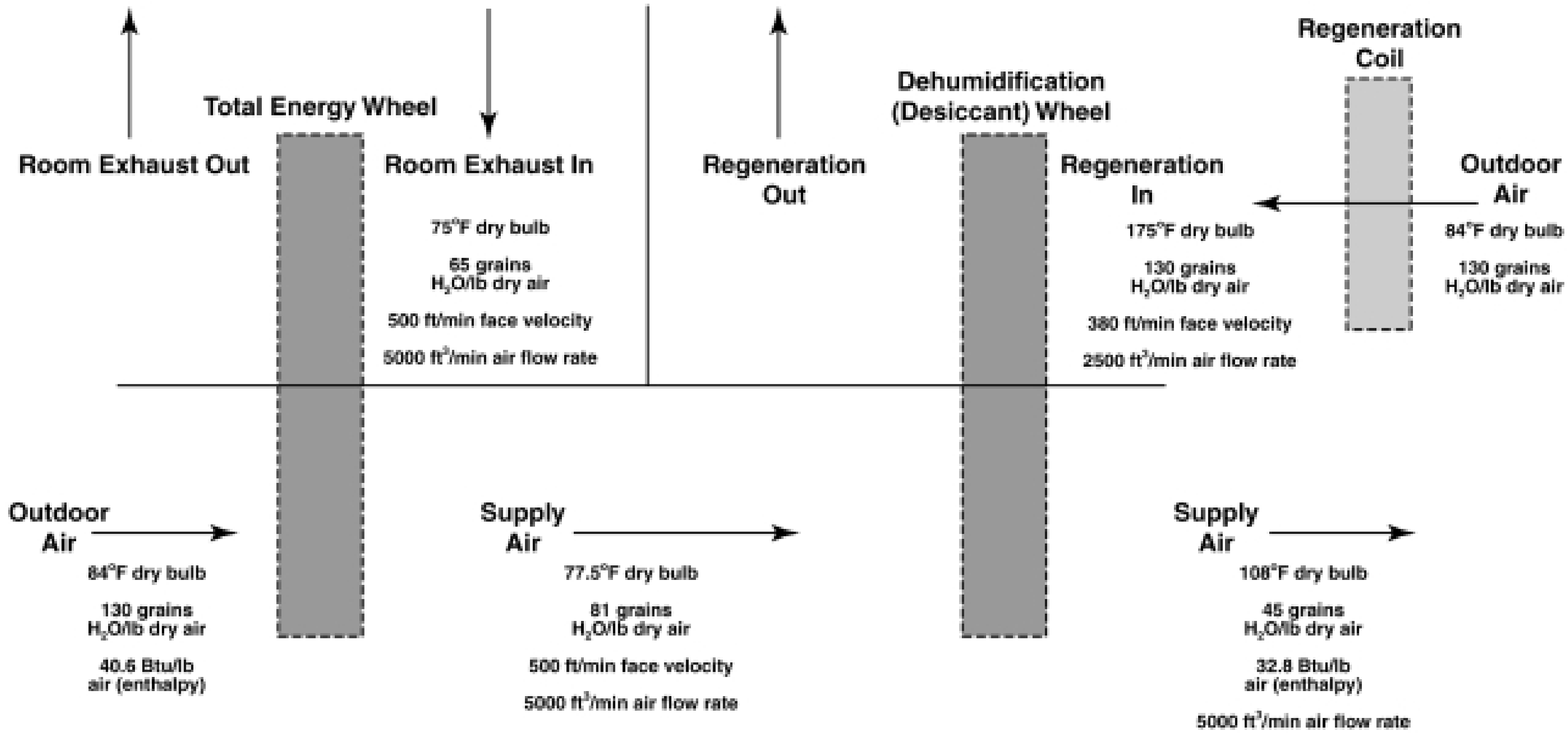


Fig. A.6. Flow schematic of typical active desiccant system performance for dew point design condition.



Cooling Capacity:	162450 Btu/hr	Total Tons	**13.5**
		Latent Tons	**15.6**
Regen. Energy:	164700 Btu/hr	Sensible Tons	**-2.0**
COP=	0.99		

Fig. A.7. Flow schematic of dehumidification–total energy recovery system performance for dry bulb BIN design condition.



Cooling Capacity:	175500 Btu/hr	Total Tons	**14.6**
		Latent Tons	**24.1**
Regen. Energy:	245700 Btu/hr	Sensible Tons	**-9.5**
COP=	0.71		

Fig. A.8. Flow schematic of dehumidification–total energy recovery system performance for dew point design condition.

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