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TECHNOLOGY REVIEW AND AVAILABILITY OF LIQUID DESICCANT SYSTEMS

Task 7

Prepared by:
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DESICCANT DEHUMIDIFICATION TO SUPPORT ENERGY EFFICIENT SPACE CONDITIONING SYSTEMS FOR HAWAII
PROJECT PHASE 1: DESIGN STUDY AND PROJECT SITE SELECTION

Project Deliverable No. 1/a:

"Technology Review and Availability Assessment of Liquid Desiccant Systems"

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ABBREVIATIONS

AC  Air-conditioning
AHU  Air-handling Unit
ASHRAE  American Society of Heating, Refrigerating, and Air-Conditioning Engineers
BRR  Benefit to Risk Ratio
CHP  Combined Heat and Power
CA  conditioned air (CA)
COP  Coefficient of performance
DP  Dew point
DX  Direct Expansion Air Conditioning System
DEAC  Direct Evaporative Air Coolers
DEC  Direct evaporative cooler
DEVAP  Desiccant Enhanced Evaporative Air Conditioning
EER  Energy Efficiency Ratio Engelard Titanium Silicate (ETS)
EAC  Evaporative Air Cooling external source of energy (Q)
GTI  Gas Technology Institute
HNEI  Hawaii Natural Energy Institute
HR  Humidity Ratio
HVAC  Heating Ventilation Air Conditioning
IC  Internal Combustion
IEAC  Indirect Evaporative Air Coolers
LDDX  Liquid Desiccant DX System
IDECOAS  Indirect and Direct Evaporative-Cooling-Assisted Outdoor Air System
IRR  Internal Rate of Return
LD  Liquid Desiccant
LDAC  Liquid Desiccant Air-Conditioning
LDDX  Liquid Desiccant Direct Expansion
IEC  Indirect Evaporative Cooler
LiCl  Lithium Chloride
LCST  Lower Critical Solution Temperature
O&M  Operations & Maintenance
OA  Outdoor (or outside) Air
ABBREVIATIONS

PV  Photovoltaic System, also solar-PV Power System
RH  Relative Humidity
SHR Sensible Heat Ratio
SDC Sustainable Design & Consulting LLC
SI  International System of Units
VAC Vapor-Compression Air Conditioning
WBT Wet-bulb temperature
VAV Variable Air Volume
VLI Ventilation Load Index

UNITS

atm  Atmosphere pressure
BTU  British Thermal Unit
BTUH/sqft  BTU per square feet
°C  Degree Celsius
CFM  Cubic feet per minute
°F  Fahrenheit
K  Kelvin
kW  kilo Watt
kJ  kilo Joule
MPa  Mega Pascale
square feet  Square Feet
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Appendix F - Selected technical information - Menerga Apparatebau, GmbH
SECTION 1 - EXECUTIVE SUMMARY AND OVERALL FINDINGS

This report, “Project Task 1: Technology Review and Availability Assessment of Liquid Desiccant Systems” is the Deliverable 1/A of the project “Desiccant Dehumidification to Support Energy Efficient Space Conditioning Systems for Hawaii – Project Phase 1: Design Study and Project Site Selection”. Sustainable Design & Consulting (SDC), LLC is performing the work under contract (RCUH P.O. #Z10143891) for Hawaii Natural Energy Institute (HNEI).

It is a technical review of state-of-the-art liquid desiccant applications. The review identifies availability of liquid desiccant (LD) technology for a future pilot HVAC installations which will be installed in Phase 2 of the project. This report also identifies and ranks companies that are active in liquid desiccant development, sales and maintenance, to be part of the cooperative project in Part 2, the pilot HVAC installation.

1.1 Approach of the Technology Review and Availability Assessment

This report provides an overview of important liquid desiccant processes and technologies which are the basis for the design of the planned pilot HVAC installation of a desiccant dehumidification and cooling system in the Project Phase 2.

Section 2 of this report starts with the discussion of important basic aspects of dehumidification and describes differences between technologies, including conventional cooling based dehumidification and solid and liquid desiccant dehumidification. Improving humidity control in buildings is gaining more importance. As the building envelope and the equipment inside the building become more energy efficient, the sensible thermal load is reduced and therefore that portion of the latent load increases. The conventional method of controlling HVAC is by temperature set-points, a method that only relies in the sensible load. In applications where the sensible heat index is small, latent load cannot be removed in an optimum manner. This requires a new approach to space conditioning where the control of sensible and latent loads must be separated so that they can be independently controlled.

Section 3 presents important aspects of liquid desiccant (LD) dehumidification for application in HVAC. While liquid desiccants are important for specific industrial processes, the use of liquid desiccants for the conditioning and/or cooling of buildings is not yet a widely-used technology in HVAC for buildings. The section discusses important design principles that have been under investigation and in active development for a longer time.

Section 4 draws builds on the conventional liquid desiccant system technologies presented in Section 3. Section 4 discusses shortcomings of conventional liquid desiccant dehumidification technology for their use in HVAC. Newer developments of LD technologies are presented that include the so-called low flow technology. This newer technology avoids problems that “older” liquid desiccant technologies have regarding using liquid desiccant in HVAC. The improvements include simplifying the design, making operation of the LD system more energy efficient, and solving major operational challenges. The section also points out further developing needs to make liquid desiccants an even better fit for HVAC applications.
Section 5 describes the liquid desiccant technologies and experiences of the seven candidate companies. The seven candidate companies were selected after an initial literature search. One company of the seven initially selected candidate companies was found to have gone out of business recently. Each company had some unique liquid desiccant experience and proprietary technology that was deemed important for the planned pilot HVAC installation in Hawaii.

Table 5.1.1: Initial selection of candidate companies

<table>
<thead>
<tr>
<th>No</th>
<th>Company</th>
<th>Country</th>
<th>Type of Products</th>
<th>Status of company</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7 AC Technologies</td>
<td>US</td>
<td>Start-up company that develops membrane base liquid desiccant dehumidification; product is based on membrane technology developed by NREL</td>
<td>Start-up company, has been in business for less than 5 years</td>
</tr>
<tr>
<td>A</td>
<td>Advantix Systems</td>
<td>US</td>
<td>Innovative solutions for liquid desiccant products;</td>
<td>Company closed and went out of business</td>
</tr>
<tr>
<td>2</td>
<td>AIL Research</td>
<td>US</td>
<td>A wide range of liquid desiccant products and technologies</td>
<td>Company has been in business for more than 10 years and is a recognized leader in innovative technology development</td>
</tr>
<tr>
<td>3</td>
<td>Be Power tech</td>
<td>US</td>
<td>Start-up company that develops membrane base liquid desiccant dehumidification; product is based on membrane technology developed by NREL. Adds fuel cell as means to provide thermal energy for regeneration</td>
<td>Start-up company, has been in business for less than 5 years</td>
</tr>
<tr>
<td>4</td>
<td>Kathabar Dehumidification Systems, Inc</td>
<td>US</td>
<td>Large range of desiccant dehumidification products which are installed in various industrial dehumidification applications</td>
<td>US internationally active company that is in business for several decades</td>
</tr>
<tr>
<td>5</td>
<td>L-dcs</td>
<td>Germany</td>
<td>Providing integrated and energy saving HVAC solutions. Included are liquid desiccant technology</td>
<td>German company has been in business for more than 10 years and is a recognized leader in innovative technology development</td>
</tr>
<tr>
<td>6</td>
<td>Menerga Apparatebau, GmbH</td>
<td>Germany</td>
<td>Large range of HVAC technologies that provide advanced technology solution for building AC and comfort</td>
<td>German internationally active company that is in business for 30 years.</td>
</tr>
</tbody>
</table>

Section 6 describes a ranking systematic of the seven candidate companies. A two-tier ranking methodology was used to identify the prospect of a company to be a strong partner in the planned pilot HVAC project, with aspects considered including applicability and flexibility of their proprietary liquid desiccant technology, their experience in HVAC applications, their technical and design support potential and their willingness to make significant contributions to the project.
1.2 Results Obtained

The six remaining candidate companies were ranked using 14 criteria, which were grouped in the following four categories, where the four categories were given different weights:

- Technology maturity (overall weight 25%)
- Prior installation/application experience of technology (overall weight 20%)
- Technology flexibility / ability to implement pilot HVAC installation (overall weight 40%)
- Communication / willingness to cooperate substantially (overall weight 15%)

Companies were evaluated against each of the 14 ranking criteria. A ranking score point score, in percentage, was determined for each criterion and the sum of all score points determined the overall ranking of the company. The higher the total score, the better the company was considered ready to contribute to the planned pilot HVAC installation.

The result of the ranking was as follows:

<table>
<thead>
<tr>
<th>Company</th>
<th>Total score</th>
<th>Rank</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 2 - AIL Research Inc.</td>
<td>96%</td>
<td>1</td>
</tr>
<tr>
<td>No. 5 - L-dcs GmbH</td>
<td>90%</td>
<td>2</td>
</tr>
<tr>
<td>No. 1 - 7 AC Technologies</td>
<td>62%</td>
<td>3</td>
</tr>
<tr>
<td>No. 6 - Menerga Apparatebau, GmbH</td>
<td>60%</td>
<td>4</td>
</tr>
<tr>
<td>No. 4 - Kathabar Dehumidification Systems</td>
<td>56%</td>
<td>5</td>
</tr>
<tr>
<td>No. 3 - Be Power Tech</td>
<td>37%</td>
<td>6</td>
</tr>
</tbody>
</table>

AIL Research and L-dcs were found to be the two companies with the most promising liquid desiccant technology for the planned pilot HVAC installation, which will be installed in Part 2 of the project.

1.3 Main Conclusions and Findings

The major findings of this report are as follows:

1. Improved humidity control in buildings becomes increasingly important as the typical thermal loads in building shift from predominately sensible to gradually more latent. The conventional HVAC technology uses temperature set points to control the total thermal load removal from the conditioned spaces. As the sensible thermal loads decrease due to more energy efficiency in envelope, lighting and equipment, more humidity enters and less humidity is rejected from the conditioned space. Humidity related indoor environmental impacts are logical consequences. Therefore, precise humidity control of conditioned spaces requires new HVAC approaches. Separate controls and rejection of sensible and latent loads is a key approach. Liquid desiccant dehumidification is a suitable technology for advanced humidity control and can provide more benefits as the conventional cooling based dehumidification.
2. Indoor comfort in the Hawaii environment does not require the cold temperatures that are common in conventional HVAC systems with cooling based dehumidification. This conventional HVAC method of reducing humidity from the indoor recirculated and fresh outside air is based on cooling the air below dew point temperature chilled coils and which causes condensation on the cooling coils. To avoid overheating spaces, additional energy has to be used to reheat the supply air, a method that is very energy intensive. This has negative occupant comfort implications. Since liquid desiccant dehumidification does not require cold process temperatures, overcooling can be avoided.

3. Conventional liquid desiccant technology has had a long and successful record of providing precise humidity control for specific industrial processes. All equipment and process issues that are part of these established liquid desiccant industrial applications have been thoroughly optimized.

4. Conventional liquid desiccant technology that was optimized for industrial applications does not automatically perform well in HVAC applications in commercial or residential buildings for several reasons. A major barrier to use liquid desiccant technologies for HVAC systems is the high maintenance requirement for industrial liquid desiccant units. Desiccant solutions are typically corrosive and can affect equipment, and droplets can be entrained into the process air flow if impacts are not mitigated. Liquid desiccant equipment that will be successfully used in HVAC applications must be corrosion resistant, and effective measures must be applied to avoid desiccant droplet carryover to the supply air stream. In addition, LD technologies must be energy efficient and provide for low first and O&M costs.

5. Recent developments of the “low-flow” LD technologies provide significant performance improvement that will make LD technologies more suitable for building HVAC applications. Low-flow conditioners (e.g. liquid desiccant absorbers) avoid desiccant droplet entrainment into and conveyance by the supply air. Therefore low-flow conditioners can be regarded as “zero carry-over” equipment. In addition, low flow LD technology keeps the desiccant flow rate low, thus avoiding large pumping power. Energy storage in form of a concentrated desiccant solution requires only a fraction of the required desiccant solution, in comparison to conventional high-flow liquid desiccant technology.

6. Low-flow technology requires internally cooled conditioners and internally heated regenerators. Recent development of internal cooled wicking plate conditioners and regenerators have solved desiccant distribution and wetting characteristics inside the conditioners and regenerator. The low-flow conditioner and regenerator are significantly smaller than the conventional packed bed (or column) process vessels, which are used for industrial LD dehumidification. This results in smaller unit sizes which can be easier integrated into retrofitted HVAC systems.

7. Recent developments of liquid desiccant technology, such as low flow, reflect significant contributions that make liquid desiccant dehumidification and cooling a recognized and trusted component of energy efficient and high comfort HVAC applications. While current advances in
Technology Review and Availability Assessment of Liquid Desiccant Systems

SECTION 1 - EXECUTIVE SUMMARY AND OVERALL FINDINGS

Liquid desiccant based HVAC technology are significant, several liquid desiccant technology innovations have been identified that will increase the performance further. Major identified innovations include advanced desiccants that are less corrosive and offer good flow and wetting characteristics, multiple-effect regenerators that will increase the coefficient of performance (COP) and improving the efficiency of using solar heat for LD processes. Research needs are described in this report.

8. Seven companies with a strong knowledge base and previous track records of designing and manufacturing liquid desiccant systems for use in building HVAC applications were selected from an initially larger pool of possible companies. The initial selection of the seven companies was based on published technical papers as well as online searches and research. Of the seven companies, one was found to have recently ceased business operations. For the remaining six companies a detailed analysis was conducted of their products, completed system installations and their willingness to cooperate on the pilot HVAC project in Hawaii. The results of the research and analysis of the seven companies are presented in this report.

9. For the remaining six companies, a ranking of their performance relative to a set of selection criteria was performed. The ranking utilized a two-tier methodology, where the first tier defines the relative importance of a selection criterion and the second tier assesses the degree to which the company satisfies the criteria statements. The scores for each selection criteria are added to yield the total score.

10. The results of the ranking indicate that two small engineering and technology companies have the highest scores. Both companies offer valuable flexibility in adapting their low-flow liquid desiccant technology to the requirements of the pilot HVAC installation. Both companies have the required design and manufacturing knowledge to provide the relatively small liquid desiccant system components for the pilot HVAC installation.

11. The results of this report will be used to provide preferred liquid desiccant technology and design strategies for the subsequent phases of the project.
SECTION 2 – INTRODUCTORY REMARKS OF DEHUMIDIFICATION TECHNOLOGIES

This section discusses basic considerations of dehumidification technologies and their application in dehumidification for industrial and commercial processes. This section also describes why dehumidification is of increasing relevance in the building industry and why liquid desiccant dehumidification technology is an important solution to provide advanced humidity control in buildings.

2.1 Dehumidification

The process by which moisture or water vapor is removed from air is called dehumidification. Often moisture or water vapor in the air is referred to as humidity. All three terms (moisture, water vapor, and humidity) stand for the same principle, where water in gaseous form is mixed with dry air, creating moist air. The energy used to evaporate water contained in moist air mixture is referred to as enthalpy. The physical and thermodynamic properties and the changes in state of air-water vapor mixtures is described in the psychrometric chart. The ideal way of dehumidification would be to reduce humid air to the desired lower humidity level and a different temperature, either higher or lower, on a straight line. But typically, this is not possible with conventional dehumidification strategies, since the move from the start state to the desired state involves multiple changes in humidity and dry bulb temperatures, which means either cooling or heating.

2.2 Cooling Based Dehumidification

The most commonly used type of dehumidification is cooling based dehumidification, using mechanical or refrigerative dehumidifiers. This basic process uses the principle of drawing moist air over a refrigerated surface, most often cold coil bundles. By exposing the moist air to a temperature below the dew point condensation occurs and water is removed as drainage of the coils. The colder the cooling coil the more humidity can be removed. This process of dehumidification works most effectively with higher ambient temperatures with a high dew point temperature. In cold climates, the process is less effective. The practically achievable dew point of the moist air is limited by the effectiveness of the condensation process on the cooling coils. At very low temperatures of the coils, ice will form on the coil surfaces, thus inhibiting effective humidity removal. Therefore, the water vapor that condenses on the coils cannot be drained and an increasingly thick layer of ice layer forms an increasing barrier to effective heat transfer.

2.3 Desiccant Dehumidification

An alternative method of removing humidity from the air is by means of desiccants. The process of desiccant dehumidification differs significantly from cooling based from cooling-based dehumidifiers. Instead of cooling the air below the dew point and removing the water as condensate, desiccants attract moisture from the air by creating an area of low vapor pressure at the surface of the desiccant material. If the vapor pressure in the air is higher humidity migrates to the desiccant; if the vapor pressure in the air is lower the opposite occurs, when moisture migrates from the desiccant surfaces to the air. Consequently, dehumidification is the process when the vapor pressure in the air is higher, so the water molecules move from the air to the desiccant and the air is dehumidified.
SECTION 2 – INTRODUCTORY REMARKS OF DEHUMIDIFICATION TECHNOLOGIES

Desiccants are hygroscopic substances used as drying agents. They attract water-vapor molecules from the air using absorptive or absorptive processes. Two types of desiccant dehumidification processes are being used, solid (dry) and liquid desiccant dehumidification.

The type of desiccant selected for the dehumidification application depends on aspects such as the required degree of dehumidification, the available heat source for desiccant regeneration or specific building geometry. Dehumidification systems are more effective to achieve lower dew points than cooling based dehumidification. Desiccant dehumidification is typically used in industrial applications to ensure appropriate relative humidity levels. Desiccant wheel and liquid desiccant dehumidification systems are among the most common desiccant technology application used.

2.4 Comparison of Cooling Based and Desiccant Dehumidification

An essential requirement for air conditioning is the control of humidity in addition to controlling sensible loads. A basic shortcoming of conventional AC is that it cannot effectively meet the latent loads from ventilation on days with high humidity. Historically, conventional chillers and air conditioners are devices intended to primarily achieve sensible cooling and only secondly to dehumidify by lowering air temperature below its dew point so that moisture is removed from the air by condensation on the cooling coils. This type of dehumidification produces wet cooling coil and the air that leaves this coil will be close to saturation.

AIL RESEARCH points out that limitations of conventional chillers and direct expansion (DX) air conditioners become evident when cooling based dehumidification, such as displacement ventilation, chilled beams, and radiant panels are used in advanced HVAC systems. The benefits of these advanced HVAC systems include energy savings by significantly reducing fan energy used in a conventional system that recirculates large volumes of air. As AIL RESEARCH points out, these advanced systems will not work with a conventional chiller or DX air conditioner that supplies relatively cold air (e.g., 50 to 55°F) that is saturated with moisture (i.e., 100% RH). These systems will however work well with cooling systems, such as desiccant systems which supply drier, but warmer air.

Figure 2.4.1 illustrates the energy penalty of overcooling that is required by conventional AC to achieve low humidity levels. In the example depicted in Figure 1, the paths in the psychrometric chart of conventional AC, with cooling based dehumidification, and desiccant dehumidification are compared. For the example, the state points of the outdoor air and the target indoor supply air are 86/78 (DB/WB; 70% RH) and 65/50 (DB/RH), respectively. The conventional AC with cooling based dehumidification must reduce the outdoor temperature to condense the moisture below 46°F and then reheat the air to 65°F. With an assumed 6,000 cfm of warm, humid outdoor air, 54.0 tons of cooling are required to approach target dew point and then 10.8 tons of reheating. For the energy of cooling (e.g. sensible load removal) alone the conventional AC would require 25% more load removal than a desiccant system. AIL Research points out that this percent of excess cooling becomes much larger during cooler, damp weather, e.g., if it were 70°F and raining, overcooling would be 42% of the required cooling.
SECTION 2 – INTRODUCTORY REMARKS OF DEHUMIDIFICATION TECHNOLOGIES

2.5 Basic Desiccant Cycle

The driving force of desiccant dehumidification is the difference between water vapor pressures at the desiccant’s surface and of the surrounding air. Sorption of moisture occurs with a positive vapor pressure gradient at the desiccant surface, which means a higher vapor pressure in the surrounding air than at the exposed desiccant surface. Desorption occurs with an opposite pressure gradient, when the desiccant surface vapor pressure is higher than that of the surrounding air.

The nonlinear relationship between desiccant moisture content and vapor pressure at the desiccant surface is depicted in Figure 2.5.1. As the moisture content of the desiccant rises, so does the water vapor pressure at its surface. At some point, an equilibrium condition occurs, when the vapor pressure at the desiccant surface is the same as that of the surrounding air. At this point neither sorption nor desorption occurs, since the pressure difference has reached a minimum threshold, which is an insufficient driving force. At this point the desiccant must be regenerated to restore the desiccant’s ability for sorption of water vapor. Figure 2.5.1 illustrates the effect of temperature on vapor pressure at the desiccant surface. Both higher temperature and increased moisture content increase surface vapor pressure. This indicates that desiccants can attract moisture better at lower temperatures, since the pressure gradient between the surrounding air and the desiccant surface is larger. At higher
temperatures, the vapor pressure at the desiccant surface is higher, which results in less sorption or, in case of desiccant regeneration, in better desorption when moisture is driven from the desiccant to the surrounding air.

Figure 2.5.1 illustrates sorption and desorption cycles used in desiccant dehumidification. The cycle starts at (1) with a cold and dry desiccant coming into contact with a moist air stream. During process step (1) to (2) moisture is attracted and the moisture content of the desiccant increases. The desiccant surface temperature increases due to heat of sorption. As the desiccant enters equilibrium conditions at (2), sorption stops. During process steps (2) to (3) external heat is added and the vapor pressure at the desiccant surface increases, creating a pressure gradient to drive moisture out of the desiccant. At (3) the moisture content of the has fallen to a desired process condition. During process step (3) to (1) the desiccant is cooled to attain the lower vapor pressure of the initial state at (1), where a new cycle starts.

The effectiveness of desiccant cycle depends on the energy required to move through this cycle. The process step (1) to (2), when dehumidification occurs, requires only relatively small fan or pump energy to bring the desiccant and the air stream into physical contact. The process steps (2) to (3) and (3) to (1) requires significant heating and cooling energy, respectively.

![Desiccant Water Vapor Pressure as Function of Moisture Content and Temperature (ASHRAE, 2009)](image1)

![Desiccant Cycle (ASHRAE, 2009)](image2)

Figure 2.5.1: Relationship between desiccant moisture content and vapor pressure at the desiccant surface

During process step (2) to (3) heating requires the sum of the following heat input:

- Sensible heat input to raise the desiccant to a temperature high enough to make its surface vapor pressure higher than that of the surrounding air, the higher the
- Latent heat input to evaporate moisture
• Latent heat for desorption of water from the desiccant (a small amount)

During process step (3) to (1) cooling requires sensible heat rejection to the surrounding air, where the required cooling capacity is proportional to the desiccant mass properties and difference between its temperature at (3) and (1). Regeneration in the desiccant cycle also works with pressure differences. This is used in certain industrial and commercial applications but not in HVAC applications.

Summarizing, the greater the difference between the air and desiccant surface vapor pressures, the greater is the ability of the desiccant material to attract moisture from the air. The selection of the most effective desiccant for a particular application depends on the range of water vapor pressures in the air, the temperature of the regeneration process, and moisture sorption and desorption characteristics of the desiccant.

2.6 Liquid Desiccant Dehumidification Systems

Section 2.5. described the basic desiccant dehumidification process which is basically the same for solid and liquid desiccants. Section 2.6. takes a closer look at the liquid desiccant systems.

Liquid desiccant systems use liquid solutions as an absorbent and cooling material. In general, liquid desiccant can remove moisture from the air more efficiently and to a lower remaining humidity than dry (solid) desiccants. All liquid desiccant systems have two process vessels, a conditioner and regeneration process vessel. In the conditioner, concentrated liquid desiccant solution absorbs moisture from the incoming air and therefore dehumidifies the air. In the exothermic absorption process, heat is liberated which heats the desiccant solution and the surrounding air. As the desiccant absorbs more humidity, the solution becomes weaker, which means more diluted. The vapor pressure decreases as the desiccant gets warmer and less concentrated. Thus, the driving force to absorb more humidity, which means the differential of vapor pressures in the moist air and the desiccant interface, decreases. This completes the conditioning process cycle.

In order to be available for the next conditioning process cycle, the diluted desiccant solution must be regenerated by driving out humidity from the desiccant and thereby lowering the vapor pressure at the desiccant interface. The diluted desiccant solution is therefore heated, which increases its vapor pressure. Heating of the desiccant can either be externally or internally of the regenerator process vessel. When the heated desiccant solution is brought into contact with air that has a lower vapor pressure, transfer of water is initiated towards the lower partial pressure, thus towards the air. As parts of the air volume in the regenerator takes up the humidity, they are being replaced by fresh air flowing through the regenerator. The concentrated desiccant then flows back to the conditioning process vessel for reuse, and a new cycle of humidity absorption.

In the process outlined above, the desiccant solution is heated. The process dynamics also work when the scavenging air stream is heated instead of the desiccant. In addition, a combination of heating desiccant and the scavenging air is possible.
Figure 2.6.1 illustrates the basic process of a liquid desiccant system, by example of a packed column system. There are two process vessels, the conditioner and the regenerator vessel, each of which has its own desiccant and air flow configuration. The depicted configuration has a counter flow configuration. In the upper right corner a process diagram shows the relationship between the desiccant moisture content and its surface vapor pressure.

In the conditioner process vessel, process air, e.g. the air that is dried to the desired dew point (or vapor pressure), moves upwards. Strong desiccant solution is distributed on a packing of some sort. The packing contributes to the mixing of the desiccant solution as it flows downwards. Surfaces are continuously refreshed, which increases the gas transfer between the gaseous and the liquid phase of the air and the desiccant fluid. In the bottom well, the desiccant solution accumulates and is either circulated to the upper distribution plate in the packed column or a portion is pumped to the regenerator. While being recirculated, the desiccant is cooled in the external cooler.

In the regenerator, also a packed column in this case, the desiccant flows downward and encounters a scavenging air flow. As the desiccant is circulated through the regenerator cycle, the desiccant is heated outside the vessel. Due to a vapor pressure differential between the desiccant and the air, humidity is transferred from the desiccant to the air, thereby drying the desiccant and increasing the strength of the solution.

The thermodynamic properties of the desiccant are illustrated in three process steps (1) through (3):

1. ABSORPTION: As the humidity in the desiccant increases the desiccant surface temperature increases due to the absorption process.
2. DESORPTION upstream of the regenerator: The heated desiccant releases water vapor thereby losing humidity.
3. COOLING: The desiccant is cooled which decreases the vapor pressure.
2.7 Dry Desiccant Dehumidification Systems

This section briefly introduces dry, or solid desiccant systems. Since dry systems are not the subject of this investigation a brief introduction suffices. In the related dehumidification process, humidity is adsorbent while the desiccant doesn’t change phase as it collects moisture. Common types of dry desiccants include zeolites, silica gel and activated alumina.

While there are numerous configurations of solid desiccant dehumidification systems regarding geometry, air flow, regeneration etc., the desiccant wheel configuration is the most widely used system. The desiccant wheel is a rotating wheel, which contains desiccants that are fixed to structural elements. As the wheel slowly turns, it rotates through two streams of air. The first air stream, which represents the air supply, goes through one section of a desiccant-coated wheel, which absorbs moisture and dries the air. As the wheel rotates further, another section of the wheel is exposed to a regenerating air stream that dries and expels, or desorbs, the moisture that it collected from the process air. The moisture transfer occurs because of vapor pressure differences between the desiccant’s surface and the surrounding air. The desiccant traps moisture when the surface’s vapor pressure is lower than the vapor pressure of the passing air. The desiccant releases the moisture when the surface’s vapor pressure rises. The direction of the moisture transfer is the result of the difference between the process air stream’s relative humidity and the regeneration air stream’s relative humidity. AIL Research suggests that dry desiccant humidification systems are good for mixed air systems and cold storage areas that require warmer air or supply-air dew points to be below 50°F. Desiccant wheel dehumidification is also good for dedicated outdoor food storage units, so the air within them is less humid.
2.8 Applications of Liquid Desiccant Systems

For the past 70 years, liquid desiccants have been used in dehumidification, mainly for industrial and institutional applications. One US market leader, Kathabar Dehumidification Systems, Inc., has provided solutions to industrial, commercial, and institutional customers, which require reliable, precise, and economical control of temperature and humidity. The liquid desiccants used in these industrial applications are typically very strong solutions of ionic salts, such as lithium chloride and calcium chloride. These ionic salts have essential zero vapor pressure, and so vapors of the desiccant will not appear in the air supplied by the LDAC. The downside of lithium and calcium chloride solutions are that they are very corrosive, which requires that all wetted parts within the liquid desiccant systems be protected and that no droplets of desiccant are entrained by the supply air.

Typical industrial, commercial and institutional applications of liquid desiccant systems are as follows:

Art Galleries & Museums, Libraries and Archives & Record Storage:

Benefits of liquid desiccant applications include:
- Prevents degradation of stored pieces
- Inhibits color fading and paper decaying
- Eliminates mold growth
- Protects integrity and value
- Provides dust- and bacteria-free air
- Energy efficient operation reduces utility consumption

Source: Kathabar Dehumidification Systems, Inc

Coating, Cooling, Storage and Drying of Food Products

Benefits of liquid desiccant applications include:
- Maintains a clean, controlled temperature and humidity level
- Improves drying time
- Prevents agglomeration
- Energy efficient operation for reduced utility consumption
- Acts as an air scrubber to neutralize airborne microorganisms for bacteria free air
- Eliminates condensation
- Maximizes production rates by eliminating costly downtime

Source: Kathabar Dehumidification Systems, Inc
SECTION 2 – INTRODUCTORY REMARKS OF DEHUMIDIFICATION TECHNOLOGIES

Chemical Industry
Benefits of liquid desiccant applications include:
- Improves product quality
- Acts as an air scrubber to neutralize airborne bacteria, viruses, and mold
- Prevents condensation and corrosion
- Energy efficient operation for reduced utility consumption
- Eliminates the need for special refrigeration
- Maintains a clean, controlled temperature and humidity level
- Prevents products from sticking to equipment and one another

Source: Kathabar Dehumidification Systems, Inc

Cold Storage
Benefits of liquid desiccant applications include:
- Precise humidity control and air quality
- Operational improvements, less equipment required
- Improves operator safety
- Operational improvements; Avoid unscheduled maintenance Decreases fog, ice & defrost cycles
- Quality improvements; Extends storage life and improves product quality; Mold and fungus prevention / abatement; Sanitary air
- Energy efficient operation; reduces tonnage required through optimal use of refrigeration
- Cold storage applications; Warehouses and Freezers

Source: Kathabar Dehumidification Systems, Inc

Hospitals and Health Institutions
Precise temperature and humidity levels for general and critical care hospital zones. Benefits include:
- Captures and kills 94% of all microorganisms entrained within the airstream.
- Reduces operating costs through energy savings and simple maintenance
- Eliminates wet cooling coils that promote reproduction of microorganisms.
- Relative humidity can be easily controlled to 45% or lower, even at temperatures below 60ºF.
- Unit is simple and silent and operates continuously with very low maintenance.
- Waste heat utilization at hospitals that has a distributed power or cogeneration system,
- Captures and kills 94% of all microorganisms entrained within the airstream. Desiccant solution is bactericidal to most airborne organisms. Bacteria, mold and viruses are automatically neutralized and washed from both the conditioner and regenerator air streams.

Source: Kathabar Dehumidification Systems, Inc
2.9 The Increasing Importance of Dehumidification in Building HVAC Applications

The widespread use of liquid desiccant systems in buildings, either as stand-alone dehumidification or integrated into AC systems, has been curtailed by the perceived high complexity of maintenance and operational challenges. The potential significant energy savings which liquid desiccant systems offer have generally not been perceived as a significant incentive to warrant higher operational complexities in HVAC systems.

The wider use of dedicated outdoor air system applications in buildings, where the separation of latent and sensible loads is more workable than in conventional AC systems, has opened new possibilities for dedicated dehumidification in the building industry. Wider application would include not only specialized industrial buildings such as hospitals, where a precise humidity control is important, but a more generalized use of dehumidification systems in commercial or residential buildings.

Of special interest is the separation of humidity and temperature control in buildings under the premise of changing thermal loads in buildings. Historically, AC systems were installed to provide desired thermal conditions, with humidity control being seen of a lower priority and occurring through passing humid air over cold coils. AC systems are typically controlled by air temperature, which means cooling only occurs when the temperature is outside a low and high indoor temperature range, referred to as set points. If ventilation occurs while the building AC system operates under partial load or is cycling, humidity can be introduced to the indoor environment thought supply air that is not dehumidified.

Table 2.9.1 describes the different sensible and latent loads that must be removed from the building indoor environment by means of the space conditioning system. Figure 2.9.1 illustrates the fact that the sensible load in buildings keeps decreasing due to lowering external and internal sensible loads due to more efficient building envelope and electric equipment, respectively. This infers that latent load removal becomes more important regarding control and efficient dehumidification.

<table>
<thead>
<tr>
<th>Type of load</th>
<th>From external loads</th>
<th>From internal loads</th>
<th>Process of load reduction by conventional AC</th>
</tr>
</thead>
</table>
| Thermal (sensible) load | • Heat conduction through opaque envelope  
                          | • Solar gain through transparent envelope  
                          | • Outdoor air ventilation (sensible portion)  
                          | • Infiltration (sensible portion)           | • Lights  
                          | • Fans & other motors  
                          | • Office equipment and electronics  
                          | • Plug loads  
                          | • People (sensible portion)            | Thermal heat load is directly absorbed into the refrigerant and rejected outdoor. |
Table 2.9.1: Typical loads in commercial or institutional buildings that are removed by conventional (AC) system to maintain heat balance

<table>
<thead>
<tr>
<th>Type of load</th>
<th>From external loads</th>
<th>From internal loads</th>
<th>Process of load reduction by conventional AC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Moisture (latent) load</td>
<td>• Outdoor air ventilation (latent portion)</td>
<td>• People (latent portion)</td>
<td>Humidity is reduced indirectly by overcooling air past condensation point and then adding reheat</td>
</tr>
<tr>
<td></td>
<td>• Infiltration (latent portion)</td>
<td>• evaporation from indoor plants</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Cooking &amp; drying processes</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Bathrooms and washing</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Miscellaneous processes</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>introducing humidity</td>
<td></td>
</tr>
</tbody>
</table>

Figure 2.9.1: Fraction of HVAC moisture load is substantially increasing in buildings (source: Advantix Systems)

Figure 2.9.2 illustrates the mechanism of condensation and re-evaporation of moisture on cooling coils. The figure demonstrates that effective dehumidification is negatively affected by cooling coils which operate at partial loads. At part load, conventional cooling coil cannot dehumidify without over-cooling, or at least initiate temperature swings. Condensation and dehumidification on the cooling coils and subsequent removal of humidity by gravity induced drainage of the condensate only commences after a
delay period. When the cooling source is removed, which means when cooling flow stops running through the coils, condensate remaining on the coils will re-evaporate into the air stream passing over the coils.

Figure 2.9.2: Illustration of cooling coil dehumidification under partial loads (source: Advantix Systems)

(A) Period of Delay of condensation to form and being removed from the air handling unit
(B) Period when humidity is removed by condensation forming and continuously being removed
(C) Period of evaporation of the condensation from non-operating cooling coil that is still wetted

(A) When condensation starts to form on the cooling coils it takes a delay time until the coil is sufficiently wetted so that drainage commences. This is due to the delay in the onset of the condensation process itself, the forming of droplets that are drained from the coils and removed of the condensate from the air handling unit.

(B) Once the condensation process has attained steady state conditions water is continuously forming on and drained from the cooling coils. This is the period of effective humidity removal by cooling the moist air below its dew point and forming condensate.

(C) Once the flow of coolant, either chilled water or refrigerant of DX systems, through the cooling coils stops the temperature rises above the dew point and condensation stops. Due to the residual wetness of the cooling coils water starts to evaporate as warmer air flows over the coils. The evaporation continuous until the coils are dry.
SECTION 3 - IMPORTANT ASPECTS OF LIQUID DESICCANT DEHUMIDIFICATION FOR HVAC APPLICATIONS

This section summarizes important processes of liquid-desiccant dehumidification technologies as they relate to HVAC applications in buildings.

3.1 Packed-bed Liquid Desiccant Dehumidifier and Cooler

Lowenstein (2008) discusses a typical industrial application of a liquid-desiccant system. The system is shown in Figure 3.1.1. The conditioner, or absorber, is the system component that cools and dries the process air. In the configuration depicted in Figure 3.1.1 the conditioner is a packed column, with a bed of structured contact fill. The concentrated liquid desiccant solution is cooled in a heat exchanger upstream of the conditioner vessel and is then sprayed onto the contact media. Lowenstein indicates that the desiccant flow rate must be sufficiently high to ensure complete wetting of the media. He suggests about 5 gpm per square foot of face area. The process air is cooled and dried as it encounters the desiccant-wetted surfaces of the packing. Heat energy is released as the desiccant absorbs water from the air. A high flow rate of the desiccant limits the temperature rise of the desiccant to a few degrees. The desiccant solution weakens as more water is absorbed.

The weak desiccant flows to the regenerator where water is removed from the that the desiccant. The desiccant passes through a heater which increases the desiccant temperature and thereby raises its equilibrium vapor pressure. The hot desiccant, typically between 160°F and 200°F, is sprayed over a bed of random fill inside the regenerator. Flooding rates must be sufficiently high to ensure complete wetting of the media, thus ensuring effective gas exchange. The hot desiccant desorbs water to the air that flows through the bed. This moisture-laden scavenging air is exhausted to ambient.  

Figure 3.1.1: Schematic of a packed-bed liquid-desiccant dehumidifier and cooler (Lowenstein, 2008); modified
Both the regenerator and conditioner require downstream mist eliminators, such as droplet filters, to ensure that desiccant droplets are not entrained in the supply air to the building.

The formation of droplets occurs in the spray distributor and also inside the flooded packing used in industrial equipment. Lowenstein suggests that droplet filters can suppress desiccant carryover to parts per billion of airflow. The downside of these filters or demisters are that they induce air-side pressure drops. In addition, filters and demisters require maintenance. An interchange heat exchanger (IHX) provides heat energy recovery by preheating the weak desiccant upstream of the regenerator using the hot, concentrated desiccant flow that leaves the regenerator. The heat exchanger reduces both the thermal energy use of the regenerator and the cooling requirements of the conditioner.

3.2 Basic Liquid-desiccant Dehumidification and Air-Conditioning System

The use of liquid desiccant dehumidification can be combined with an evaporative cooling process. This system works even in humid climates where evaporative cooling would normally not be a viable option. Figure 3.2.1 shows the basic process diagram of an integrated liquid-desiccant dehumidification and air-conditioning system. The system shown in Figure 3.2.1 has four major system components: (A) the dehumidification unit, (B) the regeneration unit, (C) the liquid-desiccant storage unit, and (D) the sensible heat handling unit.

The dehumidification unit (A) removes moisture of the inlet air by bringing it into contact with liquid desiccant that runs downward on the packing. The regeneration unit (B) applies thermal energy to regenerate the diluted solution flowing from dehumidification unit to an acceptable concentration level. These two system parts (A) and (B) are essential for the continuous process. The liquid storage unit (C) is optional, although all liquid desiccant systems have at least a small volume vessel where desiccant is held, to accommodate desiccant flow demand fluctuations. A larger volume desiccant storage tank can serve as an energy storage by storing strong solution downstream of the regenerator. A larger storage volume can serve to bridge intermittent periods of desiccant regeneration, such as using solar heat for regeneration. The storage tank is typically equipped with an ancillary heater to avoid crystallization of the desiccant solution. Crystallization can impose system inefficiencies such as clogging of desiccant flow conduits.

The sensible heat handling unit (D) downstream of the dehumidifier removes sensible heat load of the process air flowing from the dehumidification in accordance to the target air temperature to be supplied to the conditioned spaces. In the system shown in Figure 3.2.2 the sensible load is removed by an adiabatic evaporative cooler. The fact that the evaporative cooler uses process air that was dried by the dehumidifier renders this system effective also in more humid climates.

The surface vapor pressure difference between liquid-desiccant film and air is acting as the driving force for mass transfer. The diluted solution flowing from dehumidification unit requires preheating to increase its surface vapor pressure. Low-grade energy is used as the preheating source for weak desiccant as well as the driving force for the entire system. Similarly, the inlet liquid desiccant used in the dehumidifier requires pre-cooling to lower its surface vapor pressure, to achieve better dehumidification efficiency. The heat exchanger between the weak desiccant flowing out of the
dehumidification unit and strong desiccant solution flowing in is an important energy recovery for preheating and pre-cooling.

![Diagram of liquid-desiccant dehumidification system](image)

Figure 3.2.1: Schematic diagram of liquid-desiccant dehumidification air-conditioning system (Mei & Dei, 2008)

An operational difficulty is the highly corrosive nature of certain desiccants. Normal anti-corrosion methods mentioned in the literature is adding additives to the liquid desiccant, or choosing parts made of synthetic plastic for the system, which can simultaneously lower cost compared with metal parts.

Lowenstein (2008) describes the properties of liquid desiccants regarding their psychrometric properties. The desiccant’s equilibrium vapor pressure increases roughly exponentially with the temperature of the desiccant/water system, and increases as the desiccant absorbs water and becomes diluted. As illustrated in Figure 3.2.2 for solutions of lithium chloride the absolute humidity of air that has come into equilibrium with a liquid desiccant of fixed concentration closely follows a line of constant relative humidity.

Heat transfer in desiccants follow a mechanism that is the inverse of evaporative cooling. Air below saturation that flows over water surfaces cause water to evaporate and thereby lower temperature at a state the water-air interface toward the wet-bulb temperature of the air. As an example, Figure 3.2.3 shows the line of constant enthalpy passing through a state point of 80°F and 50% RH intersects the saturation line on a psychrometric chart at the wet-bulb temperature of 66.7°F. If air passes over a surface wetted with desiccant and the desiccant absorbs water from the air heat is released, which is contrary to the evaporative cooling effect at the water-air interface.

Lowenstein (2008) defines a brine-bulb temperature as the temperature that the desiccant-air interface approaches. The brine-bulb temperature is a function of a liquid desiccant’s concentration and the air’s
temperature and humidity. As shown in Figure 3.2.3, the brine-bulb temperature will always be slightly higher than the temperature at which a line of constant enthalpy from the air state point intersects the equilibrium relative humidity curve for the desiccant, because of the chemical heat of mixing between the desiccant and water is added to the vapor-liquid latent heat for the water vapor. As an example, Figure 3.2.3 shows a 118°F brine-bulb temperature for a 43% solution of lithium chloride and air at 86 and 78°F DB/WB. With an ambient wet-bulb temperature of 78°F (25.6°C), a typical cooling tower might supply water at 85°F (29.4°C). This low achievable air temperature suggests that a strong cooling effect could be achieved by wetting the surfaces of the heat exchanger with a 43% lithium chloride.

Lowenstein (2008) provides a summary of aspects that are important to the selection of liquid desiccants. Lowenstein suggests that halide salts are corrosive to most ferrous and nonferrous metals. Aqueous solutions of lithium bromide are commonly used in absorption chillers, which operate in a
Similar process of absorbing humidity. Absorption chillers, however, are closed systems, where oxygen levels can be kept low and corrosion inhibitors can be used. Titanium is one of the few metals that could be used in the high-temperature heat exchanger. The price for titanium, however, has dramatically increased in the past few years. Lowenstein (2008) estimates that the heat exchanger for a 25 ton LDAC might cost $2,500.

Glycols has been used as a liquid desiccant, specifically in industrial dehumidification equipment. Glycol based desiccants have low toxicity and are generally compatibility with most metals used in HVAC applications. The drawback of solutions contained glycoce is its volatile nature, which can cause significant entrainment of the desiccant solution into the process air. Lowenstein estimates that in an HVAC application where a 6,000 cfm LDAC operating for 2,000 hours per year, the annual loss of TEG in the conditioner could be as high as 10,000 pounds of glycol. Economic and environmental impacts of this scale of loss would be unacceptable in an HVAC system and therefore glycol is not considered for HVAC applications. Other salts of weak organic acids, such as potassium or sodium formate and acetate, are also less corrosive alternatives to halide salts but are not volatile. These desiccant solutions proposed by several developers, however, are not useful in LDAC because of the low concentration at which they are used. At higher concentration, there are potential operational problems related to impacts of trace contaminations, odor and possible simulation of biological growth.

The cost of lithium-based desiccants could be an obstacle if storage of concentrated desiccant becomes important for intermittent regeneration heat sources, such as solar cooling system that must provide cooling during hours when solar insolation is low or zero. Although relatively inexpensive uninsulated and plastic storage tank can be used for storage of concentrated desiccant, the cost of the lithium chloride by itself could discourage larger desiccant storage capacities.

Mixtures of lithium chloride and calcium chloride can provide lower-cost alternative to single desiccant lithium chloride. The cost for calcium chloride is approximately one-twentieth that of the lithium salt. By itself, calcium chloride is a moderately strong desiccant, with a 29% equilibrium relative humidity for a saturated solution at 77°F. As the fraction of calcium chloride to lithium chloride increases from 0% to 50%, the equilibrium vapor pressure for the solution more closely matches the value for pure lithium chloride. The 43% solution of the 50/50 mixture of calcium chloride to lithium chloride would behave in the LDAC the same as a 40% solution of pure lithium chloride, whereas, a 43% calcium chloride solution behaves like a 34% lithium chloride solution.

The use of a stronger desiccant than lithium chloride might have the important advantage of an air-cooled LDAC that does not need cooling water (Lowenstein, 2008). In comparison, a 43% lithium chloride solution and a 62% lithium bromide solution that processes air at 86/78°F DB/WB temperatures have a brine-bulb temperature of 118°F and 136°F, respectively. With the higher equilibrium temperature heat transfer an LDAC could operate with an air cooler and would not need a cooling tower or other source of evaporative cooling. In addition to being a simpler system to install and maintain, it would place minimal demands on water resources. Not only would the LDAC use no water for cooling, but by switching air conditioning loads from electricity to mostly thermal energy, the water demands at the power plant would be greatly reduced.
3.3 Conventional Packed Column for Desiccant Based Mass Transfer

Conventional liquid dehumidification processes have included packed columns to enhance the mass transfer between the gaseous and liquid phases of process and desiccant solutions. Packing materials enhance mass transfer in falling films of the liquid desiccant within the surrounding process air. The selection of packing materials and other characteristics of the packed column affect the performance of the packed column dehumidification unit.

Packing materials can be categorized as random packing materials and structured packing materials, which are materials without regular geometric forms and placed randomly and materials with fixed geometric form. Figure 3.3.1 shows examples of commonly used packing material.

![Random packing materials – Pall ring](image)
![Random packing materials – Rashing ring](image)
![Random packing materials – enhanced](image)
![Structured packing materials](image)

Figure 3.3.1: Typical structured and random packing materials (sources: online images)

Benefits of different packing materials vary. The following is a brief discussion. The random placement of packing materials can and result in undesirable distribution of the desiccant over the surface of the
packing materials. Furthermore, random packing may initiate wall flow and channel flow, especially with small liquid-desiccant loads. Structured packing materials regulates the flowing direction of the liquid desiccant due to their structured geometric patterns and preferred flow paths. Typically, there is considerable innovation in types of packing and different processes may increase in performances using different types of packing.

The pressure drop is one of the major criteria for choosing packing materials for liquid-desiccant system, since higher pressure drops result in larger fan energy for blowing the process air through the process vessels. Investigations by Gandhidasan (2002) suggested that structured packing materials have a larger pressure drop for desiccant dehumidification systems than random packing materials. It has been reported that structured packing materials have high efficiency and high capacity for mass and heat transfer.

Important design parameters of the packing material are volumetric area (surface area per unit volume of the packing materials), void ratio (the void volume per unit volume of the packing materials) as well as spacing intervals. The void ratio can be used to estimate air flow resistance, which decreases with the increase of void ratio. An optimum spacing interval will create less resistance on the inlet air while improving the coverage ratio of the sprinkled desiccant on the packing material.

The equivalent diameter is another important parameter for heat and mass transfer occurring within packing materials. The wetting ratio of different packing materials differs from each other because of their separate cross sections and flow channels and surface characteristics. Height and length of the packing material are controlling parameter for sizing of dehumidification systems. Apart from the selections of packing materials in terms of mass transfer ratio, the criteria for the quality of the packing material should also be noticed. The packing material should not be distorted if soaked in the liquid desiccant and the packing material layers should not be bent under the acceptable inlet air velocity.

### 3.4 Flow Patterns in Desiccant Dehumidification Process Equipment

Figure 3.4.1 illustrates the three flow patterns used in dehumidification, namely parallel flow, counter flow, and cross flow.

![Flow configurations for dehumidifiers](Mei and Dai, 2008)
SECTION 3 - IMPORTANT ASPECTS OF LIQUID DESICCANT DEHUMIDIFICATION FOR HVAC

For parallel flow in liquid-desiccant dehumidification low air flow rate and increased channel height can create better performance of dehumidification and cooling processes. Low air flow rate will increase the contacting time between air and desiccant thus improves mass transfer. Counter flow is the most widely used flow pattern for design purposes of dehumidifier. A certain minimum flow rate of the liquid desiccant is needed. Increasing channel height will enlarge the contacting area between air and desiccant, which also enhances mass transfer. Cross flow configuration for liquid-desiccant dehumidification systems have not been investigated as frequently as parallel and counter flow. It has been detected that temperature and humidity ratio of both liquid desiccant and inlet air change along horizontal and vertical axes.

3.5. Types of Liquid Desiccant Dehumidifiers

There are two types of dehumidifiers, the adiabatic and the internally cooled dehumidifier.

The adiabatic dehumidifier has a large air-desiccant contacting area and a high mass and heat transfer efficiency. It has a relatively simple process vessel geometry, but large pressure drop on process air flowing through the packing materials can be created. The increase in temperature of liquid desiccant during absorption of moisture negatively affects the performance of dehumidifier and make humidity and temperature control of the process air less accurate.

The internally cooled dehumidifier uses cooling coils inside the process vessel to remove heat generated by the absorption process. Figure 3.5.1 illustrates a conventional design of an inner cooled dehumidifier, which is configured for cross flow of air and solution flow for packing materials. The flow pattern can also be parallel and counter flow configurations. Cooling coils embedded in the packing material remove absorption heat. An outer insulation layer prevents ambient heat losses. Embedding of the cooling coils into the packing materials causes difficult installation methods. As an alternative to cooling coil embedded in packing, cooling coils can be installed in layers. The configuration with plates is illustrated in Figure 3.5.2. The layers of cooling elements remove absorption heat. Corrugated plates enlarge the air-desiccant contacting area and therefore enhance dehumidification. A third type of inner cooled dehumidifier is illustrated in Figure 3.5.3. Here cooling coils replace the packing material and provide the air-desiccant contacting surface for mass and heat transfer. A finned tube type is used to increase air-desiccant contacting area. Vertical and horizontal distances of tubes should be considered for optimum pressure loss and mass and heat transfer. Because of the corrosive nature of the salt-desiccant solution the finned tubes must be made of special anti-corrosion materials or high thermal conductivity metal of anti-corrosion coated layers.
SECTION 3 - IMPORTANT ASPECTS OF LIQUID DESICCANT DEHUMIDIFICATION FOR HVAC

Figure 3.5.1: Basic configuration of a packed inner cooled dehumidifier. (Mei and Dai, 2008)

Figure 3.5.2: Inner cooled dehumidifier with spaced parallel plate as packing materials (Mei and Dai, 2008)

Figure 3.5.3: Shell and tube dehumidifier (Mei and Dai, 2008)
3.6 Performance Metrics of Dehumidifiers

The performance of dehumidifiers is typically described by the following parameters.

**Moisture removal rate**, is the metric for the removal of the latent heat load of the process air, where \( d_{\text{ain}} \) and \( d_{\text{a,out}} \) is the air humidity ratio if the inflowing and outflowing process air.

\[
\Delta d_a = d_{\text{a,in}} - d_{\text{a,out}}
\]

**Dehumidification efficiency**, describes the efficiency of dehumidifier \( \varepsilon_{\text{de}} \) as the dimensionless humidity ratio that describes how the dehumidification process approaches the assumed optimum one; where \( d_{\text{a,eq}} \) is the humidity of air that is in vapor pressure equilibrium with the vapor surface pressure of the inlet desiccant solution.

\[
\varepsilon_{\text{de}} = (d_{\text{a,in}} - d_{\text{a,out}}) / (d_{\text{a,in}} - d_{\text{a,eq}})
\]

**Air to solution mass ratio**, MR, is an important dimensionless parameter for optimization of dehumidification, where \( m_a \) and \( m_s \) are mass flow rate (kg/s) of the process air flow and desiccant solution.

\[
MR = m_a / m_s
\]

Extreme mass flow ratios can affect the dehumidification process as follows, (1) a too small flow rate of the desiccant solution can lead to an uneven spread of desiccant solution over the packing layer, (2) a too high mass flow rate will cause small concentration difference between inlet and outlet desiccant solution. This renders regeneration of the weak solution more difficult and require more energy for pumping larger quantities of desiccant solution.

3.7 Multi-effect Liquid Desiccant Dehumidifiers

In single-stage dehumidification process a reduction of dehumidification efficiency is caused by a significant rise in temperature due to the absorption heat of the desiccant solution. Multi-effect dehumidification can overcome this inefficiency of conventional single dehumidification units. Multi-effect dehumidification lowers the adverse temperature rise by installing several single dehumidifier modules in series, where each module cools desiccants separately.

Figure 3.7.1 shows the counter flow concept of a generic multi-effect dehumidifier. As process air flows through the successive module the air is first dehumidified by the weakest desiccant solution, where the weak desiccant solution and the process air has the highest surface vapor pressure. As the air flows through successive modules process air is dried and its vapor pressure is decreased whereas the desiccant solution becomes higher resulting in lower vapor pressures. Thus, the vapor pressure difference between desiccant solution and process air is kept at a moderate level without drastic change. This moderate differential helps to keep the irreversible losses in multi-stage dehumidifier significantly lower than in conventional single-effect dehumidifiers.
SECTION 3 - IMPORTANT ASPECTS OF LIQUID DESICCANT DEHUMIDIFICATION FOR HVAC

Since the inlet and outlet concentration difference of the desiccant solution is increased regeneration is very effective, thus reducing related energy losses. A major concern is the low mass flow rate of the desiccant solution, which can impose uneven distribution of the desiccant solution on the transfer packing.

3.8 Configuration and Performance of Liquid Desiccant Regenerators

The regenerator increases the concentration of the weak desiccant solution that flows from the dehumidifier through the introduction of external heat and an effective mass transfer between scavenging air and the desiccant solution. Preferred sources of process heat are waste heat from thermal processes and solar heat. Intermittent heat sources would require an ancillary heater, such as gas or electricity. The geometry and flow process of regenerators are very similar to those of dehumidifiers, while the mass transfer process occurs in opposite directions. In regenerators humidity is driven out from the desiccant solution into the scavenging air thought thermally induced vapor pressure increases at the desiccant surfaces.

The performance of the regenerator can be expressed as a dimensionless regeneration efficiency number $\varepsilon_{re}$ (Gandhidasan, 2005). Expressed in air humidity ratios the dimensionless regeneration efficiency is defined as follows, where $d_{a,out,max}$ is the maximum humidity ratio that the outlet air from regenerator could reach as a result of the inlet conditions of weak desiccant solution

$$\varepsilon_{re} = \frac{(d_{a,out} - d_{a,in})}{(d_{a,out,max} - d_{a,in})}$$

Expressed in terms of concentration of desiccant solution dimensionless regeneration efficiency number $\varepsilon_{re}$ is defined, where $C_{s,sat}$ is the concentration of the saturated desiccant solution at representative regeneration temperature, typically some form of average process temperature.

$$\varepsilon_{re} = \frac{(C_{s,out} - C_{s,in})}{(C_{s,sat} - C_{s,in})}$$
3.6 Multi-effect Regenerator Concepts

Figure 3.6.1 shows a two-stage regenerator consisting of high-pressure and low-pressure process vessels which was proposed by Lowenstein (1995). The hot water vapor evaporated from the high-pressure regenerator heats the weak solution in low-pressure generator, where the low-pressure generator is kept at a relatively low temperature. Heat exchanger pre-heat the weak inlet solution and heat recovery from the hot water vapor before it is discharged into the ambient surroundings. The differentiating feature of this regenerator is that the high driving force for mass transfer exists in the low-pressure generator, which is evacuated before put into operation. The regenerator can be driven by low-grade energy and is less dependent on ambient climatic conditions compared with the conventional regenerators.

![Two-stage regenerator diagram](image)

Figure 3.6.1: Process of two-stage regenerator operating with process pressures (Lowenstein, 1995)

3.7 Energy Storage Capacity of Concentrated Desiccant Solution

Storage of strong desiccant solution is a form of energy storage, since the ability of dehumidification in the form of latent heat in desiccant solution is stored, rather than a quantity of heat. The energy storage capacity by desiccant solution can be quantified by the following relationship;

\[ S = \left( (m_{a,\text{in}} \cdot d_{a,\text{in}} - m_{a,\text{out}} \cdot d_{a,\text{out}}) \cdot \varphi_{\text{rs, out}} \cdot r \right) / m_{a,\text{out}} \]

where: \( m_a \) is the mass flow rate of air, \( d_a \) is the air humidity ratio, \( \varphi_s \) is the desiccant density, \( r \) is the latent heat of vaporization at the average temperature of desiccant solution in the dehumidifier. One significant operational difficulty is the possibility of desiccant crystallization if temperature fluctuations in the stored desiccant material occur.

3.8 Integration of Sensible Cooling into Humidification System

This section shows two possible options of hybrid systems which integrate sensible cooling into a liquid desiccant dehumidification process. Another preferred option would also be to add an evaporation cooling cycle downstream of the dehumidifier. This option was discussed in Section 1 of this report. The advantage of placing an evaporative cooler downstream of the dehumidifier is that evaporative coolers
will become less dependent on the climate condition and could also work effectively in humid climate that typically would exclude evaporative cooling.

Figure 3.8.1 shows the integration of a conventional vapor compression cooling (VCC) system into a liquid desiccant process (Parson, 1989). It was found that the integration of the desiccant dehumidification and cooling cycle improves the VCS’s overall performance. The hybrid systems with VCC system reduces electric power consumption, the size of VCC, and the mass rate of the condensed water because of the lower condensation temperature. In addition, the COP of the vapor compression chiller increase since the chilled water temperature or evaporator temperature of an DX AC-system operates at higher temperatures than conventional AC which dehumidifies process air with cooling coils. Parson suggested that 35% energy saving could be realized by adding a liquid-desiccant dehumidifier to a VCC cycle.

Figure 3.8.1 illustrates outside air passing through a multi effect dehumidifier and then the evaporator of a DX AC-system, which first dries and then cools the process air supplied to the conditioned space. An evaporative unit provides cooling water for desiccant solution of a multi-effect dehumidifier. The return air from the space passes through the condenser coils of DX AC-unit and subsequent through a regenerator unit that receives process heat from a solar array. Parson (1989) suggested that 35% energy saving could be realized by adding a liquid-desiccant dehumidifier to a VCC cycle and therefore separating the sensible and latent load removal. The system’s largest electricity requirement is for the vapor compressor. Part of the waste heat energy from the compressor is being recovered by heating the return air.

Figure 3.8.2 shows the integration of a conventional vapor absorption cooling (VAC) system into a liquid desiccant process (Parson, 1989). The hybrid system using an absorption chiller for sensible heat removal requires working fluids for both subsystems of liquid-desiccant dehumidification. Therefore, for the VAC system the required heat load from the regenerator is significantly larger as for the VAC system. This results in a larger heat capacity of the solar collector to realize. Differently from the VCC system the return air from the conditioned space is not pre-heated before flowing to the regenerator. The system requires no electricity only for the operation of circulation pumps. This makes the system very attractive at places with high solar gain and needs to conserve electric energy or lower peak power.
Figure 3.8.1: Liquid-desiccant dehumidification hybrid system with a conventional vapor compression cooling and a solar heating system (Mei and Dai, 2008)

Figure 3.8.2: Liquid-desiccant dehumidification hybrid system with an absorption cooling and a solar heating system (Mei and Dai, 2008)
SECTION 4 – RECENT LIQUID DESSICANT DEVELOPMENTS AND IDENTIFIED FUTURE RESEARCH NEEDS

This section presents reviewed literature that discusses required advances in liquid desiccant (LD) air dehumidification to make this new technology suitable for residential and commercial HVAC systems.

As discussed in previous sections in this report liquid desiccant dehumidification plays an important role in selected industrial applications which require very low air and/or precise humidity control independently of thermal load control. Industrial liquid desiccant applications are well established but they require a significant level of maintenance effort and sophistication. This level surpasses the typical maintenance requirements of conventional HVAC with vapor compression cooling and cooling based dehumidification. A main requirement for competitive liquid desiccant applications in the HVAC market is therefore simplification of the process equipment, in terms of configuration of the equipment, effects of the desiccant solutions on the ventilation air and removing process steps that require a higher level of maintenance.

4.1 General Considerations for Solar Cooling Applications with LD Systems

Solar cooling systems work on the basis of either thermal or electric energy conversion.

Solar electric cooling applications use photovoltaic (PV) panels to provide electricity to conventional vapor compression cycles. Energy efficient systems and compressors are using DC electricity, typically at 48V, which avoids losses from an inverter. Because of the intermittency of solar based electric supply and the need to provide electricity during off-peak solar periods, batteries are required for continuous on-demand operation. This type of solar cooling uses the conventional way of providing a combined sensible and latent load removal, with the advantages of simple operations but with the disadvantages of conventional cooling coil operation. The only benefit of a PV solar AC system is the grid independence. A typical PV solar cooling systems is shown in Figure 4.1.1.

![Figure 4.1.1: DC Powered Solar Air Conditioner](image)

- DC Powered Split Unit
- 100 % DC Powered Compressor
- DC brushless fan motors can greatly reduce energy consumption
- 280W-300W solar panels
Solar thermal cooling application: Active solar cooling applications use either closed or open-loop systems.

- Solar closed-loop absorption cooling: Closed-loop system use solar heat, thermally driven chillers and they cool the supply air indirectly. Liquid and solid desiccants are used in absorption and adsorption heat pumps, respectively. The working principle is similar to the desiccant dehumidification, but here closed systems work inside air-tight and low-pressure vessels. In the process either water or another working fluid is absorpt or adsorpt by the desiccant liquid solution or solid material, respectively. Heat of evaporation provides cooling and therefore heat is driven to the condenser side from where it is rejected to the outside environment. The closed-loop chillers can only provide cooling capacity. Dehumidification would be carried out by conventional cooling coils.

- Solar open-loop air conditioning using desiccants: Open-cycle systems have direct contact to the supply air. Cooling is provided by adiabatic evaporative cooling. Humidity (latent load) is extracted by means of strong desiccants which are regenerated by solar heat. Placing the evaporative cooler downstream of the desiccant dehumidification unit allows evaporative cooling even in high humidity climates, where stand-alone evaporative cooling would not be feasible. More advanced liquid-desiccant air conditioners (LDACs) have been developed that can compete against conventional roof-top systems. This technology, offers new opportunities for commercializing a solar cooling system.

The solar electric cooling and solar closed-loop both provide cooling capacities which are alternatives to the grid-electricity based conventional vapor compressor chillers. These two cooling technologies provide sensible and latent cooling, just the way that the conventional AC is doing. In such a way, both cooling technologies use cooling coil dehumidification. The solar open-loop air conditioning use desiccants for dehumidification and evaporative cooling for sensible heat removal.

Lowenstein (2003) suggested that the long-term goal of a solar cooling system is its competitiveness with conventional systems without relying on government or utility subsidies or unrealistically high energy prices. The author suggested that if long-term benefits offered by an emerging technology are sufficiently great, state and federal agencies in the U.S. have been willing to provide financial incentives to encourage its adoption. In the longer term, improvements in the solar and desiccant technology should be sufficient to eliminate subsidies and still render LDAC systems competitive.

4.2 Low-flow Liquid Desiccant Systems

Lowenstein (2003) emphasized that the single biggest obstacle to moving liquid desiccant technology from its well-established base in industrial applications into a wider HVAC market has been the high maintenance requirements for the technology. The most common configuration for a liquid-desiccant conditioner (i.e., the component that dries and cools the process air) is a bed of porous contact media that is flooded with desiccant. As process air moves through this bed and the air is dried by absorbing the humidity into the desiccant solution. The flow characteristics inside the flooded packed-bed conditioner is such that it can entrain small desiccant droplets which can be conveyed downstream with
the air stream; this process is also referred to as “carry-over”. A typical approach to remove the droplets from the air stream leaving the conditioner is by means of an “mist eliminator” and/or a filter of appropriate size. In industrial applications, these measures of “demisting” the supply air are maintained to insure no carry-over occurs. However, it is unlikely “industrial” maintenance procedures would be widely accepted in HVAC applications.

One of the typical characteristics of an industrial liquid desiccant dehumidifier using packed columns is that the desiccant flow rate is sufficiently high to avoid significant temperatures gains of the desiccant solution as it absorbs humidity inside the process vessel. The desiccant is cooled before it enters the conditioner.

The alternative to a conventional liquid desiccant dehumidification process is the so-called low flow process. In this process, very low flow rates of liquid desiccant are used on the contact surfaces of the conditioner, flow rates that are one-tenth to one-fiftieth the levels in typical industrial systems. Due to the very low flow rates the desiccant solution heats up significantly inside the conditioner as it absorbs water vapor from the process air. External desiccant cooling upstream of the conditioner is not sufficient. Instead, the low flow conditioner should be internally cooled.

Initial designs of the low-flow conditioner used a water-cooled heat exchanger (Lowenstein, 2003). A prototype model was successfully tested, which was comprised of 132 plates of the heat exchanger. The plates were plastic extrusions with individual length, width and thickness of 48, 12 and 0.1 inch, respectively. Process air flowed through 0.1-inch wide spaces between heat exchanger plates. Liquid desiccant was distributed to the top edges of each plate and flowed downwards over wicks on the outer surfaces of the plates create thin, uniform desiccant films.

In addition to a conditioner, the liquid-desiccant cooling system had a regenerator and an interchange heat exchanger. This configuration is shown in Figure 4.2.1. The interchange heat exchanger improves the efficiency of the overall system by using the hot concentrated desiccant that leaves the regenerator to preheat the entering weak desiccant. The regenerator was built in a similar way. But instead of internally cooled as in the conditioner, the regenerator is internally heated to remove the water that the desiccant absorbed in the conditioner. A scavenging air flow between the plates carries away the water vapor.

Lowenstein (2003) indicated that the single effect regenerator has a Coefficient of Performance (COP) that is close to 0.65. A more efficient double-effect regenerator with a COP of a high as 1.25 would have a 300 F atmospheric-pressure desiccant boiler upstream of the scavenging-air regenerator. The steam that leaves the boiler at a 212 F saturation temperature provides the thermal energy for the scavenging air regenerator.
4.3 Design Concepts of Liquid Desiccant Conditioner for HVAC Applications

Packed-column conditioners with external heat transfer have been discussed earlier in this report and are not further discussed in this section. Rather, this section only addresses design concepts of internally cooled conditioners and conditioners with a high mass ratio.

There are two types of internally cooled conditioners, internally cooled packed column conditioners and conditioners using internally cooled plate-type heat exchangers.

In the packed-column type conditioners internal cooling is accomplished by water cooled copper tubes or polypropylene tubes. With appropriate corrugation, tubes can also serve as contact surface. In an early design (Gommel et al, 2004), the copper tubes were quickly corroded by the desiccant, and the polypropylene tubes were too difficult to wet due to the surface tension of the desiccant. Lowenstein (2008) reported that industrial liquid-desiccant conditioners used water-cooled banks of metal tubes as the contacted area between the desiccant and the process air. The author suggested that when operating with halide salt solutions this type of internally cooled conditioner was too expensive for use in HVAC applications.

Lowenstein (2008) reports on two types of internally cooled conditioners using plate-type heat exchangers to reject the process heat from the desiccants:

(A) Using an integrated indirect evaporative cooler as the heat sink
(B) Using water cooled plate as the heat sink

Both (A) and (B) plate-type heat exchanger share the same working principle with air flowing on one side of the heat exchanger and a coolant on the other. The plate surfaces which are exposed to the air flow are wetted with the liquid desiccant, while the other side of the plate is in contact with the coolant, which rejects the absorption heat of the desiccant. With the absorption heat constantly removed by the coolant, the temperature of the desiccant solution remains constant, with the desiccant having only a slightly higher temperature than the coolant so that heat transfer through the plate is at steady state.
Type (A) uses a conventional indirect evaporative cooling process as the heat sink. While this type of conditioner has a higher cooling potential it is also more difficult to implement (Lowenstein, 2008). This configuration of plate-type conditioners uses alternating passages between the plates for the primary and the secondary air flow. The primary air flow comes into contact with the desiccant wetted surfaces where humidity is removed from the air by the desiccant. The secondary air flow evaporates a water film on the opposite side of the plate, thereby lowering the temperature of the plate. Thus, the primary and secondary air flows are not getting in contact. Figure 4.3.1 shows the concept of a plate-type conditioner that is cooled through adiabatic evaporation.

Several developers tested various designs of the type (A) conditioners that used coated metal and plastic plates. Where the metal plates had a better heat conduction performance, the metal plates are prone to corrosion from the desiccant. Lowenstein (2008) reported that the plastic coatings used in the tests did not adequately protect the aluminum plates, and several parts of the LDAC were seriously corroded by the desiccant. He suggested that the mass flow ratio was fairly high for an internally cooled conditioner. The author also described plastic-plate prototype tests of an evaporatively cooled, liquid-desiccant conditioner. The conditioner used a cross-flow air-to-air heat exchanger as its contact surface, where the plastic plates had dimples on the plates that maintained a constant 3 mm gap between plates. The stack of conditioner plates was tilted so that the two liquid flows tended to flow down over the surfaces of the plates. With plates diagonal in a vertical orientation which resulted in 45° flow direction from the horizontal for both primary and secondary air flow, the conditioner had a 0.75 heat/mass effectiveness. As a general observation, it was found that an evaporatively cooled conditioner can be difficult to build and operate because of possible leaks in the separating plates between the desiccant and water.

Several developers tested various designs of the type (B) internally cooled conditioner. For this design, the plates are internally cooled by means cooling water flowing through them. A concept design of a water-cooled plate-type conditioner is illustrated in Figure 4.3.2. Differently from the evaporatively cooled conditioners where two air streams as well as desiccant solution and water flows must be separated, the water-cooled conditioner has a much simpler air and liquid flow arrangement. For type (B) conditioners, each gap between plates was wetted with desiccant solution and process air was flowing through the gaps between plates.

Several designs of the internally water-cooled plate-type conditioners have been proposed and tested. Early designs had extruded plastic extrusions which created internal pathways for cooling water. A twin-wall polypropylene plate design was a hollow extrusion with two parallel walls filled with thin webs which maintained the space between the walls. A low flow of desiccant was directly delivered to the outer surfaces of the plates, which were covered with a polypropylene fleece that ensured even distribution of the desiccant on the wetted surfaces. Another design used plates made from PVC extrusions with many cooling passages (illustrated in Figure 4.3.2), running the length of the plate. The plates outer surfaces had a thin 0.5 mm wick cover to ensure even wetting by the desiccant.
For all water-cooled plate-type designs each transfer plate was connected to upper and lower manifolds, which served as the source and sink for the vertically flowing cooling water. An upper end-piece had a desiccant distributor assemble that delivered low flows of desiccant directly to the top of the plate. The capacity of conditioners can be adjusted by staking plate modules. The covers of the plate, such as the wicks, are essential for a good performance of an internally cooled conditioner. Halide salt solutions, which are the most common liquid desiccant material for HVAC applications, have high surface-tension and therefore they do not easily wet plastic walls. Therefore, wicks must be used to uniformly spread desiccant over the plastic plates; and plastic material will remain the preferred material due to the corrosive nature of the desiccant.
The heat sinks of types (B) and (B) differ. Type (A) has the evaporative cooling heat sink adjacent to every contact surface that is wetted with desiccant. This allows for a very effective cooling with little losses. Type (B) has an external heat sink, typically a conventional cooling tower. Both evaporative cooling processes in types (A) and (B) are limited by the wet bulb temperature of the ambient air. However, type (B) introduces additional losses since evaporative cooling is carried out externally by means of a conventional cooling tower. This added system components slightly reduces the cooling capacity of the conditioner type (B) and increases the complexity and therefore the costs of the overall system.

### 4.4 Design Concepts of Liquid Desiccant Regenerators for HVAC Applications

Analogous to the liquid desiccant conditioners, regenerators can also operate as packed columns and or alterative plate-type regenerators. In all types of regenerators, the liquid desiccant is heated in order to raise the vapor pressure at the desiccant surfaces and thereby initiate a mass transfer of water from the low concentration desiccant to the scavenging air stream that is in contact with the desiccant. As the desiccant solution loses humidity its concentration increases.

Packed-column scavenging-air regenerators are the established technology for industrial liquid-desiccant dehumidification application with halide salts. The significant drawback for this type of regenerators is that desiccant flow rates must be high in order to limit the change in desiccant concentration as it flows through the bed. The desiccant’s equilibrium vapor pressure, the driving potential for mass transfer during regeneration, is an exponential function of temperature. The saturation desiccant temperature, in turn, is a somewhat linear function of desiccant concentration, with higher concentrations attaining lower temperatures in the adiabatic absorption process. As the desiccant concentration increases the temperature drops, resulting in unsatisfactory process conditions if the temperature drop is too high. The conventional packed column regenerator therefore must have high flow rates, which in turn require large power requirements for pumps.

In regenerators with internal heating the temperature of the desiccant is no longer coupled to its flow rate. Therefore, the flow rate can be decreased significantly without negative temperature and performance implications. Conditioners with internal heating can operate in a low-flow process mode and high concentration differentials. The low-flow regenerator can therefore operate at average concentration levels that are significantly below high-flow regenerators. Lower desiccant solution
concentrations have higher vapor pressures, which increases the process driving force and leads to smaller regenerator sizes.

Low-flow regenerators are using configurations similar to internally heated plate-type conditioners. Due to the corrosive nature of the desiccant various forms of plastic plate-type exchangers have been used, with plates made of extruded polypropylene and polymer. Thin layers of fibers on the plastic plates ensured uniform wetting by the desiccant.

The coefficient of performance (COP) of the regenerator greatly affects the overall performance of the total system. The magnitude of the COP is directly proportional to the process temperature. Lowenstein (2008) reported a 17% increase of COP of an internally heated regenerator, that increased desiccant concentration from 36% to 40%, when the process temperature was increased from 160 to 200 F. Lowenstein (2008) suggested that the improvement in COP produced by higher temperatures in scavenging-air regenerators resulted from the desiccant’s equilibrium water vapor pressure which has an exponential relationship with temperature. Temperature losses of the desiccant that lower the COP are due to convective heat losses to the air and the desorption of the desiccant. Since convective heat losses are linear and mass transfer exponential in nature, the overall COP increases with temperature although parasitical sensible losses also increase with higher temperature. Yet, the driving forces outweigh the thermal losses with increasing temperatures. Figure 4.4.1 illustrates data by Lowenstein that shows the dependency of COP and water removal rates (WR) as a function of temperature.

![Figure 4.4.1: COP and water removal capacity of an internally heated regenerator as a function of temperature of the desiccant solution.](image)

Terms used for the data curves:
- “Exp’t” indicate experimental data
- “Model” indicates theoretical predictions

4.5 Mitigation of Desiccant Droplet Carry-over

Possible droplet entrainment into the process air (e.g. carry-over) inside the liquid-desiccant conditioners and regenerators that use halide salts represents a serious concern for the use of liquid-desiccant dehumidification in HVAC applications. Industrial applications employ maintenance intensive elimination of droplet carry-over measures that would be regarded as unreasonable for conventional HVAC applications of liquid desiccants.
SECTION 4 – RECENT LIQUID DESICCANT DEVELOPMENTS AND IDENTIFIED FUTURE RESEARCH NEEDS

Carry-over can, however, be eliminated in low-flow internally cooled conditioners and regenerators. Droplet formations and subsequent entrainment and transport in the process air are avoided, since sprays and drips are not used and the low flow rate of the air flow minimizes aerodynamic lift forces and therefore lifting of droplets. Rather than flowing downward from one contact surface to the next, like in the packed columns, the desiccant flows inside a wick. Figure 4.5.1 indicates that partial density in the observed size ranges are identical in the inlet and outlet process air flow of a low-flow liquid desiccant conditioner. The fact that particle densities in each size range are at the inlet and outlet are essentially the same serves as an indicator that no entrainment of droplets occurs inside the conditioner. While the potential for carry-over still exists, whereas very small, actual applications include filter downstream of the conditioner the process air flow.

The elimination of potential carry-over of liquid desiccant droplets into the supply air is a significant advantage of the low-flow and wicked conditioner and regenerator technology.

Figure 4.5.1: Particle counts at the inlet and outlet of a low-flow conditioner (Lowenstein, 2008)

4.6 Current Liquid Desiccant Materials

Lithium chloride has been the preferred halide salt for liquid desiccant dehumidification in HVAC applications. Lithium chloride is a strong desiccant, but its disadvantages include high corrosiveness to most ferrous and nonferrous metals and the high cost for the desiccant. The high costs are especially important when storage is required to bridge periods of intermittent heat sources, such as solar or waste heat. Titanium is one of the few metals that could be used in the high-temperature heat exchanger that supplies hot desiccant to the regenerator shown. Costs for titanium have dramatically increased in the past few years. Therefore, plastic system components are used to the extent possible.

Alternatives to lithium chloride as the liquid desiccant of choice have been investigated and there are candidates of advanced liquids desiccant that could possibly replace lithium chloride. The following describes some of the possible choices:
Glycols is routinely used in industrial equipment. Glycol has low toxicity and is compatible with most metals, which promotes glycol for HVAC applications. However, all forms of glycols are volatile and annual losses in the HVAC system would impose economic penalty and environmental impact that would make Glycol unacceptable for use in HVAC applications.

Lithium bromide is a very strong desiccant. In comparison to lithium chloride lithium bromide has about double the drying capacity. Lithium bromide is almost exclusively used in absorption chillers that use water as the refrigerant. In this application, however, the process cycle is closed and oxygen can be controlled and corrosion inhibitors can be used. This alleviates the corrosion problems. The significant disadvantage of lithium bromide is the likelihood of odor problems. In a HVAC system where the supply air comes into contact with bromine, even in traces, odor problems basically eliminate bromides as suitable desiccants.

Salts of weak organic acids, such as potassium or sodium formate and acetate, have been proposed for HVAC applications. The benefits of these advanced desiccants include their low corrosiveness and the fact that they are non-volatile. Their lower surface tension than lithium chloride causes less problems with wetting of the contact surfaces. While not available in wider use, the costs of these desiccants suggest being lower than those of lithium chloride of bromide. Although these organic acids are significantly weaker desiccants than lithium bromide or lithium chloride, the ability to dry air below 30% relative humidity while avoiding important operational considerations could make potassium formate a good candidate desiccant in some applications. Significant disadvantages, however, include possible reactions with trace contaminants that may be present in the process air, the possibility to promote biological growth and the possibility of offensive odors. These disadvantages might make these advanced desiccants unsuitable for HVAC applications.

Calcium chloride is a moderately strong desiccant, with a 29% equilibrium relative humidity for a saturated solution, this is significantly less than the 11% equilibrium relative humidity of a saturated lithium chloride solution. The significant advantage of calcium chloride are the comparative low costs which is approximately one-twentieth that of lithium chloride. With the much less stringent dehumidification requirement of HVAC applications than in selected industrial application this desiccant could be an interesting choice, especially if large volumes of the desiccant are required for energy storage.

A mixture of lithium chloride and calcium chloride can also be a suitable desiccant. Lowenstein (2008) suggested that the equilibrium vapor pressures of a mixture of lithium chloride and calcium chloride is a function of their fraction in the mixture. For example, 43% solutions of calcium chloride, lithium chloride and a 50/50 mixture at 85°F would have equilibrium dew points of 52.1°F, 33.5°F, and 40.2°F, respectively.

### 4.7 Identified Development Targets for Liquid Desiccant Cooling

Important development targets for liquid desiccant dehumidification and cooling were identified as critical for making the liquid-desiccant dehumidification a suitable HVAC technology. The liquid-desiccant technology has been proven as viable and more advantageous in specialized industrial...
SECTION 4 – RECENT LIQUID DESICCANT DEVELOPMENTS AND IDENTIFIED FUTURE RESEARCH NEEDS

dehumidification applications. The next hurdle will be to improve the overall LDAC systems in such a way that it becomes a dependable and accepted HVAC system component.

Lowenstein (2008) suggested the following areas where liquid desiccant technologies needed further improvements. Improvements of system components could significantly improve overall system performance.

The development of noncorrosive desiccant: Currently desiccants used in LDAC are nontoxic, but corrosive. Design solutions of LDAC systems have ensured that the desiccant will be contained in the system and that there will be no carry over to the conditioned spaces. The acceptance of LDAC will, however, be higher if more benign desiccant solutions would be used instead of the halide salt solutions.

The preferred desiccant should be noncorrosive and nontoxic desiccant. This desiccant solution should have low surface tension to promote good wetting performance of the contact surfaces. The desiccant should also have a not too high viscosity to ensure good flowing and pumping performance. The desiccant should be chemically stable in open systems where they are exposed to air that may have trace contaminants such as ozone, oxides of nitrogen, and volatile organic compounds.

Wetting and Rewetting of Contact Surface: The uniform desiccant wetting of contact surface is important for an efficient performance of the conditioners and regenerators. The currently used desiccants, such as halide salts, have high surface tension which complicate uniform wetting, especially in low flow applications. The wicking of plates has diminished the problem, but further development could improve wetting and therefore overall system performance. As pointed out before, an advanced low surface tension desiccant would improve wetting uniformity. In the absence of significant breakthrough in advanced desiccant development, the use of surface textures, surface material and contact surface geometry could also improve wetting performance.

Advanced Evaporative Cooling in conjunction with LDAC: Effective rejection of absorption heat generate by the desiccant in the conditioner is essential to maintain good performance of an internally cooled low flow conditioner. Typically, and in the absence of other heat sinks, evaporative cooling produces the cooling capacity. The conventional evaporative cooling is, however, limited by the wet bulb temperature and the resulting cooling is always higher than the wet bulb temperature because of thermal losses. Advanced indirect evaporative coolers have been demonstrated that can cool air to below the wet bulb temperature of the air that acts as the heat sink (Maisotsenko et al. 2004). The use of advanced evaporative cooling as a part of LDAC will increase the performance and competitiveness of the overall LDAC system.

High-Efficiency Regenerators: It is essential for good overall system performance to increase the COP of the regenerator, which for most LDAC installations is presently well below 1.0. Possible new developments to increase the COP include multiple-effect boilers and vapor-compression distillation.
**Enhanced Heat and Mass Transfer:** The heat and mass exchange of internally cooled conditioners can be increased by surface enhancement of the plates. These enhancements could include fins, spines, and other extended surfaces are commonly used in heat exchangers to reduce their size. Although the desiccant that flows on the contact surfaces can complicate the design of the regenerator and conditioner, approaches to increasing heat and mass transfer coefficients should be explored to reduce the size, pressure drop, and cost of the conditioner.
SECTION 5 – LIQUID DESICCANT TECHNOLOGIES OFFERED BY DIFFERENT VENDORS

This section introduces several companies which are in the process of developing and have a track record of developing, manufacturing and/or selling liquid desiccant technologies. A literature research had identified seven candidate companies, which have track records in desiccant dehumidification.

5.1 Initial Section of Candidate Companies

The main overall project goal is to design and eventually install a pilot HVAC installation at a selected project site in Hawaii. The pilot HVAC installation will involve partnering with one or more of the initially identified companies; this will involve certain selection criteria, such as track record and ease of communication. Review of available online product information and technical presentation given by the company led to the selection of seven candidate companies. Table 5.1.1 lists the initial selection which included the following seven companies:

Table 5.1.1: Initial selection of candidate companies

<table>
<thead>
<tr>
<th>No</th>
<th>Company</th>
<th>Country</th>
<th>Type of Products</th>
<th>Status of company</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7 AC Technologies</td>
<td>US</td>
<td>Start-up company that develops membrane based liquid desiccant dehumidification process; product is based on membrane technology developed by NREL</td>
<td>Start-up company, has been in business for less than 5 years</td>
</tr>
<tr>
<td>A</td>
<td>Advantix Systems</td>
<td>US</td>
<td>Innovative solutions for liquid desiccant products;</td>
<td>Company closed and went out of business</td>
</tr>
<tr>
<td>2</td>
<td>AIL Research</td>
<td>US</td>
<td>Established engineering and technology company with a wide range of liquid desiccant products and technologies</td>
<td>Company has been in business for more than 10 years and is a recognized leader in innovative technology development</td>
</tr>
<tr>
<td>3</td>
<td>Be Power tech</td>
<td>US</td>
<td>Start-up company that develops membrane based liquid desiccant dehumidification; product is based on membrane technology developed by NREL. Fuel cell is used to provide a constant supply of thermal energy for regeneration</td>
<td>Start-up company, has been in business for less than 5 years</td>
</tr>
<tr>
<td>4</td>
<td>Kathabar Dehumidification Systems, Inc</td>
<td>US</td>
<td>Established company with a large range of desiccant dehumidification products, which are installed in various industrial dehumidification applications</td>
<td>US internationally active company that is in business for several decades</td>
</tr>
<tr>
<td>5</td>
<td>L-dcs</td>
<td>Germany</td>
<td>Established engineering and technology company which provides integrated and energy saving HVAC solutions. Included are liquid desiccant technology</td>
<td>German company has been in business for more than 10 years and is a recognized leader in innovative technology development</td>
</tr>
</tbody>
</table>
Company listed under No. A, Advantix Systems, went out of business in 2015. Using information still posted on the web, engineers and researchers had provided valuable information on state-of-the-art liquid desiccant technologies. Therefore, while no longer in business some pertinent information is still pertinent and is presented hereafter. The following sub-sections 5.2 through 5.9 of Section 4 present summaries of pertinent products and technologies provided by the listed companies. Appendix A through F present selected product and technology descriptions as well as other information that is pertinent for the present project.

<table>
<thead>
<tr>
<th>No.</th>
<th>Company</th>
<th>Summary of company technology and products presented in Section 4.x</th>
<th>Samples of product information presented in Appendices</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7 AC Technologies</td>
<td>Section 4.2</td>
<td>A</td>
</tr>
<tr>
<td>A</td>
<td>Advantix Systems</td>
<td>Section 4.3</td>
<td>B</td>
</tr>
<tr>
<td>2</td>
<td>AIL Research</td>
<td>Section 4.4</td>
<td>C</td>
</tr>
<tr>
<td>3</td>
<td>Be Power tech</td>
<td>Section 4.5</td>
<td>Not available</td>
</tr>
<tr>
<td>4</td>
<td>Kathabar Dehumidification Systems, Inc</td>
<td>Section 4.6</td>
<td>D</td>
</tr>
<tr>
<td>5</td>
<td>L-dcs</td>
<td>Section 4.7</td>
<td>E</td>
</tr>
<tr>
<td>6</td>
<td>Menerga Apparatebau, GmbH</td>
<td>Section 4.8</td>
<td>F</td>
</tr>
</tbody>
</table>

### 5.2 Product information of 7 AC Technologies

The company web site is [http://7actech.com](http://7actech.com)

**About the company:** 7-AC-Technologies is a Beverly, MA based company which is working on the commercialization of an innovative HVAC concept that was first introduced by NREL and has been licensed to 7 AC Technologies. The NREL concept is called “Desiccant Enhanced Evaporative Air-Conditioning (DEVap)”. At the heart of this innovative HVAC concept is a membrane technology that establishes a physical separation between the supply air and the liquid desiccant solution. Water vapor can pass the membrane but liquid desiccant cannot. Appendix A shows a summary of the NREL membrane technology. Figure 4.2.1 depicts the technology that 7AC technologies has developed.

**The company summaries its philosophy as follows:** “Using proprietary membrane modules, 7AC is the first company to develop a way to use liquid desiccants in a self-contained system, a three-way
path of air, water and salt to cool and dehumidify outside air in a single step. This allows 7AC to use a “single step” cooling and dehumidifying process, avoiding the over cooling of vapor compression, the heating required by solid desiccants and the corrosion challenges of traditional liquid desiccant approaches.”

Company declared benefits and goals: The company describes building related benefits for their technology applications as follows:

Occupants: By controlling humidity, 7AC can make buildings more comfortable for occupants. 7AC also allows for increased ventilation, providing healthier living and working environments. Proper control of humidity reduces the potential of mold and sick building syndrome and can affect a variety of health issues including skin, respiratory and infectious conditions.

Commercial Building Owners & Operators: 7AC offers building operators & owners the opportunity to dramatically lower energy costs but the benefits go beyond savings. Correctly maintaining humidity levels protects real estate investments and 7AC’s system offers drop-in replacement for existing systems, ideal for retrofits.

Differentiators of technology: The membrane technology avoids direct contact between desiccant solution and the process air, thereby avoiding carry over. The advanced indirect evaporative cooling provides the cooling capacity.

Products and technologies: The company has not yet released commercially available products. Several pilot installations have been completed with, as the company suggests, significant success. Figure 5.2.1 is the company’s description of the technology. This information is publicly available on the company web site. It should be noted the basic thermal processes are the same as the DeVAP process that was developed by NREL and is now licensed to 7 AC Technologies. As a comparison, Figure 5.2.2 shows the basic process diagram of NREL’s DeVap technology.
The DeVAP technology uses a membrane to physically separate the desiccant and the process air. The DeVAP technology is the basis for the development of commercial products by 7 AC Technologies.
Pilot Installation in Jubail, Kingdom of Saudi Arabia (KSA): The company installed a pilot plant in an office building in the town of in Jubail, KSA.

Configuration of the pilot plant:
- 2 Conditioner and 2 Regenerator LDHX modules
- Supply air flow rate: 471 lps (1000 CFM)
- 1 variable speed compressor –35 kWt (10 tons)

The significant advantage of the membrane technology is the physical separation of the desiccant and process air, thus avoiding entrainment and carry over. There have been investigations suggesting the membranes can lose integrity and subsequently leak after a period of operation.

Figure 5.2.3: Working principle of the 7 AC Technology membrane technology (source 7 AC Techn.)

Figure 5.2.4: Process flow diagram pilot installation in KSA (source 7 AC Techn.)
Communication with company: The project team had very few communication exchanges with one representative of the company. The company has indicated that initial pilot plants have been installed and have successfully operated for a duration of a couple of months.

Overall assessment: The company works on commercialization of the very promising liquid desiccant AC membrane technology, the DeVAP technology introduced by NREL. In accordance with the available information the technology is scalable. The development is still in the pre-commercial phase, with apparent challenges of preventing leaking of desiccant solution through the membrane. It is not clear when the first units will be available for actual deployment.
5.3 Advantix Systems

About the company: The company, based in Miami, Florida, closed business in 2015. Before that date the company was one of the leading supplier of liquid desiccant HVAC equipment and the recipient of several industry awards for innovation in HVAC.

The company described its products and services as follow: Advantix Systems is a leading manufacturer of air conditioning systems that deliver energy savings of 30 to 80 percent in commercial and industrial applications. A pioneer in harnessing liquid desiccant, Advantix's clean tech air conditioning and dehumidification systems deliver cooling with powerful, precise humidity control, substantial energy savings and cleaner air free of bacteria and mold. With competitive first costs in both new and retrofit applications, Advantix offers compelling economics, better performance, and an improved environmental footprint.

Company declared benefits and goals: The company’s message focused on industry best practice design standards for separate equipment to treat ventilation and/or latent loads. The company cited ASHRAE: Although most centralized and decentralized systems are very effective at handling the space sensible cooling and heating loads, they are less effective (or ineffective) at handling ventilation air or latent loads. As a result, outside air should be treated separately.

The company cited significant cost savings by dealing with moisture more efficiently:

First (Capital) costs
- Comparable (or less) upfront cost to alternative equipment
- Systems sized more efficiently by handling more humidity removal than conventional units

Energy costs (O&M costs)
- 30-40% energy savings compared to conventional mechanical systems
- 30-60% energy savings compared to (solid) desiccant wheels

Products and technologies: According to still available information the company aggressively marketed the LDAC technology for the following industries:

- Hospitality
- Healthcare
- Other Commercial
- Pharma
- Food & Beverage
- Other industrial uses

In addition, the company marketed their products to the building industry.

The basic flow diagram of the LDAC technology is shown in Figure 5.3.1.
The company presented various types of integration of their LDAC into the building AC-system. The LDAC takes the entire latent load, while the RTU handles the sensible load.
Communication with company: There was no communication with the company.

Overall assessment: Before going out of business Advantix Systems was very well positioned in the industrial and commercial dehumidification market and had received several industry awards for excellence. It is not clear if the company stopped operation due to insufficient financial performance or if the products had some form of quality problems.
5.4 AIL Research, Inc.

Company web site: http://ailr.com

About the company: AIL Research Inc. is a privately held small-business R&D and engineering company based in Hopewell, NJ. The company has received numerous awards for excellence and innovation. R&D efforts have been supported by federal grants and publicly or privately projects. The company owner Dr. Andrew Lowenstein holds several patents for devices and process in liquid desiccant dehumidification. Lowenstein has been at the forefront in the development and application of liquid desiccant technology.

Company declared benefits and goals: The following goals are bring expressed by the company.

For the past 25 years, AILR's innovations have focused on technology that moves liquid desiccant from their traditional industrial base into HVAC applications. However, our more broad expertise in heat and mass transfer has led to interesting innovations in the design of plastic heat exchangers, low-cost solar thermal collectors, and high efficiency thermal distillation processes.

"The Valley of Death" is often used to describe the challenge of turning a promising technology from a laboratory prototype into a commercial product. AILR is working with one HVAC manufacturer to bring to market an air conditioner that uses the low-flow, direct-contact liquid desiccant technology. Similar partnerships are being explored with other manufacturers and venture-backed start-up companies.

As a cooling system driven by thermal energy, liquid desiccants can alleviate the high electrical demands created by compressor-based ACs. Their ability to dry air without first cooling the air is a more efficient approach to controlling indoor humidity and LD systems can have much higher thermal efficiencies for regenerating the desiccant and lower power requirements for moving air through the unit. Quoted from AI Research, Inc. web site.

Areas of applications: The company has a broad range of liquid desiccant technologies. The following applications are stated as most important for the company:

Solar Cooling: The company claims that there is a natural "fit" between solar energy and the need for cooling since they both peak during the periods of hot weather. Sorption chillers (either absorption or adsorption) are the most common technology applied to solar cooling. They however are not an attractive option for a building owner looking to reduce his carbon footprint, since they are expensive, require large cooling towers and can be difficult to maintain. LDAC can greatly improve the value proposition for installing a solar cooling system and can provide significant benefits to building owners who must supply large volumes of ventilation air.

Some of the benefits include:
• LDAC can save the building owner money by solving an indoor humidity problem or avoiding overcooling/reheat.
• Storing "cooling capacity" as concentrated desiccant is less expensive and more compact than storing hot water or chilled water.
• The cooling tower for an LDAC will be about half the size of that needed for a single-effect absorption chiller.
• The LDAC is simpler than an absorption chiller -- it does not have to maintain a vacuum vessel as does the absorption chiller.

Combined Heat and Power Applications: Combined Heat and Power (CHP), which is also commonly referred to as cogeneration, distributed generation (DG) and distributed energy (DE), is one of the most effective approaches to improving fuel-use efficiency. With CHP, the thermal energy produced when generating electricity is used for either hot water or space heating rather than being wasted.

Using the heat for cooling expands the potential of CHP installations. It allows the use of sorption chiller. However, as with solar applications, an absorption chiller may make the combined system harder rather than easier to sell. The advantages that liquid-desiccant air conditioners offer over conventional electric units reverses this situation by giving the customer more effective control over indoor humidity and allowing appropriate ventilation rates. Typically, the recovered heat from CHP systems that produce between 60 kW and 75 kW are a good match with the thermal requirements of a 6,000 cfm LDAC.

Supermarkets: Supermarkets use tremendous amounts of electricity with approximately half this power being used to cool refrigerated display cases. A store's refrigeration system must continually reject heat which cool display cases gain from the warmer and humid indoor store climate to the outdoors. In effect, the refrigerated display cases are air conditioning the store, providing both sensible and latent cooling. But they provide this cooling at a very poor efficiency, since evaporators in these display cases are running at a much lower temperature than a conventional HVAC cooling system (e.g., the evaporators for display cases for frozen foods can run as low as -25 F and for non-frozen product, between 20 F and 35 F versus 45 F to 50 F for the evaporator of the store's air conditioner). A refrigeration system's compressor must work harder, and less efficiently, when it pumps heat over a larger temperature rise.

Pools: Indoor pools, water parks and natatoriums can benefit from the efficient dehumidification provided by LDACs. In the application illustrated in Figure 5.4.2 continuous evaporation from the pool created a high demand for dehumidification within the building as well as for direct heating of the pool to keep it at 80 F. Before the installation of the LDAC system, a 11,000 cfm chilled water coil system provided 22 and 14.5 tons of sensible and latent tons for dehumidification and temperature control within the pool building. The LDAC provided 11 tons of latent cooling to the space, which directly reduced the thermal load on the chilled water coil. The LDAC dried the 3,240 cfm supplied to the perimeter air curtain. The supply dew point for this air was below 45 F, effectively avoiding condensation on the windows without heating the air. Pool water directly cooled the LDAC, so that the latent load served by the LDAC, which approximately equals the
evaporative heat loss from the pool, was returned to the pool. The heat energy needed to regenerate the liquid desiccant was met by recovering heat from a 75 kW CHP system that was installed at the site. The annual savings were significant and amounted to approximately 600 million Btu saved for the pool heating and dehumidification.

Figure 5.4.1: LDAC system installed in a supermarket.

Example of an LDAC installation in a supermarket. The main system components such as conditioner and regenerator were supplied by AIL Research. (source AIL Research Inc.)

Figure 5.4.2: LDAC application for dehumidification and temperature control of indoor pools (source AIL Research Inc.)

The installation of LD technologies from AIL Research Inc. provided significant energy savings for the operation of an indoor pool at the Recreation Center at the Stevens Institute of Technology.

**Products and technologies:** The range of technologies offered by AIL Research includes, but is not limited to, the following LD technologies:

**Compressor Based Liquid Desiccant Air Conditioner (LDDX)** is an efficient, high latent cooling system that can adjust its Sensible Heat Ratio (SHR: the fraction of total cooling that is sensible, the balance being latent) between 0.35 to 0.75. The LDDX with wicking-fin technology is a compressor-based liquid-desiccant air conditioner in which a solution of lithium chloride floods the surfaces of the air conditioner’s evaporator and condenser providing direct contact between the desiccant and the air flowing through these two coils.
Integrated into this refrigerant circuit is a liquid-desiccant circuit that supplies strong desiccant to the top of a wicking-fin evaporator and weak desiccant to the top of a wicking-fin condenser. The strong desiccant absorbs water vapor from the process air flowing through the evaporator. The weak desiccant flowing off the evaporator is warmed in the interchange heat exchanger before it is delivered to the top of the condenser. The heat rejected in the condenser further warms the weak desiccant which then desorbs water to the cooling air that flows through the condenser. The warm, strong desiccant that flows off the condenser is cooled in the interchange heat exchanger before it is supplied to the evaporator.

The essential characteristic of the LDDX is its ability to supply cool, dried air without reheat. Whereas a conventional DX air conditioner without reheat might have a sensible heat ratio (SHR) of 0.75 (i.e., 75% of the cooling it provides is sensible temperature reduction and 25% is latent dehumidification), the LDDX can have an SHR as low as 0.35. The SHR adjustment is achieved by varying the amount of liquid desiccant that is recirculated over the evaporator. As shown in Figure 5.4.3, a diverting valve splits the weak desiccant that leaves the evaporator into a stream that flows to the inlet of the pump that supplies desiccant to the condenser and a stream that is returned to the evaporator. As the fraction of desiccant which is returned to the top of the evaporator increases, the concentration of the desiccant becomes weaker and the amount of moisture absorbed from the process air decreases. The overall effect of increasing the recirculation rate to the evaporator is to increase the LDDX’s SHR.

**Plastic Interchange Heat Exchanger:** The interchange heat exchanger (IHE) is an essential system component since it recovers thermal energy and increases the overall efficiency of the LD system. The regenerator concentrates a solution of lithium chloride from to 39% to 43% at approximately 200 °F. This equals 70% of the thermal energy required to remove the water from the desiccant. Thermal energy in the hot, strong desiccant that flows from the regenerator to the conditioner is recovered to preheat the weak desiccant flowing to the regenerator. Figures 5.4.4 and 5.4.5 show a typical LD process diagram with an IHE and a photographic rendering of interchange heat exchanger. Due to the corrosive nature of the liquid desiccant a plastic heat exchanger is a cost-effective alternative to stainless and titanium, respectively. AILR has developed liquid-to-liquid heat exchangers with plates made from thin, thermoformed films of polymers that can operate at over 200 °F. The thin films (typically 5 mil thick) have good conductive heat transfer allowing heat exchanger to have a high effectiveness of approximately 80%.
The LDDX can adjust to a wide range of sensible heat ratio (SHR) by varying the amount of desiccant recirculating through the wicked material covering the evaporator.

Membrane Heat and Mass Exchanger: The solutions of lithium chloride that are most commonly used in liquid-desiccant systems do present a practical maintenance concern. As with all strong halide salt solutions, lithium chloride is corrosive to metals commonly used in HVAC installations. AILR’s wicking-fin heat exchangers addresses possible maintenance concerns by (1) operating with very low flow rates of desiccant so that possible droplet formation is suppressed, and (2) safeguarding that all wetted surfaces will resist corrosion by the lithium chloride. One solution of these challenges is to use less expensive metals for heat transfer surfaces. The second solution for liquid desiccant technology is the development of heat and mass exchangers that isolate the liquid desiccant behind a membrane. The membrane is designed to be highly permeable to water vapor so it introduces only a small resistance to the transport of water between the liquid desiccant and the process air. Figure 5.4.6 and 5.4.7 show membrane-based conditioners with water cooled plates.
that were designed, fabricated and tested by AIL. Water cooling the membranes is a different approach than the evaporative cooling approach used in the membrane DeVAP technology developed by NREL. Problems with the membrane technology include potential membrane fouling or otherwise being compromised by trace contaminants in the air or desiccant. Another problem that was identified is the "weeping" of lithium chloride through a membrane that was a hydrophobic, microporous, polypropylene film (Figure 5.4.8).

![Figure 5.4.6:](image1)
![Figure 5.4.7:](image2)
![Figure 5.4.8](image3)

**Scavenging-Air Regenerator:** With the equilibrium vapor pressure of a 42% solution of lithium chloride at 285 °F at 14.7 psi, the desiccant would have to be heated to 285 °F. An alternative to a desiccant boiler is a regenerator in which the desiccant is heated to a temperature at which its equilibrium vapor pressure is significantly higher than the partial pressure of water vapor in ambient air. At desiccant temperatures above 160 °F (preferably above 180 °F if the regenerator is to supply lithium chloride at 40% or higher), a scavenging-air regenerator, in which ambient air flows over surfaces that are wetted with hot desiccant, can efficiently remove water from the weak desiccant. A low flow scavenging-air regenerator is preferred and has important advantages including (1) lower pump power, (2) lower air-side pressure drop, and (3) better suppression of droplet carryover. (High-flow and low-flow technology
as applied to conditioners are compared in the liquid-desiccant tutorial.) As a low-flow device, the regenerator is a heat exchanger with hot water flowing through its internal passages and ambient air flowing over its desiccant-wetted exterior surface. Figure 5.4.9 shows types of scavenging-air regenerators developed and sold by AIL.

![Figure 5.4.9: Types of scavenging-air regenerator: As plastic-plate heat exchangers (left) and wicking-fin heat exchangers (above) (source AIL Research)](image)

**Wicking-fin technology:** Wicking-fin technology surrounds the heat exchanger surfaces with a layer of porous contact media. This configuration of heat and mass transfer integration allows the temperature of a liquid that flows over the contact media to be controlled while the liquid desiccant exchanges gas (e.g. air) that flows through the contact media. Figure 5.4.10 shows the concept flow diagram of the wicking-fin heat and mass exchanger (HMX). In the application of the wicking-fin technology the heat exchanger that is surrounded by the contact media is the evaporator of a vapor-compression air conditioner. Figure 5.4.11 shows a typical detail of the fin embedded into the contact media. The figure illustrates a low flow of liquid desiccant delivered to the top of the HMX. The liquid desiccant (green arrow) is first cooled as it flows over the uppermost refrigerant tubes (brown). The cool desiccant then flows from the tubes onto the first row of fins. The wicking surfaces of the fins uniformly spread the desiccant. The process air that flows horizontally between the fins is simultaneously cooled and dried as it comes into contact with the desiccant-wetted surfaces. The fin length is designed so that the desiccant’s temperature rises only a few degrees before it flows onto the next lower row of cooling tubes.

When properly designed, the convective heat transfer of the desiccant on the fin is an effective substitute for the conductive heat transfer of the aluminum fins used in a conventional finned-tube heat exchanger. Two important operating parameters for a wicking-fin HMX are its air-side pressure drop and avoidance of droplet entrainment and carryover. Due to the corrosive nature of the lithium chloride used in a wicking-fin HMX, the embedded heat exchanger, which typically is metallic since heat fluxes on
the tubes are high, must resist corrosion by the strong salt solution. Depending on operating temperatures, either 90/10 or 70/30 cupronickel (also referred to as copper-nickel) has demonstrated good corrosion resistant to lithium chloride and is considered a good candidate for this application.

![Concept flow diagram of the wicking-fin heat and mass exchanger (HMX)](image1)

![Detail of the fin embedded into the contact media (source AIL Research)](image2)

**Figure 5.4.10:** Concept flow diagram of the wicking-fin heat and mass exchanger (HMX)

**Figure 5.4.11:** Detail of the fin embedded into the contact media (source AIL Research)

**Two Stage Regenerator:** Two effect (stage) liquid-desiccant regenerator significantly increases the COP compared to a solid-desiccant system or a single effect liquid desiccant regenerator. Figure 5.4.12 illustrates the process of weak desiccant boiling within the first stage heat exchanger (gas-fired heat exchanger of a boiler). A mixture of steam and strong desiccant leave the first stage and enters a spin separator. Concentrated desiccant leaves at the bottom of the spin separator, and steam leaves at the top. The steam is being recovered as the thermal source for additional regeneration in a conventional scavenging-air regenerator. Therefore, each amount unit of thermal energy supplied to the regenerator is captured twice: once to remove water from the desiccant in the boiler and once to remove water in the scavenging-air regenerator. Typically, a single stage scavenging-air regenerator operating on 200 °F might have a gas-based COP in the range of .6 to .76, whereas the two-stage regenerator will have a COP of 1.08.
Steam-Generating Solar Collector: AIL Research has developed an innovative solar collector that converts solar radiation into atmospheric pressure steam. Figure 5.4.13 shows steam-generating solar collector (SGSC), which is an array of Dewar-type evacuated tubes. The Dewar-type evacuated tubes design resembles vacuum thermos bottle with an inner tube positioned within an outer tube and the space between the two tubes evacuated. The collectors are installed horizontally. Before the collector is subjected to solar heat, this means in the morning, the tubes are filled approximately half-full of water and the solar heat converts the water to steam. The steam from each tube is collected in a common manifold and delivered to the end use (e.g. desiccant regeneration for a LDAC). As the steam condenses heat at 212 °F is released. The hot condensate is stored in an insulated tank overnight and returned to the collectors the next morning.

This new type of solar collectors was tested and the thermal performance was like conventional evacuated-tube solar collectors. However, the very simple design for the SGSC should permit a dramatic reduction in first-cost. Furthermore, the horizontal orientation will reduce installation costs since racking will not be needed. The direct supply of steam to the regenerator eliminates the cost and power for a hot-water circulating pump. The horizontal orientation for the tubes of a SGSC will penalize performance when the sun is low in the sky. Thus, the SGSC is not a good source of thermal energy for mid and high latitude locations in the winter. However, the SGSC is an excellent, low-cost source of high-grade thermal energy in climates that require cooling.
The steam-generating tubes are converting liquid water into steam. The steam flows to the end use and condenses, creating a very effective heat transfer mechanism. The collector has a comparable thermal performance to other evacuated tubes, but the steam generating tube collectors provide significant savings in regard to tube installation and avoidance of hot water recirculation pump.

Example/Pilot Installations: AIL Research has several very successful pilot and commercial installations. Details can be obtained from the company’s web site.

As an example installation, which is not presented on the company web site, the authors of this report visited the Whole Foods Market (WFM) store in Kailua, Oahu, where earlier versions of the wicking-fin conditioner and regenerator were installed. The LD system in Kailua is currently not in operation. The LD-system was designed and installed to provide approximately 6,000 cfm of dried outside air supply to the WFM store with a total area of approximately 35,000 sqft. The system was sized to assist in latent heat removal and lower the energy requirements for the store refrigeration displays. The LD-system augmented the store’s rooftop DX-systems.

Figures 5.4.14 through 5.4.17 show several images of the LD-system of Whole Food Markets, in Kailua, Oahu. The entire site visit documentation is presented in Appendix XX.
The LD system is installed on the roof of the Whole Foods Market. The largest system component is the 80-solar collector rack system with 30 evacuated solar tubes per rack. All system components are located on the roof, except for the two hot water storage tanks, which are installed on ground level and connected to the roof top components with insulated hot water pipes.
Communication with company: The HNEI project team has had numerous communications with the principal of AIL Research, Dr. Andrew Lowenstein. He has been very forthcoming and helpful to provide information and to discuss with the project team various topics about LDAC technologies and related technology supports.

Overall assessment: AIL Research, led by its founder Dr. Andrew Lowenstein has been a leading authority for the development of liquid desiccant applications in HVAC. The company has obtained several US patents for LD-system parts, including wicking-fin conditioner and two-effect regenerator, steam generating evacuated solar panels, and other LD technologies. AIL Research technologies and system integrations have pioneered the use of liquid desiccant specifically for HVAC in buildings. Thus, AIL Research has been instrumental to create LD technologies that will lead to energy efficient HVAC applications which can separate latent from sensible load removal.

5.5 Be Power Tech, Inc

Company web site https://www.bepowertech.com/

About the company: Be Power Tech, Inc is a privately held small-business R&D company based in Deerfield Beach, FL. The company has received numerous awards for excellence and innovation. The company’s R&D efforts are supported by privately investment. The company is developing system integration of a LDAC and Fuel Cell (FC) technology for use in the commercial and residential HVAC market.
The company statement is as follows.

*Be Power Tech is leading the way in a new generation of energy-efficient air conditioning systems that produce electricity while cooling (or heating) your building. Our innovative BeCool™ system is designed to bring dramatic energy and cost savings, unparalleled performance, and restore the environment through its reduction of harmful emissions. The BeCool™ system can be used in commercial and industrial applications.*

*We believe that innovation begins with the customer’s needs and is followed by a solution that exceeds their expectations. Our state-of-the-art BeCool™ air conditioner delivers the best in HVAC performance and goes a step further to produce electricity that the customer uses to power their building. The value of the electricity produced is higher than the cost of natural gas consumed, turning energy savings into energy earnings.*

**Company declared benefits and goals:** Company summarizes the benefits of its packaged LDAC technology as follows:

- Reduces building energy cost by more than 67%
- Produces 5 kW of high quality electricity to power your building or sell back to the grid
- Cooling or heating mode
- Peak electricity demand reduced by more than 70%
- Easy to install packaged rooftop system
- No harmful chemicals (uses NO refrigerant)
- Protects and restores the environment by reducing and eliminating harmful emissions
- Water neutral (no hook up or condensate drain lines required)

**Products and technologies:** The company refers to its innovative HVAC technology as “BeCool™”. The system uses the thermal process heat of a natural gas powered fuel cell for regeneration heat of the liquid desiccant. Latent heat is removed with a liquid desiccant system and sensible cooling is achieved by evaporative cooling. Figure 5.5.1 shows a basic process diagram of the Be Cool technology. Figure 5.5.2 presents a 3D-illustration of the 10-ton BeCool prototype system. Figure 5.5.3 shows the final product.
Figure 5.5.1: BeCool technology basic process diagram

The differentiator of the BeCool system is the integration of a natural gas (NG) fuel cell, which generates electricity and provides waste process heat for desiccant regeneration.

The sensible heat is rejected by an evaporative cooler.

(source: Be Power Tech, Inc)

Figure 5.5.2: BeCool technology system 3D-rendering

(source: Be Power Tech, Inc)
Commercial / Pilot Installations: The company has only tested the technology in a lab-setup. The company has announced that it will enter the commercial building market in 2019. Field trials are expected to begin in 2018. The company is presently inviting applications for field trial partners.

Communication with company: Several attempts have been made to communicate with the company but no response was received.

Overall assessment: Be Power Tech, Inc offers an interesting combination of a co-generation plant which uses fuel cell technology for energy generation and provides the fuel cell process heat for desiccant regeneration. The “BeCool™” system, the trade mark under which the company advertises its technology latent load, will remove the latent load with liquid desiccants and the sensible load with an evaporative cooling process. No detail description of the finished product has been released on the company web site. The company promotes field pilot installation and the target to have initial commercial products available in 2018. The company is a startup technology company that is funded by an investment group led by Flagship Pioneering.
5.6 Kathabar Dehumidification Systems, Inc

Company web site: http://www.kathabar.com

About the company: Kathabar Dehumidification Systems is a publicly held company that designs and manufactures liquid and dry desiccant systems for dehumidification and energy recovery applications. The company has served primarily the manufacturing or processing industry as well as health care institutions, which have stringent requirements for humidity-, temperature-, or microorganism-sensitive operations. The company was established 75 years and is a recognized leader in the liquid desiccant industry.

Company declared benefits and advantages: The company states the following key benefits for the liquid desiccant systems

Key benefits:
- High energy efficiency
- Can make use of low cost coolant and heat sources
- Provides anti-microbial filtration of incoming air without the need of added filtration & pressure drop
- Applicable for relative humidity's between ~18-80%

Technology Advantages:
- Simultaneous dehumidification and direct air cooling – simple design provides high energy efficiency.
- Microbiological decontamination – effective biocide captures and neutralizes airborne pathogen
- Performance reliability – non-vaporizing desiccant has infinite life.
- Energy savings – 100% modulation capacity; less energy is required to operate than dry desiccant or mechanical refrigeration systems.
- Precise humidity control (+/-1%RH) – fully adjustable humidity level based on liquid desiccant concentration and temperature.
- Frost-free cooling – temperatures as low as -60°F with no coil to freeze up or defrost.
- FRP (fiberglass) non-metallic industrial construction – long equipment life, reliability and reduced maintenance.
- Design flexibility – Use of hot water or low-pressure steam for regeneration (including waste heat); multiple conditioners with single centralized regenerator; vertical and horizontal airflow orientations available

Basic technology and processes: The company provides a general process for the liquid desiccant (Figure 5.6.1):
SECTION 5 – LIQUID DESICCANT TECHNOLOGIES OFFERED BY DIFFERENT VENDORS

Figure 5.6.1: Basic process diagram for the Kathabar liquid desiccant dehumidification (source: Kathabar Dehumidification Systems)

Figure 5.6.2 suggests preferred applications for liquid desiccant systems. The figure suggests that the company regards liquid desiccant systems well suited for the range of required humidity reductions in building HVAC applications.

Figure 5.6.2: Desiccant application (source: Kathabar Dehumidification Systems)

The red shaded area represents the typical application for liquid desiccant systems offered by Kathabar Dehumidification Systems.

Products and technology application pertaining to building conditioning: The company offers dehumidification products for a wide array of applications.
Applications: The following are applications that pertain to building conditioning. The dehumidification performance for building applications are different from more stringent industrial and commercial drying operations.

Archive Solutions, including Libraries:

Benefits:
- Prevents degradation of stored pieces
- Inhibits color fading and paper decaying
- Eliminates mold growth
- Protects integrity and value
- Provides dust- and bacteria-free air
- Energy efficient operation reduces utility consumption

Green Building Solutions – Options to reduce HVAC energy in buildings:

Benefits:
- Superior energy recovery
- Substantial utility cost savings
- Significantly averts refrigeration equipment cost
- Independent control of temperature and humidity
- No wet cooling coils
- Scrubs and microbiologically cleans building makeup air
- Remote supply and exhaust airstreams
- No cross contamination
- Prevents sick building syndrome
- Effective use of waste heat
- Integrated systems approach: supports on

Process characteristics:
- LDAC, when using distributed or district power can achieve large scale energy savings in air conditioning.
- Heat recovery by LDAC systems creates high building energy efficiency and significant cost and utility savings, specifically for applications that requiring large amounts of outside air and having recovered heat available.
- Building exhaust air can supply be conveyed through a cooling tower generating cool water.
Hospital Solutions

Benefits:

- Maintains a clean, controlled temperature and humidity level for the ultimate comfort of surgeons and patients
- Improves indoor air quality
- Provides bacteria-free air
- Acts as an air scrubber to remove and neutralize airborne bacteria, viruses, and mold
- Energy efficient operation reduces utility consumption

Process advantages:

- Captures and kills 94% of all microorganisms entrained within the airstream.
- Reduces operating cost through lower utility consumption.
- Eliminates wet cooling coils that promote reproduction of microorganisms.
- Relative humidity can be easily controlled to 45% or lower, even at temperatures below 60°F.
- Highest energy efficiency of any desiccant unit.
- New fiberglass construction which is simple, silent, trouble-free, and operates continuously with very low maintenance.
- Automatic air humidification in winter without a dry steam humidifier
- Microbiological Decontamination — desiccant solution is bactericidal and viricidal to most airborne organisms. Bacteria, mold and viruses are automatically neutralized and washed from both the conditioner and regenerator air streams.

Two methods are involved:

- The process scrubs the air as it passes through the spray, removing airborne particulates, germs, viruses, etc.
- All airborne organisms washed from the airstream are neutralized by the liquid desiccant solution.

Products: The company offers a wide array of commercial and industrial products. The following is an example for a HVAC system that can serve smaller buildings. This product could be a candidate for the pilot HVAC installation in Honolulu.
Industrial and Commercial Installations: Since the company has been in business for 75 years there are numerous commercial and industrial installation.

Communication with company: The project team could communicate with the HQ on the East Coast as well as with the regional sales and technical support team in Los Angeles.

Overall assessment: Kathabar Dehumidification Systems has been developing, manufacturing and selling liquid desiccant systems for several decades. The technology and products have been performing well for the established applications in the manufacturing and process industry and more niche applications for specialty institutional buildings. Therefore, their products are adjusted to satisfy the stringent humidity and temperature requirements of these established application. Regarding a broader use of their product in general HVAC and “green buildings” applications, the company sees potential, while recognizing that their products would have to be optimized for that new market.
5.7 Liquid Desiccant Cooling Systems (L-DCS)

Company web site: http://www.l-dcs.de (only German version available at this time)

**About the company:** The company is a small German engineering and technology company that works in energy efficient buildings and systems, with a specialty in liquid desiccant dehumidification and cooling systems. The company was founded in 2003 and has offices and manufacturing facilities in Munich, Germany. The company has been working on numerous privately and governmental products. The company founder and president Mathias Pelzer, is a recognized expert in energy efficient building systems integration.

**Company slogan:** Company theme: “Helping our clients breath greener”

**L-DCS describes the following advantages and benefits:** The company states that dehumidification and integrated heat recovery systems are advanced technology applications with a range of functionalities which are unique and require expert technology knowledge. The company suggests that its team of engineers have such specialty knowledge.

Advantages of L-DCS technology and systems integrations are as follows:

- Low process temperatures of the LD-systems, between 130 and 170 °F, allow the use of low temperature resources, without a loss in performance. Possible thermal resources that can be utilized by L-DCS LD-systems:
  - Solar heat
  - Co-generation
  - Heat recovery from compressed air systems
  - Low temperature heat recovery from industrial processes
- Ultra-Low-Flow© Technology:
  - Reduces pump energy
  - Enables energy storage as concentrated desiccant solution with up to 280 kWh/cbm
  - Effective energy transport without losses
- Corrosion resistant construction of system components
- Zero carry over of desiccants to process air flow due to low flow conditions
- Separate process vessels for conditioner and regenerator: Both process components, e.g. conditioner and regenerator, can be located at different points of the building. This allows important design flexibility to accommodate especially building retrofitting.
- Modular arrangement of systems parts allows flexibility to adjust size to customer requirements and performance.
- Lithium chloride desiccant solution acts as anti-viral, bactericide, fungicide agent. Therefore, the liquid desiccant dehumidification is very suitable for applications where hygiene is important, such as hospitals
SECTION 5 – LIQUID DESICCANT TECHNOLOGIES OFFERED BY DIFFERENT VENDORS

Products and technologies:

**Dehumidification:** L-DCS dehumidification systems are completely made of polypropylene (PP) and therefore they are recyclable. All system enclosure housings are available for both indoor and outdoor installation. Air dehumidification L-DCS products are based on an open-cycle sorption technology developed by L-DCS. Process air conditioning consists of an absorber (dehumidifier), a desiccant regenerator and, if applicable, an evaporation cooler as a re-cooling component for removing the absorption heat. Depending on the customer’s fresh air requirements, individually sized air dehumidifiers are manufactured from individual, identical dehumidifying modules with a unit air capacity of 2000 m³/air/h (1,200 cfm). The lithium chloride can be supplied from a sorbent storage tank which, due to the Low-Flow® technology developed by L-DCS, serves as energy storage with an energy density of up to 280 kWh/m³. The capacity of the energy storage can be sized to the specific project needs. Typically, a L-DCS evaporation cooler is added to obtain high-performance process cooling of the conditioner unit. Local conditions can be integrated into the design to optimize the overall system. Thus, if the local climate and existing conditions permits, an existing cooling tower, or alternative cooling sources, such as well water, can be used directly for re-cooling, instead of the L-DCS evaporative cooler

![Figure 5.7.1: L-DCS packaged dehumidification unit.](source: L-DCS)  
![Figure 5.7.2: L-DCS evaporative cooler (source: L-DCS)]

**Evaporative cooler:** L-DCS provides indirect evaporation chiller units that work on the basis of countercurrent flow. These units can be used either for the production of cold water or cold air. The cold-water evaporation coolers are, like all L-DCS components, manufactured as individual modules. This allows for a flexible sizing of units to meet customer’s needs. Operating adiabatically in the building air exhaust, the evaporator units are ideal re-coolers for the L-DCS dehumidifiers. The evaporation coolers are equipped with the L-DCS Low-Flow® technology and therefore have low pressure losses and a very low secondary power consumption.
**Commercial / pilot installations:**

**Major retrofit of HVAC adding evaporative cooling:** L-DCS was contracted to perform a commercial building retrofit. The objective was to increase the cooling capacity for an entertainment center in Munich, Germany. Due to space constraints, the building duct system could not be expanded. Fan coil units were added in the space with hydronic system delivering chilled water from an evaporative cooler. The space return air was dried with liquid desiccant and the dried air was provided to the indirect evaporative cooler. The installation was intended as proof of concept for a retrofitted and upgraded HVAC system using evaporative cooling for sensible heat removal. Figure 4.7.3 show the process diagram and several images of the installed system.

![Process diagram and images of installed system](image-url)

**Figure 5.7.3:** Liquid desiccant cooling system for a commercial entertainment center in Munich, Germany (source: L-DCS)

(a) Process diagram  
(b) Absorber-regenerator-unit

(a) Process diagram of Liquid desiccant cooling system  
(b) Absorber-regenerator-unit by L-DCS Technology for an air flow of 4000 m³/h  
(c) Desiccant handling equipment installed above the tanks

(c) Desiccant handling equipment

**New LDAC installation for energy efficiency center:** L-DCS designed a LDAC system to provide space dehumidification and sensible cooling for the Energy Efficiency Center (EEC) Würzburg, Germany. The
EEC is depicted in Figure 4.7.4. The process diagram of the LDAC system with energy storage is shown in Figure 4.7.5.

![Figure 5.7.4: Energy Efficiency Center (EEC) Würzburg, Germany (source: EEC)](image)

EEC houses offices and conference facilities and is a showcase for innovative and highly energy efficient building design and systems. The building main focus is placed on implementing novel, innovative, and energy efficient materials, systems, and technologies, as well as on exemplarily demonstrating their applicability. EEC web site: [http://www.energy-efficiency-center.de](http://www.energy-efficiency-center.de)

![Figure 5.7.5: LDAC system with energy storage (source: L-DCS)](image)

Outside air is dehumidified in an absorber by a concentrated LiCl solution and subsequently cooled in an air cooler to supply air temperature. The return air of the building is used in indirect evaporative coolers to produce cooling water by evaporating water from wetted heat exchanger surfaces. The cooling water cools the supply air cooler and the absorber. The diluted desiccant is regenerated by evaporating water at temperatures of 70 to 80 °C (160 to 180°F). Concentrated desiccant tanks provide energy storage.

**Communication with the company:** The HNEI project team had several communications with the principal of L-DCS, Mr. Matthias Peltzer. The long conversations provided significant and valuable application knowledge for the project.
Overall assessment: L-DCS is a German innovative engineering and technology company. The company has successfully tested and implemented its innovative liquid desiccant technology, in commercial and test projects. The company has the design and technology capacity and flexibility to assist the HNEI team with the pilot HVAC installation. The company has been responsive and shared important application knowledge.

5.8 Menerga Apparatebau, GmbH (Menerga)

Company web site: http://www.menerga.com

About the company: Menerga is a HVAC engineering and technology company, with the HQ located in Muehlheim, Germany. The company has sales and service representative in 30 European countries. The manufacturing of the plant equipment is carried out at the HQ in Germany. Menerga has been developing and producing innovative ventilation and air conditioning technology for diverse fields of application since 1980. The company slogan is “Creating a good indoor climate – through Minimal Energy Application “. Menerga is part of the internationally operating Systemair group. The company has a wide range of proprietary technology products for integrated humidity and thermal control applications.

Company declared benefits and advantages: The company suggests that it becomes engaged in the building design at an early stage and thus the company is able to create complete solutions that consider first costs as well as subsequent O&M costs. The company states that it can adapt the technology to the requirements of every single project and offers flexible and modular systems.

The company expresses the following benefits and advantages:

- Intelligent technology leads to lastingly low operating costs
- Focus on utilization of regenerative energy sources
- Design of units are compact and durable
- Offering complete, and packaged units – with integrated control
- Every unit that is delivered is thoroughly tested with a comprehensive factory test runs
- The units are delivered ready-for-connection to the local infrastructure
- The company offers comprehensive maintenance concepts.

Products and technologies: The company’s core competencies include the swimming pool hall dehumidification, which was the main market at the start of the company. Since then the company has expanded and offers a wide range of comfort and process air conditioning. The main applications areas are:

- Swimming pool hall dehumidification (public and private)
comfort air conditioning

- Schools and universities
- Sport facilities
- Trade and commerce
- Hotels and restaurants
- Production and logistics
- Offices
- Health service

- Process climate and cold water
  - Industrial climate
  - Data centers

The company offers 13 different dehumidification and integrated dehumidification and cooling systems. Among these there is one liquid desiccant system, the Sorpsolair system. The unit has an air volume flow rate range from 2,900 to 14,900 m³/h (1,700 to 8,800 cfm).

**Sorpsolair Series (72 and 73)** was developed specially to utilize regenerative energy. The air conditioning concept combines liquid desiccant based dehumidification, adiabatic evaporative cooling and an efficient heat recovery system in a compact comfort air conditioning unit. The 72 series, without a brine accumulator, is suitable for directly utilizing the waste / process heat e.g. from combined heat and power system (CHPS). The desiccant accumulator integrated into the 73 series provides the storage of e.g. solar energy and hence increases the total efficiency of your installations. Sorpsolair systems are designed for all office and business buildings, as well as many other forms of application. System features include:

- Integrated absorber (conditioner) and desorber (regenerator)
- Desiccant accumulator (73 series) for storage of thermal heat allows discontinuous dehumidification operation (e.g. use with solar thermal system)
- Filtering the air in any operating mode
- Corrosion-free heat exchanger made from polypropylene
- Pumped hot water heating coil
- Individually controllable performance parameters
- Complete unit, ready to connect, contains all structural elements for comfort air conditioning, including all control and regulation fittings
- Outdoor installation

Figure 5.8.1 illustrates the working principle of the Sorpsolair. The sorption-based air conditioning takes place in two stages, air dehumidification and air cooling. For air dehumidification, warm outside air (OA) is passed through contact surfaces wetted with lithium chloride. The dried outside air then flows through the double plate heat exchanger with indirect evaporative cooling. Significant cooling takes place in the process. The diluted brine is thermally regenerated by heat sources that can come from solar thermal plants, district heat networks or waste heat from combined heat and power plants or industrial processes. The air dehumidification and the desiccant regeneration take place in separate
Process cycles. Energy, in the form of concentrated desiccant solution can be stored almost indefinitely and without loss in a liquid form. Storage is particularly important for intermittent heat supply.

Figure 5.8.1: Working principle of the Menerga working principle of the Sorpsolair (source: Menerga)

Example Installations of Sorpsolair 72 or 73 series:

Example installation of a Sorpsolair unit:

**Hospital**: LDAC system was installed for the outpatient and emergency room areas of the Universitätsklinikum Freiburg, Germany.

**Heat source** is a co-generation plant that is located adjacent the hospital.

Example installation of a Sorpsolair unit:

**Cafeteria**: Freight staff canteen of Munich Airport, Germany.

**Heat source** is a solar thermal array located on the building roof.
Example installation of a Sorpsolair unit:

**Car dealer ship:** Toyota Frey: The "green" car dealer with the goal of zero emission.

**Heat source** is a solar thermal array located on the building roof.

**Communication with the company:** There have been no direct communication with the company.

**Overall assessment:** Menerga is a German engineering and technology company that sells packaged plants for dehumidification and cooling for the building industry. The company was founded 37 years ago, and has been a leader in the specialized field of building humidity control. Out of a 13 products series the company offers one liquid desiccant unit that works with low temperature sources such as solar or waste heat. The company would be a candidate to provide turn key solution.
SECTION 6 - RANKING OF COMPANIES AND TECHNOLOGIES

6.1 Methodology of Two-tiered Ranking

This section performs an initial ranking of the companies and their technologies regarding their potential involvement in the planned pilot HVAC installation. The ranking methodology is a two-tiered approach to identify the level of conformance to a stated group of ranking criteria described below.

- **First tier:** Figure 6.1.1 shows the weighting logic. As shown, there are two tiers of overall weights. The 14 ranking criteria used are grouped into four categories, or first tier (or level) level. In the second level, each of ranking criteria are given percentage weights indicating relative importance to the category. The overall weight of the individual criteria = the first weight of the category \times the second level weight of the criteria, assigned to the category.

```
| No | Criteria categories (1st level) | Ranking criteria (2nd level) | 1st level | 2nd level | Overall weight 
<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>Technology maturity</td>
<td></td>
<td>25%</td>
<td>40%</td>
<td>10%</td>
</tr>
<tr>
<td></td>
<td>1.1 Technology status is mature</td>
<td></td>
<td></td>
<td>30%</td>
<td>8%</td>
</tr>
<tr>
<td></td>
<td>and tested in real-world setting</td>
<td></td>
<td></td>
<td>30%</td>
<td>8%</td>
</tr>
<tr>
<td></td>
<td>sum 2nd level &gt;&gt;&gt;</td>
<td></td>
<td>100%</td>
<td></td>
<td>25%</td>
</tr>
<tr>
<td>2</td>
<td>Prior installation/application experience of technology</td>
<td></td>
<td>20%</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>2.1 Products have been installed in Hawaii (tropical) climate</td>
<td></td>
<td></td>
<td>35%</td>
<td>7%</td>
</tr>
<tr>
<td></td>
<td>2.2 Technology has been tested in Hawaii (tropical) climate</td>
<td></td>
<td></td>
<td>25%</td>
<td>5%</td>
</tr>
<tr>
<td></td>
<td>2.3 Ability to use heat source specific to Hawaii (i.e. solar)</td>
<td></td>
<td></td>
<td>40%</td>
<td>8%</td>
</tr>
<tr>
<td></td>
<td>sum 2nd level &gt;&gt;&gt;</td>
<td></td>
<td>100%</td>
<td></td>
<td>20%</td>
</tr>
<tr>
<td>3</td>
<td>Technology flexibility/ability to implement pilot installation</td>
<td></td>
<td>40%</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>3.1 Flexibility do apply in smaller installations</td>
<td></td>
<td></td>
<td>25%</td>
<td>10%</td>
</tr>
<tr>
<td></td>
<td>3.2 Ability to deviate from standard &amp; prefabricated (larger) systems</td>
<td></td>
<td></td>
<td>15%</td>
<td>6%</td>
</tr>
<tr>
<td></td>
<td>3.3 Ability to retrofit an existing HVAC</td>
<td></td>
<td></td>
<td>30%</td>
<td>12%</td>
</tr>
<tr>
<td></td>
<td>3.4 Ability to build/configure HNEI hybrid system</td>
<td></td>
<td></td>
<td>30%</td>
<td>12%</td>
</tr>
<tr>
<td></td>
<td>sum 2nd level &gt;&gt;&gt;</td>
<td></td>
<td>100%</td>
<td></td>
<td>40%</td>
</tr>
<tr>
<td>4</td>
<td>Communication/willingness to cooperate substantially</td>
<td></td>
<td>15%</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>4.1 Ease of communication</td>
<td></td>
<td></td>
<td>25%</td>
<td>4%</td>
</tr>
<tr>
<td></td>
<td>4.2 Willingness &amp; ability to provide technical support for pilot install.</td>
<td></td>
<td></td>
<td>30%</td>
<td>5%</td>
</tr>
<tr>
<td></td>
<td>4.3 Ease to transport system to Hawaii</td>
<td></td>
<td></td>
<td>20%</td>
<td>3%</td>
</tr>
<tr>
<td></td>
<td>4.4 Ease to purchase domestic (US) products</td>
<td></td>
<td></td>
<td>25%</td>
<td>4%</td>
</tr>
<tr>
<td></td>
<td>sum 2nd level &gt;&gt;&gt;</td>
<td></td>
<td>100%</td>
<td></td>
<td>15%</td>
</tr>
<tr>
<td></td>
<td>sum 1st level &gt;&gt;&gt;&gt;&gt;&gt;</td>
<td></td>
<td>100%</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
```

Figure 6.1.1: Ranking of companies and technologies – First tier overall weights
**Second tier:** In the second tier, each company is ranked regarding how well they comply with each ranking criterion. The level of compliance is designated with a discrete grade from 1 to 3, with 1, 2 and 3 implying low, medium and high compliance with each criterion. The grades 1, 2 and 3 are assigned the percentages 10%, 50% and 100%, respectively. Figure 6.1.2 illustrates the ranking procedure with a generic example.

Figure 6.1.2 shows a generic a sample of overall weights calculated in the multiple-tier method. The criteria category “N” has a 1st level weight of 35% of total weight. The criteria N.1 through N.3 are each assigned individual 2nd level weights which add up to 100%. Each of the overall weights of criteria N.1 through N.3 is calculated by multiplying the of 1st level weight (e.g. the weight of the criteria category) with 2nd order weights of the respective criterion. For example, the criterion N.1 is assigned a 2nd level weight of 40%. The product of 35% and 40% is 14%. Therefore, the overall weights of Criteria N.1, N.2 and N.3 are 14%, 11% and 11%, respectively. In the next step the company X is ranked on its compliance to the three criteria N.1 through N.3 with the discrete grade 1 through 3, where 1 through 3 are assigned percentage weights, in our example 1 = 10%, 2 = 50% and 3 = 100%. Consequently, if criterion N.3 is given a grade number 2 the weight grade percentage 50% is assigned. Therefore, in our example, the resulting ranking of criterion N.3 for company x is 11% multiplied by 50% equal 5.3 %. In the same fashion, all ranking criteria are calculated and the sum of the ranking results for all criteria is the total ranking percentage for company x.

<table>
<thead>
<tr>
<th>No.</th>
<th>Criteria category (1st level)</th>
<th>1st level weight</th>
<th>2nd level weight</th>
<th>Overall weight (1st * 2nd level)</th>
<th>Example Ranking of Company X</th>
</tr>
</thead>
<tbody>
<tr>
<td>N</td>
<td></td>
<td>35%</td>
<td></td>
<td></td>
<td>Grade</td>
</tr>
<tr>
<td>N.1</td>
<td>Criterion N.1</td>
<td></td>
<td>40%</td>
<td>14%</td>
<td>1</td>
</tr>
<tr>
<td>N.2</td>
<td>Criterion N.2</td>
<td></td>
<td>30%</td>
<td>11%</td>
<td>3</td>
</tr>
<tr>
<td>N.3</td>
<td>Criterion N.3</td>
<td></td>
<td>30%</td>
<td>11%</td>
<td>2</td>
</tr>
</tbody>
</table>

**Check A:** This number must be the same as the 1st level weight

*Figure 6.1.2: Example of ranking with the two-tiered ranking methodology*
6.2 Definition of Ranking Criteria

Table 6.2.1 defines the ranking statements of all individual ranking criteria.

**Table 6.2.1: Definition of ranking criteria**

<table>
<thead>
<tr>
<th>Criterion No.</th>
<th>Title / description</th>
<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td><strong>Category 1 - Technology maturity:</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>This category combines ranking criteria to assess the maturity status of the LD technology. The objective of this criteria category is to define whether the technology products offered by the company is a mature technology or if the technology is more of a concept or vision.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.1</td>
<td><strong>Technology status is mature and tested in real-world setting:</strong> The company has a good track record of the technology product in real-world (commercial) installations.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.2</td>
<td><strong>Technology has passed the level of concept:</strong> The technology product has successfully passed the level of concept and its processes are well understood and there is little uncertainty that the technology can serve specific needs of the customer.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>The technology product has not been installed or only to very limited extent in real world settings.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>The technology products have been installed in several real-world settings</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>The technology product is still at the concept level, even if it has already been installed; with several processes still not fully develop and/or understood.</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>The technology product is at an advanced stage and all processes are fully understood and can be reliably sized and applied.</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Table 6.2.1: Definition of ranking criteria

<table>
<thead>
<tr>
<th>Criterion No.</th>
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<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.3</td>
<td>Technology is manageable and has exciting application potential: The technology promises significant innovation and application potential and that the promising potential outweighs identified risks.</td>
<td>The technology product appears to have only limited potential and benefit of using technology seems unclear in respect to risk of investing in it.</td>
<td>The technology product has very high potential and its processes and equipment seem manageable, while the risks seem low.</td>
</tr>
<tr>
<td>2</td>
<td><strong>Category 2 - Prior installation/application experience of technology:</strong> This category combines ranking criteria to assess prior installation experience in Hawaii or similar tropics climate and energy (utility) logistics. The hot and humid climate in Hawaii has high latent loads.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.1</td>
<td>Products have been installed in Hawaii (tropical) climate: The technology has been installed in real-world applications in a hot and humid climate that is similar of Hawaii</td>
<td>The technology product has not been installed in hot and humid climate similar to Hawaii</td>
<td>The technology product has been installed several times in hot and humid climate similar of Hawaii, and has had a good performance record.</td>
</tr>
</tbody>
</table>
### Table 6.2.1: Definition of ranking criteria

<table>
<thead>
<tr>
<th>Criterion No.</th>
<th>Title / description</th>
<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.2</td>
<td>Technology has been tested in Hawaii (tropical) climate: The technology has been tested in the hot and humid climate of Hawaii; yet real-world and commercial applications have not been completed</td>
<td>The technology product has not been tested in hot and humid climate, although commercial projects have been completed elsewhere.</td>
<td>The technology product has been successfully tested in hot and humid climate, and commercial projects have been completed.</td>
</tr>
<tr>
<td>2.3</td>
<td>Ability to use heat source specific to Hawaii (i.e. solar): The technology appears to be able to use heat sources for desiccant regeneration that are unique to Hawaii. This implies that the selection of heat sources must account for the specific utility conditions found in Hawaii.</td>
<td>The technology product cannot or not readily use typical heat sources for desiccant regeneration that are available in Hawaii.</td>
<td>The technology product is flexible enough to use all typical heat sources for desiccant regeneration that are available in Hawaii.</td>
</tr>
</tbody>
</table>

**Category - Technology flexibility / ability to implement pilot HVAC installation:**

This category combines ranking criteria to assess the flexibility of the company and technology products regarding building the pilot HVAC installation. Therefore, the application potential of the technology products should not be limited by the size of the required supply air flow rate and a minimum system capacity. Nor should the application be limited to new installations, but allow use in retrofitting.
<table>
<thead>
<tr>
<th>Criterion No.</th>
<th>Title / description</th>
<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td><strong>Flexibility to apply in smaller installations:</strong> The statement implies that the technology can be successfully adjusted in size or capacity, to allow seamless integration for the pilot HVAC installation.</td>
<td>The technology product only is available for large air flow rate. Therefore, flexibility is low.</td>
<td>The technology product can be adjusted to any reasonable air flow rate. While this might create inefficiency because of limited economy of scale, design / application flexibility is secured.</td>
</tr>
<tr>
<td>3.2</td>
<td><strong>Ability to deviate from standard &amp; prefabricated (larger) systems:</strong> The statement implies that the technology system components can be successfully installed outside a standardized or packaged unit. This infers that the pilot HVAC installation can deviating from a standardized solution that the company offers.</td>
<td>The technology product only is available in standardized or packaged units. Therefore, system components cannot be installed in pilot applications using high flexibility in the design of the pilot installation.</td>
<td>The technology product can be adjusted and is not limited to standardized plant configurations. Therefore, system components can be installed assuring high design flexibility.</td>
</tr>
</tbody>
</table>
### Table 6.2.1: Definition of ranking criteria

<table>
<thead>
<tr>
<th>Criterion No.</th>
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<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3</td>
<td><strong>Ability to retrofit an existing HVAC:</strong> The statement implies that the technology product can be readily used to retrofit the site of the pilot HVAC installation in a flexible way, allowing design and operation freedom.</td>
<td>The technology product cannot easily be used for HVAC retrofit.</td>
<td>The technology product cannot readily be used for HVAC retrofit. Allowing high design flexibility for system configuration and location of systems components.</td>
</tr>
<tr>
<td>3.4</td>
<td><strong>Ability to build / configure HNEI hybrid system:</strong> The statement implies that the system components or the entire technology product can readily be used to configure the hybrid HNEI space conditioning system for the pilot HVAC installation. In addition, the company can provide application and design knowledge to support the HNEI project team.</td>
<td>The technology product does not offer flexibility to configure the hybrid system and the company appears not able or willing to assist in applying the technology product in a different than the company’s standard application.</td>
<td>The technology product does offer significant flexibility to configure the hybrid system and the company appears able and willing to assist in applying the technology product in a different than the company’s standard application.</td>
</tr>
</tbody>
</table>
Table 6.2.1: Definition of ranking criteria

<table>
<thead>
<tr>
<th>Criterion No.</th>
<th>Title / description</th>
<th>Low grade “1” implies no or insignificant compliance to criteria statement</th>
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</tr>
</thead>
<tbody>
<tr>
<td>4</td>
<td><strong>Category - Communication / willingness to cooperate substantially:</strong> This category combines ranking criteria to assess the expected ease of communicating and working with the company to design the pilot HVAC installation, transport the technology product to Hawaii and assist in the installation process.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4.1</td>
<td><strong>Ease of communication:</strong> The statement evaluates that, based on previous experience, communicating with the company is expected to be easy and effective.</td>
<td>The communication with the company is expected to be difficult.</td>
<td>The communication with the company is expected to be easy.</td>
</tr>
<tr>
<td>4.2</td>
<td><strong>Willingness &amp; ability to provide technical support for pilot HVAC installation:</strong> The statement evaluates that it is expected that the company will be available for technical and design support. This expectation implies further that the company will be interested to cooperate in a pilot HVAC installation project that will serve as prove of concept for a new type of energy efficient space conditioning, as envisioned by the HNEI project team.</td>
<td>It is expected that the company will not be interested and willing to provide substantial design and technology support.</td>
<td>It is expected that the company will be very interested and willing to provide substantial design and technology support.</td>
</tr>
</tbody>
</table>
Table 6.2.1: Definition of ranking criteria

<table>
<thead>
<tr>
<th>Criterion No.</th>
<th>Title / description</th>
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<th>High grade “3” implies strong or significant compliance to criteria statement</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.3</td>
<td>Ease to transport system to Hawaii: The statement implies that the efforts to transport the technology product to the pilot HVAC installation project site in Hawaii will be not be high and the company will fully cooperate. This also implies that transporting system components from far away will be more difficult and costly than from a shorter distance.</td>
<td>Long and costly transport. Since system parts will be large and heavy product must be transported via ground and sea.</td>
<td>Relative short and cost-effective transport. Since system parts will be small and light product can be transported via air.</td>
</tr>
<tr>
<td>4.4</td>
<td>Ease to purchase domestic (US) products: The statement implies that procurement will be easier and more cost effective if technology product is acquired domestically.</td>
<td>The technology product is acquired from abroad.</td>
<td>The technology product is acquired within the US.</td>
</tr>
</tbody>
</table>
6.3 Results of Ranking

Table 6.3.1 and Figure 6.3.1 shows the overall scores of the ranking.

<table>
<thead>
<tr>
<th>Company</th>
<th>Total score</th>
<th>Rank</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. 1 - 7 AC Technologies</td>
<td>62%</td>
<td>3</td>
</tr>
<tr>
<td>No. 2 - AIL Research Inc.</td>
<td>96%</td>
<td>1</td>
</tr>
<tr>
<td>No. 3 - Be Power Tech</td>
<td>37%</td>
<td>6</td>
</tr>
<tr>
<td>No. 4 - Kathabar Dehumidification Systems, Inc</td>
<td>56%</td>
<td>5</td>
</tr>
<tr>
<td>No. 5 - L-dcs GmbH</td>
<td>90%</td>
<td>2</td>
</tr>
<tr>
<td>No. 6 - Menerga Apparatebau, GmbH</td>
<td>60%</td>
<td>4</td>
</tr>
</tbody>
</table>

The results of the ranking indicate that based on the ranking criteria and the assessment of how well the companies might perform in accordance to the ranking statement the **two companies AIL and L-DCS have the highest total scores**, with more than 25% points higher than the third-place company, AC Technologies. The advantages of AIL and L-DCS include their flexibility to adapt their proven technology products to a narrow design envelope, and their willingness to cooperate in fitting their technology to a suitable test site for the pilot HVAC installation in Hawaii.
REFERENCES


REFERENCES


Vandermeulen, P. (2013) "Desiccant Enhanced Air Conditioning", ASHRAE 2013 Winter Conference, Dallas, Texas


The following web sites were accessed and used in the preparation of this report:

<table>
<thead>
<tr>
<th>No.</th>
<th>Company</th>
<th>Web site</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>7 AC Technologies</td>
<td><a href="http://7actech.com">http://7actech.com</a></td>
</tr>
<tr>
<td>2</td>
<td>AIL Research</td>
<td><a href="http://ailr.com">http://ailr.com</a></td>
</tr>
<tr>
<td>3</td>
<td>Be Power tech</td>
<td><a href="https://www.bepowertech.com/">https://www.bepowertech.com/</a></td>
</tr>
<tr>
<td>4</td>
<td>Kathabar Dehumidification Systems, Inc</td>
<td><a href="http://www.kathabar.com">http://www.kathabar.com</a></td>
</tr>
<tr>
<td>5</td>
<td>L-dcs</td>
<td><a href="http://www.l-dcs.de">http://www.l-dcs.de</a></td>
</tr>
<tr>
<td>6</td>
<td>Menerga Apparatebau, GmbH</td>
<td><a href="http://www.menerga.com">http://www.menerga.com</a></td>
</tr>
</tbody>
</table>
APPENDIX A

SELECTED INFORMATION BY CANDIDATE COMPANIES

7 AC TECHNOLOGIES
Desiccant Enhanced Air Conditioning

Sponsor: TC 08.12 Desiccant Dehumidification Equipment and Components

Peter Vandermeulen, President and CEO
7AC Technologies, Inc.
peterv@7actech.com
(781) 574-1348

Learning Objectives

• Explain how liquid desiccants can efficiently produce cool, dry air
• Explain how a membrane can prevent desiccant carryover
• Provide an overview of a liquid desiccant DX air conditioner that can provide high latent loads without the need for reheat
• Explain how liquid desiccants can outperform vapor compression
• Explain how liquid desiccant systems work
• Describe a refrigerant free liquid desiccant based cooling system

AGENDA FOR SESSION

• Introduction to liquid desiccant systems
• Membrane properties and theory
• Liquid desiccant system design
• Desiccant regeneration and waste heat
• Solar Integration
• Energy recovery and Winter Heating
• Conclusions

Liquid Desiccants Remove Humidity

• Membranes prevent desiccant carry-over (corrosion)
• Low regeneration temperature – waste heat
• Lower costs (simple plastics)

Cooling Air

- Conventional HVAC
  • Oversizes to dehumidify
  • Reheats for comfort
- 3-Way Membrane Plates
  • Direct cool/dehumidify
  • No reheat
  • Saves >50% energy

Membrane Liquid Desiccants
Membrane Structure

- Super-hydrophobic material holds desiccant
  - 85 psi breakthrough, <1 psi operating
- Mean Free Path of Molecules = Gap Opening
  - 65% open area

Membrane Plate Concept

- How it works:
  - Water vapor in air is absorbed through membrane into salt solution
  - Latent heat is absorbed into cooling water
  - Sensible heat from air is also absorbed into cooling water
  - Simultaneous cooling and dehumidification

Chiller or Indirect Evaporator

- Cool and dehumidify in 1 step
  - 100% Latent or 100% Sensible Cooling
  - Or anything in between
- Supply Air independent of Outdoor Air
  - Humidity and Temperature of SA
    - Independently settable
    - Variable
  - Efficient in high humidity
  - No reheat needed

Membrane Acts Like Filter

- Chiller set-point sets supply air temperature
  - LiCl Concentration determines humidity ratio of supply air

Membrane Dehumidifier/Cooler

- Characteristics:
  - >65% Efficiency
  - Counter flow design
  - Variable output temperature
  - Variable output humidity
  - 25 Membrane plates
  - 3 Tons of cooling
  - 1 x 2 x 2 ft.
  - 7.8 Year membrane life
System Integration

- Compressor assisted: IEER 23
  - RTU DROP IN

- Geothermal or district loop assisted: IEER 90
  - * No refrigerants

- Indirect evaporative assisted: IEER 70
  - * No refrigerants

Regeneration Heat

- From compressor
- Electric 6 kW
- Heat 16 kW
- Chilled Water

- From external sources
  - Solar
  - Industrial Waste Heat
  - Generator heat
  - Gas boiler

- Electric 2 kW
- Heat 16 kW
- Water 26 gal/hr

Solar Air Conditioning

- Waste Heat Integrated
  - Low regeneration temperature
  - Utilize low enthalpy of return air
  - Solar thermal or PV/thermal hybrid modules

Energy Recovery, Winter Heating

- Outside Air
- Energy Recovery
  - From return air is possible
  - Bypass heat exchanger

- Evaporator
- Condenser
- Desiccant HX

- Compressor
- Exhaust
- Supply Air
- Return Air

- Energy Recovery
  - Winter Heating
    - Reverse compressor
    - No coil freezing issues
    - Desiccant freezes at very low temperatures

Conclusions

- Liquid desiccant systems can cool and dehumidify simultaneously
- Membrane ensures zero carry-over of desiccant on conditioner and regenerator
- Chiller assisted system can use compressor heat for regeneration
- Indirect evaporator system requires no refrigerants
- Low regeneration temperatures allow for waste heat integration
- Energy recovery is possible during shoulder seasons
- Prototype systems have been build and are under test

Acknowledgements

- Eric Kozubal and Jason Woods, NREL
- Tom Hamlin, 3M Purification and Filtration

Questions?

Peter Vandermeulen
peterv@7actech.com
Seminar 30 – Liquid Desiccant Dehumidification
As a Way to Enhance IAQ and Dedicated Outdoor Air System Performance

First Results of Testing and Demonstration of a Membrane Liquid Desiccant DOAS
Learning Objectives

• Explains how a liquid desiccant dedicated outside air system (DOAS) can significantly outperform a compressor-based DOAS system.
• Describes performance modeling of a membrane encased liquid desiccant system.
• Describes a novel liquid desiccant system that utilizes micro-porous membranes to contain the liquid desiccant in a three fluid heat and mass exchanger coupled to a water to water source heat pump.

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This program is registered with the AIA/ASHRAE for continuing professional education. As such, it does not include content that may be deemed or construed to be an approval or endorsement by the AIA of any material of construction or any method or manner of handling, using, distributing, or dealing in any material or product. Questions related to specific materials, methods, and services will be addressed at the conclusion of this presentation.
Outline/Agenda

• Technology Overview
• Demonstration Overview
• Demonstration Results
• Outcome / Future Improvements
What problems are liquid desiccants solving?

Credit: iStock, Photo ID 425081
Liquid Desiccants Can Be The Solution:

- Inefficient humidity control
  - Over cool + reheat
  - Typical DOAS:
    - Annualized EER: 6 – 10 Btu/Wh

- Liquid desiccant systems:
  - Powerful dehumidification
    - Supply air dewpoint < 45°F
    - Annualized EER > 12 Btu/Wh
  - Control latent and sensible separately
Liquid Desiccant Cooling Process
Cooling Processes for DOASs

Vapor Compression + Reheat

- Typical Comfort Zone
- Building Air
- SHR = 0.5

Temperature (°F) vs. Humidity Ratio (gr/lb)
Cooling Processes for DOASs

Solid Desiccant A/C (w/condenser heat)

- Typical Comfort Zone
- Building Air
- SHR = 0.5

Diagram showing relationship between temperature and humidity ratio.
Cooling Processes for DOASs
## Temperature Lift Comparison

<table>
<thead>
<tr>
<th>Method</th>
<th>( T_{\text{cold}} )</th>
<th>( T_{\text{hot}, 1} )</th>
<th>( T_{\text{hot}, 2} )</th>
<th>Carnot Efficiency</th>
<th>Cooling Loss</th>
<th>Net EER (45% x Carnot)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Vapor Compression + Reheat</td>
<td>46°F</td>
<td>100°F</td>
<td>-</td>
<td>31.8°F</td>
<td>15%</td>
<td>12.2°F</td>
</tr>
<tr>
<td>Solid Desiccant A/C w/Condenser Heat</td>
<td>48°F</td>
<td>120°F</td>
<td>100°F</td>
<td>30.4°F</td>
<td>5%</td>
<td>12.9°F</td>
</tr>
<tr>
<td>Liquid Desiccant A/C</td>
<td>54°F</td>
<td>104°F</td>
<td>-</td>
<td>35.0°F</td>
<td>0%</td>
<td>15.7°F</td>
</tr>
</tbody>
</table>
Membrane Plate Assembly

Humid Air

Strong Desiccant

Water Vapor

Heat

Cooling Water

Dilute Desiccant

Supply Air – controlled to:
DB = 60-75°F
DP = 45-55 °F

SEM photo of vapor permeable membrane
Credit: Celgard, LLC
Module Construction

Credit: Peter Vandermeulen
Outside Air
DB = 89°F
DP = 79°F

Conditioner Membrane Modules

Supply Air
DB = 65°F
DP = 55°F
5200 CFM

Desiccant Heat Exch.

Evap.

Cond.

38 TON Water Chiller

Regenerator Membrane Modules

Outside Air

Exhaust Air
6000 CFM

Credit: Peter Vandermeulen
Beta Prototype

Credit: Colton Cmar

Fresh Air Inlet

Air Outlet
Demonstration Site

Credit: Google
Demonstration Site

Credit: Google
Anderson Air Force Base, Guam

Credit: Google
Demonstration Site

• **Fitness center:**
  – General fitness area
    • e.g. weights, cardio, yoga, etc
  – Office spaces
  – Racket ball courts

• **DOAS Requirement**
  – 5180 CFM of fresh air
  – Supply air conditioned to 55°F dewpoint
  – EER goal > 11.4 Btu/Wh

Credit: Lesley Herrmann
Demonstration Layout

Credit: Marjorie Schott
Optional Layout

Credit: Marjorie Schott
DOAS Application

Credit: Marjorie Schott
Demonstration Layout

Credit: Marjorie Schott
Numerical System Model
Pre Demonstration Modelling
Estimated EER = 12.8 Btu/Wh
Laboratory Testing (Pre-demo)
Measured EER = 9.7 Btu/Wh

Psychrometric Chart at 12 psia - Test: 7AC_Gen2_v52_Test16

- Precooled Air
- Inlet Air (OA)
- Supply Air
- Regenerator Air Outlet
- Regenerator Potential

Enthalpy lines:
- Enthalpy = 25 BTU/lbm
- Enthalpy = 30 BTU/lbm
- Enthalpy = 35 BTU/lbm
- Enthalpy = 40 BTU/lbm
- Enthalpy = 45 BTU/lbm

Desiccant Potential lines:
- Desiccant Potential - LCI @ 26.3%

Process Air
Regen Air
Conditioner Potential
Regenerator Potential
Demonstration Measurements
Optimal operational EER = 9.7 Btu/Wh
LDAC Airstream Measurements

Measured Saturated Evaporator = 44°F
Modeled: 54°F

Modeled: 117°F

Measured Saturated Condensing = 124°F
Comfort Conditions

Office Space Conditions Comparison with LDAC On & Off

- Typical Warm Climate Office Comfort Zone
- Outdoor - LDAC On
- Supply Air - AHU 1 - LDAC Off
- Office - LDAC On
- Supply Air - AHU 1 - LDAC On
- Office - LDAC Off

Temperature (Degrees F) vs Humidity Ratio (gr/lb)
Performance Improvement

- Current system design: 9.7
- Correctly spec’d and sized chiller: 2.05
- Improved desiccant distribution: 0.81
- Reduced fan power: 0.39
- Improved system design: 12.8

EER (Btu/Wh)

0.0 1.0 2.0 3.0 4.0 5.0 6.0 7.0 8.0 9.0 10.0 11.0 12.0 13.0 14.0

- Current system design
- 1. Correctly spec’d and sized chiller
- 2. Improved desiccant distribution
- 3. Reduced fan power
- Improved system design
Conclusions

• Beta prototype demonstration
  – Used 32% more energy than expected
    • Requires custom chiller design – 63%
    • Refine desiccant distribution – 25%
    • Horizontal air flow + reduce external static – 12%
  – Reduce package size
  – Remove unnecessary pre-cool coil

• Technology potential is high
  – Excellent supply air capabilities
  – EER > 12 Btu/Wh
    • Energy savings > 30%
Acknowledgements

NAVFAC
   – Florence Ching, Wah-Cheong Sze, Aaron Kam, Jim Low and Kevin Hurley

Anderson AFB, Guam
   – Omar Guiao and John San Nicolas

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   – Joel Yuen, Raymond Cheng, Kevin Ku and Ross Takai

Black Construction
   – Don McCann and Angelo Armes

7AC Technologies
   – Mark Allen, Josh Roberts, Scott Rowe, Peter Luttik, Mark Rosenblum, and Jed Swan

Mountain Energy Partnership
   – Mark Eastment, Greg Barker, Ed Hancock

National Renewable Energy Laboratory
   – Michael Deru, Jason Woods, Lesley Herrmann, Jeff Dominick, Gene Holland, Karri Bottom and Ron Judkoff
Questions?

Eric Kozubal

Eric.Kozubal@NREL.gov
APPENDIX B

SELECTED INFORMATION BY CANDIDATE COMPANIES

ADVANTIX SYSTEMS
Liquid Desiccant: How it works
What is liquid desiccant?
Start with a very salty solution...
Which will create the “Dead Sea Effect” of absorption
A two part system enables transport...

Any imbalance will create a driving force for equilibrating the solutions between the two parts.
...and finally adding a heat source creates a continuous dehumidification process
Basic Function: thermal energy can be derived from many sources

Thermal Transfer Source
- Heat Pump - Electricity only
- Electricity and external hot/cold water
- Hybrid - Heat pump AND external/renewable sources

- Heat Pump models maximize convenience (plug & play)
- External hot/cold models take advantage of existing or renewable thermal sources/sinks to provide maximum possible energy savings
How it works
Background: The science of humidity control
A building’s air conditioning load comes from a variety of sources

<table>
<thead>
<tr>
<th></th>
<th>External</th>
<th>Internal</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Thermal</strong></td>
<td>• Heat conduction through envelope</td>
<td>• Lights</td>
</tr>
<tr>
<td>(Sensible)</td>
<td>• Fenestration</td>
<td>• Fans &amp; other motors</td>
</tr>
<tr>
<td></td>
<td>• OA Ventilation (sensible portion)</td>
<td>• Office equipment and electronics</td>
</tr>
<tr>
<td></td>
<td>• Infiltration (sensible portion)</td>
<td>• Miscellaneous plug loads</td>
</tr>
<tr>
<td><strong>Moisture</strong></td>
<td>• OA Ventilation (latent portion)</td>
<td>• People (latent portion)</td>
</tr>
<tr>
<td>(Latent)</td>
<td>• Infiltration (latent portion)</td>
<td>• Plants</td>
</tr>
<tr>
<td></td>
<td>• Permeation</td>
<td>• Cooking</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Pools, showers, spa</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Washing/ Washdowns</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Drying processes</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Other wet processes</td>
</tr>
</tbody>
</table>
Air conditioning loads require both temperature and humidity control

Primary Sources
- Lighting
- Thermal conduction
- Solar gains
- Plug loads
- Occupants
- Outside air ventilation & infiltration
- Occupants

Typical Building A/C Load

Temperature Control
(sensible load)

Humidity Control
(latent load)

Conventional A/C Process

Temperature
Reduced directly by absorbing heat into the refrigerant

Humidity
Reduced indirectly by overcooling air past condensation point and then adding reheat which demands a lot of energy

Though not apparent on the thermostat, humidity control is equally important to temp control for maintaining comfort, indoor air quality, and building integrity
The fraction of moisture load in HVAC is substantially increasing in building design standards.

Source: TIAAX
Traditional “design” conditions do not reflect the true challenge of moisture control in modern buildings.

Realistically, the worst-case conditions are already at about 50% - smart designers are increasingly moving away from the “cooling design” load.

Source: TIAX
... resulting in a difficulty controlling moisture below full load conditions

- Duty cycle, aka “Runtime Fraction” is important for conventional coils
- At part load, it is very difficult for standard cooling coil to dehumidify without over-cooling, or at least large temperature swings (long-cycles)
- True for BOTH chilled water and DX coils, when cooling source is removed, the condensed water will re-evaporate into the air stream
… And can be even more challenging in facilities with more stringent requirements

<table>
<thead>
<tr>
<th>High ventilation requirements</th>
<th>Low dewpoint requirements</th>
<th>High internal humidity load</th>
</tr>
</thead>
<tbody>
<tr>
<td>Health Care</td>
<td>Supermarkets</td>
<td>Indoor pools</td>
</tr>
<tr>
<td>Hospitality (hotels, restaurants, auditoriums)</td>
<td>Hospitals (esp. operating rooms)</td>
<td>Food Products/Processing</td>
</tr>
<tr>
<td>Schools</td>
<td>Pharmaceutical / Nutraceutical production</td>
<td>Health Clubs</td>
</tr>
<tr>
<td>Labs</td>
<td>Plastic Molding</td>
<td></td>
</tr>
<tr>
<td>Electronics manufacturing</td>
<td>Painting &amp; Printing</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Cold / Dry storage</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Ice rinks</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Any space using chilled beams or VRF systems</td>
<td></td>
</tr>
</tbody>
</table>
Conventional equipment does not sufficiently address the humidity control challenge in modern buildings

1980

2010

<table>
<thead>
<tr>
<th>EER</th>
<th>8</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>Ozone-depleting R-22</td>
<td>R-410a, R-407c environmentally-friendly</td>
</tr>
<tr>
<td>Humidity Control/SHR</td>
<td>~20% of load @ full capacity (less at part load)</td>
<td>~20% of load @ full capacity (less at part load)</td>
</tr>
</tbody>
</table>

• Though efficiency has improved, conventional equipment is fundamentally limited in its ability to treat moisture load
• The SHR limits the moisture removal capability required to maintain required IAQ of modern buildings

Failure to respond has caused...

- oversizing
- inadequate ventilation
musty odors
mold & bacteria
maintenance issues
ASHRAE best practice design standards call for separate equipment to treat ventilation and/or latent loads

ASHRAE Handbook Ch. 6.7: Although most centralized and decentralized systems are very effective at handling the space sensible cooling and heating loads, they are less effective (or ineffective) at handling ventilation air or latent loads. As a result, outside air should be treated separately.
Dealing with the latent load
## Basic approaches to humidity control

<table>
<thead>
<tr>
<th>Approach</th>
<th>Technology</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dedicated Outside Air System (DOAS)</td>
<td>• <strong>Overcools</strong> as above, has packaged <strong>hot gas reheat</strong></td>
</tr>
<tr>
<td></td>
<td>• Specialized coils to allow greater moisture removal</td>
</tr>
<tr>
<td>Solid Desiccant</td>
<td>• Hygroscopic chemistry adsorbs moisture</td>
</tr>
<tr>
<td></td>
<td>• <strong>Heat addition</strong> necessitates <strong>pre-cooling</strong> and/or <strong>post-cooling</strong> of air</td>
</tr>
<tr>
<td>Liquid Desiccant</td>
<td>• Hygroscopic chemistry absorbs moisture</td>
</tr>
<tr>
<td></td>
<td>• <strong>Cools</strong> and <strong>dries</strong> air simultaneously</td>
</tr>
</tbody>
</table>
An example to compare the approaches

Example 1: treating 100% outdoor air

Requirements: bring 3000 CFM of humid outdoor air to room-neutral conditions

<table>
<thead>
<tr>
<th>Approach</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dedicated Outside Air System (DOAS)</td>
</tr>
<tr>
<td>Solid Desiccant</td>
</tr>
<tr>
<td>Liquid Desiccant</td>
</tr>
</tbody>
</table>
Mechanical dehumidification/DOAS equipment

Advantages

• Meets moisture load without overcooling the space
• Refrigerant-based systems familiar to contractors and consumers

Limitations

• Energy intensive
• Latent degradation at part load
• High maintenance requirements
Adding reheat enables humidity control with mechanical refrigeration, but at a cost

Example 1: bringing 3000 CFM of humid outdoor-air to room-neutral

Approach 1: Mechanical refrigeration + reheat

Total: 291 MBH
Solid desiccant wheel dehumidification

**Advantages**
- Able to reach extremely low dew points (< 10 gr/lb)

**Limitations**
- Expensive
- Energy intensive
- High maintenance requirements
- Usually requires pre-cooling and/or post cooling equipment
A desiccant wheel also requires significant excess energy input

Example 1: bringing 3000 CFM of humid outdoor-air to room-neutral

Method 2: Solid desiccant

Note: Pre + Post cool configuration (not shown) requires similar energy input

Total: 445 MBH

193 MBH (16 tons)

252 MBH* (For regeneration)
Alternate path of solid desiccant – some energy savings possible
To reach San Diego from Miami, why would you connect through Anchorage?
Only liquid desiccant can use the thermodynamic minimum energy in low-SHR tasks

Example 2: bringing 3000 CFM of humid outdoor-air to room-neutral

Approach 3: Liquid Desiccant

Note: Desiccant regeneration accomplished entirely through condenser heat

Total: 143 MBH
## Liquid Desiccants Do Less Work for the Same Task

### Base processes

<table>
<thead>
<tr>
<th>Units: MBH</th>
<th>Conventional</th>
<th>Solid desiccant</th>
<th>Liquid desiccant</th>
</tr>
</thead>
<tbody>
<tr>
<td>Process</td>
<td>Overcool</td>
<td>Reheat</td>
<td>Desiccant wheel dehumidification</td>
</tr>
<tr>
<td>Work required</td>
<td>217</td>
<td>74</td>
<td>252</td>
</tr>
<tr>
<td>Total energy</td>
<td>291</td>
<td></td>
<td>445</td>
</tr>
</tbody>
</table>

### With “site”energy recovery

<table>
<thead>
<tr>
<th>Optimization</th>
<th>Condenser hot-gas reheat</th>
<th>Condenser heat for regen</th>
<th>on-site heat sink</th>
</tr>
</thead>
<tbody>
<tr>
<td>MBH savings</td>
<td>-74</td>
<td>-200</td>
<td>-70</td>
</tr>
</tbody>
</table>

### Total Energy

- **Base case**
  - Overcool: 291 MBH
  - Reheat: 217 MBH
  - Condenser hot-gas reheat: 245 MBH
  - Condenser heat for regen: 445 MBH
  - Liquid desiccant cooling: 142 MBH

- **Optimized**
  - Overcool: 291 MBH
  - Reheat: 217 MBH
  - Condenser hot-gas reheat: 245 MBH
  - Condenser heat for regen: 445 MBH
  - Liquid desiccant cooling: 142 MBH
There are also non-energy, IAQ considerations to compare between approaches.

**The conventional approach contributes to IAQ issues**

- Cooling coil
- Fungi
- Bacteria
- Viruses

Wet coils & condensate system form a veritable petri dish that the treated air flows over.

**In contrast, tests and field data demonstrate liquid desiccant’s positive effect on IAQ**

- Laboratory testing shows desiccant solution killing 99%+ of microorganisms it contacts.
- Field testing shows 89-98% reduction in airborne microorganisms after install.
- Allergens, particulates, and odor causing molecules also captured by the process.
Flexible in installation: Good for both new build and retrofits

- **Series install (rooftop)**
- **Thermal-driven renewable (mechanical room)**
- **“Tight-spaces” (indoors)**
- **Industrial ventilation (rooftop)**
- **Parallel install (rooftop)**
- **Commercial ventilation (pad-mounted)**
Save green by going green – superior economics of LDAC

By dealing with moisture more efficiently:

- 30-40% lower energy consumption than conventional mechanical systems
- 30-60% lower energy consumption than desiccant wheels

- Comparable (or less) upfront cost to alternative equipment
- Systems sized more efficiently by handling more humidity removal than conventional units
A viable, sustainable solution for many global brands...

**Hospitality**
- The Ritz-Carlton Hotel Company, LLC
- Hard Rock Hotel
- Hilton
- Hooters

**Healthcare**
- Lilavati Hospital
- St. Vincent's Hospital
- Recovery Place

**Other Commercial**
- SeaWorld
- CrossFit
- Ericsson
- Charter Schools USA

**Pharma**
- GlaxoSmithKline
- MERCK
- Teva
- Glenmark
- Zydus

**Food & Beverage**
- Tyson
- Kraft
- P&G
- Cadbury Adams
- MillerCoors
- MeadJohnson

**Other Industrial**
- Siemens
- NASA
- Huawei
- Firestone
How to Apply LDAC
Application: flexible in installation (1/5)

Outdoor air, in parallel

OA → LDAC → RTU → conditioned space

Commercial (School), Pad mounted

Commercial (Restaurant), Rooftop

Industrial Ventilation (Food Processing) Pad mounted
Application: flexible in installation (2/5)

Outdoor air, in series

OA → LDAC

AHU → conditioned space

Industrial Ventilation (Food processing), Rooftop

Commercial (Multifamily), Rooftop

Commercial (Hospital), Rooftop
Application: flexible in installation (3/5)

Internal latent load, in parallel

OA

LDAC

RTU

conditioned space

Commercial (Retail), Rooftop

Industrial (Plastic Molding), Indoor

Commercial (Fitness Center), Indoor
Application: flexible in installation (4/5)

Internal latent load, in series

LDAC  →  AHU

conditioned space

Commercial (Restaurant) Pad mounted

Commercial (School) Rooftop

Industrial (Pharma) Mechanical Room
Application: flexible in installation (5/5)

Other Common Installations

Problem Spaces

Cafeteria

Hotel, Ballroom

Cold/Dry Storage

Indoor Pool

Thermally-Driven Renewable (Office Building)
LIQUID DESICCANT AIR CONDITIONING

Saves energy, Controls humidity, Cleans air
Case Studies
Pharma Production
Environmental control example – neutral conditions

- Compression area requires moderate humidity with strict control during the production process.

- Existing conventional A/C system had high operating costs, owner was seeking a more economical solution for the plant’s air treatment.

- Owner requested that replacement be done with minimum modification to the existing system and no compromise over the desired conditions.

**Design Requirements:**
- 75°F, 50% RH

**Ambient Conditions:**
- 88°F, 80% RH
LDAC solution is less expensive in first cost and operating costs

<table>
<thead>
<tr>
<th>Feature</th>
<th>Conventional A/C System</th>
<th>Liquid Desiccant System</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tons Conventional Cooling</td>
<td>30</td>
<td>20</td>
</tr>
<tr>
<td>Reheat System (kW)</td>
<td>21</td>
<td>-</td>
</tr>
<tr>
<td>Liquid Desiccant System</td>
<td>-</td>
<td>Liquid Desiccant Unit</td>
</tr>
<tr>
<td>Hourly Operating Cost</td>
<td>$5.10</td>
<td>$3.60</td>
</tr>
<tr>
<td>Annual Operating Time (hours)</td>
<td>5,000</td>
<td>5,000</td>
</tr>
<tr>
<td>Annual Operating Cost</td>
<td>$25,500</td>
<td>$18,000</td>
</tr>
<tr>
<td>Annual Operational Savings</td>
<td>None</td>
<td>$7,500</td>
</tr>
</tbody>
</table>

Liquid Desiccant equipment was less expensive than the more energy-intensive outdoor air unit it replaced.
Powder Processing
Low humidity – industrial process example

- 65,000 sq. ft plant
- Powder production / packaging line
- Powder processing requires low and precise humidity control,
- Initial design called for a solid desiccant wheel in additional to conventional A/C (chilled water system) to reach desired environmental control

**Design Requirements:**
- 81 °F, 20% RH

**Ambient Conditions:**
- 88 °F, 80% RH
Operating cost advantage is even greater for low humidity

<table>
<thead>
<tr>
<th></th>
<th>Solid Desiccant Wheel &amp; Chilled Water</th>
<th>Liquid Desiccant &amp; Chilled Water</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tons of Conventional Cooling</td>
<td>93</td>
<td>33</td>
</tr>
<tr>
<td>Approx. Cost of Conventional Equip.</td>
<td>$74,400</td>
<td>$26,400</td>
</tr>
<tr>
<td>Desiccant Equipment</td>
<td>10,000 CFM desiccant wheel</td>
<td>3 x Liquid Desiccant Units</td>
</tr>
<tr>
<td>Cost of Desiccant Equipment</td>
<td>$86,000</td>
<td>$70,000</td>
</tr>
<tr>
<td>Total First Cost</td>
<td>$160,400</td>
<td>$96,400</td>
</tr>
<tr>
<td>Annual Energy Consumption (kWh)</td>
<td>2,348,690</td>
<td>1,139,381</td>
</tr>
<tr>
<td>Annual operating costs</td>
<td>$399,277</td>
<td>$193,695</td>
</tr>
<tr>
<td>5-Year Total Cost</td>
<td>$2,156,787</td>
<td>$1,064,874</td>
</tr>
<tr>
<td></td>
<td></td>
<td>-51%</td>
</tr>
</tbody>
</table>
Big Box Store
Economics still favorable replacing inexpensive packaged DX

Before: 280 tons conventional

After: 150 tons + 18,000 CFM LDAC

Liquid Desiccant conditioned space RTU infiltration door OA
## Systems for comparison

<table>
<thead>
<tr>
<th></th>
<th>Conventional Rooftop DX Units</th>
<th>Liquid Desiccant Units + Rooftop DX Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tons of Conventional Cooling</td>
<td>280</td>
<td>150</td>
</tr>
<tr>
<td># of Conventional Rooftop Units</td>
<td>13</td>
<td>7</td>
</tr>
<tr>
<td># of Liquid Desiccant Units</td>
<td>-</td>
<td>6</td>
</tr>
<tr>
<td>Total Equipment Cost</td>
<td>~$255,000</td>
<td>~$275,000</td>
</tr>
<tr>
<td>Humidity Control</td>
<td>Overcooling to achieve target conditions</td>
<td>Overcooling not required to achieve target conditions</td>
</tr>
<tr>
<td>Projected Savings</td>
<td>$15,000-35,000/year</td>
<td></td>
</tr>
</tbody>
</table>

For a 9% premium in equipment costs, payback is under 1 year.
APPENDIX C

SELECTED INFORMATION BY CANDIDATE COMPANIES

AIL RESEARCH INC.
A Zero Carryover Liquid-Desiccant Air Conditioner for Solar Applications

Preprint

A. Lowenstein
AIL Research, Inc.

S. Slayzak and E. Kozubal
National Renewable Energy Laboratory

To be presented at ASME International Solar Energy Conference (ISEC2006)
Denver, Colorado
July 8–13, 2006
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Printed on paper containing at least 50% wastepaper, including 20% postconsumer waste
A ZERO CARRYOVER LIQUID-DESICCANT AIR CONDITIONER FOR SOLAR APPLICATIONS

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ail@ailr.com

STEVEN SLAYZAK AND ERIC KOZUBAL
NATIONAL RENEWABLE ENERGY LABORATORY
GOLDEN, CO

ABSTRACT
A novel liquid-desiccant air conditioner that dries and cools building supply air has been successfully designed, built, and tested. The new air conditioner will transform the use of direct-contact liquid-desiccant systems in HVAC applications, improving comfort and indoor air quality, as well as providing energy-efficient humidity control.

Liquid-desiccant conditioners and regenerators are traditionally implemented as adiabatic beds of contact media that are highly flooded with desiccant. The possibility of droplet carryover into the supply air has limited the sale of these systems in most HVAC applications. The characteristic of the new conditioner and regenerator that distinguishes them from conventional ones is their very low flows of liquid desiccant. Whereas a conventional conditioner operates typically at between 10 and 15 gpm (630 and 946 ml/s) of desiccant per 1000 cfm (0.47 m3/s) of process air, the new conditioner operates at 0.5 gpm (32 ml/s) per 1000 cfm (0.47 m3/s). At these low flooding rates, the supply air will not entrain droplets of liquid desiccant. This brings performance and maintenance for the new liquid-desiccant technology in line with HVAC market expectations.

Low flooding rates are practical only if the liquid desiccant is continually cooled in the conditioner or continually heated in the regenerator as the mass exchange of water occurs. This simultaneous heat and mass exchange is accomplished by using the walls of a parallel-plate plastic heat exchanger as the air/desiccant contact surface. Compared to existing solid- and liquid-desiccant systems, the low-flow technology is more compact, has significantly lower pressure drops and does not “dump” heat back onto the building’s central air conditioner. Tests confirm the high sensible and latent effectiveness of the conditioner, the high COP of the regenerator, and the operation of both components without carryover.

Keywords: Dehumidifier, Liquid Desiccant, Air Conditioner, HVAC, Solar Cooling

INTRODUCTION
The 20th century was a period during which the cooling and dehumidification of homes and commercial buildings switched from being a luxury to a necessity. In the U.S. alone, air conditioning is a $10 billion industry that uses over 4.3 quads (4.54 billion GJ) of primary energy, almost all of which comes from non-renewable sources. Perhaps equally as important as its energy use, air conditioning is often the single largest cause of overloaded electric transmission and distribution systems.

Now, at the start of the 21st century, there is growing awareness that our approach to air conditioning must change if its benefits are to continue and even expand into the developing regions of the world. One obvious change is to design buildings so that comfortable conditions can be maintained with less active cooling and dehumidification. A second is to develop air conditioners that run on renewable energy sources.

But these changes are not enough. Other challenges now face the industry that provides systems for heating, ventilation and air conditioning (HVAC). Indoor environments are often uncomfortable and unhealthy because humidity is too high. The fundamental problem is that a cold heat exchanger, whether it is a chilled-water coil or a DX evaporator, is a poor way to dehumidify air. A 45°F (7.2°C) heat exchanger will typically provide 70% of its total cooling as sensible cooling (i.e., temperature reduction) and 30% as latent cooling (i.e., dehumidification). In many applications, this latent/sensible split must be reversed if indoor humidity is to be adequately controlled.

Desiccants—which are materials that have a high affinity for water vapor—can be part of a sustainable approach to maintaining healthy and comfortable indoor environments. Desiccants are unique in that they can dry air without first cooling the air below its dewpoint. Latent cooling can be more than twice sensible cooling. Once the desiccant is loaded with water, heat is used to return the desiccant to its “dry” state. The high electrical demand of the compressor in a conventional air...
conditioner is replaced by the need for thermal energy to regenerate the desiccant. This creates an important opportunity to use solar thermal energy for air conditioning.

PAST WORK ON SOLAR DESICCANT COOLING

There have been numerous attempts at capturing the benefits of desiccants in a solar air conditioner. In one of the earliest efforts, Löf proposed a solar air conditioner that used triethylene glycol (Löf, 1955). In the early 1980s, American Solar King manufactured and sold a residential solar cooling system that used a lithium-chloride solid-desiccant rotor (Coellner, 1986). When energy prices declined in the late 1980s, American Solar King converted their product to a gas-fired unit. Robison conducted a 2-year field test of a solar cooling system that used a calcium-chloride liquid-desiccant conditioner (Robison, 1983). The test demonstrated the technical feasibility of this solar cooling system, but there was no attempt to commercialize the technology. Schlepp and Schultz have summarized the experiences of many solar desiccant cooling activities that followed the energy crisis of the 1970s (Schlepp and Schultz, 1984).

In addition to AIL Research, there are now at least two companies that are commercializing liquid-desiccant technology that can be used for solar cooling. L-DCS Technology is now commissioning a 350-kW solar cooling system in Singapore (L-DCS, 2006). In 2005, Jilier Technology Development introduced the American Genius line of liquid-desiccant air conditioners at the International Air-Conditioning, Heating and Refrigeration Exposition (Jilier, 2005).

STATE OF THE ART OF DESICCANT TECHNOLOGY

Desiccant systems are commonly categorized as either solid or liquid types. Solid-desiccant systems most commonly use a porous rotor with face seals that create two isolated air paths through the rotor. The process air moves through one sector of the rotor, while at the same time, hot regeneration air moves through another. The rotation of the rotor permits continuous dehumidification of the process air without any valves or dampers periodically redirecting the air flows. Because there is no active cooling within the rotor and the rotor itself transfers some heat from the regeneration air to the process air, the dry process air leaves the rotor at a higher enthalpy than it entered.

In most HVAC applications, the process air leaving the desiccant rotor must be cooled before it is supplied to the building.

Figure 1 shows the configuration of most liquid-desiccant systems now being sold for industrial applications. Both the conditioner and regenerator are porous, adiabatic beds that are flooded with desiccant. The desiccant is first cooled before it is sprayed onto the bed of the conditioner. The process air flows through this bed and is both cooled and dried by the desiccant.

A slip stream of desiccant (typically an order of magnitude smaller than the flooding rate) is continually recirculated between the conditioner and a regenerator where the desiccant is re-concentrated using thermal energy. Again, the desiccant flows over a porous bed of contact media. However, the desiccant is now first heated, typically to between 180°F and 210°F (82°C and 99°C), before it is sprayed onto the bed. Air flows through the bed, scavenges the water vapor that is desorbed from the desiccant, and rejects it to ambient.

The flooding rate in both the conditioner and regenerator of a conventional liquid desiccant system is relatively high for two reasons: (1) the entire internal area of the contact bed must be well wetted, and (2) the desiccant flow must have sufficient thermal capacity to ensure that the temperature of the desiccant does not increase or decrease significantly as water is absorbed or desorbed. At the high flooding rates, small droplets of desiccant will be created as the desiccant cascades down through the bed. These small droplets are entrained by the air flowing through the bed. Consequently, a conventional liquid-desiccant system must use a droplet filter or demister to prevent carryover of desiccant out of the conditioner and regenerator. In well-maintained systems, the droplet filter/demister will essentially eliminate desiccant carryover.

A LOW-FLOW ZERO-CARRYOVER LIQUID DESICCANT CONDITIONER AND REGENERATOR

The present need for increased ventilation and better humidity control within residential and commercial buildings has spurred interest in desiccant systems. However, most sales have been solid-desiccant systems. While the sales of both systems are limited by their higher costs, liquid-desiccant systems are perceived as having more intensive maintenance requirements, which further depress sales.1

A new generation of liquid-desiccant conditioners and regenerators that meets the needs of HVAC applications has been developed and proven. The two most important improvements are (1) desiccant flooding rates have been decreased by a factor of 10 to 20, and (2) contact surfaces are no longer adiabatic, being continually cooled in the conditioner and continually heated in the regenerator. These two changes are related in that when the desiccant flooding rate is decreased, the thermal capacitance of the flow is proportionately

1 A dedicated outdoor air system based on vapor-compression technology will cost on the order of $6 per cfm ($12.70 per l/s), while one based on liquid or solid desiccants will cost closer to $10 per cfm ($21.20 per l/s).
decreased. If the contact surface was adiabatic, the desiccant’s temperature would either rapidly increase in the conditioner or rapidly decrease in the regenerator, and the driving potential for the exchange of water vapor would be lost.

The preceding two improvements in liquid-desiccant technology lead to a much more competitive cooling system. Compared to the technology now in use, a low-flow liquid-desiccant air conditioner (LDAC) will

• have much lower pressure drops
• be more compact
• produce a greater cooling effect (e.g., lower cfm/ton)
• more deeply dry the process air, and
• have a higher COP.

Perhaps most importantly, both the low-flow conditioner and regenerator will operate without the entrainment of desiccant droplets by the air streams (i.e., zero desiccant carryover).

As shown in Figure 2, a LDAC that uses the low-flow technology has three main components: (1) the conditioner, (2) the regenerator, and (3) the interchange heat exchanger. The conditioner is a parallel-plate heat exchanger in which the plates are water-cooled. Films of desiccant flow in thin wicks on the outer surfaces of the plates. The process air (horizontal arrows) flows through the gaps between the plates and comes in contact with the desiccant. The desiccant absorbs water vapor from the air, and the heat that is released is transferred to the cooling water. The air leaves the conditioner drier and at a lower enthalpy (i.e., cooling occurs, although most of the cooling may be latent rather than sensible).

The water absorbed by the desiccant in the conditioner is desorbed in the regenerator. This component is again a parallel-plate heat exchanger, but now hot water (or other heat-transfer fluid) flows within the plates. The hot desiccant films that flow on the outer surfaces of the plates desorb water to a flow of scavenging air (horizontal arrows) that rejects the water to ambient.

The interchange heat exchanger, which transfers heat from the hot, strong desiccant leaving the regenerator to the cool, weak desiccant flowing to the regenerator, performs a dual function. It improves the efficiency of the regenerator by preheating the weak desiccant. It also increases the cooling provided by the conditioner by reducing the heat load imposed by the strong desiccant.

THE IMPLEMENTATION OF LOW-FLOW TECHNOLOGY

Although many liquids have desiccant properties, solutions of halide salts, particularly lithium chloride and calcium chloride, are the most viable liquid desiccants for solar applications. However, the high chloride concentrations in solutions of these salts eliminate even most stainless steels from service in contact with the desiccant. If maintenance is to be acceptable, all wetted surfaces of a LDAC should be a plastic with suitable properties.

Figure 3 – A 6,000 cfm low-flow liquid-desiccant conditioner

Figure 3 shows a plastic-plate heat exchanger that functions as a 6,000-cfm liquid-desiccant conditioner. The plates are made from a plastic extrusion. The cross-section of each plate, which is shown in Figure 4, is 0.1 in. by 12.0 in. (2.5 mm by 305 mm), with 110 cooling passages running the
length of the extrusion. The plates have a thin—approximately 0.020 mil (0.5 mm)—wick covering their surfaces to ensure even wetting by the desiccant. Each plate is bonded to an upper and lower end-piece. For the conditioner shown in Figure 3, 198 plate/end-piece assemblies are stacked and bonded together. When stacked and bonded, the upper end-pieces form two isolated flow circuits: one for distributing desiccant onto the plate surfaces, and the other for circulating a cooling fluid within the plates. In a similar fashion, the lower end-pieces form a collection sump for the desiccant that flows off the plates. Additional features of the conditioner are described in U.S. Patent 6,745,826 and several pending foreign patents.

The preceding conditioner can operate effectively at desiccant-to-air mass flow ratios 20 to 30 times less than those in a conventional liquid-desiccant conditioner. At these low desiccant flows, the liquid films on the plates of the conditioner are contained within the wicks that cover the plates. As described in a later section, this design for the conditioner has a large operating envelope within which the process air does not entrain droplets of desiccant. Furthermore, because droplets are not created when the desiccant is either delivered to or collected from the plates, droplet carryover is completely suppressed during normal operation.

A low-flow regenerator functions similarly to a conditioner, the major difference being that now a hot fluid flows within the plates instead of a coolant. The high operating temperatures forces several design changes.

As with the conditioner, polymers can best deal with the corrosiveness of the liquid desiccant. Because both the efficiency and water-removal capacity of a scavenging-air regenerator increase with operating temperature, a polymer should be selected that withstands high temperatures (e.g., temperatures on the order of 212°F [100°C]). Polymers in the polysulfone family can meet this temperature requirement.

Thermal expansion is more of an issue in designing the regenerator. Polymers have coefficients of thermal expansion (CTE) that are an order of magnitude greater than metals. The design previously described for the conditioner would make a poor regenerator because it fixes both ends of the plates to common manifolds. A non-uniformity in temperature between neighboring plates will induce stresses that could break the adhesive seals within the structure. The design for the regenerator locates the inlet and outlet manifolds for the hot heat transfer fluid at the same end of the plates. The passages within the plates create a two-pass flow circuit between the inlet and outlet manifolds. With both fluid connections at the same end, the opposite ends of the plates are unconstrained. Each plate can expand independently of its neighbor.

Figure 5 shows a low-flow scavenging-air regenerator with “hanging” plates. Each plate is 0.12 in. thick, 4.5 in. wide and 24 in. long. (3 mm x 11.4 cm x 61 cm) The 21 plates of the regenerator provide a design water-removal capacity of 18 lb/h (8.2 kg/h).

**A COMPARISON OF THE LOW-FLOW CONDITIONER WITH CONVENTIONAL DESICCANT TECHNOLOGY**

A unique feature of the low-flow liquid-desiccant conditioner is the integration of heat and mass transfer in one low pressure-drop component. In contrast to this dual-function configuration, the rotor of a solid-desiccant system does only adiabatic drying with the process air being cooled in a separate heat exchanger. For a conventional liquid-desiccant conditioner, the process air is both dried and cooled, but a separate heat exchanger must be used to cool the desiccant before it flows onto the contact bed.

By combining heat and mass transfer into a single component, the low-flow liquid-desiccant conditioner will be more compact and have lower air-side pressure drops than existing desiccant technologies. The lower desiccant flow rate compared to a conventional liquid-desiccant conditioner also reduces pump power by close to an order of magnitude.

The performance of a low-flow liquid-desiccant conditioner is next compared with that of conventional liquid-desiccant and solid-desiccant systems. Manufacturer’s data is used to predict the performance of the two conventional systems.

All three systems are designed to meet the following constraints:

- 6,000 scfm (2.8 m$^3$/s) of outdoor air at 95°F and 118 gr/lb (35°C and 16.9 g/kg) are processed
- Cooling tower water supplied at 86°F (30°C) is used for cooling
- Air velocities at the face of the rotor or conditioner are 400 fpm (2 m/s).

With the preceding constraints, the low-flow liquid-desiccant conditioner that has been described in the preceding section and which operates with 44% lithium chloride solution will supply air at 93.3°F, 24.4% rh, and 57.0 gr/lb (34.1°C and 8.14 g/kg). The air-side pressure drop will be about 0.3 in. w.c. (75 Pa), and the water-side pressure drop will be less than 1 psi (6,900 Pa). If two conditioners are placed in series, air will be supplied at 92.6°F, 18.3% rh, and 41.7 gr/lb (33.7°C and 5.95 g/kg), and the air-side pressure drop will be doubled. The desiccant circulator pump will have a 1/5th HP motor that will draw 200 W (one pump/motor per conditioner).

For the conventional, high-flooding-rate conditioner, the cooling-tower water cools the liquid desiccant before it is sprayed onto the contact bed. Assuming a conditioner configuration in which the process air flows horizontally
through the bed, a representative supply air condition will be 97°F, 20.3% rh, and 53.0 gr/lb (36°C and 7.57 g/kg). The air-side pressure drop through the conditioner will be 1.3 in. w.c. (324 Pa). The desiccant recirculator pump will have a 2 HP motor that will draw 1.5 kW.

The conventional conditioner will also be larger than the low-flow conditioner. Not including inlet and outlet plenums, a conventional conditioner that processes a nominal 7,500 cfm will be 61 in. x 60 in. x 92 in. (W x D x H; 1.55 m x 1.52 m x 2.34 m). For the same air flow, the low-flow conditioner will be 65 in. x 40 in. x 77 in. (1.65 m x 1.02 m x 1.96 m).

The psychrometric performances of the preceding four systems are compared in Figure 6, in which the x-axis is absolute humidity expressed in mass of water per mass of dry air. The single low-flow liquid-desiccant conditioner, the solid-desiccant rotor, and the conventional liquid-desiccant conditioner all supply air at about the same humidity. The single low-flow liquid-desiccant conditioner does deliver air at a lower temperature and slightly lower enthalpy. A more important difference is that the low-flow liquid desiccant conditioner delivers this cooling with only a 0.3 in. w.c. (75 Pa) pressure drop, while the other two systems have pressure drops between 1.3 and 1.7 in. w.c. (324 and 423 Pa). The use of two low-flow conditioners in series increases latent cooling by 25% and total cooling by 14%, but still has a pressure drop that is less than one-half that of the alternative systems.

**Figure 6 – Comparative performance of solid- and liquid-desiccant systems**

For the solid desiccant system, the rotor is 400 mm deep and regenerated at 250°F (121°C). The hot, dry process air leaving the rotor is cooled in a 4-row finned tube heat exchanger that has an 80% effectiveness and is cooled with cooling tower water. At these operating conditions, the process air leaves the rotor at 151°F and 55.0 gr/lb (66.1°C and 7.85 g/kg). The heat exchanger downstream of the rotor cools the supply air to 99°F and 19.8% rh (37.2°C) with no change in humidity. The pressure drop through the rotor is about 1.5 in. w.c. (373 Pa) and through the heat exchanger, 0.24 in. w.c. (60 Pa). The drive motor for the rotor draws on the order of 200 W (at 6,000 cfm [2.83 m³/s], each 1.0 in. w.c. [250 Pa] air-side pressure drop increases fan power by about 1.3 kW).

**A COMPARISON OF THE LOW-FLOW REGENERATOR WITH CONVENTIONAL DESICCANT TECHNOLOGY**

Both the solid-desiccant and liquid-desiccant systems can use solar thermal energy for regeneration. This thermal energy can be provided by either glazed, flat-plate collectors or evacuated-tube collectors. (A third type of collector—concentrating, tracking collectors—tend to be used in very large systems and are not compared here.) Flat-plate collectors are less expensive, but they supply thermal energy at a lower temperature: their installed cost will be on the order of $25 to $40 per square foot ($269 to $431 per square meter), and at peak summer conditions, they will deliver between 50% and 60% of the incident solar radiation as hot water at 180°F (82°C). Evacuated-tube collectors will have an installed cost that is 1.5 to two times that of a flat-plate collector, but they will achieve the same 50% to 60% collection efficiency when operating at 250°F (121°C) or higher.

The selection of the collectors for a solar cooling system is a trade-off between their cost and performance. In general, the Coefficient of Performance (COP) for desiccant regeneration—defined as the thermal energy needed to evaporate a unit mass of pure water divided by the thermal energy supplied to the regenerator to remove the same mass of water from the desiccant—increases at higher temperatures. For liquid-desiccant systems, the improvement in COP with increasing regeneration temperature is most dramatic when the desiccant is regenerated in two stages (similar to the double-effect generator of an absorption chiller). While gas-fired two-stage desiccant regenerators have been developed, no comparable technology is available for solar applications. When used with a single-stage liquid-desiccant regenerator, such as the scavenging-air regenerator shown in Figure 5, the higher operating temperature of the evacuated-tube collector will not justify its higher cost.

As discussed in a later section, the 44% lithium-chloride solution that produces the performance shown in Figure 6 for the low-flow liquid-desiccant systems can be regenerated at a thermal COP of 0.80 in a low-flow regenerator that operates at 180°F (82°C). On a peak summer day, a glazed, flat-plate collector can deliver between 50% and 60% of the incident solar energy to the regenerator at this temperature. Thus, based on incident solar energy, the regeneration COP for the liquid desiccant will be between 0.40 and 0.48. Assuming a collector installed cost of $32.50 per square foot ($350 per square meter) and a peak solar insolation of 317 Btu/hr-ft² (1,000 W/m²), the solar collectors cost $2,800 per peak ton ($796 per peak kW) of latent cooling.

A conventional packed-bed regenerator that operates at the same conditions as the preceding low-flow regenerator will have a thermal COP of 0.55. The flat-plate solar collectors that provide thermal energy to this regenerator will cost $4,070 per peak ton ($1,158 per peak kW) of latent cooling.

The performance for the solid-desiccant cooling system shown in Figure 6 assumes that the desiccant is regenerated at 250°F. This relatively high temperature can be supplied by evacuated-tube collectors, but not the lower cost flat-plate collectors. At this temperature, the solid-desiccant regeneration COP will be 0.5, and the efficiency of the solar
collectors on a peak summer day will be between 50% and 60% (i.e., percent of incident solar radiation delivered to the heater for the desiccant regenerator). Thus, based on incident solar energy, the regeneration COP for the solid desiccant will be between 0.25 and 0.30. Assuming a collector installed cost of $65.00 per square foot ($700 per square meter) and a peak solar insolation of 317 Btu/hr-ft² (1,000 W/m²), the solar collectors cost $9,000 per peak ton ($2,560 per peak kW) of latent cooling.

ENERGY STORAGE WITH LIQUID DESICCANTS

A competitive solar cooling system must store energy if it is to effectively use the thermal energy provided by its solar collectors. Peak solar insolation will occur mid-day, while cooling loads for the building peak in the afternoon and extend into the early evening. At a minimum, several hours of storage are needed to accommodate this mismatch.

A liquid-desiccant cooling system has an important advantage over all alternatives in solar applications because of the ease with which concentrated desiccant can be stored (Kessling et al., 1998). All PV-based cooling systems will be penalized for the expense and inefficiency of battery storage. Although these systems can store “cooling” as either chilled water or ice, both options impose additional economic penalties. Chilled-water storage requires very large, insulated storage tanks. Ice storage systems can be much smaller, but they are inefficient. (In conventional applications, ice can be made at night when electric rates are low, and the lower ambient temperatures compensate for the low evaporator temperatures needed to make ice. For a PV-based cooling system, ice storage would require making ice during the higher ambient temperatures of mid-day.)

Solar thermal cooling systems that use absorption chillers, adsorption chillers, or solid-desiccant systems must store energy as hot water which can later be used to run the cooling system. For single-effect technologies, hot water must be stored at between 190°F and 210°F (88°C and 99°C). Because the COP for single-effect technologies will be on the order of 0.6, approximately 60% more thermal energy must be stored than the cooling that is eventually provided. The higher COP of double-effect technologies (COPs closer to 1.0), greatly reduces the quantity of thermal energy that must be stored, but now the storage temperature must be over 320°F (160°C).

Compared to batteries, hot water, ice, and chilled water, the storage of concentrated desiccant imposes a relatively modest economic penalty and no efficiency penalty on the liquid-desiccant cooling system. Concentrated desiccant can be stored in uninsulated plastic tanks with no loss in cooling potential over time. For cooling systems that use a solution of lithium chloride that cycles between 38% and 44%, the density of storage will be 8.3 gallons per ton-hour latent cooling (2.48 liter/MJ). This is a lower volumetric requirement than the 10 gallons per ton hour (2.99 liter/MJ) that is typical of ice storage. At $2.50 per pound ($5.50 per kg) for anhydrous lithium chloride, the cost for storage will be about $80 per ton-hour ($6.32 per MJ), a value that is comparable to ice storage.

A solar cooling system that uses liquid desiccants can dramatically reduce its cost for storage by replacing lithium chloride with calcium chloride. The change does reduce the cooling capacity for the system because calcium chloride is a significantly weaker desiccant than lithium chloride. At typical high-load conditions, the switch to a 44% calcium chloride solution will decrease the total cooling effect by between 25% and 30%. While this loss is significant, the switch will reduce storage costs to less than $15 per ton-hour ($1.19 per MJ). More storage can then be part of the solar cooling system, which will greatly improve the utilization of the solar collectors.

PERFORMANCE TESTING OF THE LOW-FLOW CONDITIONER

The pre-production prototype described in this section is the product of a 5-year development effort that has progressively improved the cooling performance, pressure drop, and carryover suppression of low-flow liquid-desiccant conditioners. The 1,200-cfm (0.566-m³/s) prototype had 42 plates, each plate 4 feet in length, and a 3.47-ft² (0.323-m²) face area.

Testing was conducted at the Advanced Thermal Conversion Lab of the National Renewable Energy Laboratory’s (NREL) Center for Buildings and Thermal Systems. This research facility accelerates development of high efficiency HVAC concepts by rapidly and accurately evaluating the thermodynamic performance and design features of full-scale prototypes and comparing them to the state-of-the-art. The lab’s current technical specifications and unique capabilities are detailed in Slayzak and Ryan (2004). Airflows in these experiments were measured to ±2%; drybulb and dewpoint temperatures to ±0.3°F (±0.2°C). Uncertainties for the resulting grain depressions are therefore approximately ±2 gr/lb (0.3 g/kg) for dry inlet air and ±3 gr/lb (0.4 g/kg) at the most humid conditions examined. Desiccant concentrations were monitored manually throughout testing by a temperature-compensated hydrometer with 0.1% concentration graduations and were controlled to within ±0.25 concentration points of reported values.

The performance of a low-flow prototype and a conventional packed-bed conditioner were measured at this facility. The prototype was tested under the following conditions:

- Desiccant – commercial lithium chloride and water solution
- Inlet air to conditioner drybulb temperature – 86°F (30°C).
- Cooling water inlet temperature and desiccant inlet temperature – set to provide an outlet drybulb temperature from the conditioner equal to the inlet drybulb temperature
- Cooling water flowrate – 15 gpm (56.8 l/min)
- Desiccant flowrate – 0.5 gpm (1.9 l/min).

An important objective in developing the low-flow liquid-desiccant technology was to reduce the air-side pressure drop through the conditioner. The data in Figure 7 show that at the design face velocity of 400 fpm (2.0 m/s), the pressure drop for the prototype is approximately one-tenth that for the conventional conditioner: 0.3 in. w.c. versus 3.4 in. w.c. (75 Pa versus 846 Pa).

Desiccant concentration has a negligible effect on pressure drop at the design face velocity. At twice the design
flow rate of desiccant, an increase in desiccant concentration from 36% to 44% produced a 15% increase in air-side pressure.

**Figure 7. Comparison of the pressure-drop characteristics of a low-flow conditioner and a conventional conditioner**

Figure 8 compares the dehumidification provided by the low-flow prototype with that of a conventional packed-bed conditioner. (The 100 and 150 grain/pound full scales of the x-axis and y-axis in Figure 8 correspond to 14.3 and 21.4 g/kg.) Both units operated at a face velocity of 400 sfpm (2.0 m/s). The temperatures of the cooling water to the low-flow prototype and the desiccant to the industrial unit were adjusted so that at each operating point, the supply air temperature was 86F (30C).

As shown in Figure 8, when processing air at 115 gr/lb and 86F (16.4 g/kg and 30C), the low-flow prototype provided 57 gr/lb (8.1 g/kg) of dehumidification when the inlet desiccant concentration was 44%. The packed-bed unit provided almost the same dehumidification when the inlet desiccant concentration was 39%.

The packed-bed conditioner matched the dehumidification of the low-flow prototype when supplied with weaker desiccant. Although this may appear to be an advantage for the packed-bed conditioner, the advantage disappears when considering the concentration that must be returned from the regenerator. The low-flow prototype operates in a “once through” mode: the conditioner is supplied with concentrated desiccant from the regenerator, and the entire flow of the weak desiccant leaving the conditioner is sent to the regenerator. The packed-bed conditioner operates with high recirculation. The concentration of the desiccant that floods the packed bed is close to that of the weak desiccant that drains off it. In the preceding comparison, there is a 5.5-point change in desiccant concentration across the low-flow conditioner. The regenerator, therefore, must supply 44.5% desiccant to the packed-bed conditioner if it is to match the performance of the low-flow conditioner operating with 44% desiccant.

At the 115 gr/lb (16.4 g/kg) operating point, the low-flow conditioner was supplied with cooling water at 79F (26.1C), and the industrial unit was supplied with desiccant at 81F (27.2C). Assuming that the desiccant is cooled in a 67% effective heat exchanger, then this heat exchanger must be supplied with cooling water at 75F (23.9C).

The laboratory tests illustrate the advantages offered by the low-flow desiccant technology. For two conditioners that provide identical cooling, the low-flow conditioner operates with weaker desiccant, higher temperature cooling water, and, most importantly, an air-side pressure drop that is about one-tenth that of the conventional flooded-bed conditioner and its mist eliminators.

Additional performance data for the low-flow liquid-desiccant conditioner from tests at the Advanced Thermal Conversion Laboratory appear in the report by Lowenstein et al. (2005).

**VERIFICATION OF THE ZERO-CARRYOVER OPERATION OF THE LOW-FLOW CONDITIONER**

Tests at the Advanced Thermal Conversion Lab verified that at design operating conditions droplets of desiccant are not entrained by the air flowing through the low-flow liquid-desiccant conditioner. Two approaches were used to map the operating envelope for the prototype: visual inspection for desiccant bridging between the plates and laser particle counting/sizing.

Particle counting was accomplished using a single LasairII Model 310 laser particle counter capable of counting total airborne particle concentrations and grouping them in bins by aerodynamic diameter. The bins for this unit were 0.5-0.7 microns, 0.7-1 microns, 1-2 microns, 2-5 microns, 5-10 microns, and >10 microns.

Outdoor air supplied to the prototype during testing was filtered through a 90% effective pleated box filter to establish a particle challenge that was well below the sensor’s saturation limit of >375,000 particles/ft³ (1.3 x 10⁷ particles/m³). Two isokinetic sampling probes were positioned upstream and downstream from the prototype. They were attached via 6-foot (1.8-m) sampling tubes to the sensor so that inlet and outlet concentrations could be alternately measured. The sampling tubes were designed so that they did not trap particles. The pressure differential between the duct from which sample air was drawn and the room to which the sensor discharged sample air was managed to allow the sensor’s internal sampling fan to maintain its continuous sample flow of 1 cfm (0.47 l/s). Each sample point was averaged over approximately 30 seconds. The sensor does not distinguish between solid particles expected in ambient air at the inlet and any liquid droplets entrained into the outlet airflow. The sensor...
rarely indicated any particles greater than 10 microns; this is reasonable considering the filtration implemented, the air supply duct flow conditions, and the settling time for such large airborne particulates.

Figure 9 shows the particle counts for steady-state operation at 400 sfpm (2.0 m/s) and 0.5 gpm (1.89 l/min) desiccant flow. (In Figure 9, the 5,000 per ft³ that is the full scale of the x-axis equals 176,600 per m³.) The outlet particle count closely tracks the inlet challenge. (Similar tests of a flooded packed bed showed that the air downstream of the mist eliminators contained tens to hundreds of thousands of droplets per cubic foot of air in the 0.5-0.7 micron size range.) Because negligible particle arrestance is expected within the prototype as a result of laminar airflow between the parallel plates, droplet generation is inferred to be zero. Because of the need to switch sampling tubes, purge them, reach steady state in the sensor, and then average a sample reading, the time between points in the figure varied from 2 to 3 minutes.

The primary mechanism for droplet generation in the parallel-plate prototype is bridging of desiccant between the plates and subsequent shattering of the liquid bridge by the airflow. Other mechanisms include air bubbles in the desiccant feed line that sputter as they exit the distribution header onto the wicks and high air and/or desiccant flow rates that lead to thick desiccant films at the trailing edges of the plates that bridge the air gaps. These mechanisms, which can be visually detected, were not present during the particle counting tests.

Figure 10 summarizes a rough operating envelope for zero desiccant carryover as determined by visual inspection. (In Figure 10, the 1.25-gpm and 800-fpm full scales of the x-axis and y-axis equal 4.72 l/min and 4.07 m/s.) Desiccant flow, desiccant concentration, and airflow were varied to see under what combinations the system was able to suppress carryover. At 44% concentration, the highest tested, and 400 afpm (2.0 m/s) face velocity droplet generation was not observed below 0.75 gpm (2.84 l/min). At 500 afpm (2.5 m/s), the operating envelope became more restricted with carryover suppressed at desiccant flows below 0.5 gpm (1.89 l/min). Pushing the conditioner to 750 afpm (3.8 m/s) required a further reduction to 0.25 gpm (0.95 l/min). The operating envelope for desiccant flow could be extended by 0.25 gpm (0.95 l/min) at all air flow when the concentration was reduced to 40%. A further 0.25 gpm (0.95 l/min) increase was possible at 36% concentration. A 1.0-gpm (3.78 l/min) desiccant flow appeared to be the prototype’s limit under all operating conditions because at this rate, the desiccant flow was no longer completely contained within the wicks that cover the plate surfaces. Once the desiccant film is thicker than the wick, the fluid dynamic shear of even a low airflow can move desiccant to the plate trailing edges, causing bridging and carryover.

These tests demonstrate an operating envelope for the low-flow conditioner that allow it to meet equipment size and performance requirements (e.g., nominal operation at 400 fpm [2.0 m/s] and 0.6 gpm [2.28 l/min] for the 1,200-cfm [0.56 m³/s] prototype, while effectively suppressing droplet carryover).

**PERFORMANCE OF A LOW-FLOW LIQUID-DESICCANT REGENERATOR**

The low-flow technology that has been successfully applied to the liquid-desiccant conditioner will also improve the performance of the regenerator. In addition to eliminating desiccant carryover and reducing pressure drops, low-flow technology will increase the regenerator’s COP beyond that of conventional packed-bed regenerators. Furthermore, these efficiency improvements extend to lower regeneration temperatures, making the low-flow liquid-desiccant air conditioner attractive in distributed generation applications with both engines and PEM fuel cells, as well as solar thermal collectors.

The 21-plate model of the low-flow regenerator that is shown in Figure 5 was operated under controlled conditions at NREL’s Advanced Thermal Conversion Lab. Its performance was mapped over a range of desiccant concentrations, operating temperatures, air velocities, and water flow rates.
Figure 11 shows the water removal (WR) and the COP of the low-flow regenerator when concentrating a solution of lithium chloride from 36% to 40%. The air velocity at the face of the regenerator is 100 sfpm (0.51 m/s), and inlet air conditions are 86°F, 0.01649 lb/lb and 12.1 psi (30°C, 16.5g/kg, and 83.4 kPa). The test was conducted to simulate operation with an interchange heat exchanger that had an effectiveness between 65% and 80%. Also shown on this figure are the predictions of AILR’s computer model for the regenerator.

The measured COP for the regenerator ranged from 0.62 with 160°F (71.1°C) hot water to 0.73 with 200°F (93.3°C) hot water. Both the measured water removal rate, and the COP agreed well with the computer predictions.

The operation of the regenerator with an air-to-air heat exchanger (AAHX) that recovers thermal energy from the regenerator exhaust air to preheat the incoming scavenging air was simulated by increasing the inlet air temperature to the regenerator without changing its humidity. As shown in Figure 12, these tests show that a 50% effective AAHX would increase the regenerators COP from 0.73 to 0.79 when operating with 200°F (93.3°C) hot water and a 100-sfpm (0.51-m/s) face velocity. (The 30 lb/hr that is the full scale for the x-axis in Figure 12 equals 13.6 kg/hr.) The data in this figure also show the effect of face velocities of 100, 130, and 160 sfpm (0.51, 0.66, and 0.81 m/s) on both the rate of water removal (MRR, which is the same as the parameter WR that was used earlier) and COP.

THE FIELD OPERATION OF A 6,000-CFM ROOFTOP LIQUID-DESICCANT AIR CONDITIONER

Figure 13 shows a rooftop liquid-desiccant air conditioner that is designed to cool and dry 6,000-cfm (2.83 m³/s) of ventilation air. The air conditioner includes a low-flow conditioner and regenerator. It also includes a 400,000-Btu/h (117.2-kW) gas-fired hot-water heater that meets the thermal requirements of the regenerator at 250 lb/h (113.6 kg/h) of water removal. In solar applications, this hot-water heater may be retained as a back-up to the solar collectors or it may be eliminated. A 25-ton (87.9-kW) cooling tower provides 75 gpm (4.73 l/s) of cooling water to the conditioner.

The field operation of the rooftop liquid-desiccant air conditioner began in late September 2005 and continued for 4 weeks at which time ambient conditions in New Jersey became too cold and dry to permit meaningful testing. During the test, the air conditioner operated completely under automatic control, including PID loops for ventilation airflow, boiler temperature, and desiccant concentration. The controller also sequenced all startup and shutdown procedures for the conditioner, regenerator, boiler and cooling tower.

The liquid-desiccant air conditioner operated throughout the test with a one-half scale regenerator. When the air conditioner first operated in the fall of 2004, it included a first-generation regenerator that eventually proved not sufficiently reliable for a commercial product. The hanging-plate regenerator that was described in an earlier section was developed to replace the older design. However, because the new regenerator was not ready until the fall of 2005, when cooling loads would be well below peak summer values, the process of retrofitting the air conditioner with the new regenerator was simplified by installing a unit with one-half the required water-removal capacity.

The highest latent cooling during the test, 141 lb/h (64.1 kg/h) of water removal, occurred on October 5. Table 1 summarizes the air conditioner’s performance for a 43-minute period on that day when ambient conditions averaged 77.7°F and 0.01229 lb/lb (25.4°C and 12.29 g/kg).
The COP of the regenerator during the test period was 0.699. This good COP was achieved despite the poor performance of the IHX. Also, the regenerator did not use an air-to-air heat exchanger to preheat the scavenging air using the warm, humid exhaust from the regenerator.

At full load, the parasitic power requirements for the roof-top air conditioner are

<table>
<thead>
<tr>
<th>Power Requirement</th>
<th>Power (W)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling tower fan</td>
<td>1,600</td>
</tr>
<tr>
<td>Coolant pump</td>
<td>1,100</td>
</tr>
<tr>
<td>Strong desiccant pump</td>
<td>200</td>
</tr>
<tr>
<td>Weak desiccant pump</td>
<td>200</td>
</tr>
<tr>
<td>Hot-water pump</td>
<td>700</td>
</tr>
<tr>
<td>Process fan (at 0.5 in. w.c. external pressure)</td>
<td>2,200 W</td>
</tr>
</tbody>
</table>

During this period, the latent and total cooling supplied by the conditioner were within 3% of the values predicted by the computer models that were used to design the unit.

Both the efficiency of the boiler and the effectiveness of the interchange heat exchanger (IHX) were lower than expected. A possible cause of the IHX’s poor performance may have been that it was not completely purged of air. A new design for the IHX will be installed in the rooftop air conditioner before tests begin in 2006, and better performance is expected.

The nominal efficiency of the boiler is 79% at full firing. During the period reported in Table 1, the boiler operated at about 50% of full firing. It is probable that at part-load firing, the air-to-fuel ratio of the combustor is too lean, which is degrading efficiency.

The COP of the regenerator during the test period was 0.699. This good COP was achieved despite the poor performance of the IHX. Also, the regenerator did not use an air-to-air heat exchanger to preheat the scavenging air using the warm, humid exhaust from the regenerator.

A new generation of liquid-desiccant cooling systems is now being commercialized that will greatly expand the market for solar cooling. The distinguishing characteristic of the new technology is a desiccant flooding rate that is a factor of 10 to 20 lower than the rates now used in conventional packed-bed systems.

Compared to the technology now in use, the low-flow liquid-desiccant air conditioner will:
- have much lower pressure drops
- be more compact
- produce a greater cooling effect (e.g., lower cfm/ton)
- more deeply dry the process air, and
- have a higher COP.

Perhaps most importantly, both the low-flow conditioner and regenerator will operate without the entrainment of desiccant droplets by the air streams (i.e., zero desiccant carryover, which greatly reduces maintenance).

The advantages of low-flow liquid-desiccant technology have been demonstrated in laboratory and field operation. In a controlled laboratory test, a low-flow conditioner matched the dehumidification provided by a conventional packed-bed conditioner, but with less than one-tenth the air-side pressure drop.

The low-flow technology used by the new liquid-desiccant conditioner was proven effective at suppressing droplet carryover, without the use of separate droplet filters or demisters, over a wide range of operating conditions. For operation with a 44% solution of lithium chloride, a face velocity of 400 afpm (2.0 m/s), and a total air flow of 1,388 acfm (0.656 m³/s) droplet carryover was suppressed for all desiccant flows less than about 0.75 gpm (2.84 l/min).

Tests of a scavenging-air regenerator that uses the low-flow technology showed that both water removal and COP were close to the predictions of the computer model that was used to design the regenerator. This regenerator could effectively run on heat provided by either glazed, flat-plate collectors, or evacuated-tube collectors. With an air-to-air heat exchanger that recovers thermal energy from the regenerator’s exhaust air, the regenerator will have a COP of 0.79 when concentrating a solution of lithium chloride from 36% to 40% and supplied with hot fluid at 200F (93.3C).

Although not unique to the low-flow technology, the storage of concentrated liquid desiccant provides an effective means to match cooling loads with the availability of solar energy. The relatively low cost for desiccant storage, particularly systems that use calcium chloride, will ensure a high utilization of the thermal energy provided by the collectors, thereby improving the competitiveness the solar cooling system.

**ACKNOWLEDGEMENTS**

The authors wish to thank the U.S. Department of Energy’s Distributed Energy Program, which is headed by Ms. Patricia Hoffman. The support of this program was critical to the successful development of this technology for use in natural gas fired and distributed generator waste heat applications.
REFERENCES
### Title and Subtitle
A Zero Carryover Liquid-Desiccant Air Conditioner for Solar Applications: Preprint

### Author(s)
A. Lowenstein, S. Slayzak, and E. Kozubal

### Abstract
A novel liquid-desiccant air conditioner that dries and cools building supply air will transform the use of direct-contact liquid-desiccant systems in HVAC applications, improving comfort, air quality, and providing energy-efficient humidity control.

### Subject Terms
Dehumidifier; Liquid Desiccant; Air Conditioner; HVAC
Solar-Powered, Liquid-Desiccant Air Conditioner for Low-Electricity Humidity Control

Energy and Water Projects Demonstration Plan SI-0822

TP-7A40-56437-1

November 2012

Jesse Dean, Eric Kozubal, Lesley Herrmann (NREL)
Jeff Miller and Andy Lowenstein (AIL Research)
Greg Barker (Mountain Energy Partnership)
Steve Slayzak (Coolerado)

Link to Summary Report
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# Acronyms and Abbreviations

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<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AAHX</td>
<td>air-to-air heat exchanger</td>
</tr>
<tr>
<td>AC</td>
<td>air conditioning</td>
</tr>
<tr>
<td>AFB</td>
<td>air force base</td>
</tr>
<tr>
<td>AFRL</td>
<td>Air Force Research Laboratory</td>
</tr>
<tr>
<td>AHU</td>
<td>air-handling unit</td>
</tr>
<tr>
<td>AILR</td>
<td>AIL Research</td>
</tr>
<tr>
<td>ASHRAE</td>
<td>American Society of Heating, Refrigerating, and Air-Conditioning Engineers</td>
</tr>
<tr>
<td>Btu</td>
<td>British thermal units</td>
</tr>
<tr>
<td>CaCl₂</td>
<td>calcium chloride</td>
</tr>
<tr>
<td>cfm</td>
<td>cubic feet per minute</td>
</tr>
<tr>
<td>CHP</td>
<td>combined heat and power</td>
</tr>
<tr>
<td>CPVC</td>
<td>chlorinated polyvinyl chloride</td>
</tr>
<tr>
<td>CoC</td>
<td>cycles of concentration</td>
</tr>
<tr>
<td>COP</td>
<td>coefficient of performance</td>
</tr>
<tr>
<td>DAS</td>
<td>data acquisition system</td>
</tr>
<tr>
<td>DOAS</td>
<td>dedicated outdoor air system</td>
</tr>
<tr>
<td>DoD</td>
<td>Department of Defense</td>
</tr>
<tr>
<td>DOE</td>
<td>Department of Energy</td>
</tr>
<tr>
<td>DX</td>
<td>direct expansion</td>
</tr>
<tr>
<td>EER</td>
<td>energy efficiency ratio</td>
</tr>
<tr>
<td>EISA</td>
<td>Energy Independence and Security Act</td>
</tr>
<tr>
<td>E.O.</td>
<td>executive order</td>
</tr>
<tr>
<td>EPAct</td>
<td>Energy Policy Act</td>
</tr>
<tr>
<td>ESTCP</td>
<td>Environmental Security Technology Certification Program</td>
</tr>
<tr>
<td>ft</td>
<td>foot</td>
</tr>
<tr>
<td>ft²</td>
<td>square foot</td>
</tr>
<tr>
<td>FY</td>
<td>fiscal year</td>
</tr>
<tr>
<td>gal</td>
<td>gallon</td>
</tr>
<tr>
<td>GHI</td>
<td>global horizontal irradiance</td>
</tr>
<tr>
<td>gpm</td>
<td>gallons per minute</td>
</tr>
<tr>
<td>hr</td>
<td>hour</td>
</tr>
<tr>
<td>EUI</td>
<td>energy use intensity</td>
</tr>
<tr>
<td>HMX</td>
<td>heat and mass exchanger</td>
</tr>
<tr>
<td>hp</td>
<td>horsepower</td>
</tr>
<tr>
<td>HVAC</td>
<td>heating, ventilating, and air conditioning</td>
</tr>
<tr>
<td>LDAC</td>
<td>liquid-desiccant air conditioner</td>
</tr>
<tr>
<td>kW</td>
<td>kilowatt</td>
</tr>
<tr>
<td>kWh</td>
<td>kilowatt-hour</td>
</tr>
<tr>
<td>lb</td>
<td>pound</td>
</tr>
<tr>
<td>LiCl</td>
<td>lithium chloride</td>
</tr>
<tr>
<td>MBtu</td>
<td>million British thermal units</td>
</tr>
<tr>
<td>MEP</td>
<td>Mountain Energy Partnership</td>
</tr>
<tr>
<td>NREL</td>
<td>National Renewable Energy Laboratory</td>
</tr>
<tr>
<td>PLC</td>
<td>programmable logic controller</td>
</tr>
<tr>
<td>PPSU</td>
<td>polyphenylsulfone</td>
</tr>
<tr>
<td>PVC</td>
<td>polyvinyl chloride</td>
</tr>
<tr>
<td>RH</td>
<td>relative humidity</td>
</tr>
<tr>
<td>VFD</td>
<td>variable-frequency drive</td>
</tr>
<tr>
<td>W</td>
<td>watt</td>
</tr>
<tr>
<td>yr</td>
<td>year</td>
</tr>
</tbody>
</table>
ACKNOWLEDGEMENTS
The authors would like to thank all of the Environmental Security Technology Certification Program (ESTCP) project team members for their creativity, persistence, and willingness to support this project. Bruce Nielsen at Tyndall Air Force Base was instrumental in setting up the demonstration and has provided countless hours assisting with the installation of the data acquisition system (DAS) and multiyear performance testing. Mountain Energy Partnership provided invaluable assistance with the design and installation of the DAS, as well as data analysis support. Jeff Miller and Andy Lowenstein provided countless hours designing, testing, and commissioning the liquid-desiccant system at Tyndall Air Force Base. Various members of the National Renewable Energy Laboratory’s Commercial Buildings Research team—including James Page, Andrew Parker, and Michael Deru—provided laboratory testing assistance, field testing assistance, and modeling support. Finally, the project would not have been possible without financial support from the ESTCP, which not only provided funding but also provided valuable insights into the types of data analysis procedures and results that would be most beneficial to Department of Defense facilities and engineers.
EXECUTIVE SUMMARY

Today’s air-conditioning (AC) technology is primarily based on direct expansion (DX) or the refrigeration process. It is now so prevalent that it is considered a necessity for the majority of residential and commercial buildings throughout the United States. During the 100-plus years of development, DX AC has been optimized for cost and thermodynamic efficiency, both of which are nearing their practical limits. Nevertheless, AC accounts for approximately 15% of all source energy used for electricity production in the United States alone [nearly 4 quadrillion British thermal units (Btu)], which results in the release of about 343 million tons of carbon dioxide into the atmosphere every year. (DOE 2011)

The Department of Defense (DoD) occupies over 316,000 buildings and 182,000 structures on 536 military installations worldwide, and accounts for about 64% of the energy consumed by federal facilities. This makes the DoD the largest energy consumer in the United States. In Fiscal Year (FY) 2007, the DoD consumed 218 trillion Btu in site-delivered energy, 26.2 trillion Btu for AC alone. This cooling cost equates to an estimated $413 million per year. (Pacific Northwest National Lab, undated)

In hot, humid climates, conventional AC units expend excess energy to sensibly overcool the air for dehumidification. As a result, excess energy must be used to reheat the air to a more comfortable supply temperature (overcool/reheat cycle). The use of desiccant-based AC systems decouples the latent and sensible loads of an airstream, enabling higher efficiency cooling and improved thermal comfort conditions. The following is a list of criteria that can be used to identify feasible sites for liquid-desiccant air conditioner (LDAC) applications:

- Hot and humid climate – latent cooling required most of the year
- One hundred percent outdoor air ventilation requirements
- Significant reheat loads on current heating, ventilation, and AC (HVAC) system
- Heat source available or suitable installation identified for desiccant regeneration
- Current issues with humidity control – comfort, sick building syndrome, mold, etc.

The primary objective of this project was to demonstrate the capabilities of a new high-performance, liquid-desiccant dedicated outdoor air system (DOAS) to enhance cooling efficiency and comfort in humid climates while substantially reducing electric peak demand at Tyndall Air Force Base (AFB), which is 12 miles east of Panama City, Florida. The new type of LDAC invented by AIL Research (AILR) has higher thermal efficiency than any other LDAC on the market today. The technology was recently invented, and only six active units were operating at the time of this report, four of which are demonstration projects funded by the Department of Energy (DOE) with the purpose of demonstrating different applications and resolving new-product technical issues. Broader application is expected soon after technical reliability and manufacturing costs become acceptable. Seeing the technology’s potential, Munters Corporation recently purchased AILR’s LDAC technology and will commercialize it in areas with low thermal energy costs compared to electricity (e.g., low natural gas cost or waste-heat applications).

This was the first solar-powered demonstration of the technology. The goal of the project was to quantify energy and water consumption, solar energy utilization, and cost savings relative to DX air conditioners. The LDAC system that was installed at Tyndall AFB was a pre-commercial
technology, and given that it was the first solar-powered demonstration, a fundamental objective of the demonstration was to evaluate system performance and use the lessons learned to develop design/manufacturing guidance for future commercial LDAC systems. Each demonstration of this new technology is expected to reveal technical issues related to the specific application. This demonstration was also the first to integrate the LDAC as a retrofit into an existing air handler. Lessons learned from these experiences are expected to improve product design and create a methodology for determining suitable retrofit applications.

Performance evaluation of the LDAC began in the summer of 2010. Only 3 weeks of continuous operation were recorded in 2010 due to system malfunctions and limited run-time. Roughly 5 months of operation were recorded between April and September in 2011. Table 1 describes the performance objectives that were evaluated during the demonstration.

<table>
<thead>
<tr>
<th>Performance Objective</th>
<th>Metric</th>
<th>Success Criteria</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Improve humidity control and comfort (energy efficiency)</td>
<td>Hours outside psychrometric comfort zone</td>
<td>&lt;1% of hours outside ASHRAE summer comfort zone</td>
<td>Achieved but inconclusive cause</td>
</tr>
<tr>
<td></td>
<td>Chiller power</td>
<td>Reduce chiller/reheat run-time</td>
<td>Achieved but inconclusive cause</td>
</tr>
<tr>
<td></td>
<td>Reheat run-time</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Provide high-efficiency dehumidification (energy efficiency)</td>
<td>EER</td>
<td>&gt;40 (Btu/hr)/W EER</td>
<td>Not achieved</td>
</tr>
<tr>
<td></td>
<td>COP</td>
<td>Thermal COP &gt;0.7</td>
<td>Achieved</td>
</tr>
<tr>
<td>Sustain high-dehumidification performance (energy efficiency and maintenance)</td>
<td>Conditioner heat exchange effectiveness</td>
<td>&lt;Once-per-year desiccant/buffer adjustment</td>
<td>Achieved; no degradation of desiccant during operation</td>
</tr>
<tr>
<td></td>
<td>Desiccant charge</td>
<td>&lt;5% degradation of HX eff. over 3 years</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Supply-air pressure drop</td>
<td>Negligible increase in air/water pressure drop</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Conditioner cooling-water pressure drop</td>
<td>Above criteria should support &gt;10-yr service life projection</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Projected service life</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Performance data—including energy efficiency ratio (EER), kilowatt (kW)/ton, and coefficient of performance (COP) for 2010 and 2011—are summarized in Table 2 and Table 3, respectively. It is clear that the electrical and thermal efficiency improved throughout the summer of 2011.

### Table 2. Summer 2010 (3 Weeks) Performance Summary

<table>
<thead>
<tr>
<th>Date</th>
<th>Cooling (ton-hr)</th>
<th>EER [(Btu/hr)/W]</th>
<th>kW/ton</th>
<th>Solar heat (MBtu)*</th>
<th>COP*</th>
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<tr>
<td>7/21/10</td>
<td>-</td>
<td>1982</td>
<td>14.7</td>
<td>0.82</td>
<td>3.1</td>
</tr>
<tr>
<td>8/14/10</td>
<td></td>
<td></td>
<td>0.82</td>
<td>3.1</td>
<td>0.85</td>
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</table>

*Solar thermal generation only recorded for 3 days (7/21-7/23)

### Table 3. Monthly (Averaged) Performance for Summer 2011

<table>
<thead>
<tr>
<th>Month</th>
<th>Cooling (ton-hr)</th>
<th>EER [(Btu/hr)/W]</th>
<th>kW/ton</th>
<th>Solar heat (MBtu)</th>
<th>COP</th>
</tr>
</thead>
<tbody>
<tr>
<td>April</td>
<td>667</td>
<td>7.8</td>
<td>1.54</td>
<td>18.1</td>
<td>0.44</td>
</tr>
<tr>
<td>May</td>
<td>1565</td>
<td>8.2</td>
<td>1.47</td>
<td>39.9</td>
<td>0.5</td>
</tr>
<tr>
<td>June</td>
<td>1837</td>
<td>12.4</td>
<td>0.97</td>
<td>35.4</td>
<td>0.62</td>
</tr>
<tr>
<td>July</td>
<td>1142</td>
<td>14.6</td>
<td>0.82</td>
<td>19.4</td>
<td>0.71</td>
</tr>
<tr>
<td>August</td>
<td>1916</td>
<td>18.8</td>
<td>0.64</td>
<td>32.2</td>
<td>0.73</td>
</tr>
<tr>
<td>September</td>
<td>1300</td>
<td>15.1</td>
<td>0.79</td>
<td>26.7</td>
<td>0.73</td>
</tr>
</tbody>
</table>

Table 4 summarizes the displaced load on the existing chiller and the approximate energy and cost savings from the LDAC.
Table 4. Energy and Cost Savings from the LDAC in 2011

<table>
<thead>
<tr>
<th>Month</th>
<th>Cooling (ton-hr)</th>
<th>Chiller Elec. (kWh)</th>
<th>LDAC Elec. (kWh)</th>
<th>Elec. Savings (kWh)</th>
<th>Elec. Cost Savings ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>April</td>
<td>667</td>
<td>890</td>
<td>1,026</td>
<td>-137</td>
<td>-14</td>
</tr>
<tr>
<td>May</td>
<td>1,582</td>
<td>2,110</td>
<td>2,325</td>
<td>-215</td>
<td>-21</td>
</tr>
<tr>
<td>June</td>
<td>1,837</td>
<td>2,449</td>
<td>1,774</td>
<td>676</td>
<td>68</td>
</tr>
<tr>
<td>July</td>
<td>1,239</td>
<td>1,652</td>
<td>1,131</td>
<td>521</td>
<td>52</td>
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<tr>
<td>Aug</td>
<td>1,916</td>
<td>2,554</td>
<td>1,223</td>
<td>1,331</td>
<td>133</td>
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<tr>
<td>Sept</td>
<td>1,333</td>
<td>1,778</td>
<td>1,099</td>
<td>678</td>
<td>68</td>
</tr>
</tbody>
</table>

The total cost savings for the 2011 cooling season was $321. The installed costs for the solar thermal system were $170,000, and the installed costs for the LDAC components were $40,000, for a total installed cost of $210,000 and a simple payback of 654 years. Because this was a pre-commercial system, the simple payback is not indicative of the payback period of a commercial system. If the system would have operated per design intent, the cost savings would be substantially higher. In addition, in building types with electric reheat, the zone-level overcooling and reheat savings dwarf the energy savings from the mechanical chiller. Reheat energy use in hospitals, for example, has been documented to account for over 30% of the total energy use. Finally, when the system is coupled with solar thermal, the solar thermal component becomes the most expensive part of the system, and solar incentives or high utility rates are required to offset the increased costs of the solar thermal system.

In general, the LDAC system did not perform as well as expected due to design, installation, and operation issues. Consequently the project’s focus was necessarily changed to focus on discovery of technical issues with this new emerging technology. Many of the issues arose because the installation had many unique features including the following:

- The demonstration was the first combination of solar heat with this type of LDAC system.
  - Due to initial budgetary constraints, the LDAC relied solely on solar heat, with no natural gas backup to ensure that the unit operated throughout the cooling season. A properly designed system that uses solar heat will have backup. Due to this, the system did not achieve peak-cooling capacity for significant hours of operation. Because the system largely has static electrical power draw, this resulted in a low average EER.
  - The solar field design and LDAC system design were not tightly coordinated by the prime installation contractor (Regenesys). This resulted in a design that did not consider the frequency and duration of stagnation periods for the solar field. The collector design was not designed to withstand more than about two stagnations per year. Furthermore, the collector system was not initially designed to withstand the massive volume of steam from these collectors when stagnation occurred. The solar field required significant redesign. The end result was
workable for the demonstration, despite being problematic and suboptimal in operation.

- The demonstration was the first to create a split system where the conditioner and regenerator were contained in separate packages and separated by a distance of approximately 120-foot (ft). This technical challenge resulted in a suboptimal pumping design because of the necessary pump size to transfer desiccant this distance. Future designs should reduce the distance from the regenerator and conditioner.

- This demonstration was the first to have 10 hours of desiccant storage using calcium chloride (CaCl₂). Tuning the storage to achieve optimal efficiency was required. The desiccant charge and the tank’s low and high levels have significant impact on efficiency, capacity, and solar utilization. These variables were tuned as the demonstration progressed.

- This demonstration required the placement of the conditioner unit about 100 feet from the outdoor intake to the building. This required significant fan power to move the air from the mechanical yard to the building. Future designs and applications should consider the duct length reduce the duct run from the conditioner to the outdoor air intake as much as possible.

- The demonstration did not treat 100% of the outdoor air, thus limiting the benefit to energy savings from offset cooling. In order to offset the reheat for such an installation, a system should be designed to ensure that the LDAC meets a significant portion of the latent load. Typically, the LDAC can meet 100% of a building’s latent load if designed to treat 100% of the outdoor air.

This report outlines lessons learned that should be applied to future projects in order to ensure successful design, installation, and operation of a solar-powered LDAC system.

At the end of 2011, the LDAC technology was sold to Munters Corporation, one of the largest HVAC manufacturers in the United States. The first demonstration of a commercial LDAC system is being evaluated at the Coral Reef Fitness and Sports Center on Andersen AFB in Guam. A 6,000 cubic-feet-per-minute (cfm) conditioner was designed for this system. The power requirements per ton of cooling for the existing building level chiller and LADC are 1.05 kW/ton and 0.3 kW/ton, respectively. Note that the power requirement of the chiller does not account for the chiller water pumps, so the power requirement may be slightly greater in reality. The system is designed with an evacuated-tube solar thermal field supplying 80% of the thermal power and a backup diesel-powered boiler providing 20% of the thermal power. The system is expected to reduce HVAC energy use by 34% and save $145,395 per year with an estimated simple payback of 11.6 years.
1.0 INTRODUCTION

1.1 BACKGROUND

Today’s AC is primarily based on the DX or refrigeration process, which was invented by Willis Carrier more than 100 years ago. It is now so prevalent and entrenched in many societies that it is considered a necessity for maintaining efficient working and living environments. DX AC has also had 100-plus years to be optimized for cost and thermodynamic efficiency, both of which are nearing their practical limits. However, the positive impact of improved comfort and productivity does not come without consequences. Each year, AC accounts for approximately 15% of all source energy used for electricity production in the United States alone (nearly 4 quadrillion Btu), which results in the release of about 343-million tons of carbon dioxide into the atmosphere every year. (DOE 2011)

R-22 (Freon) as a refrigerant for AC is quickly being phased out because of its deleterious effects on the ozone layer. The most common remaining refrigerants used today (R-410A and R-134A) are strong contributors to global warming; their global warming potentials are 2,000 and 1,300, respectively. (Owen 2010) Finding data on refrigerant release rates for air conditioners is challenging as they are generally serviced only when broken, and refrigerant recharge is not accurately accounted for. The limited data that does exist indicates that typical refrigerant release rates for supermarket refrigeration equipment are 10% to 15% per year. (Baxter et al. 1998) A typical residential-size AC unit may contain as much as 13 pounds of R-410A, and a 10-ton commercial AC will contain as much as 22 pounds.

Water is not commonly considered to be a refrigerant, but the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) recognizes it as the refrigerant R-718. Evaporative cooling uses the refrigerant properties of water to remove heat the same way DX systems use the refrigeration cycle. Water evaporates and drives heat from a first heat reservoir; water vapor is then condensed into a second reservoir. The water used in this process is delivered to the building as a liquid via the domestic water supply. Evaporative cooling is so efficient because the Earth’s atmosphere and nature cycles, rather than a compressor and condenser heat exchanger, perform the energy-intensive process of recondensing the refrigerant.

The National Renewable Energy Laboratory’s (NREL’s) thermally activated technology program has been working closely with AILR as an industry partner for more than 15 years to develop an LDAC. The technology uses liquid desiccants to enable water as the refrigerant in lieu of chlorofluorocarbon-based refrigerants to drive the cooling process. The desiccants are strong saltwater solutions. In high concentrations, desiccants can absorb water from air and drive dehumidification processes; thus, evaporative cooling devices can be used in novel ways in all climates. Thermal energy dries the desiccant solutions once the water is absorbed. LDACs substitute most electricity use with thermal energy, which can be powered by many types of energy sources, including natural gas, solar thermal, biofuels, and waste heat. The benefits include generally lower source energy use, much lower peak-electricity demand, and lower carbon emissions, especially when a renewable fuel is used.

The LDAC technology deployed in this demonstration was invented by AILR, and was the result of collaborative effort with NREL, and was funded by DOE. The LDAC technology developed
by AILR is the result of a 10-year, $5 million DOE R&D effort to increase the efficiency of the LDAC technology on the market and decrease maintenance concerns related to legacy problems with desiccant carryover into the product airstream. The technology is emerging and at the writing of this report, six active demonstrations had been deployed. Munters Corporation has seen the promise of the technology and has purchased the rights from AIL Research. The six active demonstrations are focused on providing cooling to grocery stores where the benefits from drying the space to sufficient levels reduce refrigeration evaporator-coil frosting due to water condensation and freezing. Energy is reduced by less defrosting and a lower load on the refrigeration system. Munters Corporation has taken on the task to manufacture the LDAC technology. The demonstration to date, including the Tyndall demonstration, has shown a critical level of reliability of the LDAC system and identified points of improvement. The sale of the technology shows that Munters Corporation is satisfied with the current state of reliability and willing to commercialize it.

AC is also the single largest contributor to peak demand on electric grids and is a primary cause of grid failure resulting in blackouts. Power generators and electric air conditioners are least efficient at high ambient temperatures, when cooling demand is highest, leading to increased pollution, excessive investment in standby generation capacity, and poor utilization of peaking assets. This LDAC approach—the result of a 10-year, $5 million DOE R&D effort—increases the efficiency of LDAC technology on the market and decreases maintenance concerns related to legacy problems with desiccant carryover into the product airstream.

1.2 OBJECTIVE OF THE DEMONSTRATION

The primary objective of this project was to demonstrate the capabilities of a new high-performance, liquid-desiccant DOAS to enhance cooling efficiency and comfort in humid climates while substantially reducing electric peak-demand. This was the first solar-powered demonstration of the technology. The goal of the project was to quantify energy and water consumption, solar energy utilization, and cost savings relative to DX air conditioners. The LDAC system installed at Tyndall was a pre-commercial technology and given that it was the first solar-powered demonstration, a fundamental objective of the demonstration was to evaluate the performance of the system and use the lessons learned to develop design/manufacturing guidance for future commercial LDAC systems.

At the end of 2011, the LDAC technology was licensed to Munters, one of the largest HVAC manufacturers in the United States, and the first demonstration of a commercial LDAC system is being evaluated on the Coral Reef Fitness and Sports Center on Andersen AFB in Guam.

1.3 REGULATORY DRIVERS

The DoD Environmental Security Technology Certification Program awarded this new technology demonstration project as a means to identify programmatic changes that could be applied to the design and construction of energy-efficient, DOAS AC systems for humid environments. A new low-energy use LDAC unit could be implemented throughout ASHRAE climate zones 1, 2, and 3 to help the agency meet or exceed the various requirements set forth in Executive Order (E.O.) 13423, the Energy Policy Act (EPA) Act of 2005, and the Energy Independence and Security Act (EISA) of 2007.
EPAct 2005 requires the U.S. secretary of energy to ensure that not less than 7.5% of total electricity consumed by the federal government comes from renewable sources in FY 2013; and thereafter, to the extent economically feasible and technically practicable in FY 2013, and thereafter, of the total electricity consumed by the federal government comes from renewable energy. If the thermal portion of the LDAC unit is driven by a solar thermal source, this technology would help DoD meet its renewable energy goals.

The key features of EISA 2007 that pertain to this technology are outlined in section 431 and requires a reduction in energy use intensity (EUI) \[1,000 \text{ (k)Btu/square foot (ft}^2\text{/year (yr)})\] of federal buildings by 3% per year, from a 2003 baseline, resulting in a 30% reduction in EUI by 2015. The EISA 2007 legislation has superseded all previous EUI reduction mandates.

E.O. 13423 provides requirements for water conservation at federal facilities, mandating federal agencies reduce potable water consumption intensity 2% annually through FY 2020. This would result in a 26% reduction by the end of FY 2020, relative to a FY 2007 baseline. E.O. 13514 also mandates a reduction in industrial, landscaping, and agricultural water consumption by 2% annually, or 20% by the end of FY 2020, relative to a FY 2010 baseline.

The LDAC unit can substantially reduce energy use and peak demand, which will help meet EISA 2007 requirements, but it also has the potential to increase potable water consumption, which will be detrimental to the E.O. 13514 requirements. Each DoD base is encouraged to try to identify alternative sources of cooled water for the conditioner, such as geothermal-based cooling.
2.0 TECHNOLOGY DESCRIPTION

2.1 TECHNOLOGY OVERVIEW

Desiccants reverse the paradigm of standard DX AC by first dehumidifying and then sensibly cooling the outside airstream to meet a given cooling load. Desiccant at any given temperature has a water-vapor pressure equilibrium that is roughly in line with constant relative humidity (RH) lines on a psychrometric chart, as shown in Figure 1. The green lines show the dehumidification potential for two common types of liquid desiccants: lithium chloride (LiCl) and CaCl₂. If the free surface of the desiccant is kept at a constant temperature, the ambient air will be driven to the dehumidification potential line. If used with an evaporative heat sink at temperatures between 55°F and 85°F, the air can be significantly dehumidified, and dew points less than 32°F are easily achieved. The blue arrow in Figure 1 shows the path of ambient air driven to equilibrium with CaCl₂ with the use of an evaporative heat sink. At this point, the air can be sensibly cooled to the proper supply temperature. This type of desiccant AC system decouples sensible and latent cooling by controlling each independently.

During the dehumidification process, the liquid desiccant (about 43% salt concentration by weight in a water solution) absorbs water vapor in an exothermic reaction. The heat released by the desiccant is carried away by a heat sink, usually cooled water from a cooling tower. As water vapor is absorbed from the ambient air, it dilutes the liquid desiccant, and decreases its vapor pressure and its ability to absorb additional water vapor. Lower concentrations of desiccant come into equilibrium at higher ambient air RH levels. Dehumidification can be controlled by the desiccant concentration supplied to the device. The outlet humidity level of the processed ambient air can be controlled by the desiccant concentration and/or the flow of highly concentrated desiccant. The latter allows the highly concentrated desiccant to quickly be diluted and thus “act” as a weaker desiccant solution in the device.
Absorption of water vapor will eventually weaken the desiccant solution and reduce its dehumidifying potential; the desiccant must then be regenerated to drive off the absorbed water. Thermal regeneration is the reverse process of vapor absorption. In this process, the desiccant is heated to a temperature at which the equilibrium vapor pressure is above ambient vapor pressure. The water vapor desorbs from the desiccant and is carried away by an airstream (see Figure 2). Figure 2 shows how a scavenging airstream picks up heat and moisture from a regenerator. The green line represents the psychrometric condition of air in equilibrium with a CaCl₂ solution at the given temperature. Sensible heat is recovered by first preheating the ambient air using an air-to-air heat exchanger (AAHX). The air comes into contact with the desiccant in the heat and mass exchanger (HMX)—in this example at 190°F—and carries the desorbed water vapor away from the desiccant. Sensible heat is recovered by taking the hot humid air to preheat the incoming air through the AAHX. The change in enthalpy of the airstream as it passes through the regenerator represents the majority of the thermal input.
The process uses hot water or steam to achieve a latent COP between 0.8 and 0.94, depending on desiccant concentration. Latent COP is defined as:

$$COP_{\text{Latent}} = \frac{(\text{Moisture Removal Rate}) \times (\text{Heat of Vaporization})}{\text{Heat Rate (Higher Heating Value)}}$$

COP is maximized by maximizing the regeneration temperature and change in concentration while minimizing the desiccant concentration. Including the COP of the water heater (about 0.82), a typical combined latent COP for the LDAC systems is $0.82 \times 0.85 = 0.7$. If the heating source is derived from solar thermal technologies, the COP of the water heater would be the efficiency of the solar collectors (the benefit here being that there is no fuel cost penalty for the heat conversion efficiency).

The AILR technology innovations result in higher thermal efficiency when compared to other technologies on the market. NREL tests have shown that the Kathabar and other high-flow systems achieve a latent COP of about 0.4 to 0.55.

The LDAC technology developed by AILR uses novel HMXs to perform these two processes as shown in Figure 3, which illustrates the desiccant conditioner and scavenging air regenerator. The liquid desiccant is dispensed over the plates in the conditioner (absorber) where the inlet ambient air is dehumidified. This technology is called low-flow, liquid-desiccant AC because the desiccant flow is minimized in the HMXs of the conditioner and regenerator to the flow rate needed to absorb the necessary moisture from the airstream, which eliminates liquid desiccant
carryover into the supply airstream. The HMXs must therefore have integral heating and cooling sources; 55°F – 85°F cooling tower water is supplied to the conditioner, and the regenerator uses hot water or hot steam at 160°F – 200°F. The cooling or heating water flows internal to the heat exchange plates while the desiccant flows on the external side of the HMX plates. The plates are flocked, which effectively spreads the desiccant and creates direct-contact surfaces between the air and desiccant as the air passes between the plates.

2.2 FUTURE DEVELOPMENTS

NREL is working with AILR to develop a double-effect regenerator that expands on the scavenging air regenerator by first boiling the water out of the liquid desiccant solution (250°–280°F) and reusing the steam by sending it through the scavenging air regenerator. This two-stage regeneration system can achieve a latent COP of 1.05–1.2. A typical solar regenerator would consist of a hot water supply and a scavenging regenerator (which would result in a single-effect device that would have about a 60% solar conversion efficiency based on absorber area).

Table 5. Technology Options for Residential and Commercial Buildings*

<table>
<thead>
<tr>
<th>Regenerator</th>
<th>Performance Rating</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solar Array</td>
<td>60% solar conversion</td>
</tr>
<tr>
<td>Single effect using natural gas</td>
<td>COP 0.7</td>
</tr>
</tbody>
</table>

*Based on NREL calculations and laboratory data, available upon request
For the low-flow LDAC, the regenerator and conditioner systems are shown connected in Figure 4, which illustrates the three thermal energy sources that can be used to regenerate the desiccant: solar thermal, traditional water heaters, and a double-effect technology. The water heater or boiler can be fueled by many sources, including natural gas, combined heat and power (CHP), or even biofuels. Also shown is the option for desiccant storage; storage allows an AC system to effectively bridge the time gap between thermal energy source availability and cooling load. A desiccant system that uses strong desiccant at 43% by weight and weakens the desiccant to 35% will be able to achieve storage density of 5 gallons (gal) of desiccant per latent ton-hour·(hr) of cooling provided. This is most easily accomplished using LiCl as the desiccant, as 35% desiccant still retains significant dehumidification potential. Desiccant storage is simple and theoretically lossless. Storage consists of a sealed, uninsulated tank with proper liquid connections. In comparison, ice storage is approximately 13–16 gal/ton-hr (theoretically 10 gal/ton-h, but in practice only 67% of the volume is frozen. (Ice Energy 2012) This storage can be useful to enable maximum thermal use from solar or on-site CHP.

![Figure 4. LDAC schematic.](Illustration by NREL)

### 2.3 TECHNOLOGY DEVELOPMENT

Since its founding in 1988, the primary mission of AILR has been to develop and commercialize high-efficiency, end-use products for heating and cooling applications. For the past 14 years, AILR has focused on the parallel activities of developing plastic heat exchangers and applying these heat exchangers in HVAC products that use advanced liquid-desiccant technology.
From October 1990 to October 1991, AILR conducted research for the Gas Research Institute on a project entitled, “The Effect of Material Properties on the Performance of Liquid Desiccant Air Conditioners and Dehumidifiers.” (AILR, undated) In addition to investigating alternative desiccants to LiCl, AILR studied novel configurations of the regenerator and the conditioner of an LDAC. An important conclusion from this work, which was reported in ASHRAE Paper No. AN-92-3-3, was that the desiccant flow rate in a packed-bed conditioner (which was the dominant technology at the time) is set by the requirement to limit the desiccant’s temperature rise. (Lowenstein and Gabruk 1992) By embedding cooling within the conditioner heat and mass exchanger, the desiccant’s temperature could be controlled independently of the amount of water absorbed. The desiccant flow rate could then be set by the need to limit the concentration change of the desiccant, a requirement that allows the desiccant flow to be reduced by over an order of magnitude compared to packed-bed conditioner designs.

In 1994, AILR received a patent that covered the low-flow, liquid-desiccant technology. (Lowenstein 1994) The patent was assigned to the Gas Research Institute, the organization that sponsored the research. Shortly after receiving the patent for low-flow, liquid-desiccant technology, AILR began to explore ways to practically capture the benefits of the technology. In September 1998, AILR delivered a 1,000-cfm, liquid-desiccant conditioner to NREL that used low-flow technology. (NREL 1997) The conditioner, which is shown in Figure 5, was composed of 75 polypropylene extruded plates that had been modified so that cooling water made six passes within each plate. A woven cotton fabric sleeve was slipped over each plate to provide a wicking surface for the desiccant.

![Figure 5. The first implementation of a low-flow conditioner.](Photo from AILR)

Following the successful testing of the liquid-desiccant conditioner at NREL (shown in Figure 5), AILR began to develop a manufacturable design for a low-flow, liquid-desiccant conditioner with additional support from NREL. (NREL 1998 and 1999) A low-flow, liquid-desiccant conditioner composed of extruded polyvinyl chloride (PVC) plates bonded to injection-molded manifold pieces was developed in this follow-on work. A 40-plate prototype of this conditioner
was successfully tested at NREL in June 2004. This manufacturable design for the liquid-desiccant conditioner, shown in Figure 6, is used in the Tyndall solar LDAC.

Figure 6. The upper end of a manufacturable low-flow conditioner.

(Photo from AILR)

AILR’s development of a low-flow conditioner was complemented by a parallel effort to develop a manufacturable, low-flow regenerator. Several approaches to a low-flow regenerator were explored under sponsorship by NREL. (NREL 2001) Prototypes were built using extruded chlorinated PVC (CPVC) plates and coated aluminum plates. The initial operation of both prototypes was good, but within several hundred hours of operation, both prototypes failed. A third prototype composed of extruded polyphenylsulfone (PPSU) plates, which is shown in Figure 7, proved successful operation for thousands of hours. A prototype of the PPSU regenerator was tested by NREL in February 2006. This PPSU regenerator is used in the Tyndall LDAC. PPSU is a plastic that can withstand temperatures as high as 250°F, but is substantially more expensive than other plastics. AILR and NREL continue to investigate regenerator designs with lower cost materials.
In 2003, AILR built the first prototype of a 6,000-cfm roof-top LDAC under a subcontract to Kathabar, Inc., as part of a larger effort of Oak Ridge National Laboratory. This prototype originally used a CPVC regenerator that failed after several hundred hours of operation.

In 2005, AILR built, installed and operated a second 6,000-cfm LDAC prototype, again under sponsorship of NREL. (NREL 2005) This prototype, shown in Figure 8, was installed on a machine shop in Wrightsville, Pennsylvania, where it successfully processed ventilation for 2 years.

**Figure 7. A PPSU regenerator (similar to the one installed in the Tyndall LDAC).**

(Photograph from AILR)

**Figure 8. The second prototype of a low-flow LDAC processing ventilation air for a machine shop in Wrightsville, Pennsylvania.**

(Photograph from AILR)
The 3,000-cfm solar LDAC built for Tyndall AFB was installed in spring 2010 and operated during the summers of 2010 and 2011. The Tyndall LDAC was the first implementation of a low-flow LDAC driven solely by solar thermal energy for regeneration. It was also the first AILR LDAC to operate in the field using a solution of CaCl₂ as the desiccant, which is more cost effective than LiCl as a means to store cooling, but it also does not provide the same dehumidification as LiCl, and is thus a compromise.

In May 2009, PAX Streamline (a venture-backed startup company) established a memorandum of understanding with AILR to transfer the low-flow technology to PAX for commercialization. Working together, PAX and AILR built a 6,000-cfm and 3,000-cfm LDAC and installed them on separate supermarkets in the Los Angeles area. Figure 9 shows the 6,000-cfm installation. Unfortunately, PAX Streamline failed in April 2010, despite the successful operation of the two supermarket LDACs.

Following the failure of PAX, AILR continued to work with two former employees of PAX to build and install three more supermarket LDACs: two in California and one in Hawaii. These LDACs were installed between October 2010 and April 2011.

In July 2011, all intellectual property and know-how developed by AILR for building liquid-desiccant conditioners and regenerators that use low-flow technology were sold to the Munters Corporation. Munters is now in the early stage of commercializing the technology.
2.4 ADVANTAGES AND LIMITATIONS OF THE TECHNOLOGY

LDACs are a new breed of AC technology that decouples the latent load from the total load (sensible + latent) normally done by a refrigeration or chilled water system. De-coupling of these loads enables independent temperature and humidity control in a space. Also, lower humidities in a space can be achieved more efficiently by avoiding the energy intensive processes. Examples of these avoided processes and systems include:

- Overcooling and then reheating (which reverses the sensible cooling by the refrigeration system, thus lowering efficiency)
- Solid-desiccant wheels with natural gas or condenser heat regeneration. These systems generally increase energy use by the HVAC system due to high air-pressure losses and natural gas use.
- Ultra-low apparatus dew-point temperatures, which increase energy use by the refrigeration system.

LDACs largely switch much of the energy to condition air to thermal sources, such as natural gas, solar thermal, or waste heat. High-density storage can be employed to bridge thermal energy source profiles with cooling profiles, such as the case with solar thermal or even waste heat. Using waste heat is the most cost effective way to power an LDAC unit and should be considered first if a waste heat source is available. Natural gas or propane is economically utilized when dehumidification requirements are high. Supermarkets are a typical case where decreased store humidity drastically improves the efficiency of the food refrigeration systems. Thus store humidity levels are generally kept as low as possible. LDACs enable lower store humidity levels than other available humidity-control technologies and are just now being adopted at major supermarket chains as a result. For example, Whole Foods has recently included the LDAC technology in its HVAC specification in humid climates. Solar energy for LDACs can become economical if the relative cost of solar thermal energy is competitive with natural gas or propane. This is often the case on islands such as Hawaii, Guam, and many other tropical island nations. Solar thermal systems should always be used to offset fossil fuel use but not as the primary source of energy. Designs that attempt to get a solar fraction of 1.0 inherently will have solar fields and desiccant storage tanks that are much larger and more expensive than practical.

LDACs primarily use cooling towers for their cooling sink. If cooling towers are compared to air-cooled AC systems, site water use will increase. However, many chiller systems use cooling towers, and LDAC technology would use about the same amount of water for cooling as these systems do. The electric power grid also uses water to cool thermal power plants. The avoided use of electric power can result in substantial regional water savings. Case-by-case analysis is required to calculate these savings. However, typical thermal power generation station produce about 1.0 to 2.0 kilowatt-hours (kWh) of electricity per gallon of water evaporated (0.5 - 1 gal/kWh).

Water use is dictated by how much energy is removed per pound (lb) of water evaporated. Water’s heat of vaporization is about 1,060 Btu/lb, which is equivalent to 1.37 gal/ton-hr of
cooling load. However, because evaporative cooling is an open-cycle process where mineral content of domestic water must be removed, the water use will be higher by the cycles of concentration (CoC) required to prevent mineral buildup in an evaporative system. CoC is defined as the ratio of mineral concentration in the blow-down water divided by the initial concentration. CoC is dependent on water quality, but typically range from 2-7 where a CoC of 2 is typically associated with facilities that have extreme water hardness. A typical water-draw rate for a typical cooling tower will be 1.57 to 2.74 gal/ton-hr. In the case of the LDAC technology, a cooling tower must only remove the cooling load. In the case of a water-cooled DX system, the cooling tower must remove the cooling load plus the compressor load. For a DX cycle with a COP of 4, a cooling tower would thus draw 25% more water or typically 1.96 to 3.42 gal/ton-hr.

The preceding analysis does not include the complicated weather effects on a cooling tower, but is approximate for most conditions where the cooling tower’s airstream becomes fully saturated and leaves at the same temperature as the inlet air. However, the comparison with DX cooling remains accurate in relative amounts. LDAC technology will, in general, use about 25% less water than a water-cooled DX system. The net regional water impact by using an LDAC system will typically be small or even positive in some cases.

LDACs are now being employed to treat dedicated outdoor airstreams to control humidity in a space. The highest benefit thus will be for humid climates with large yearly humidity loads and applications where reheat energy is high.

High-value applications include buildings with large outdoor air loads that have the highest levels of reheat or benefit from decreased humidity in the space such as the following:
- Hospitals (avoiding massive amounts of reheat)
- Supermarket humidity control
- School buildings in humid regions
- Buildings with waste heat available
- Indoor pools.

LDACs are an emerging technology and have not seen the level of refinement that economy of scale has bestowed upon vapor-compression technology. The technology is still in a pre-commercial state; the systems are more complex than traditional vapor-compression systems and require custom engineering in most new applications. This is a major hurdle that now faces this technology as research funding will inherently drop and market pull must pick up. Thus LDACs will be first introduced in the highest-value applications, where market pull for the benefits is large enough. In its current state the LDAC technology cannot compete in facilities that do not over-cool/re-heat supply air and are also in locations with lower electricity rates.
### 3.0 PERFORMANCE OBJECTIVES

Table 6 describes the performance objectives, metrics, and data requirements to determine the performance objective results; and the criteria for achieving the objectives.

<table>
<thead>
<tr>
<th>Performance Objective</th>
<th>Metric</th>
<th>Data Requirements</th>
<th>Success Criteria</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Quantitative Performance Objectives</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Improve humidity control and comfort (energy efficiency)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Hours outside psychrometric comfort zone</td>
<td>• EER</td>
<td>• Supply-air temp./humidity</td>
<td>• &lt;1% of hours outside ASHRAE summer comfort zone</td>
</tr>
<tr>
<td>• Chiller power</td>
<td>• COP</td>
<td>• Supply-air flow rate</td>
<td>• Reduce chiller/reheat run-time</td>
</tr>
<tr>
<td>• Reheat run-time</td>
<td></td>
<td>• Ambient temp./humidity</td>
<td></td>
</tr>
<tr>
<td>• Indoor temp./humidity</td>
<td></td>
<td>• Power consumption</td>
<td></td>
</tr>
<tr>
<td>• Chiller power</td>
<td></td>
<td>• Heat consumption</td>
<td></td>
</tr>
<tr>
<td>• Reheat coils</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Provide high-efficiency dehumidification (energy efficiency)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Conditioner heat-exchange effectiveness</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Desiccant charge</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Supply-air pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Conditioner cooling-water pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Projected service life</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Supply air temp/hum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Ambient temp/hum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Desiccant chemistry and concentration</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Conditioner core-air and water-pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• EER &gt; 40 (Btu/hr)/W</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• &gt;0.7 Thermal COP</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Sustain high-dehumidification performance (energy efficiency and maintenance)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Conditioner heat-exchange effectiveness</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Desiccant charge</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Supply-air pressure drop</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>• Conditioner cooling-water pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Projected service life</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>• Supply air temp/hum</td>
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<td></td>
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<tr>
<td>• Ambient temp/hum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Desiccant chemistry and concentration</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Conditioner core-air and water-pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• &lt;5% degradation of HX eff. over 3 years</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• &lt;Once-per-year desiccant/buffer adjustment</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Negligible increase in air/water pressure drop</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Above criteria should support &gt;10-yr service life projection</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Qualitative Performance Objectives

| Maintainability (ease of use) | Ability of an HVAC technician to operate and maintain the technology | Standard form feedback from the HVAC technician on usability of the technology and time required to maintain | A single facility technician able to effectively operate and maintain equipment with minimal training |

3.1 IMPROVE HUMIDITY CONTROL AND COMFORT

Conventional AC has fundamental limitations regarding humidity control. Mechanical AC provides coincidental drying when it lowers air below its dew-point temperature, condensing water vapor on its cooling coils. In order to provide separate control over temperature and humidity, it must overcool the air and then reheat it so as not to overcool the space. This can be excessively energy intensive. Mechanical AC runs into additional problems with dehumidification when humidity loads are large compared to sensible loads, such as in the spring and fall, and under any conditions where short-cycling occurs, which leads to re-evaporation of water from the cooling coils.

This objective will demonstrate that desiccant drying effectively manages humidity independently of sensible loads throughout the cooling season. A comfort zone of combined temperature and humidity ranges has been established by ASHRAE; a technology’s ability to provide comfort is typically represented by the number of hours per year indoor conditions leave this established zone. NREL monitored chiller power and reheat coil run-times both with and without desiccant system operation to show effects on comfort and overcool/reheat. NREL used its testing experience to ensure that sensors were positioned to read representative supply-air temperatures that were not skewed by spatial nonuniformities or radiative heat exchange.

3.2 PROVIDE HIGH-EFFICIENCY DEHUMIDIFICATION

It is critical to establish the efficiency advantages of desiccant dehumidification over conventional mechanical AC. The cooling effect is determined by the change in enthalpy of the supply air relative to ambient, multiplied by the total mass flow of supply air. The range of psychrometric conditions that is monitored allows high-quality temperature and RH sensors to adequately make these enthalpy measurements. Mass flows are measured with pitot tube-in-duct measurements, inferred from fan power draw and checked against pressure drop measurements calibrated at NREL’s test facility. Energy balance for the cooling effect can be checked against the measured heat gain of the cooling water flowing through the conditioner.

Power measurements to monitor fan and pump energy are straightforward. Thermal energy consumption is also conveniently measured by the drop in temperature of the solar collector fluid through the regenerator multiplied by its mass flow rate and specific heat. All of these parameters are straightforward to determine in real-time.
3.3  SUSTAIN HIGH-DEHUMIDIFICATION PERFORMANCE

A common concern regarding liquid-desiccant devices is the carryover of desiccant droplets into the supply (or scavenging) airflows, exhibited as decreasing desiccant charge and corrosion of metal ductwork where these droplets settle. One strategy for avoiding this loss of desiccant is the deployment of high-pressure drop demisters in the supply duct. However, this approach is not desirable because it imparts a significant energy penalty in fan power and requires frequent maintenance to remain effective. Alternatively, the low-flow liquid desiccant uses minimal mist eliminators, relying on its low-flow feature to prevent carryover and preserve desiccant charge. This low-flow feature is enabled by the flocked surfaces of its conditioner component. These desiccant-wetted wicks must keep themselves relatively free of dirt buildup to maintain their zero-carryover feature and the effectiveness of the HMX. The conditioner’s heat exchange effectiveness and pressure drop were monitored to ensure that ambient dirt was being successfully managed to preserve performance over a projected service life of 10-15 years. In general, because the conditioner is water cooled, mineral fouling of its internal water passages is possible, albeit improbable due to its all-plastic construction.

3.4  MAINTAINABILITY

There are many ways for a building technology to fail if its operation requires more than the current HVAC standard maintenance requirements, which is minimal. The success criteria for this performance objective were determined if the demonstration unit settled into autonomous operation after the first season following Tyndall’s standard HVAC maintenance schedule.
4.0 FACILITY/SITE DESCRIPTION

The selected laboratory at Tyndall AFB is located in Panama City, Florida, on the Gulf Coast. Its high temperatures are typically in the 80-89°F range in the summer, rarely above 90°F, with ambient humidity in excess of 0.02 lbs of water per lb of dry air through portions of the cooling season. Its design wet-bulb temperatures are very high, ranging from 70°F to over 80°F, with humidity extremes up to 90%. Figure 10 illustrates the wide range of temperatures and humidity that Tyndall experiences throughout the year.

![Psychometric plot Tyndall AFB](image)

Figure 10. Psychometric plot Tyndall AFB

In addition to the hot and humid summer days, the cool temperature days from the late fall to early spring allow for a robust system performance evaluation through the observation of operation in non-ideal weather conditions and taking proper measures to avoid damage from freezing. Altogether, the site provides the necessary spectrum of ambient conditions to characterize the LDAC performance sufficiently for predicting performance across most, if not all, conditions in the United States, U.S. territories, and countries with active U.S. military operations.

4.1 FACILITY/SITE LOCATION AND OPERATIONS

The Air Force Research Laboratory (AFRL) building at Tyndall AFB is a mix of laboratory and office space. Three main air-handling units (AHUs), serve the laboratory and office space. A satellite image of Tyndall AFB and the LDAC system is provided in Figure 11.
Typical configuration may include a single-packaged unit or a split-system arrangement, where the regenerator cabinet is physically split from the conditioner cabinet. The requirement to place the solar system in the nearby field rather than roof mounting meant that the regenerator is best placed in the field adjacent to the solar array for minimizing heat loss. The conditioner, cooling tower, and desiccant storage tank are located in an enclosed HVAC area, so the conditioned air can be supplied into the building via ductwork.

Figure 12 shows the Tyndall AFB building layout, with the space apportioned by office, laboratory, and mechanical rooms. The red box highlights the predominantly laboratory space for which the LDAC system provides ventilation air. Reheat coils in terminal units in each zone activate if the air has been overcooled, and using measured data from AHU #3, the reduction of reheat can be determined.
The laboratory wing served by the low-flow, liquid-desiccant unit underwent a chiller upgrade in 2008 because cooling loads were going unmet. The building was also recommissioned to balance the outdoor air to ensure positive pressure within the building to eliminate condensation due to infiltration. As a result, reheat coils were not being actuated because indoor temperature set points were not being reached. This implies that indoor humidity was not being controlled and that conditions were likely uncomfortably humid. The chiller upgrade provides sufficient capacity to properly dehumidify (overcool) the space, and therefore, required reheat coil operation. The upgrade also included condenser heat recovery to offset reheat energy use.

A DX, air-cooled chiller (Table 7) supplies chilled water to all three AHUs. The LDAC system conditions ventilation air that serves AHU #3 (Table 8).
### Table 7. Air-Cooled Chiller Schedule

<table>
<thead>
<tr>
<th>Air cooled chiller</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Evaporator Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Total Capacity (tons)</td>
<td>180</td>
</tr>
<tr>
<td>Entering Water (°F)</td>
<td>56</td>
</tr>
<tr>
<td>Leaving Water (°F)</td>
<td>44</td>
</tr>
<tr>
<td>P.D. (Ft)</td>
<td>15</td>
</tr>
<tr>
<td>GPM</td>
<td>370</td>
</tr>
<tr>
<td><strong>Compressor Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Type</td>
<td>Rotary Screw</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>22</td>
</tr>
<tr>
<td>Number of Compressors</td>
<td>3</td>
</tr>
<tr>
<td><strong>Electrical Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Compressor and Fan KW</td>
<td>205</td>
</tr>
<tr>
<td>EER</td>
<td>9.3</td>
</tr>
</tbody>
</table>

### Table 8. AHU #3 schedule

<table>
<thead>
<tr>
<th>AHU #3</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Fan Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Fan Type</td>
<td>Forward Curve</td>
</tr>
<tr>
<td>Supply Air (cfm)</td>
<td>11,710</td>
</tr>
<tr>
<td>Outside Air (cfm)</td>
<td>7,000</td>
</tr>
<tr>
<td>Static Pressure (in. H2O)</td>
<td>2.5</td>
</tr>
<tr>
<td>Motor Size (HP)</td>
<td>20</td>
</tr>
<tr>
<td>Configuration</td>
<td>Blow-Thru</td>
</tr>
<tr>
<td>Volt/Phase/Cycle</td>
<td>460/3/60</td>
</tr>
<tr>
<td><strong>Cooling Coil Performance</strong></td>
<td></td>
</tr>
<tr>
<td>Max Face Velocity (fpm)</td>
<td>520</td>
</tr>
<tr>
<td>Max Air P.D. (in. H2O)</td>
<td>1</td>
</tr>
<tr>
<td>Max Water P.D. (Ft. H2O)</td>
<td>15</td>
</tr>
<tr>
<td>Entering DB/WB (°F)</td>
<td>86.5/71.1</td>
</tr>
<tr>
<td>Leaving DB/WB</td>
<td>54.6/53.5</td>
</tr>
<tr>
<td>Entering Water</td>
<td>44</td>
</tr>
<tr>
<td>Leaving Water</td>
<td>56</td>
</tr>
<tr>
<td>GPM</td>
<td>127</td>
</tr>
<tr>
<td>Total Heat (BTUH)</td>
<td>758,400</td>
</tr>
</tbody>
</table>
5.0 TEST DESIGN

5.1 CONCEPTUAL TEST DESIGN

NREL installed instrumentation and a data acquisition system for one SOA3000 dehumidiifier, powered by a 1,300-ft², evacuated-tube solar thermal array. The solar array and regenerator components are oversized relative to the conditioner’s average dehumidification output in order to generate and store excess desiccant during the day. An 800-gal uninsulated storage tank fully utilizes the solar arrays excess heat output and allows for a few hours of average cooling operation without solar input. The unit is designed to operate continuously at maximum airflow in order to serve fume-hood makeup air needs. The LDAC technology was characterized in NREL’s Advanced HVAC Systems Laboratory in 2004 and 2006. The laboratory test results are invaluable in interpreting the field results, particularly with regard to critical airflow rates, which are notoriously difficult measurements to make in the field. The unit was monitored for two cooling seasons, and its annual and peak energy use was compared to conventional AC.

5.2 BASELINE CHARACTERIZATION

The installation had the potential to generate compelling side-by-side test results in that the recent chiller upgrade should allow operation with or without desiccant unit operation. Circumstances that complicate comparison include the facts that: 1) the chiller serves the entire building; and 2) the disparity in capacities between the chiller and the desiccant system is approximately 10:1. Chiller power and calls for reheat were measured in one wing of the facility. Because mechanical AC is a well-understood technology, baselines for individual sites were not critical to project energy savings relative to conventional equipment at various efficiency levels. Once the efficiency of the desiccant system was established, comparisons of energy use relative to mechanical AC were straightforward over the full range of building applications and climates.

5.2.1 Laboratory Testing

NREL was directly involved in the development of the LDAC technology and funded the development of the three core HMXs now used: the conditioner, regenerator, and interchange heat exchanger. These HMXs were developed by AILR and NREL determined the performance of the units through testing at the Advanced HVAC Systems Test Laboratory. The final conditioner and regenerator HMX designs were tested in 2004 and 2006 respectively. The data from these tests proved invaluable in the demonstration as baseline performance was determined and verified against laboratory data. Airflow, pressure drop, and exchange effectiveness from the laboratory proved invaluable in the calibration of airflow and pressure drop measurements in the field, which improved the uncertainty of the field data. There are no standard methods for testing LDAC systems. However, NREL is a world leader in the development of desiccant technologies and draws upon sound practices in order to determine the performance of these HMXs. To determine effectiveness, each HMX was instrumented at the NREL laboratory with the following parameters and accuracy:

- Air and liquid temperature measurements have an accuracy of ± 0.36°F with <0.18°F deviation.
- Airflow measurements have an accuracy of ± 2%.
- Humidity is calculated with dew-point hydrometers and has an accuracy of 0.27°F.
• Differential pressure measurements have an accuracy of ± 0.025 inches water column (W.C.)
• Barometric pressure has an accuracy of 0.15%.
• Water flow-rate measurements have an accuracy of 0.5%.
• Liquid desiccant flow-rate measurements have an accuracy of 1.0%.
• Desiccant concentration by weight is measured to within 0.0025 lbs salt per lb of solution.

A diagram of the HVAC laboratory in the Thermal Test Facility at NREL is provided in Figure 13.

Accuracy: Chilled mirror dewpoint, Laminar Flow Element (LFE) flow measurement, Energy Balance within 5%

Speed: Performance map vs. single rating point

Scale: Prototypes, Residential, Commercial. Up to 10 ton cooling capacity, 50 to 20,000 CFM

The laboratory testing resulted in the development of a regression of cooling effectiveness and capacity versus airflow rate; conditioner and regenerator water flow rate and temperature; and desiccant concentration. These data were also shared with AILR to calibrate its internal modeling of the LDAC system for design purposes. It is, however, impractical to measure every combination of independent variables to determine a performance map of the LDAC system.
Thus the calibration/validation of the AILR model using NREL data improves the accuracy of the model, making it a suitable tool to determine LDAC performance using building energy simulators such as EnergyPlus.

5.3 DESIGN AND LAYOUT OF TECHNOLOGY COMPONENTS

Figure 14 illustrates the design and layout of the LDAC system. Demonstration equipment was placed in two locations at the test site. The solar array and regenerator components were collocated in the open field to the north of the building. The desiccant storage, conditioner component, and cooling tower were placed within the walled HVAC equipment area on the west end of the subject wing with the chiller. Piping connecting the storage tank and conditioner supply strong desiccant for the dehumidification process, and a 100-ft-long duct run was installed across the roof to connect the conditioner to the fresh air intake of the building.

![LDAC/Solar System Schematic](image)

**Figure 14. LDAC system and supply air layout.**
(Illustration by NREL)

5.3.1 Conditioner

The conditioner component (Figure 15) is a proprietary plastic heat-exchanger design that brings liquid desiccant into direct contact with ventilation air to dehumidify the outside air, while simultaneously rejecting heat to the cooling tower water. The system was designed to operate with CaCl₂ as the liquid desiccant (up to 43% salt and 57% water), which is suitable for solar applications. The conditioner is designed to reduce the RH of the conditioned air to 35% - 40% RH. A 3.5-horsepower (hp) fan with a variable-frequency drive (VFD) is designed to provide 3,000 cfm of treated air to the building. Two fractional (1/4) hp pumps circulate ~ 3 gal per minute (gpm) of desiccant between the storage tank and the conditioner. A 1-hp pump supplies 37 gpm of cooling tower water to the conditioner.
The cooling tower (Figure 16) is an EVAPCO model ICT 3-63 packaged unit with a 1-hp fan with a VFD.
5.3.2 Regenerator
The water vapor absorbed by the liquid desiccant is carried away in the humid exhaust air exiting the regenerator (Figure 17). The regenerator installed at Tyndall AFB is designed to utilize 165-190°F water to drive the dehumidification process. The water is heated by a plate and frame heat exchanger with the solar thermal heating system providing the heat. A 0.7-hp fan provides the 900 cfm of scavenging air needed to reject the moisture to the ambient air. A fractional (1/4) hp pump circulates ~4 gpm of desiccant between the storage tank and the regenerator. A 1-hp pump supplies 18 gpm of solar-heated water to the regenerator.
5.3.3 Solar Collectors

The solar array (Figure 18) is a Viessmann 200-T model evacuated-tube collector using 29 30-tube bundles covering 1,350 ft² of area. The 200-T model evacuated-tube collector is a tube-in-tube flow through design, which has different performance characteristics than the standard heat pipe design. The solar collector transfers the collected solar energy to a heat transfer fluid that is then transferred through a plate and frame heat exchanger to the regenerator. Because the solar collector stagnates the entire liquid volume internal to the collector, this solar collector requires unusually high expansion tank volumes to deal with stagnation pressures as well as a heat transfer fluid that can resist 450°F stagnation temperatures. For this reason, water was used as the heat transfer fluid. Under normal conditions, propylene glycol could be used with these collectors, provided that stagnation is considered an uncommon occurrence (occurring less than about three times per year). However, the LDAC system would frequently be shut down for maintenance or lack of cooling load throughout its normal operation. The use of water then necessitates draining the unit prior to freezing conditions every year and startup must occur after the last expected freeze date. For this reason, this particular collector design was a poor choice to be paired with an LDAC system in a freezing climate. Furthermore, heat-pipe, evacuated-tube designs require far less expansion tank volume than do the 200-T. A solar system is best used
when there is a year-round thermal load to maximize this more expensive thermal energy source. Currently, islands with high fuel costs are the target application for pairing LDAC with solar thermal energy.

**Figure 18. Image and design of solar collector array.**
(Photo by NREL)

### 5.3.4 Desiccant Storage Tank
Desiccant storage is provided in an 800-gal, uninsulated, plastic tank (Figure 19). A proprietary plastic interchange heat exchanger is plumbed between the tank and the regenerator. It precools hot, strong desiccant coming from the regenerator and preheats weak desiccant going to the regenerator. This process reduces the heat input required to the regenerator and supports concentration stratification in the storage tank. This enhances performance by supplying the conditioner with the strongest desiccant possible at all times.
Regenerator and conditioner operation are essentially independent. Regeneration components are activated when sensors indicate the solar array is capable of generating minimum productive temperatures (~160°F). The control sequence for the conditioner was modified throughout the 2011 cooling season in an attempt to evaluate the technology in a variety of operating modes.

The system components have pumps and fans to drive liquid and air through the system. Table 9 lists the size and power draw of the pumps and fans that are used on the LDAC system.
Table 9. LDAC System Pump and Fan Motor Schedule

<table>
<thead>
<tr>
<th>Function</th>
<th>Type</th>
<th>Motor Nameplate Data</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Power (hp)</td>
</tr>
<tr>
<td>Conditioner</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Supply Pump</td>
<td>Centrifugal</td>
<td>0.25</td>
</tr>
<tr>
<td>Return Pump</td>
<td>Vertical</td>
<td>0.25</td>
</tr>
<tr>
<td>Process Fan</td>
<td>Motorized Impeller</td>
<td>3.45</td>
</tr>
<tr>
<td>Cooling Tower</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pump</td>
<td>Jet Pump</td>
<td>1</td>
</tr>
<tr>
<td>Fan</td>
<td>Axial Propeller</td>
<td>1</td>
</tr>
<tr>
<td>Regenerator</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Supply Pump</td>
<td>Centrifugal</td>
<td>0.25</td>
</tr>
<tr>
<td>Return Pump</td>
<td>Centrifugal</td>
<td>0.25</td>
</tr>
<tr>
<td>Loop Pump</td>
<td>Vertical</td>
<td>0.24</td>
</tr>
<tr>
<td>Fan</td>
<td>Motorized Impeller</td>
<td>0.71</td>
</tr>
<tr>
<td>Solar Array</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Loop Pump</td>
<td>Circulating</td>
<td>0.24</td>
</tr>
<tr>
<td>Loop Pump</td>
<td>Circulating</td>
<td>0.33</td>
</tr>
</tbody>
</table>

5.4 OPERATIONAL TESTING

Field testing was conducted in two phases: startup and monitoring. During startup, NREL and Mountain Energy Partnership (MEP) installed sensors and confirmed HVAC/data system operation on-site. Startup commenced as the equipment installation proceeded in winter 2009 and concluded 2 weeks later. The performance of the system was monitored over the 2010 and 2011 cooling season, and the unit was shut down and winterized each winter.

5.5 SAMPLING PROTOCOL

An initial site visit to Tyndall AFB was made in December 2009 to install the monitoring system for the LDAC system. All of the sensors and data loggers were installed at that time; however, the solar collector and LDAC were not functioning properly due to the improperly designed stagnation strategy with the Viessmann solar collectors. Modifications to the original solar collector design were required to accommodate normal stagnation conditions of the system. The monitoring system could not be fully commissioned until normal operation of the LDAC was achieved in July 2010. A site visit was made in July 2010 to complete the monitoring system installation.
Normal operation of the LDAC and a complete monitoring system facilitates an initial estimate of uncertainty in the measured LDAC heat flows. The sensible and latent capacities of the LDAC are of primary importance in evaluating system performance. The monitoring system directly measured the latent and sensible capacities on the airside of the conditioner by measuring the air velocity with a pitot tube, the change in dry-bulb temperature using two resistance-temperature device sensors, and the change in humidity ratio using two capacitive RH sensors. Flow rates through the conditioner were inferred using the pressure drop through the conditioner and the fan-speed indication, measured by the conditioner’s programmable logic controller (PLC) and transferred to the data logger via Modbus communication. The heat removed by the cooling tower was intended to be a direct measurement of the total capacity of the conditioner. Water-side measurement using a turbine flow meter and two thermistors were used because it was expected to have lower uncertainty than the air-side measurement. A third measurement of latent capacity was estimated by determining the change in liquid volume of the desiccant storage tank during periods when only the conditioner was in operation.

A list of monitoring points and sensor accuracy is provided in Table 10. In addition to sensors installed by MEP, outputs from the LDAC controller were transferred via Modbus communication and recorded by a Campbell Scientific CR1000 data logger.
### Table 10. Sensor Accuracy Summary

<table>
<thead>
<tr>
<th>Sensor</th>
<th>Location</th>
<th>Vendor</th>
<th>Model</th>
<th>Accuracy Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Immersed thermistor</td>
<td>Collector loop</td>
<td>Omega Engineering</td>
<td>TJ36-44004</td>
<td>±0.18 °F</td>
</tr>
<tr>
<td>Immersed thermistor</td>
<td>Cooling tower</td>
<td>Omega Engineering</td>
<td>ON-910-44006</td>
<td>±0.18 °F</td>
</tr>
<tr>
<td>Temperature and RH</td>
<td>Duct mount</td>
<td>Vaisala</td>
<td>HMD40Y</td>
<td>±0.36 °F, ±2 % RH</td>
</tr>
<tr>
<td>Temperature and RH</td>
<td>Wall mount</td>
<td>Vaisala</td>
<td>HMW40Y</td>
<td>±0.36 °F, ±2 % RH</td>
</tr>
<tr>
<td>Temperature</td>
<td>Supply register</td>
<td>Cantherm</td>
<td>MF52</td>
<td>±0.36 °F</td>
</tr>
<tr>
<td>Turbine flow meter</td>
<td>Collector</td>
<td>Omega Engineering</td>
<td>FTB1431</td>
<td>1 % of reading</td>
</tr>
<tr>
<td>Turbine flow meter</td>
<td>Cooling tower</td>
<td>Omega Engineering</td>
<td>FTB8015B-PT</td>
<td>1.5 % of reading</td>
</tr>
<tr>
<td>Turbine flow meter</td>
<td>Cooling makeup</td>
<td>Omega Engineering</td>
<td>FTB602B-T</td>
<td>1 % of reading</td>
</tr>
<tr>
<td>Turbine flow meter</td>
<td>Desiccant</td>
<td>Omega Engineering</td>
<td>FTB6207-PS</td>
<td>1.5 % of reading</td>
</tr>
<tr>
<td>Differential pressure</td>
<td>Conditioned makeup</td>
<td>Setra</td>
<td>264</td>
<td>1 % of full scale</td>
</tr>
<tr>
<td>Differential pressure</td>
<td>Total makeup</td>
<td>Setra</td>
<td>264</td>
<td>1 % of full scale</td>
</tr>
<tr>
<td>Differential pressure</td>
<td>LDAC unit</td>
<td>Setra</td>
<td>264</td>
<td>1 % of full scale</td>
</tr>
<tr>
<td>Ambient pressure</td>
<td>Outdoor</td>
<td>Setra</td>
<td>276</td>
<td>1 % of full scale</td>
</tr>
<tr>
<td>Electrical energy</td>
<td>Regen and conditioner</td>
<td>Continental Controls</td>
<td>WNB-3D-240-P</td>
<td>0.5 % of reading</td>
</tr>
<tr>
<td>Current transformer</td>
<td>Regen and conditioner</td>
<td>Continental Controls</td>
<td>CTS-0750-30</td>
<td>1 % of reading</td>
</tr>
<tr>
<td>Pyranometer</td>
<td>Horizontal</td>
<td>Campbell Scientific</td>
<td>CS300</td>
<td>5 % of daily total</td>
</tr>
<tr>
<td>Level transmitter</td>
<td>Desiccant tank</td>
<td>Omega Engineering</td>
<td>LVU 109</td>
<td>+/- 0.6 cm</td>
</tr>
</tbody>
</table>

#### 5.6 EQUIPMENT CALIBRATION AND DATA QUALITY ISSUES

#### 5.6.1 Calibration of Equipment

The primary measurement instrument was the Campbell CR1000 data logger. It measured temperature, humidity, flow rate, pressure, and electric power; stored the data; and automatically relayed the data to NREL for analysis. These systems were maintained and calibrated in accordance with manufacturer’s requirements.
5.6.2 Quality Assurance

Quality assurance was provided primarily by NREL’s field and lab experience testing this type of device. All sensors were sampled every 10 seconds, and any mathematical manipulations of those primary measurements were also made on the same 10-second interval. Data were stored as averages or totals in four separate data tables identical in field description but varying in storage interval: 1-minute, 15-minute, 60-minute, and 24-hour (midnight-to-midnight).

Thermistor probes installed in pipes and immersed in water or glycol solution were used in calculating some of the primary energy flows of the collector and cooling tower of the LDAC system. The specified accuracy of 0.18 °F for the immersed sensors is the best normally available accuracy for this type of sensor. The uncertainty in the energy measurement for the collector or the cooling tower depends on the temperature difference between two fluid streams. As that temperature difference becomes small, the accuracy of the energy measurement can become unacceptably large. In this application, the temperature difference for the collector loop was approximately 18 °F on a sunny day. The temperature difference of the cooling-tower loop was typically greater than 5.4 °F, which provides sufficient temperature difference to limit accuracy problems.

Duct-mounted temperature and RH sensors were used to calculate the sensible and latent energy removed from the conditioned airstream. The sensor accuracy of the device is ±0.63 °F. The wall-mounted temperature and RH sensor, and the supply register temperature sensor were not used in calculation of primary energy flows, but were used as an indication of indoor comfort conditions.

A pyranometer was used to quantify the efficiency of the solar collector array. Two measurements of electric energy were made: one for the regenerator unit, including the collector, and one for the entire conditioner unit.

Differential pressure sensors were used to measure airflow rates of the LDAC unit and the main air handler. One pressure sensor was used with a pitot tube to measure the airflow rate exiting the conditioner. Another pressure sensor measured the pressure difference across the LDAC core as an additional indication of airflow rate and heat exchanger fouling over time. The measurement of liquid flow rate in the collector loop and the cooling tower loop were used in calculating primary energy flows for the LDAC system. Relative difference between the pitot tube and conditioner pressure drop is a good indicator to show consistent airflow performance.

5.7 SAMPLING RESULTS

A detailed summary of the sampling results is provided in the report’s performance assessment section.
### 6.0 PERFORMANCE ASSESSMENT

Performance evaluation of the LDAC began in the summer of 2010. Three weeks of continuous operation was recorded during the 2010 cooling season, and around 5 months of operation were recorded for 2011. Because the majority of the LDAC system operation occurred during the summer months of 2011, the performance assessment is based on summer 2011 data. The 2010 performance data are presented to illustrate performance variability. Representative performance assessment metrics for each objective are summarized in Table 10.

Table 11 describes the performance objectives, metrics, and data requirements to determine the performance objective results, and the criteria for achieving the objectives.

<table>
<thead>
<tr>
<th>Performance Objective</th>
<th>Metric</th>
<th>Data Requirements</th>
<th>Success Criteria</th>
<th>Results</th>
</tr>
</thead>
<tbody>
<tr>
<td>Improve humidity control and comfort (energy efficiency)</td>
<td>Hours outside psychrometric comfort zone</td>
<td>&lt;1% of hours outside ASHRAE summer comfort zone</td>
<td>Achieved but inconclusive cause</td>
<td>Achieved but inconclusive cause</td>
</tr>
<tr>
<td></td>
<td>Chiller power</td>
<td>Reduce chiller/heat runtime</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Reheat runtime</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Provide high-efficiency dehumidification (energy efficiency)</td>
<td>EER</td>
<td>EER &gt;40 (Btu/hr)/W</td>
<td>Not achieved</td>
<td>Achieved</td>
</tr>
<tr>
<td></td>
<td>COP</td>
<td>&gt;0.7 Thermal COP</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 11. Performance Objectives
| Sustain high-dehumidification performance (energy efficiency and maintenance) | • Conditioner heat exchange effectiveness  
• Desiccant charge  
• Supply air pressure drop  
• Conditioner cooling water pressure drop  
• Projected service life | • Supply-air temp./humidity  
• Ambient temp./humidity  
• Desiccant chemistry and concentration  
• Conditioner core-air and water-pressure drop | • <5% degradation of HX eff. over 3 years  
• <Once-per-year desiccant/buffer adjustment  
• Negligible increase in air/water pressure drop  
• Above criteria should support >10yr service life projection | • Achieved; no degradation of desiccant during operation  
• Duration of performance evaluation too small to determine |

| Qualitative Performance Objectives | Maintainability (ease of use) | Ability of an HVAC technician to operate and maintain the technology | Standard form feedback from the HVAC technician on usability of the technology and time required to maintain | A single facility technician able to effectively operate and maintain equipment with minimal training | • Many unforeseen maintenance issues occurred during initial demonstration  
• Many lessons learned for design and ease of operation |
6.1 IMPROVE HUMIDITY CONTROL AND COMFORT

The LDAC provides primarily latent cooling to the air, so the total cooling is representative of dehumidification. To show the effect of latent cooling, Figure 20 and Figure 21 illustrate the decrease in humidity of the air.

Figure 20. Outdoor air conditions and conditioner-outlet air conditions for summer 2010 operation
Figure 21. Outdoor air conditions and conditioner-outlet air conditions for summer 2011 operation

During the 2010 cooling season, the unit dehumidified the air into the 40% to 55% RH range and was not as close to the RH equilibrium point of ~35% RH for CaCl₂ desiccant solution because the system was operating on a weaker desiccant.

The LDAC unit noticeably reduced the enthalpy of the outside air during the 2011 cooling season, and the RH of the outside air was reduced to 35% to 60% RH throughout the summer. As noted above, the drying potential of the system is based on the concentration of the desiccant entering the conditioner, and the system is operating with a weaker desiccant when the conditioned air RH is in the 40% to 60% range.

Figure 22 illustrates the daily cooling supplied by the LDAC during the 3 weeks of summer 2010 operation. The daily average cooling was 79.3 ton-hr.
Figure 22. Daily cooling load supplied by the LDAC during the 3 weeks of summer 2010 operation

Figure 23 shows the total cooling load provided by the LDAC during the summer of 2011. Several days were omitted from Figure 23 because of low or no operation on those days. The average daily cooling is 56 ton-hr, which is representative of a 10-ton cooling system that operates for 5.6 hr during the day. This is below the expected cooling load achievable from this system, which is greater than 100 ton-hr/day. The reason the cooling load was lower than anticipated was primarily based on the fact that the supply fan for the conditioner was operated at less than 75% of the rated flow rate (3,000 cfm) for the 2011 cooling season, and the unit was not providing the full capacity that it was designed to provide. In addition, the control sequence for the desiccant storage tank was set up to allow for more total hours of cooling per day at a weaker desiccant concentration, which also had the effect of reducing the cooling potential of the unit.
6.1.1 ASHRAE Comfort Zone

The criterion for improved comfort is the number of hours indoor air conditions (temperature and RH) fall outside of the “comfort zone,” which has been established by ASHRAE, as shown in Figure 24. Because the Tyndall AFB also uses a chiller for conditioning the air with a capacity much larger than the LDAC (about 10 times larger), the resulting supply air conditions are not fully dictated by the LDAC.

The success criterion for comfort is achieving less than 1% of hours outside of the comfort zone. The comfort zone for the summer is shifted to warmer air temperatures because less clothing is worn during the summer; for a laboratory and AFB, more clothing is worn than at civilian facilities, so the winter comfort zone is more applicable for the Tyndall AFB for certain occupants. This success criterion was not met for 2010 cooling season but was met for the 2011 cooling season.

To illustrate the performance of the chiller alone, Figure 24 shows the outdoor air conditions and the laboratory air conditions from summer 2010.
The existing HVAC system and air-cooled chiller have the ability to maintain the space within the comfort zone for the majority of the summer; the lab is outside of the comfort zone 4.3% of the time (when the LDAC is not operating). During the 3 weeks of LDAC operation in 2010, the system met comfort conditions 10.6% of the time. It is not apparent from this data that the LDAC system has a significant effect on the indoor conditions for the 2010 cooling season. There are two reasons that this could be the case: the first is related to the fact that the system was operating on weak desiccant and wasn’t drying the air to its full potential; and secondly, the LDAC unit is treating less than one-third of the outside air for this AHU. If it was treating 100% of the outside air, the total moisture removal rate would be significantly higher.
In the summer of 2011, the LDAC operated for most of the summer; the outdoor air and laboratory conditions are shown in Figure 26. Although the outdoor air conditions are slightly different for the 2010 and 2011 summers, it is clear that the indoor air is drier and cooler. The lab is outside of the comfort zone only 0.2% of the time, which is below the success criterion of 1%. This includes both the summer and winter ASHRAE comfort zone as it is difficult to differentiate which one should be applied on a military base because a number of occupants are wearing full military gear during the summer months, which would indicate that the winter comfort zone is more appropriate than the summer comfort zone for those individuals. As mentioned, the chiller is used to cool the air in each zone’s AHU, and the overcool/reheat cycle is also used to control humidity. Therefore, it cannot be conclusively stated that the LDAC system credibly achieved the comfort criterion.
6.1.2 Reduce Reheat Run-Time

While there is evidence that the LDAC system achieves a better range of indoor air conditions, the LDAC system was undersized for this particular application, and the chiller also controls humidity by overcooling the air to condense out the water and reheating it to a suitable supply temperature. The overcooling and reheating process adds significant cooling load to the chiller when there is a high ventilation requirement. The LDAC mitigates this process by drying the humid air before being sensibly cooled with the chiller.

The overcool/reheat process only occurs when the humidity is high. The amount of time this occurs can be determined by comparing the discharge air temperature from the main AHU to the supply-air temperature into the space (after the terminal-unit reheat coil), which activates the terminal-unit reheat coils when the temperature is too low for comfort. The discharge-air temperature was only measured after one terminal unit; and if the supply-air temperature after the terminal unit was greater than 3.5 °F above the discharge air/supply air temperature from the AHU, the reheat coils were assumed to be activated, meaning the overcool/reheat process occurred. The 3.5°F temperature differential represents about 10 times the normal differential from ductwork heat gain that was monitored in this particular facility. This method was used to approximate the reheat run-time rather than measuring the hot water reheat system directly.

In the summer of 2010, while the LDAC was off, the reheat coils activated 8.9% of the time. During the 3 weeks when the LDAC was running, the reheat coils activated 2.4% of the time. The LDAC operated for the majority of the 2011 summer, and the reheat coils activated just
1.4% of the time. Although these data are limited, they indicate that the LDAC has mitigated the necessity of the overcool/reheat cycle, and this performance metric was successfully met.

6.2 PROVIDE HIGH-EFFICIENCY DEHUMIDIFICATION

While it is apparent that the LDAC system can dehumidify air to improve comfort and mitigate the overcool/reheat cycle, it must do so while improving the efficiency of the total AC system. Several common metrics are commonly used to characterize the performance of AC systems, such as EER, kW/ton, and thermal COP. The LDAC system aims to increase the overall efficiency of the existing cooling process.

6.2.1 Electrical Efficiency: EER and kW/ton

EER is defined as the ratio of cooling supplied (Btu/hr) to the electricity [watt (W)] required by the system:

$$EER = \frac{\text{Cooling Supplied (Btu/hr)}}{\text{Electricity Required (W)}}$$

Figure 27 illustrates the daily electrical performance of the LDAC for the 3 weeks of summer 2010 operation. The average EER for the 3 weeks was 14.7 (Btu/hr)/W, a 63% improvement over the rated EER of 9.0 (Btu/hr)/W of the existing chiller.

Figure 27. Daily average of LDAC EER for 3 weeks of summer 2010 operation

Figure 28 shows the daily EER performance of the LDAC system for summer 2011. It is apparent that the performance improved steadily throughout the summer. This is because various adjustments were made to the system to operate at near maximum capacity while limiting auxiliary power consumption. As indicated by the red star in Figure 28, a VFD was installed on the cooling tower pump to lower power consumption during lower capacity
operation; this modification was critical to lowering the power consumption from the cooling tower and raising the EER.

![Figure 28. Daily and monthly average of LDAC EER for summer 2011](image)

The red star indicates when the higher efficiency cooling tower pump motor and VFD were installed. As shown in Figure 28, the EER of the LDAC is between 15 and 20 for August with an average of 18.8. The best single-day performance was achieved in July at 25 (Btu/hr)/W. Although the LDAC outperformed the existing chiller in terms of EER, the success criterion for electrical efficiency is an EER of 40, so the system did not meet this objective.

Figure 29 shows the dependence of EER on cooling load.
While there is much scatter in the data due to variability in outdoor air conditions, solar availability, and operational adjustments, there is a positive linear correlation between EER and cooling load. Above a daily cooling load of 70 ton-hr, the system operates at an average EER of about 15 (Btu/hr)/W. The following list characterizes the reasons why the unit was not able to operate at an EER of 40:

- **Cooling tower fan/pump** - Several issues caused the system to operate at lower than expected efficiency during April and May of 2011. The pump and fan of the cooling tower were set at a fixed flow and speed, which led to excessive electrical energy use to meet a given cooling load. The cooling tower pump required 1.2 kW of electric power, and the motor had an efficiency of 62%. The temperature difference across the cooling tower was also small (~2°F), which indicated that excessive flow rates were present. A new cooling tower pump was installed, in conjunction with a VFD in July 2011. A new control sequence was written that maintained the cooling tower fan at 75% fan speed and modulated the cooling tower pump flow rate to maintain a 10°F temperature difference between the cooling tower supply and return water temperatures. This resulted in over a 65% reduction in cooling tower energy use.

- **Conditioner supply fan settings** – The conditioner supply fan was operated at partial fan speeds for the 2011 cooling season and was not providing the full 3,000 cfm of conditioned air. This limited the cooling capacity of the unit and the overall efficiency. Although this was fixed towards the end of the 2011 cooling season, the cooling season was coming to an end and the amount of cooling that could be accomplished was reduced.

- **Conditioner fan power** – The long 100-ft duct run from the conditioner to the outside air intake resulted in a significant pressure drop and the need for a large supply fan. The
supply fan was rated at 3.45 hp and 2,570 W. If the conditioner was located directly at the outside air intake, it could reduce the supply fan horsepower to below 1 hp and the fan power by 70%.

- **Desiccant concentration** – The system was operated with a weaker desiccant during the majority of the demonstration as a way to ensure the system would work because of problems with the solar field. With this particular system, the rate of water absorption from the conditioner and removal from the regenerator is significantly different because of the variability in total heat production from the solar field; and the storage tank has to operate as a buffer between the two components. The settings were modified towards the end of the 2011 cooling season to operate on a stronger desiccant, but the conditioner unit did not operate on a fully strong desiccant throughout the day.

- **Pump and fan efficiency and part-load performance** – Because this was a solar demonstration, additional fans and pumps were needed to move the desiccant around, and many of them had really low electrical efficiencies (less than 60%). In addition, a number of pumps and fans were constant volume units and used the same amount of power regardless of the total cooling capacity. Because the unit was only providing about one-half of its rated cooling capacity and using the same amount of electrical power as it would if it was providing 100% of the cooling capacity, the total electrical efficiency is reduced.

In previous non-solar LDAC demonstrations, the conditioner and regenerator are located in the same enclosure, and the solar pumps are eliminated from the system. These systems, when located directly next to the outside-air intake, have successfully demonstrated electrical efficiencies above EER = 60. These issues and recommendations for future installations are provided in the lessons learned section of the report.

### 6.2.2 Thermal Efficiency – Coefficient of Performance

The daily thermal performance of the LDAC system during the summer of 2011 is shown in Figure 30. This performance metric accounts for the thermal input and output of the LDAC unit, which represents the solar heat utilization of the system.
While there is no fuel cost from this heat source, it is still beneficial to utilize the maximum amount of the captured heat. The regenerator heat measurement for the summer 2010 operation was limited to 3 days, so a graphic is not shown here; the average COP of these 3 days was 0.85, where \( \text{COP} \) is defined as:

\[
\text{COP}_{\text{thermal}} = \frac{\Delta h_{\text{air}} (\text{Cooling Supplied})}{\Delta h_{\text{Hot Water}} (\text{Solar Thermal Energy})}
\]

It should be noted that the system achieves a COP of greater than 1 on a few days due to excess desiccant storage from the previous day. The COP increased throughout the summer due to an increase in solar availability and the amount of cooling the system achieves. The success criterion for thermal COP of 0.7 was achieved for the system.

Another metric to compare the efficiency of the heat source usage is the solar efficiency. While the solar heat source is “free” after the capital costs and operational costs are accounted for, it is useful to know how efficiently the system is using the available solar energy to maximize its value. The solar efficiency is defined as the ratio of the heat transferred to the regenerator from the solar loop to the global horizontal irradiance (GHI) in W/meter\(^2\) times the solar array absorber area:

\[
\text{Solar Efficiency} = \frac{\text{Heat to Regenerator}}{\text{GHI} \times A_{\text{absorber}}}
\]

The solar efficiency was relatively constant at an average value of 68% throughout the summer, as shown in Figure 31.
6.2.3 Water Consumption

Figure 32 illustrates the monthly water usage (gal) for each ton-hr of cooling. Several issues in the first 2 months, such as blow-down pipe leakage and over-circulation of cooling tower water (evaporation losses), caused the system to use a large amount of water. The issues were corrected in June and July 2011, and the average water usage for the rest of the summer was 1.3 gal/ton-hr. This is in line with existing arguments, but also suggests that the air temperature out of the cooling tower was higher than ambient because water consumption was less than 1.55 gal/ton-hr.
6.3 PERFORMANCE SUMMARY

The 3 weeks of performance data for 2010 are summarized in Table 12.

Table 12. Summer 2010 (3 Weeks) Performance Summary

<table>
<thead>
<tr>
<th>Date</th>
<th>Cooling (ton-hr)</th>
<th>EER [(Btu/hr)/W]</th>
<th>kW/ton</th>
<th>Solar heat (MBtu)*</th>
<th>COP*</th>
</tr>
</thead>
<tbody>
<tr>
<td>7/21/10-8/14/10</td>
<td>1982</td>
<td>14.7</td>
<td>0.82</td>
<td>3.1</td>
<td>0.85</td>
</tr>
</tbody>
</table>

*Solar thermal generation only recorded for 3 days (7/21-7/23)

Table 13 provides a listing of electrical EER, electrical kW/ton, and thermal COP for each month in 2011. It is clear that the performance improved throughout the summer, both in electrical and thermal efficiency.

Table 13. Monthly (Averaged) Performance for Summer 2011

<table>
<thead>
<tr>
<th>Month</th>
<th>Cooling (ton-hr)</th>
<th>EER [(Btu/hr)/W]</th>
<th>kW/ton</th>
<th>Solar heat (MBtu)</th>
<th>COP</th>
</tr>
</thead>
<tbody>
<tr>
<td>April</td>
<td>667</td>
<td>7.8</td>
<td>1.54</td>
<td>18.1</td>
<td>0.44</td>
</tr>
<tr>
<td>May</td>
<td>1565</td>
<td>8.2</td>
<td>1.47</td>
<td>39.9</td>
<td>0.5</td>
</tr>
<tr>
<td>June</td>
<td>1837</td>
<td>12.4</td>
<td>0.97</td>
<td>35.4</td>
<td>0.62</td>
</tr>
<tr>
<td>July</td>
<td>1142</td>
<td>14.6</td>
<td>0.82</td>
<td>19.4</td>
<td>0.71</td>
</tr>
<tr>
<td>August</td>
<td>1916</td>
<td>18.8</td>
<td>0.64</td>
<td>32.2</td>
<td>0.71</td>
</tr>
<tr>
<td>September</td>
<td>1300</td>
<td>15.1</td>
<td>0.79</td>
<td>26.7</td>
<td>0.73</td>
</tr>
</tbody>
</table>

Figure 33 annotates the timeline of the LDAC performance to synchronize the events and changes that occurred to the LDAC system throughout the summer of 2011 with the performance metrics described in the previous sections.
Figure 33. Timeline with electrical (EER) and thermal (COP) efficiency performance labeled with LDAC system events and changes
6.4 SUSTAIN HIGH-DEHUMIDIFICATION PERFORMANCE

Several metrics were defined for the quantification of sustained performance: conditioner heat exchange effectiveness; desiccant charge; supply air and cooling water pressure drop; and projected service life. Because the system only operated for about 6 months at various operating points, these metrics are difficult to quantify.

6.4.1 Conditioner Heat Exchange Effectiveness
The conditioner heat exchange effectiveness is defined as the change in enthalpy between the outside air and discharge air from the conditioner divided by the outside air enthalpy minus the desiccant enthalpy:

\[ \varepsilon = \frac{\Delta h_{air}}{h_{air, in} - h_{des, Tcold}} \]

Because the unit only operated for a couple of weeks during the 2010 cooling season, it is not possible to calculate a change in heat exchanger effectiveness.

6.4.2 Desiccant Charge
This metric quantifies the addition or removal of desiccant that must occur during the course of a year. Because the system only operated for one complete cooling season, there is not enough statistical data to quantify this metric. Additional salt (in solution form) was added to the system in June 2011 because the original concentration calculations overestimated the desiccant concentration. This addition was not associated with a loss of desiccant concentration over time. Kathabar liquid-desiccant systems have run for many years without requiring replacement or adjustment of the desiccant charge; however, it is appropriate to inspect the desiccant at least once a year.

There are also times when the desiccant solution picks up contaminates from the air or from cooling water or boiler (solar) water leaks. For example, the 3,000-cfm Kathabar unit that ran for a number of years at the University of Maryland CHP test facility was located too close to the exhaust of a natural gas engine. The desiccant in that unit absorbed combustion products and became very acidic over time. These issues can be properly addressed with adequate planning and design.

6.4.3 Pressure Drop Increases
It is necessary to sustain low-pressure drops in the system to maintain high-efficiency cooling. The quantification and statistical certainty of this metric also suffers from the relatively low time span in which the system was operational. Several maintenance issues can be addressed to ensure low-pressure drop for both the process air and the water flows.

Air filters for the regenerator and the conditioner must be changed regularly, depending largely on the quantity of outdoor air particulates. The filters at Tyndall AFB have lasted between 2 and 3 months, while filters on a system in Los Angeles, California, must be changed every month.

Desiccant and water filters can also cause an increase in pressure drop if not inspected. One major issue, which increased the pressure drop across the cooling tower unit, was due to not
maintaining biological growth control. This issue caused the cooling tower water filter to clog and required replacement every 4-6 weeks. This issue was resolved by adding biocide to the water. Additionally, a new strainer replaced the filter, which decreased the pressure drop. With proper design and filter selection, pressure drop can be reduced and maintained without intensive man-hours.

6.4.4 Service Life Projection
In order to achieve attractive life cycle cost and energy savings, the system service life must be greater than 10 years. While demonstrating a new technology over the course of 2 years, this projection does not come without uncertainty. It is expected that the service life for the LDAC system is greater than 10 years, and proper design will support this projection.

In particular, it is expected that the non-desiccant components in a well-installed and maintained system—including the solar field, the cooling tower, the storage tank, and the balance of the balance systems (enclosures, fans, pumps, controls, etc.)—would have a service life greater than 10 years. It is expected that some component failures would occur over time, such as loss of a pump, an instrument failure, etc. This is not uncommon for many HVAC systems.

Of the three key desiccant components (i.e., conditioner, regenerator, and interchange heat exchanger), the system that should be closely monitored is the regenerator (the actual plastic parallel-plate HMX inside the regenerator unit). Some degradation in the flock on the plates running in the field has been observed on the Los Angeles Whole Foods LDAC units. This problem has not been seen on two earlier prototypes, and so far there is no sign of the same problem with the Tyndall regenerator. The conditioners and interchange heat exchanger have held up well in the field on all previously demonstrated units up to this point.

6.5 MAINTAINABILITY

Ideally, regular maintenance would be limited to tasks such as inspecting and changing filters; in practice, this system has required much more attention. The largest share of those issues stem from the poor initial design and installation of the solar field. The most significant issue is that the solar system has and continues to experience stagnation problems. With no secondary hot-water load and no heat dump built into the system, stagnation happens occasionally. A system change was made to run the collectors on water and not glycol, which means that the system is no longer freeze protected; and therefore, must be shut down and drained during the winter. The cooling tower is also not freeze protected. Protecting the system against both stagnation and freezing is essential to run the system over the course of 10 years.

There were also a number of issues with the control of the system that required attention. For example, there have been zero offset shifts on the analog-to-digital cards that read the 4-20 milliamp signal from the desiccant storage tank level sensor. This seemed to occur at times after an electrical storm and was resolved by adding a lightning surge arrestor.

There are other issues with the VFD/fan motor/PLC proportional-integral-derivative loop for the process fan. The VFD/fan motor has faulted and failed to automatically reset on several occasions. The fault occurs infrequently and is difficult to reproduce for determining the source of the problem. As a result, the process fan was set at a fixed speed.
Many of the performance and maintenance issues, were identified and resolved throughout the 2011 summer of operation. This demonstration has undoubtedly been a “lessons learned” experience, and many issues will be improved upon to reduce or eliminate unnecessary maintenance and/or repair. A single facility technician should be able to maintain the system with a minimal amount of training for future projects, but that was not the case for this particular project.
7.0 MARKET ANALYSIS

7.1 COST MODEL

Table 14 summarizes the displaced load on the chiller and the approximate energy and cost savings from the LDAC. It should be noted that these savings may slightly underestimate the actual savings because excess cooling due to the overcool/reheat cycle, which is mitigated by the LDAC, is not accounted for in the analysis.

Improved performance in August 2011 led to the largest energy and cost savings, which is indicative of the performance potential of the LDAC system. Unforeseen maintenance and operation issues arose during the summer months, and this hindered the sustained high performance of the system.

<table>
<thead>
<tr>
<th>Month</th>
<th>Cooling (ton-hr)</th>
<th>Chiller Elec. (kWh)</th>
<th>LDAC Elec. (kWh)</th>
<th>Elec. Savings (kWh)</th>
<th>Elec. Cost Savings ($)</th>
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</thead>
<tbody>
<tr>
<td>April</td>
<td>667</td>
<td>890</td>
<td>1,026</td>
<td>-137</td>
<td>-14</td>
</tr>
<tr>
<td>May</td>
<td>1,582</td>
<td>2,110</td>
<td>2,325</td>
<td>-215</td>
<td>-21</td>
</tr>
<tr>
<td>June</td>
<td>1,837</td>
<td>2,449</td>
<td>1,774</td>
<td>676</td>
<td>68</td>
</tr>
<tr>
<td>July</td>
<td>1,239</td>
<td>1,652</td>
<td>1,131</td>
<td>521</td>
<td>52</td>
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<tr>
<td>Aug</td>
<td>1,916</td>
<td>2,554</td>
<td>1,223</td>
<td>1,331</td>
<td>133</td>
</tr>
<tr>
<td>Sept</td>
<td>1,333</td>
<td>1,778</td>
<td>1,099</td>
<td>678</td>
<td>68</td>
</tr>
</tbody>
</table>

The total cost savings for the 2011 cooling season was $321. The installed costs for the solar thermal system were $170,000, and the installed costs for the LDAC components were $40,000, for a total installed cost of $210,000, and a simple payback of 654 years. Because this was a pre-commercial system, the simple payback is not indicative of the paybacks of a commercial system. If the system would have operated per design intent, the cost savings would be substantially higher. In addition, in building types with electric reheat, the zone-level reheat savings dwarf the energy savings from the mechanical chiller. Reheat energy use in hospitals for example has been documented to account for over 30% of the total energy use. Finally, when the system is coupled with solar thermal, the solar thermal component becomes the most expensive part of the system and solar incentives or high utility rates are required to offset the increased costs of the solar thermal system.

One of the first commercial LDAC systems is being installed at the Coral Reef Fitness and Sports Center on Andersen AFB in Guam. A 6,000-cfm conditioner was designed for this system. The power requirements per ton of cooling for the existing building level chiller and LDAC are 1.05 kW/ton and 0.3 kW/ton, respectively. Note that the power requirement of the chiller does not account for the chiller water pumps, so the power requirement may be slightly greater in reality. The system is designed with an evacuated-tube solar thermal field supplying 80% of the thermal power and a backup diesel-powered boiler providing 20% of the thermal
power. The system is expected to reduce HVAC energy use by 34% and save $145,395 per year with an estimated simple payback of 11.6 years.

7.2 RELEVANT MARKETS

The LDAC system typically used for outdoor air dehumidification, and an electric chiller is typically required to sensibly cool the air to the desired temperature. The energy consumption from the LDAC includes heat for regeneration and electricity for the pumps and fans in the system. The LDAC is most suitable where:

- The existing HVAC system is not able to meet latent loads on a facility
- Humidity control is required
- Overcool/ reheat strategies are used in traditional HVAC systems
- Large quantities of ventilation air are needed

The LDAC should be applied to hot/humid climates that require year-round cooling and dehumidification. Future installations should focus on facilities in ASHRAE climate zones 1A and 2A. A full market analysis with a listing of appropriate building types, locations, design characteristics, and life cycle costs will be provided in the latest DoD LDAC demonstration in Guam.
8.0 IMPLEMENTATION ISSUES
The project’s focus was necessarily changed to focus on discovery of technical issues with this new emerging technology. Many of the issues arose because the installation had many unique features including the following:

- The demonstration was the first combination of solar heat with this type of LDAC system.
  - Due to initial budgetary constraints, the LDAC relied solely on solar heat with no natural gas backup to ensure that the unit operated throughout the cooling season. A properly designed system that uses solar heat will have backup. Due to this, the system did not achieve peak-cooling capacity for significant hours of operation. Because the system largely has static power draw, this resulted in a low average EER.
  - The solar field designer and LDAC system design were not tightly coordinated by the prime installation contractor (Regenesys). This resulted in a design that did not consider the frequency and duration of stagnation periods for the solar field. The collector design was not designed to withstand more than about two stagnations per year. Furthermore, the collector system was not initially designed to withstand the massive volume of steam from these collectors when stagnation occurred. The solar field required significant redesign. The end result was workable for the demonstration despite being problematic and suboptimal in operation.

- The demonstration was the first to create a split system where the conditioner and regenerator were contained in separate packages and separated by around a 100-ft distance. This technical challenge resulted in a suboptimal pumping design configuration because of the necessary pump size to transfer desiccant this distance. Future designs should reduce the distance from the regenerator and conditioner.

- This demonstration was the first to have 10 hours of desiccant storage using CaCl₂. Tuning the storage to achieve optimal efficiency was required. The desiccant charge and the tank’s low and high levels have significant impact on efficiency, capacity, and solar utilization. These variables were tuned as the demonstration progressed.

- This demonstration required the placement of the conditioner unit about 100 feet from the outdoor intake to the building. This required significant fan power to move the air from the mechanical yard to the building. Future designs and applications should consider the duct length reduce the duct run from the conditioner to the outdoor air intake as much as possible.

- The demonstration did not treat 100% of the outdoor air, thus limiting the benefit to energy savings from offset cooling. In order to offset the reheat for such an installation, a system should be designed to ensure that the LDAC meets a significant portion of the latent load. Typically, the LDAC can meet 100% of a building’s latent load if designed to treat 100% of the outdoor air.
8.1 SUMMER 2010

Many issues came up in the summer of 2010, which led to limited operation during the 2010 cooling season. The Viessmann 200-T solar collector array caused the internal working fluid (Tyfocor HTL glycol) to heat up to greater than the designed limitation (339°F) during stagnation, which caused two main issues for the system: 1) the burning of the glycol renders its antifreeze properties less effective, and makes the fluid acidic; and 2) high-pressure buildup triggers the pressure release valve. These issues could be avoided by using a heat-pipe (Viessmann 300-T) design instead of a direct-flow design because the heat-pipe design has a stagnation temperature of 302°F, which is below the upper limit for the glycol. Also, the glycol has a vapor pressure of 4 bar at 302°F, which is below the release pressure of the relief valve. An alternative solution was used to replace the glycol solution with water, which eliminated the risk of corroding the piping from burned glycol.

It was found that the dead-bands used to limit short cycling of the system for the upper and lower limits of the storage tank were larger than necessary; the dead-bands were decreased accordingly. It was also determined that decoupling the conditioner and regenerator operation allowed for optimal control on the system by allowing each unit to operate independently based on the availability of concentrated desiccant and solar radiation, respectively.

8.2 SUMMER 2011

The LDAC system successfully operated for the majority of the 2011 cooling season (May through September), but several issues arose regarding the operation and performance monitoring. During the start-up period, several maintenance procedures were required; there were pressure leaks in the solar collector array piping and a collector tube required replacement. It was observed that the system was running on lower desiccant concentrations than the design condition, so several drums of concentrated salt solution were ordered to be added to the system. The storage tank volume limits were also lowered in effort to run the system with higher concentrated desiccant, which increased the cooling capacity and efficiency of the system.

Several issues caused the system to operate at lower than expected efficiency during April and May 2011. The pump and fan of the cooling tower were set at a fixed flow and speed, which led to excessive electrical energy use to meet a given cooling load. The cooling tower pump required 1.2 kW of electric power, and the motor had an efficiency of 62%. The water temperature change across the cooling tower was also small (~2°F), which indicated that excessive flow rates were present. A new cooling tower pump was installed, in conjunction with a VFD. A new control sequence was written that maintained the cooling tower fan at 75% fan speed and modulated the cooling tower pump flow rate to maintain a 10°F temperature difference between the cooling tower supply and return water temperatures. This resulted in over a 65% reduction in cooling tower energy use.

The concentration of the liquid desiccant was initially below the desired value. Ideally, the desiccant would be 43% concentrated, which is the upper limit of concentration before crystallization occurs. The liquid level in the storage tank indicates the desiccant concentration; the lower level means the desiccant concentration is at a maximum. The concentration associated with the lower level was initially under the desired maximum concentration, so the
level was decreased accordingly (allowing the tank to fall to a lower level, which corresponds to a more concentrated solution). Increasing the concentration of the desiccant solution increased the drying rate of the air and increased the cooling capacity and overall efficiency.

The solar array used for desiccant regeneration was oversized for the system and was designed to allow for excess desiccant to be stored during the day. This regenerator flow rate exceeded the rate at which the conditioner weakened the desiccant (by absorbing water from the air) unless the conditioner was running at near maximum capacity.

Under normal operation, the desiccant in the tank is sufficiently weak such that a full day of regeneration plus operation of the conditioner system should maximize solar utilization. However, in the event the desiccant tank is fully strong and the latent load is small, the solar regeneration system may have to shut off. This occurrence creates stagnation in the solar field. The solar field stagnates in the following manner:

1. The solar water pump shuts off.
2. The fluid internal to the evacuate tubes raises in temperature and turns to steam.
3. The steam displaces the fluid in the system, which is forced into expansion tanks.
4. Continued steam generation is dissipated by a passive heat-dump radiator installed in-line between the collectors and the expansion tank.
5. The fluid pumps are locked out until the system temperature drops well below the boiling point of the fluid in the system to avoid flashing the fluid and creating a pressure spike. The pump is usually locked out until after sunset.

The solar loop was initially directly connected to the regenerator. There was far too little expansion capacity in the system to allow the collectors to stagnate properly. This resulted in an overheating/over-pressurizing event that caused steam and burnt propylene glycol to leak into the regenerator slump the first time the collectors were charged with glycol. As a result, some initial charged desiccant was lost. The final design included a heat exchanger between the solar field and regenerator to fix this issue.

8.3 LESSONS LEARNED

There are a number of lessons that should be learned from this project and applied to future projects in order to ensure successful design, installation, and operation of a solar-powered LDAC system.

8.3.1 Climate
The climate in Tyndall only required dehumidification for a few months per year (3-4 months) and experienced freezing temperatures during the winter. This limited the need for an outside air dehumidification system. Future installations should focus on climates that require year-round cooling and dehumidification, or on facilities that have problems with maintaining interior humidity levels.

8.3.2 Building Type
The LDAC system should be applied to 100% outside air AHUs that are set up to overcool and reheat the supply air. The LDAC should also be sized to meet 100% of the outside air ventilation requirements. Because this system was only sized to meet 33% of the ventilation air,
it limited the overall impact the system has on removing the appropriate amount of moisture from the outside air.

8.3.3 Solar Field
In the future all solar field installations should include the use of heat-pipe based, evacuated-tube collectors. The cheaper draw through evacuated-tube collectors used on this project caused the majority of the problems. In addition, the system should include an appropriately sized heat-dump radiators or be connected to a secondary heating system so that the heat can be used when the dehumidification loads are reduced. This will mitigate the stagnation problems and provide a means for using the waste heat.

Solar should be used where electricity and thermal energy is expensive, as with island nations. Otherwise, natural gas should be used.

8.3.4 LDAC Fans and Pumps
The efficiency of the fans and pumps should be carefully considered when designing a system. Thought should also be given to the part-load efficiency of the pumps and fans. In the future all motors should have a minimum 80% electrical efficiency, and all of the pumps and fans should either utilize an electronically commutated motor or VFD to allow for part-load operation and much better part-load electrical efficiencies. This is the new “normal” in the LDAC design. All pumps and fans are on VFDs.

8.3.5 Cooling Tower
A cooling tower water treatment program should be implemented when the cooling tower is installed. In addition the cooling tower should come equipped with high-efficiency motors and VFDs on both the cooling tower fans and pumps.

8.3.6 Sequence of Operation
The sequence of operation should modulate the flow of all of the fans and pumps during part-load operation to increase the part-load electrical efficiency. In addition, the system should be set up to shut off the system when there is not a sufficient need for dehumidification.
9.0 REFERENCES


**APPENDIX – POINTS OF CONTACT**

List all the important points of contact (POC) involved in the demonstration, such as co-investigators, sponsors, industry partners, and regulators. The list should include the following information: (1) full name; (2) complete mailing and FedEx addresses (if different); (3) telephone number, fax number, and e-mail address; and (4) the role of the individual in the project.

Use the tabular format below:

<table>
<thead>
<tr>
<th>POINT CONTACT Name</th>
<th>ORGANIZATION Name</th>
<th>Phone</th>
<th>Fax</th>
<th>E-mail</th>
<th>Role in Project</th>
</tr>
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<tbody>
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<td>Data Acquisition System</td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX D

SELECTED INFORMATION BY CANDIDATE COMPANIES

KATHABAR DEHUMIDIFICATION SYSTEMS, INC
Liquid Desiccant Engineering Reference Guide

Reliable, high efficiency desiccant dehumidification systems
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Kathabar Dehumidification Systems

Kathabar Dehumidification Systems, Inc., the world leader in industrial humidity control, has manufactured dehumidification equipment for more than 75 years. The name "Kathabar," which is derived from Greek words meaning clean or pure air, describes what Kathabar equipment does best. The primary use of Kathabar is to provide precise and energy efficient air temperature and humidity control. Kathabar maintains the process or space at the required condition regardless of weather or process variations. The "bacteria-free" benefit of Kathabar is an added feature to temperature and humidity control.

Over the last 75 years, the design of Kathabar equipment has been continually evolving. Advances in heat and mass transfer technology and advances in construction materials have been incorporated. New product lines have been developed to serve the changing needs of industrial, institutional, and commercial users as well as to reflect changes in the cost and availability of energy. Energy cost and availability issues have resulted in the development of the Twin-Cel air-to-air enthalpy recovery system. Today, KDS offers you the unbiased choice between liquid and dry desiccant systems to best meet your needs.

Kathapac FRP Series

The latest product of this design evolution, the Kathapac FRP Series, is designed to provide additional values for all dehumidification applications. This completely corrosion-resistant line of dehumidification equipment utilizes an external heat exchanger with extremely efficient packing for greater longevity and performance than previous designs. Its values and benefits include the following:

- Simple, accurate control of performance
- Large airflow capacity
- Simple integration
- Low utility consumption
- Low maintenance cost
- Improved air quality
- Insensitivity to airborne contamination
- Long equipment and desiccant life
Kathapac systems operate on the principle of chemical absorption of water vapor from air. The absorbent or desiccant solution used, Kathene®, is a water solution of lithium chloride salt. Kathene solution is non-toxic, will not vaporize, and is not degraded by common airborne contaminants.

- The ability of Kathene to remove or add water vapor from the air is determined by the temperature and concentration of the solution. The concentration of Kathene can be adjusted so the conditioner delivers air at any desired relative humidity between about 18% and 90%. For a given Kathene concentration, lower solution temperatures enable the conditioner to deliver cooler, dryer air.
- The diagram above shows the basic elements of a Kathabar system. In operation, air to be conditioned is cooled and dehumidified by contacting Kathene in the conditioner. By continuously circulating the desiccant through a heat exchanger, energy is extracted from the air and transferred to a coolant. The amount of heat extracted by the Kathabar dehumidifier is modulated by controlling coolant flow through the heat exchanger.
Sample Problem Design Data

Outside air requirements 1,000 SCFM
Outside air summer design 95°F DB, 78°F WB
Space maintained conditions 75°F, 30% R.H. 39 Gr/Lb
Internal sensible load (including fan heat) 450,000 BTU/Hr
Internal latent load 325,000 BTU/Hr
Maximum diffusion temperature difference 20°F
Available coolant 45°F chilled water
Available heat source 200°F hot water

A. Determine conditioner leaving air temperature and airflow
Leaving temperature = 75°F maintained - 20°F diffusion = 55°F
Airflow = 450,000 BTU/Hr ISL = 20,833 SCFM
20°F diffusion x 1.08

B. Select conditioner size from Engineering Data Table, page 10
Unit size 2000 will accommodate 20,833 SCFM

C. Determine maximum diffusion humidity difference
Difference = 325,000 BTU/Hr ILL = 22.9 Gr/Lb
20,833 SCFM x .68

D. Determine conditioner leaving air humidity
Leaving air humidity =
39 Gr/Lb space maintained - 22.9 Gr/Lb diffusion difference = 16.1 Gr/Lb

E. Check conditioner leaving air temperature and humidity to be sure that the desired performance falls within the conditioner performance envelope
Desired performance is 55°F, 16.1 Gr/Lb per Psychometric Chart, page 16, at a leaving air temperature of 55°F the conditioner can deliver air as dry as 11 Gr/Lb Therefore, conditioner can meet desired performance

F. Determine air temperature and humidity entering conditioner
1,000 SCFM outside air @ 95°F DB, 78°F WB, 118 Gr/Lb
20,833 SCFM - 1,000 SCFM = 19,833 SCFM return air @ 75°F, 39 Gr/Lb
Mix air temperature = 75°F + 1,000 SCFM (95°F - 75°F) = 76°F
20,833 SCFM
Mix air humidity = 39 Gr/Lb + 1,000 SCFM (118 Gr/Lb - 39 Gr/Lb) = 42.8 Gr/Lb
20,833 SCFM

Therefore, air enters at 76°F, 42.8 Gr/Lb
G. **Determine maximum coolant supply temperature** that will achieve the desired conditioner performance
Air enters conditioner at 76°F, 42.9 Gr/Lb
Air leaves conditioner at 55°F, 16.1 Gr/Lb
Air temperature depression = 76°F - 55°F = 21°F
Air humidity depression = 42.8 Gr/Lb - 16.1 Gr/Lb = 26.7 Gr/Lb
See Air to Coolant Approach Curves (see Figures 3 and 4, page 7)
With Kathapac FV, approach = 7.8°F
Maximum coolant supply temp. = 55°F - 7.8°F = 47.2°F
With Kathapac FH, approach = 10.4°F
Maximum coolant supply temp. = 55°F - 10.4°F = 44.6°F
Therefore, Kathapac FV Conditioner can provide desired performance with 45°F chilled water

H. **Determine the design moisture removal (MR) load on the conditioner**
Air humidity depression = 26.7 Gr/Lb
Airflow = 20,833 SCFM
20,833 SCFM x .643 x 26.7 Gr/Lb = 358 Lbs/Hr (MR)

I. **Determine regenerator capacity**
Lbs/Hr/Ft²
Air leaves conditioner @ 55°F, 16.1 Gr/Lb (25% R.H.)
Kathapac Regenerator Capacity Curve (see Figure 6, page 8)
Therefore, with 200°F hot water and 25% R.H. air, regenerator capacity = 40 Lbs/Hr/Ft²

J. **Calculate minimum regenerator face area** required to handle design moisture removal load
\[ \frac{358 \text{ Lbs/Hr}}{40 \text{ Lbs/Hr/Ft}^2} = 9.0 \text{ Ft}^2 \] min. face area

K. **Select regenerator having sufficient face area** using Kathapac Regenerator Engineering Data Table, page 10
Per table, select a 10 FP Regenerator with 10 Ft² face area

L. **Determine regenerator load** using design moisture removal and face area of selected regenerator
\[ \frac{358 \text{ Lbs/Hr}}{10 \text{ Ft}^2} = 35.8 \text{ Lbs/Hr/Ft}^2 \]

M. **Determine regenerator heat requirements** at design load, using Kathapac Regenerator Heat Requirements (see Figure 8, page 9)
Regenerator load = 35.8 Lbs/Hr/Ft²
Conditioner leaving humidity = 25% R.H.
Conditioner leaving temperature = 55°F
Therefore, 2,075 BTU/Lb x 358 Lbs/Hr MR = 743,000 BTU/Hr regenerator heat input
N. Determine conditioner cooling load at design conditions as follows:
Calculate sensible cooling loads
\[ 20,833 \text{ SCFM} \times 1.08 \times (76^\circ\text{F} - 55^\circ\text{F}) = 472,500 \text{ BTU/Hr} \text{ sensible load} \]
Calculate latent cooling load using design moisture removal and Kathapac Conditioner “L” Factor (see Figure 5, page 8)
Regenerator load = 35.8 Lbs/Hr/Ft²
Conditioner leaving humidity = 25% R.H.
Conditioner leaving temperature = 55°F
Therefore, “L” Factor = 1,320 BTU/Lb
358 Lbs/Hr x 1,320 BTU/Lb = 472,600 BTU/Hr latent load
Total cooling load = 472,500 BTU/Hr + 472,600 BTU/Hr = 945,100 BTU/Hr (78.8 tons)

**FIGURE 2**
System Flow Diagram
Kathapac Performance Curves

**FIGURE 5**
Kathapac Conditioner “L” Factor

**FIGURE 6**
Kathapac Regenerator Capacity
FIGURE 7
Pressure Drop through Kathapac Conditioners

FIGURE 8
Kathapac Regenerator Heat Requirements
### Kathapac Small Packaged Unit

#### Conditioner and Regenerator Engineering Data

<table>
<thead>
<tr>
<th>Kathapac Small Packaged Units (SP Series)</th>
</tr>
</thead>
</table>

<table>
<thead>
<tr>
<th>Conditioner</th>
<th>Regenerator</th>
</tr>
</thead>
</table>

#### Conditioner Units

<table>
<thead>
<tr>
<th>Unit Size</th>
<th>Airflow Min. CFM</th>
<th>Max. CFM</th>
<th>Air Face Sq. Ft.</th>
<th>Nom. Pump HP</th>
<th>Inlet SCFM</th>
<th>Outlet ACFM</th>
<th>E.S.P. Avail. in. W.C.</th>
<th>Nom. Fan HP</th>
<th>E.S.P. Nom. Pump HP</th>
<th>Unit Weight, Lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>240SP</td>
<td>1,500</td>
<td>3,000</td>
<td>6</td>
<td>6</td>
<td>1.5</td>
<td>630</td>
<td>830</td>
<td>2.0</td>
<td>1</td>
<td>1.9</td>
</tr>
<tr>
<td>400SP</td>
<td>2,500</td>
<td>5,000</td>
<td>10</td>
<td>10</td>
<td>1.5</td>
<td>950</td>
<td>1,260</td>
<td>3.2</td>
<td>1.5</td>
<td>2.0</td>
</tr>
<tr>
<td>600SP</td>
<td>3,750</td>
<td>7,500</td>
<td>15</td>
<td>2</td>
<td>1.5</td>
<td>1,480</td>
<td>2,000</td>
<td>4.7</td>
<td>3</td>
<td>2.0</td>
</tr>
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</table>

#### Regenerator Units

<table>
<thead>
<tr>
<th>Unit Size</th>
<th>Airflow Min. CFM</th>
<th>Max. CFM</th>
<th>Air Face Sq. Ft.</th>
<th>Nom. Pump HP</th>
<th>Inlet SCFM</th>
<th>Outlet ACFM</th>
<th>E.S.P. Avail. in. W.C.</th>
<th>Nom. Fan HP</th>
<th>E.S.P. Nom. Pump HP</th>
<th>Unit Weight, Lbs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5FP</td>
<td>475</td>
<td>630</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
<td>1.5</td>
<td>2.0</td>
<td>1.5</td>
<td>650</td>
</tr>
<tr>
<td>3FP</td>
<td>950</td>
<td>1,200</td>
<td>3</td>
<td>1.5</td>
<td>1.5</td>
<td>2.0</td>
<td>2.0</td>
<td>3</td>
<td>2</td>
<td>900</td>
</tr>
<tr>
<td>6FP</td>
<td>1,900</td>
<td>2,500</td>
<td>6</td>
<td>3</td>
<td>3</td>
<td>1.6</td>
<td>2</td>
<td>2</td>
<td>1.6</td>
<td>1,200</td>
</tr>
<tr>
<td>10FP</td>
<td>3,160</td>
<td>4,200</td>
<td>10</td>
<td>5</td>
<td>2</td>
<td>5</td>
<td>5</td>
<td>7.5</td>
<td>3</td>
<td>2,100</td>
</tr>
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<td>15FP</td>
<td>4,700</td>
<td>6,300</td>
<td>15</td>
<td>7.5</td>
<td>2</td>
<td>5</td>
<td>5</td>
<td>7.5</td>
<td>3</td>
<td>2,700</td>
</tr>
<tr>
<td>20FP</td>
<td>6,300</td>
<td>8,400</td>
<td>20</td>
<td>10</td>
<td>2</td>
<td>5</td>
<td>5</td>
<td>7.5</td>
<td>3</td>
<td>3,400</td>
</tr>
<tr>
<td>30FP</td>
<td>9,500</td>
<td>12,600</td>
<td>30</td>
<td>15</td>
<td>2</td>
<td>10</td>
<td>10</td>
<td>7.5</td>
<td>3</td>
<td>4,600</td>
</tr>
<tr>
<td>40FP</td>
<td>12,800</td>
<td>16,800</td>
<td>40</td>
<td>20</td>
<td>2</td>
<td>15</td>
<td>15</td>
<td>7.5</td>
<td>3</td>
<td>5,800</td>
</tr>
</tbody>
</table>

### ENGINEERING DATA NOTES

1. Nominal horsepowers listed are for typical installations. Actual horsepowers may be higher or lower depending on performance requirements.
2. Normal operating weight should be used for sizing vibration isolators, if required.
3. Maximum operating weight should be used for structural calculations.
4. All weights are approximate.
5. All regenerators are furnished with FRP exhaust plenums.
Kathapac Equipment Pictures

Kathapac Conditioners

3000 FV

4000 FH

Kathapac Regenerator

Kathapac Small Packaged (SP) Unit

6 FP

240 SP
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.

### Inlet Plenum Notes

1. Ductwork entering the inlet plenums must be designed for a maximum velocity of 15 ft/min.
2. Inlet plenums must extend a minimum of 36 inches from the unit to provide adequate room for maintenance.

### Kathapac Small Packaged Unit

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>COND. FAN HP</th>
<th>NOMINAL FAN CFM</th>
<th>E.S.P. AVAIL. @ NOM.CFM</th>
</tr>
</thead>
<tbody>
<tr>
<td>240SP</td>
<td>5</td>
<td>2,400</td>
<td>3.4” WC</td>
</tr>
</tbody>
</table>
NOTES
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.

---

FIGURE 10
400 & 600 SP Kathapac Units

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>COND. FAN HP</th>
<th>NOMINAL FAN CFM</th>
<th>E.S.P. AVAIL. @ NOM.CFM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A       B       C</td>
<td>D   E   F</td>
<td></td>
<td></td>
</tr>
<tr>
<td>400SP</td>
<td>119     125     48</td>
<td>30  12  1</td>
<td>7 1/2</td>
<td>4,000</td>
</tr>
<tr>
<td>600SP</td>
<td>163     132     72</td>
<td>45  14  1 1/4</td>
<td>10</td>
<td>6,000</td>
</tr>
</tbody>
</table>

INLET PLENUM NOTES
1. DUCTWORK ENTERING THE INLET PLENUMS MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 15 FT/MIN.
2. INLET PLENUMS MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE.
Vertical Kathapac Conditioner

**FIGURE 11**
240 FV Kathapac Conditioner

**INLET AND DISCHARGE PLENUM NOTES**

1. DUCTWORK LEAVING THE DISCHARGE PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1000 FT/MIN.

2. DISCHARGE PLENUM MUST BE ONE-HALF INCH LARGER THAN THE OUTLET OPENING. DISCHARGE PLENUM MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE. LARGER PLENUMS MAY BE REQUIRED TO MEET THE DUCT VELOCITY CRITERIA IN NOTE NUMBER ONE ABOVE.

---

**UNIT SIZE** | **NOMINAL PUMP HP** | **NOMINAL FAN HP** | **NOMINAL FAN CFM** | **E.S.P. AVAIL. @ NOM.CFM**
--- | --- | --- | --- | ---
240FV | 2 | 5 | 2400 | 2.0” WC

**NOTES**

1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.

2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.

3. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
### Vertical Kathapac Conditioner

#### FIGURE 12
400 & 600 FV Kathapac Conditioners

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>NOMINAL PUMP HP</th>
<th>NOMINAL FAN HP</th>
<th>NOMINAL FAN CFM</th>
<th>E.S.P. AVAIL. @ NOM.CFM</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
<td>C</td>
<td></td>
<td></td>
</tr>
<tr>
<td>400FV</td>
<td>74</td>
<td>14 1/2</td>
<td>48</td>
<td>2</td>
<td>5</td>
</tr>
<tr>
<td>600FV</td>
<td>98</td>
<td>16 1/2</td>
<td>72</td>
<td>3</td>
<td>7 1/2</td>
</tr>
</tbody>
</table>

**NOTES**

1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. FAN AND FAN DRIVE SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
Vertical Kathapac Conditioner

FIGURE 13
800 through 1600 FV Kathapac Conditioners

INLET AND DISCHARGE PLENUM NOTES
1. DUCTWORK ENTERING THE INLET PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1500 FT/MIN.
   DUCTWORK LEAVING THE DISCHARGE PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1000 FT/MIN.
2. DISCHARGE PLENUM MUST BE ONE-HALF INCH LARGER THAN THE OUTLET OPENING. INLET AND DISCHARGE PLENUMS MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE. LARGER PLENUMS MAY BE REQUIRED TO MEET THE DUCT VELOCITY CRITERIA IN NOTE NUMBER ONE ABOVE.

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>NOMINAL PUMP HP</th>
<th>NOMINAL FAN HP</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
<td>C</td>
</tr>
<tr>
<td>800FV</td>
<td>55-1/2</td>
<td>88</td>
<td>60</td>
</tr>
<tr>
<td>1200FV</td>
<td>85-1/2</td>
<td>118</td>
<td>90</td>
</tr>
<tr>
<td>1600FV</td>
<td>115-1/2</td>
<td>148</td>
<td>120</td>
</tr>
<tr>
<td>2000FV</td>
<td>147-1/2</td>
<td>180</td>
<td>150</td>
</tr>
</tbody>
</table>

NOTES
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. OPTIONAL FANS WITH ADAPTERS FOR MOUNTING ON UNIT DISCHARGE.
5. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
Vertical Kathapac Conditioner

FIGURE 14
2000 through 7000 FV Kathapac Conditioners

INLET AND DISCHARGE PLENUM NOTES
1. DUCTWORK ENTERING THE INLET PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1500 FT/MIN. DUCTWORK LEAVING THE DISCHARGE PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1000 FT/MIN.

2. DISCHARGE PLENUM MUST BE ONE-HALF INCH LARGER THAN THE OUTLET OPENING. INLET AND DISCHARGE PLENUMS MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE. LARGER PLENUMS MAY BE REQUIRED TO MEET THE DUCT VELOCITY CRITERIA IN NOTE NUMBER ONE ABOVE.

UNIT SIZE | DIMENSIONS (INCHES) | NOMINAL PUMP HP
--- | --- | ---
A | B | C | D | E | F | HP
--- | --- | --- | --- | --- | --- | ---
2000FV | 89 3/4 | 132 1/4 | 32 1/2 | 4 | 96 | 46 1/2 | 10
2500FV | 113 3/4 | 156 1/4 | 32 1/2 | 4 | 120 | 46 1/2 | 10
3000FV | 137 3/4 | 180 1/4 | 32 1/2 | 4 | 144 | 46 1/2 | 15
4000FV | 185 3/4 | 232 1/4 | 36 1/2 | 4 | 192 | 52 1/2 | 15
5000FV | 233 3/4 | 280 1/4 | 36 1/2 | 6 | 240 | 56 1/2 | 20
6000FV | 281 3/4 | 328 1/4 | 36 1/2 | 6 | 288 | 56 1/2 | 20
7000FV | 329 3/4 | 376 1/4 | 36 1/2 | 6 | 336 | 56 1/2 | 25

NOTES
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
5. FAN ADAPTERS AND FANS ARE AVAILABLE FOR 2000 THROUGH 3000FV UNITS.
Horizontal Kathapac Conditioner

FIGURE 16
800 through 1600 FH Kathapac Conditioners

INLET AND DISCHARGE PLENUM NOTES
1. DUCTWORK ENTERING THE INLET PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1500 FT/MIN. DUCTWORK LEAVING THE DISCHARGE PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1000 FT/MIN.
2. INLET AND DISCHARGE PLENUMS MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE. LARGER PLENUMS MAY BE REQUIRED TO MEET THE DUCT VELOCITY CRITERIA IN NOTE NUMBER ONE ABOVE.

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>NOMINAL PUMP HP</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>800FH</td>
<td>60 3/4</td>
<td>93 1/4</td>
</tr>
<tr>
<td>1200FH</td>
<td>90 3/4</td>
<td>123 1/4</td>
</tr>
<tr>
<td>1600FH</td>
<td>120 3/4</td>
<td>153 1/4</td>
</tr>
</tbody>
</table>

NOTES
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
INLET AND DISCHARGE PLENUM NOTES

1. DUCTWORK ENTERING THE INLET PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1500 FT/MIN. DUCTWORK LEAVING THE DISCHARGE PLENUM MUST BE DESIGNED FOR A MAXIMUM VELOCITY OF 1000 FT/MIN.

2. DISCHARGE PLENUM MUST BE ONE-HALF LARGER THAN THE OUTLET OPENING. INLET AND DISCHARGE PLENUMS MUST EXTEND A MINIMUM OF 36 INCHES FROM THE UNIT TO PROVIDE ADEQUATE ROOM FOR MAINTENANCE. LARGER PLENUMS MAY BE REQUIRED TO MEET THE DUCT VELOCITY CRITERIA IN NOTE NUMBER ONE ABOVE.

Horizontal Kathapac Conditioner

FIGURE 17
2000 through 7000 FH Kathapac Conditioners

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>NOMINAL PUMP HP</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>A</td>
<td>B</td>
</tr>
<tr>
<td>2000FH</td>
<td>96 3/4</td>
<td>139 1/4</td>
</tr>
<tr>
<td>2500FH</td>
<td>120 3/4</td>
<td>163 1/4</td>
</tr>
<tr>
<td>3000FH</td>
<td>144 3/4</td>
<td>187 1/4</td>
</tr>
<tr>
<td>4000FH</td>
<td>192 3/4</td>
<td>235 1/4</td>
</tr>
<tr>
<td>5000FH</td>
<td>240 3/4</td>
<td>287 1/4</td>
</tr>
<tr>
<td>6000FH</td>
<td>288 3/4</td>
<td>335 1/4</td>
</tr>
<tr>
<td>7000FH</td>
<td>336 3/4</td>
<td>383 1/4</td>
</tr>
</tbody>
</table>

NOTES
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
NOTES

1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.

FIGURE 18
1.5 through 6 FP Kathapac Regenerators

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>DIMENSIONS (INCHES)</th>
<th>NOMINAL PUMP HP</th>
<th>NOMINAL FAN HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.5FP</td>
<td>A 41 B 37 C 10 D 15 E 12 F 9 3/4 G 9 1/2</td>
<td>1 1/2</td>
<td>1 1/2</td>
</tr>
<tr>
<td>3FP</td>
<td>A 56 B 37 C 12 D 30 E 12 F 17 1/4 G 9 1/2</td>
<td>1 1/2</td>
<td>1 1/2</td>
</tr>
<tr>
<td>6FP</td>
<td>A 62 B 50 C 16 D 36 E 16 F 20 1/4 G 14</td>
<td>2</td>
<td>3</td>
</tr>
</tbody>
</table>
###DIMENSIONS (INCHES)

<table>
<thead>
<tr>
<th>UNIT SIZE</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
<th>NOMINAL PUMP HP</th>
<th>NOMINAL FAN HP</th>
</tr>
</thead>
<tbody>
<tr>
<td>10FP</td>
<td>46 3/4</td>
<td>79 1/4</td>
<td>42</td>
<td>22</td>
<td>26</td>
<td>5</td>
<td>5</td>
</tr>
<tr>
<td>15FP</td>
<td>70 3/4</td>
<td>103 1/4</td>
<td>49</td>
<td>26</td>
<td>38</td>
<td>5</td>
<td>7 1/2</td>
</tr>
</tbody>
</table>

**NOTES**
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
1. A CLEARANCE OF TWO FEET MINIMUM IS REQUIRED FOR MAINTENANCE ACCESS.
2. ALL PIPING, DUCTWORK, AND CONDUIT TO RUN CLEAR OF ALL ACCESS DOORS.
3. HOUSING AND PUMP TANK SHIP SEPARATELY.
4. TOWER AND SUMP ARE PERMANENTLY BONDED AND SHIPPED AS ONE PIECE.
1. HEAT EXCHANGER VENT TO BE LOCATED AT HIGH POINT OF KATHENE SUPPLY LINE.
2. USE THREE-BOND SEALANT ON THREADS IN KATHENE PIPING.
3. KATHENE PIPING AND FITTINGS ARE CPVC OR FRP.
4. ALLOW AT LEAST THREE FEET OF CLEARANCE ALONG ONE SIDE OF HEAT EXCHANGER FOR MAINTENANCE ACCESS.
1. USE DRIP LEG FROM STEAM MAIN WHEN:
   A: STEAM SUPPLY TO CONTROL VALVE EXCEEDS 50 FEET.
   B: STEAM SUPPLY IS BRANCHED FROM SIDE OR BOTTOM OF MAIN.
   C: STEAM SUPPLY IS LOWERED AND THEN LIFTED BETWEEN STEAM MAIN AND CONTROL VALVE OR LOWERED AND STOPPED BY CONTROL VALVE.
2. USE PRESSURE REDUCING VALVE WHEN STEAM SUPPLY PRESSURE EXCEEDS 30 PSIG.
3. WHEN CONDENSATE RETURN LINE IS OVERHEAD, A CONDENSATE RECEIVER AND PUMP MUST BE USED.
4. USE DRIP LEG AHEAD OF HEATER WHEN PIPE RUN FROM CONTROL VALVE EXCEEDS 50 FEET OR WHEN STEAM SUPPLY IS LOWERED AND THEN LIFTED BETWEEN CONTROL VALVE AND HEATER.
5. SERVICE HAND VALVES ARE REQUIRED IN KATHENE SUPPLY AND DISCHARGE LINES IF HEATER IS INSTALLED AT AN ELEVATION LOWER THAN THE REGENERATOR PUMP TANK.
6. SEE COOLER PIPING SCHEMATIC FOR INSTALLATION DETAILS OF PRESSURE GAUGE AND THERMOMETER.
7. ELEVATE HEATER TO PROVIDE SUFFICIENT HEIGHT FOR 12 INCHES VERTICAL CONDENSATE DROP AND PITCH OF GRAVITY RETURN LINE TO CONDENSATE PUMP OR RETURN MAIN.
8. VENT VALVE TO BE LOCATED AT HIGH POINT OF KATHENE SUPPLY LINE.
9. HEATER KATHENE PIPING AND FITTINGS ARE FRP.
10. USE THREE-BOND SEALANT ON THREADS IN KATHENE PIPING.
11. ALLOW AT LEAST THREE FEET OF CLEARANCE ALONG ONE SIDE OF HEAT EXCHANGER FOR MAINTENANCE ACCESS.
**Kathene Control and Piping Schematic**

**Gravity Equalized System**

**LEGEND**

**BT Bubbler Tube** - Used in conjunction with LCP to pneumatically sense solution level in the pump tank.

**FM Flow Meter** - Measures and controls the transfer of concentrated solution from the regenerator to the conditioner.

**LCP Level Control Panel** - Senses solution level and provides on/off functions such as high level, low level and water makeup. Also provides modulating control to maintain a constant solution level in the pump tank.

**TC3 Conditioner Leaving Air Temperature Controller** - Receives a proportional signal from TT3 and sends a modulating signal to operate coolant valve V3.

**TT3 Conditioner Leaving Air Temperature Transmitter** - Senses air temperature leaving the conditioner and sends a proportional signal to TC3.

**V3 Conditioner Solution Cooler Valve** - Receives a modulating signal from TC3 and controls the coolant flow entering the solution cooler.

**V4 Regenerator Solution Heater Valve** - Receives a modulating signal from LCP and controls the heating media flow entering the solution heater.

**V5 Water Makeup Valve** - Receives an on/off signal from LCP and allows makeup water to enter the conditioner pump tank.
Kathene Control and Piping Schematic
Pump In/Pump Out System

**BT Bubbler Tube** - Used in conjunction with LCP to pneumatically sense solution level in the pump tank.

**FM Flow Meter** - Measures and controls the transfer of concentrated solution from the regenerator to the conditioner.

**KV7 Conditioner Pump Out Valve** - Receives a modulating air signal from the LCP-1 and controls the solution flow to the regenerator pump tank.

**LCP Level Control Panel** - Senses solution level and provides on/off functions such as high level, low level and water makeup. Also provides modulating control to maintain a constant solution level in the pump tank.

**TC3 Conditioner Leaving Air Temperature Controller** - Receives a proportional signal from TT3 and sends a modulating signal to operate coolant valve V3.

**TT3 Conditioner Leaving Air Temperature Transmitter** - Senses air temperature leaving the conditioner and sends a proportional signal to TC3.

**V3 Conditioner Solution Cooler Valve** - Receives a modulating signal from TC3 and controls the coolant flow entering the solution cooler.

**V4 Regenerator Solution Heater Valve** - Receives a modulating signal from LCP-2 and controls the heating media flow entering the solution heater.

**V5 Water Makeup Valve** - Receives an on/off signal from LCP-1 and allows makeup water to enter the conditioner pump tank.
The following equipment is normally supplied by Kathabar:

- Kathapac conditioner
- Conditioner fan (3000 and smaller units)
- Kathene solution cooler
- Kathapac regenerator with fan
- Regenerator exhaust plenum
- Kathene solution heater & control valve
- All hand valves in Kathene solution piping
- Kathene solution transfer meter
- Electric control panel
- Level control panel

The following items are required for a complete system installation but are not normally supplied by Kathabar:

- Conditioner fan (4000 and larger units)
- Interconnecting ductwork with access doors for servicing eliminators and diffusers
- Interconnecting Kathene solution piping (except SP units factory piped)
- Conditioner coolant piping and control valve
- Regenerator steam or hot water piping
- Makeup water piping and control valve
- Interconnecting electrical wiring (except SP units factory wired)
- Insulation
- Sound attenuator and vibration isolation, if required

The following optional equipment can be supplied by Kathabar:

- Transfer interchanger
- Pre conditioning module, which may contain inlet air louver, air filters, winter preheat coil, summer precool coil and controls, complete with appropriate access doors and access plenums
- Post-conditioning module, which may contain afterheat coil, aftercool coil, final filters and controls, complete with appropriate access doors and access plenums
- Factory packaging of Kathabar conditioner and regenerator, which may include mounting of all components on a curbed FRP platform, factory installation of Kathene piping, power wiring and controls, and single-point connections for utilities
- Factory insulation of conditioner, regenerator, and optional modules

**Pre-Installation Storage**

Equipment should be protected from the weather prior to installation. Indoor storage is preferred. If indoor storage cannot be arranged, the equipment should be set on blocking and securely covered with tarpaulins. Kathene solution must be stored in an area having a minimum temperature above freezing.

**Rigging and Handling**

Kathapac equipment should be lifted only from the bottom of the unit. When lifting lugs or eyes are provided, they should be used for lifting by crane. If lifting lugs are not provided, slings should be used. Spreader bars must be used to prevent equipment damage. If the equipment must be laid on its back or side during movement, Kathabar Engineering should be consulted for advice to prevent equipment damage.

**Equipment Location**

Conditioner and regenerator units need not be installed in the same location, and may be located wherever convenient. Units may be installed outside if adequate freeze protection is provided for water, steam and condensate piping and weatherproof insulation is provided as needed.
The Engineering Data Tables should be used to obtain the operating weights of the conditioner and regenerator units for structural design. Conditioner and regenerator units should be set on a level concrete floor, housekeeping pad, or piers. If piers are to be used, contact Kathabar for recommendations on pier size and location. Before the equipment is set in place, the floor should be sealed with an epoxy sealant. To facilitate startup and normal maintenance procedures, the equipment should be surrounded by curbing. A floor drain should be located inside the curbing and near the conditioner pump tank. If the equipment cannot be surrounded by curbing, piping should be run from the safety drain connection to a floor drain or a suitable container.

Adequate area should be provided around the conditioner, regenerator, and heat exchangers for maintenance. Recommended maintenance access areas are shown on the equipment drawings.

**Plenums & Ductwork**

Kathapac units are provided with flanges on the air openings for duct connection. Inlet and outlet plenums should be bolted to the flange with a gasket between the connection. Closed-cell foam gasketing at least one-fourth inch thick is recommended.

**Access doors, for servicing diffusers and eliminators, must be provided in the inlet and outlet plenums.** See the equipment drawings for recommended access door size and location. Inlet ductwork must be designed to allow uniform distribution of the air across the entire opening.

Outlet plenums and ductwork must be designed to allow adequate room for servicing the eliminators and to provide proper airflow through the equipment. See the equipment drawings for recommended plenum and ductwork sizes.

**Regenerator Exhaust Ductwork**

Because the regenerator exhaust air is hot and humid, the regenerator exhaust ductwork should be made of fiber-reinforced polyester (FRP). The material should be rated for continuous duty at 180°F. Duct joints should be of watertight construction. Long horizontal duct runs should be pitched slightly in the direction of airflow, and should incorporate low-point condensate drains.

**Kathene Solution Piping**

The conditioner piping should be CPVC or FRP. If the design Kathene temperature in the conditioner is below freezing, FRP pipe should be used because CPVC pipe becomes brittle at low temperatures. Only FRP piping is recommended for the regenerator.

**Black iron, galvanized and stainless steel pipe must not be used for Kathene piping.** CPVC piping should be Schedule 80, Type IV, Grade 1, 4120, in accordance with ASTM Standard 1784. FRP piping should be Fibercast Centricast III EP, Smith Green Thread, or other equivalent epoxy resin pipe and fittings with an interior corrosion barrier and rated for continuous service at 225°F with chloride brines. Valves in the conditioner Kathene piping should be made of CPVC or thermoplastic-lined cast iron with non-metallic disc. Consult Kathabar Engineering for more detailed materials and construction information.

Thermowells in the Kathene piping should be titanium (available from Kathabar). Stainless steel thermowells must not be used.

All pipe fittings should be socket fittings, and all connections with valves and other components must be flanged. Threaded fittings and connections...
should be avoided because the pipe is significantly weakened, and threaded joints in non-metallic pipe are difficult to make leak-tight. Red rubber or neoprene full-face gaskets are recommended in flanged connections. The piping must be supported so that no stress is placed on connections to the Kathapac equipment. Pipe supports and anchors may require closer spacing than with metal pipe. Piping should be installed at least two feet away from all maintenance access openings and belt guards.

If the Kathene piping cannot drain completely by gravity, low-point drains with lined metal or non-metallic hand valves must be provided. Kathene pump discharge piping must be arranged to allow removal of the pump from the pump tank. The pump discharge piping should incorporate a 90° elbow or a vertical spool piece at least four feet long so the pump can be lifted vertically from the tank.

**Insulation**

To prevent surface condensation (sweating) and minimize coolant use, conditioners should be insulated whenever a coolant other than cooling tower water is used. The entire unit including Kathene and coolant piping should be insulated. Flexible rubber, rigid foam plastic, or other non-permeable, vapor-tight insulation material is recommended for conditioners. When the equipment is installed outside, an ultraviolet and weather protective coating should be applied to the insulation. Regenerators need not be insulated unless heat gain in the equipment location is a concern. Steam or hot water piping should be insulated with at least two inch thick rigid plastic faced fiberglass or equal. Kathene solution piping should also be insulated if required for personnel protection. If the equipment is installed outside, weather protective covering should be applied.

The outer casing of the conditioner and regenerator must not be penetrated with insulation fasteners. The use of contact cement or other adhesive is recommended for insulation fastening. Contact the insulation manufacturer for adhesive recommendations for an FRP substrate. All piping should be pressure tested for leaks before insulating.
**General**
The humidity conditioning system shall be of the liquid desiccant type, as manufactured by Kathabar. The system shall be capable of simultaneous air cooling and dehumidification, as described in the Performance section of this specification. The system shall automatically, fully modulate the usage of conditioner coolant and regenerator heat to match the system cooling and dehumidification loads.

The desiccant solution, Kathene, shall consist of a water solution of lithium chloride salt. The solution shall be stable and non-toxic and shall not exist in the vapor phase in the conditioned airstream. The manufacturer shall provide the end user with analysis and recommendations for maintenance of the desiccant solution six times yearly free of charge for the life of the equipment.

The humidity conditioning system shall consist of separate conditioning and regeneration units, providing complete separation of conditioned and regeneration airstreams. The manufacturer shall guarantee that there will be no cross leakage of conditioner and regenerator airstreams under any circumstances.

**Equipment**

**SP UNITS**
The equipment supplied shall consist of a conditioner, conditioner fan, conditioner Kathene cooler, regenerator, regenerator fan, regenerator Kathene heater, control panel and base platform. The conditioner and regenerator shall each consist of a water-tight housing containing the sump, inlet air diffuser, Kathene-to-air contact surface, Kathene distribution system and mist eliminator system. The housing shall be constructed of vinylester FRP. Internal parts shall be made of non-metallic corrosion proof materials. The conditioner and regenerator Kathene circulating pumps shall be of the sealless magnetic-drive type, with wetted parts made of glass-filled PVDF or glass-filled polypropylene. The conditioner fan shall consist of a galvanized steel housing containing a steel, forward-curved fan, and motor and drive. The fan housing shall be Heresite coated on the inside and painted with a prime and finish coat of industrial enamel on the outside. The conditioner fan shall be shipped loose for field mounting.

The regenerator fan shall consist of a vinylester FRP housing, glass-filled polyamide backward-inclined wheel and inlet cone, and direct-drive motor. The fan shall be factory-mounted on the regenerator. The conditioner Kathene cooler and regenerator Kathene heater shall be of the plate-and-frame type, with steel frame, carrier bars and tiebolts, titanium plates and nitrile or EPDM gaskets. The solution heater shall be supplied with the heating fluid control valve. The control panel shall consist of a NEMA 12 FRP enclosure containing fused disconnect, motor starters for all motors supplied with the equipment, start/stop buttons, system status indicator lamps and PLC controller with alphanumeric display and keypad user interface. The panel shall be factory mounted on the conditioner unit.

All the above equipment shall be mounted on a FRP-clad structural steel base. All Kathene piping shall be factory-installed using FRP and CPVC pipe and fittings. All wiring between the control panel, motors and controls...
shall be factory-installed using PVC conduit. All FRP components shall contain additives to achieve a U.L. class 1 flame spread rating. The exterior surfaces of all FRP components shall be pigmented and UV stabilized for exposure to direct sunlight.

**ALL OTHER EQUIPMENT**
The major items of equipment shall consist of a conditioner unit, a conditioner Kathene cooler, a regenerator unit, a regenerator Kathene heater, an electrical control panel and a level control panel.
The conditioner and regenerator units shall each consist of a watertight housing containing the sump, Kathene-to-air contact surface, Kathene distribution system and mist eliminator system. 240 through 600FV conditioners and 1.5 through 6FP regenerators shall be mounted on a FRP-clad platform along with sealless magnetic-drive Kathene pump having all wetted parts made of glass-filled PVDF or polypropylene. All other units shall be supplied with a freestanding pump assembly with tank, vertical sealless Kathene pump and motor, full-flow filter screen and bypass polishing filter. The pump shall be made with a titanium shaft and hardware and all other wetted parts of vinylester FRP. Unit housings and pump tanks shall be made of vinylester FRP, with additives to achieve a U.L. class 1 flame spread rating. All internal parts shall be made of nonmetallic corrosion-proof materials. All external FRP surfaces shall be pigmented and UV stabilized for exposure to direct sunlight. The conditioner shall be supplied with a discharge air plenum with mist eliminator access door, and a fan box assembly consisting of housing, forward-curved fan, motor and drive. The fan shall be made of steel. The discharge air plenum and fan shall be shipped loose for field-mounting.
The regenerator shall be supplied with a fan and fan box assembly consisting of housing, forward-curved fan, motor and drive. The fan shall be made of steel. The fan box shall be made of galvanized steel, with interior surfaces Heresite coated and exterior surfaces painted with a prime and finish coat of industrial-grade acrylic machine enamel. The fan and fan box assembly shall be shipped mounted on 6FP and smaller regenerators. The fan and fan box assembly shall be shipped loose for field mounting on 10FP and larger regenerators.
The regenerator shall be supplied with a vinylester FRP discharge plenum with eliminator access door, duct attachment collar and condensate collection ring. The plenum shall be shipped loose for field mounting.
The conditioner Kathene cooler and regenerator Kathene heater shall be of the plate-and-frame type, with steel frame, carrier bars and tiebolts, titanium plates and nitrile or EPDM gaskets. The Kathene heater shall be supplied complete with heating fluid control valve. The heat exchangers shall be shipped loose for field installation.
The electrical control panel shall consist of safety interlock relays and circuitry, motor starters for all motors supplied with the equipment, hand-off-auto switch, start-stop buttons, and system status indicator lamps, all contained in a NEMA 12 enclosure with lockable, fixed disconnect. The panel shall be shipped loose for field installation.
The level control panel shall consist of safety interlock pressure switch, unit pressure drop indicator, bubbler tube supply pneumatics, P/I transducer, I/P transducer, and PID single-loop microprocessor-based controller, all contained in a NEMA 12 fiberglass enclosure. The level control panel shall be shipped mounted.
Equipment Performance and Utilities Requirements

The Kathapac system specified herein shall provide the following design performance when furnished with the specified peak utilities:

**Design Performance**

- **Conditioner airflow:** ___ SCFM, ___" W.C. Pressure Drop
- **Conditioner fan:** ___" T.S.P., ___" Available E.S.P.
- **Summer inlet conditions:** ___°F DB, ___°F WB ___Gr/Lb
- **Summer delivered conditions:** ___°F DB, ___°F WB ___Gr/Lb
- **Summer coolant:** ___GPM of ___°F _____, _____°F T.R., _____Ft. P.D.
- **Winter inlet conditions:** ___°F DB, ___°F WB ___Gr/Lb
- **Winter delivered conditions:** ___°F DB, ___°F WB ___Gr/Lb
- **Winter heat requirements:** ___Lbs/Hr ___psig Steam

- **Regenerator airflow:** ___ SCFM, ___" W.C. Pressure Drop
- **Regenerator fan:** ___"T.S.P., ___" Available E.S.P.
- **Summer inlet conditions:** ___°F DB, ___°F WB ___Gr/Lb
- **Summer heat requirements:** ___Lbs/Hr ___psig Steam
  
  Or ___GPM of ___°F _____, _____°F T.R., _____Ft. P.D.

**Summary of Peak Utilities**

- **Coolant:** ___ Tons of ___°F _____
- **Summer heat:** ___ Lbs/Hr ___psig Steam
- **Winter heat:** ___ Lbs/Hr ___psig Steam
- **Control air:** 20 psig, Instrument Quality
- **Electrical characteristics:** ___V, ___Ph, ___Hz

- **Electrical total connected load:**
  - **Conditioner fan:** ___ HP
  - **Conditioner pump:** ___ HP
  - **Regenerator fan:** ___ HP
  - **Economizer pump:** ___ HP
APPENDIX E

SELECTED INFORMATION BY CANDIDATE COMPANIES

L-DCS GMBH
Air dehumidification systems

L-DCS Technology
System configuration for humid and tropical climate

For individual information please visit our web site www.L-DCS.com, try our System Configuration Tool, or contact us directly!

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Base process

A Liquid Desiccant Cooling System (L-DCS) is used for dehumidification and/or cooling of the fresh air intake of air conditioning systems. These systems are based on the absorption of water vapour by a hygroscopic fluid (sorbent), mostly highly concentrated aqueous solution of Lithium Chloride (H2O-LiCl). The sorbent absorbs water from the air and gets diluted. The absorbed water can be stripped from the sorbent by heating it up with low temperature heat at a supply temperature range in between about 55-75°C (temperature change about 15-20°C). Afterwards the sorbent can be used again for the absorption process. In doing so the sorbent is not used up or deteriorated in any way and can be reused for an unlimited time. Possible heat sources are i.e.:

- Low temperature heat from solar collectors, \( (T_1=75°C; T_2=65°C) \)
- Waste heat from CHP Technology (preferably coolant only) , \( (T_1=80°C; T_2=60°C) \)
- Waste heat from district heating systems (return flow) , \( (T_1=80°C; T_2=60°C) \)
- Waste heat from compressed air compressors, \( (T_1=65°C; T_2=45°C) \)

The innovative part within the process is the actual air dehumidification step. A subsequent cooling of the airflow is optional and can be done by any cooling technology, i.e. adiabatic or indirect evaporative cooling.

The overall system and its components, functional description

The system described below complies with the configuration recommended by L-DCS Technology for humid and tropical climates. It consists of the following subsystems:

1. **Ventilation**, consisting of:
   - **Fresh air supply system:**
     - Absorber, for dehumidification and Pre-cooling of the fresh air intake
     - Subsequent heat exchanger, to post-cool the dehumidified fresh air flow coming from the absorber
   - **Exhaust air system**
     - Indirect evaporative cooler, for energy recovery from the exhaust air.
   The units of the supply- and exhaust air subsystems are connected through a closed cycle system preferably containing pure water.

2. **Energy storage system**, consisting of two independent tanks the for storage of the liquid sorbent:
   - **Tank I**, for diluted sorbent
   - **Tank II**, for concentrated sorbent

3. **Heat intake system** consisting of:
   - **Regenerator**, for stripping the sorbent of water picked up during the dehumidification process
   - **Heat recovery system**, to preheat the regenerator air intake to increase the thermal efficiency of the process \( (COP_{th}) \).
Air dehumidification systems

The ventilation system consists of an absorber and a heat exchanger in the supply air flow and a Counter-flow-evaporative cooler in the exhaust air flow. Before entering the building the fresh air intake from the ambient is dehumidified in the absorber at almost constant air temperature and subsequently cooled in the heat exchanger. The heat, released during the dehumidification- and the cooling process, is picked up in the absorber by cooling water and transferred to the evaporative cooler in the exhaust air section. The evaporative cooler transfers the heat into the exhaust air flow, which is finally released into the ambient. The special feature of this evaporator is its counter-flow characteristic, the prerequisite of maximum recovery of cooling potential from the exhaust air flow.

The water vapour removed from the supply air flow is picked up by the sorbent. To achieve this concentrated sorbent is intermittently pumped from tank I of the energy storage to the absorber. A specially developed, proprietary Ultra-Low-Flow® sorbent distribution system in the absorber then only supplies a minimal amount of sorbent to the process. Subsequently the diluted sorbent leaving the process is pumped back to tank II of the energy storage. As long as tank I holds dehumidification energy in form of concentrated sorbent, fresh air can be continuously dehumidified. Only through the application of the L-DCS Ultra-Low-Flow® technology it is possible to store dehumidification- and cooling energy with a storage density of up to 280 kWh/m³ at extremely low cost. The capacity of this energy storage system is adapted to the customer needs simply by adjusting the tank size. The dehumidification energy is only released in case the sorbent gets in contact with humid air. Therefore this energy storage- and transport system is free of any losses as long as the tanks and pipelines are air tight. Consequently dehumidification energy can be stored over unlimited time, for hours, weeks, month or even seasonal. To be able to reuse the diluted sorbent after the dehumidification process the sorbent has to be stripped of the excess water. This desorption process takes place in the regenerator of the heat intake system. The diluted sorbent, supplied to the regenerator from storage tank I, is being internally heated up in the regenerator by hot water of about 55°-80°C and brought into contact with the regenerator air flow. During the contact of the two media the excess water is evaporated, whereupon the heating water gets cooled down by 15-20°C. The evaporated water is then released into the ambient by the regenerator air flow. The concentrated sorbent laves the regenerator for storage tank II and that way it is available again for the dehumidification process. The heat recovery in between the regenerator supply air- and exhaust air flow limits the thermal losses, hence increasing the thermal efficiency of the overall system (COPn).

The following graphics and diagrams document the performance of the L-DCS Technology under tropical conditions. The measurements were taken by the Centre for applied Energy Research (ZAE Bayern e.V.) on the 1.8.2014 on their own 6000m³/h L-DCS system at the Energy Efficiency Center in Würzburg, Germany. The tests and measurements were conducted within the framework of a research and development project supported by the German government (BMWI). The tropical conditions for the tests were artificially created by a climate simulator. The following pages show the measured values taken of the states of all mass flows in- and out of the absorber and the counterflow-evaporative cooler of the system described above during the performance test under tropical conditions on the 1.8.2014. Along the perpendicular line through the diagrams at t=13:53 [hh:mm the respective values for each mass flow are shown exemplarily.
Air dehumidification systems

Heat Intake
- Regenerator
- Waste Heat Source
  - 55/70°C 41/60°C

Energie Storage
- Tank I
- Tank II

Ventilation System
- Absorber
  - C_mf = 26.2%
  - Verd
  - C_mf = 39.7%
  - Konc

Fresh Air
- T = 34.2°C
- X = 24.5 g/kg
- T_m = 27.5°C
- h = 97.3°C
- r.F. = 68.6%

Exhaust Air
- T = 31.8°C
- X = 24.5 g/kg
- T_m = 28.0°C
- h = 96.0°C
- r.F. = 81.0%

Counter-Flow Evaporative Cooler
- F_k
- T = 31.2°C
- X = 9.6 g/kg
- T = 12.7°C
- h = 55.9°C
- r.F. = 32.2%

Return Air
- T = 26.4°C
- X = 10.4 g/kg
- T = 14.0°C
- h = 53.2°C
- r.F. = 46.4%

Air Flow Absorber Inlet
- T = 34.2°C
- X = 24.5 g/kg
- T = 27.5°C
- h = 97.3°C
- r.F. = 68.6%

Air Flow Absorber Outlet
- T = 31.4°C
- X = 9.7 g/kg
- T = 12.9°C
- h = 50.3°C
- r.F. = 32.2%

Water Flow Absorber
- T = 37.5°C
- X = 9.6 g/kg
- T = 27.6°C
- h = 40.0°C
- r.F. = 46.4%
Air dehumidification systems

The performance tests show, that the L-DCS Technology easily dehumidifies fresh air of tropical conditions ($T=34.5^\circ\text{C}; W=24.5\text{g/kg}$) to humidity levels well under 10g/kg without the use of conventional chiller power. At the same time the change of concentration in the sorbent was about 13.5%, relating to a storage density of 270 kWh of dehumidification energy per m³ of diluted sorbent.

The diagrams on this page show the L-DCS process values at 13:53 in the Mollier-Diagramm used in Europe (above) and the psychrometric chart used in the English speaking world (below).
Air dehumidification systems

The following diagrams show the calculated humidity and temperature of the air leaving the absorber at varying temperatures of the cooling water inlet.

The measurements taken from the real life process at 13:53 (see above) are reproduced with high precision. From this parameter variation of the temperature of the cooling water into the absorber the corresponding air temperature and humidity ratio at the absorber outlet can be predicted for different operation conditions respectively for different system configurations. For example using a standard cooling cooling tower rather than a counter-flow evaporative cooler in the exhaust air, as suggested above, results in an absorber outlet humidity ratio of approx. 12.5 [g/kg] (assuming the cooling water leaves the cooling tower at 33°C, about 4°C above wet bulb temperature).
Conclusion:
The measurements recorded during the performance tests impressively document the capability of the L-DCS technology under harsh tropical conditions. Nonetheless, they represent only one specific operating point of many possible. The dehumidification performance of the system mostly depend on the following variables:

- **Cooling water temperature** entering the absorber:  
  Temperature ↓  Dehumidification performance ↑

- **Sorbetent concentration** entering the absorber  
  Concentration ↑  Dehumidification performance ↑  
  Maximum limit: 42%, usual operation: 38-40% (mass concentration [kg salt /kg sorbent])

- **Sorbetent mass-flow** entering the absorber:  
  Mass-flow ↑  Dehumidification performance ↑  
  Alert! Consequence: Energy storage density ↓, required pumping power ↑

In addition it should be noted that the heat capacity flow within the run-around coil system, supplying the dehumidification process with cooling water from the energy recovery process in the exhaust air, needs to be balanced to yield the best system performance.

Addendum
L-DCS dehumidification system for tropical conditions with values measured during performance tests.
L-DCS Technology
Systemkonfiguration für feuchte und tropische Klima

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Systeme zu Luftentfeuchtung


- Niedertemperaturwärme aus thermischen Solarkollektoren, \((T_1=75°C; T_2=65°C)\)
- Abwärme aus BHKW Technik (Motorkühlwasser), \((T_1=80°C; T_2=60°C)\), und
- Abwärme aus Fernwärmenetzen (Rücklauf), \((T_1=80°C; T_2=60°C)\), und
- Abwärme von Pressluftkompressoren, \((T_1=65°C; T_2=45°C)\), zu nennen.

Der innovative Teil des Prozesses ist dabei die eigentliche Luftentfeuchtung. Eine anschließende Kühlung der Luft ist optional und kann durch konventionelle Technik, z.B. adiabatische oder indirekte Verdunstungskühlung erfolgen.

Das Gesamtsystem und seine Einzelkomponenten, Funktionsbeschreibung

Die im folgenden beschriebene Anlage entspricht der von L-DCS Technology vorgeschlagen Konfiguration zum Einsatz in tropischen Klimaten. Sie besteht aus den folgenden Teilsystemen:

1. **Lüftungssystem**, mit:
   - **Zuluftsystem**: Absorber zur Entfeuchtung und Vorkühlung der Zuluft
   - Nachgeschalteter Wärmetauscher zur Nachkühlung der Luft hinter dem Absorberaustritt
   - **Abluftsystem**: Indirekter Verdunstungskühler zur Energierückgewinnung aus der Abluft.

Die Apparate der Zuluft- und Abluftsysteme sind durch ein Wasser führendes Kreislaufverbundsystem gekoppelt.

2. **Energiespeichersystem**, bestehend aus zwei unabhängigen Tanks zur Speicherung des flüssigen Sorbens:
   - **Tank I**, für verdünntes Sorbens
   - **Tank II**, konzentriertes Sorbens

3. **Wärmeabnahmesystem** bestehend aus:
   - **Wärmerrückgewinnungssystem** zum Vorwärmen der Regenerationsluft zur Erhöhung der thermischen Leistungszahl \((\text{COP}_{\text{th}})\).

L-DCS System: Konfiguriert als Frischluftentfeuchtung für hohe Luftfeuchtigkeit mit Energierückgewinnung aus der Abluft

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**Grundprozess**

**L-DCS System: Konfiguriert als Frischluftentfeuchtung für hohe Luftfeuchtigkeit mit Energierückgewinnung aus der Abluft**

<table>
<thead>
<tr>
<th>Zustand</th>
<th>Zuluft</th>
<th>Abluft</th>
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<td>C_{\text{verd}}</td>
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<td>38.5%</td>
</tr>
<tr>
<td>C_{\text{konz}}</td>
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<td>26.2%</td>
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<tr>
<td>T</td>
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<td>10.4 g/kg</td>
</tr>
<tr>
<td>h</td>
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<td>53.2°C</td>
</tr>
<tr>
<td>r.F.</td>
<td>46.4%</td>
<td>46.4%</td>
</tr>
</tbody>
</table>

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Systeme zu Luftentfeuchtung

Dieser Test zeigt, dass die L-DCS Technologie in der Lage ist ohne den Einsatz von elektrisch angetriebener Zusatzkühlung Frischluft unter tropischen Bedingungen (T=34.5°C; X=24.5g/kg) auf Werte deutlich unter 10g/kg zu entfeuchtet. Gleichzeitig wurde eine Konzentrationsänderung im Sorbens von 13.5%, und damit eine Speicherdichte von 270 kWh Entfeuchtungsleistung pro m³ verdünntes Sorbens erreicht.

Die Diagramme auf dieser Seite zeigen den L-DCS Prozess mit den Werten von 13:53 im HX-Diagramm nach Molliere (oben) und dem im angelsächsischen Sprachraum üblichen Diagramm nach Carrier (unten).
Die folgenden Diagramme zeigen die berechneten Luftzustände am Austritt aus dem Absorber bei variierender Kühlwassertemperatur am Absorbereintritt.

Systeme zu Luftentfeuchtung

Fazit:

Die in den Leistungstests gemessenen Werte dokumentieren die Leistungsfähigkeit der Anlagen unter tropischen Bedingungen, sie stellen aber nur einen möglichen Betriebszustand dar. Die erreichbaren Entfeuchtungsleistungen werden im Wesentlichen von folgenden Faktoren beeinflusst:

- Kühlwassertemperatur am Eintritt in den Absorber: Entfeuchtungsleistung ↑
- Anfangskonzentration des Sorben am Eintritt in den Ansorber Konzentration ↑ Entfeuchtungsleistung ↑
- Maximale Grenze: 42%, Normalbetrieb 38-40%
- Sorbensmassenstrom am Eintritt in den Ansorber: Sorbensmassenstrom ↑ Entfeuchtungsleistung ↑

Achtung! Konsequenz: Speicherdichte ↓, Pumpenleistung ↑

Des weiteren sollten zur Optimierung der Anlagenleistung der Wärmekapazitätenstrom im Wasserkreislauf zwischen Rückkühlung und Entfeuchtung korrekt angepasst werden (Kreislaufverbundsystem).

Anhang

L-DCS Entfeuchtungssystem für feuchtes Klima mit gemessenen Leistungsdaten.
1. INTRODUCTION

Liquid desiccant cooling systems (LDCS) are a first class option for the use of waste heat in air conditioning processes. LDCS dehumidify (outside-) air by means of an aqueous salt solution, such as a Lithiumchloride solution (LiCl-H₂O), and cool it in subsequent direct or indirect evaporative coolers. The salt solution is diluted while drying the air and has to be regenerated at temperatures of 60 °C to 80 °C. If concentrated and diluted solutions are stored separately, energy for dehumidification can be stored thermochemically for some time or can be transported over some distance. A cooled dehumidifier and a low flow technique, however, are required to achieve a high energy storage density, making energy storage economically attractive, Hublitz (2005).

The Bavarian Center for Applied Energy Research, ZAE Bayern, has developed a cooled low flow desiccant dehumidification technology within the recent years and documented the high energy storage potential, Kessling (1998). The L-DCS Technology GmbH, Ismaning, Germany, has set out to make this technology commercially available. A development and demonstration project started in 2001 aimed at the following objectives:

- Development of system components such as absorber, regenerator, indirect evaporative cooler and storage equipment
- Test of system components
- Demonstration of the present state of liquid desiccant storage technology
- Cooling of a Munich jazz club

Project partners are ZAE Bayern as project manager, the Muenchner Gesellschaft für Stadtneuerung as building owner, HOWATHERM Kimatechnik GmbH, Bruecken, Germany, as supplier of the evaporative coolers and L-DCS Technology GmbH as supplier of the absorber-regenerator unit. The project was funded by the German Ministry of Economics and Labor.

The system components have been designed and manufactured. A liquid desiccant cooling system has been installed to cool the Munich jazz club. It has been designed as a test plant and extensive measurement instrumentation has been installed. The performance of the components has been tested and the results are reported here.

2. SYSTEM DESCRIPTION

Liquid desiccant cooling systems are advantageous when providing cool and dry air. The supply air ducts of the jazz club, however, are too small to allow an air flow sufficiently large to remove the cooling load. Therefore the liquid desiccant cooling system installed, shown as a sketch in Figure 1, provides cooling water to be used in fan coil units cooling the jazz club. This is not a suggestion how to produce cold water using liquid desiccants but a design for a plant where the essential components can be tested and run under non laboratory conditions.
Return air from the building is dehumidified by a LiCl-H$_2$O solution in a cooled absorber-regenerator unit. An indirect evaporative cooler (IEC1) cools the dry air by water evaporation, producing cold water for the fan coil units of the jazz club in a wetted heat exchanger. The dry air is pre-cooled by a cold recovery system consisting of two heat exchangers (HX1 and HX2) connected by a water loop. A second indirect evaporative cooler (IEC2) is used to cool the absorber, before the warm and humid air leaves the system as exhaust air. The desiccant solution is diluted in the dehumidification process and is stored in a separate tank. When the jazz club is closed, district heating regenerates the desiccant solution in the absorber-regenerator unit.

3. COMPONENT DEVELOPMENT

The absorber-regenerator-unit, the desiccant storage and handling equipment and the indirect evaporative coolers are non standard components.

The absorber-regenerator-unit has been newly developed by L-DCS Technology GmbH. The essential parts of the unit are heat and mass exchanging plates and special solution distributors. The plates are internally cooled or heated by a water loop. The solution distributors must be able to distribute a solution flow as low as 0.5 l/h per meter of distributor length, keeping the plates properly wetted in order to provide a high energy storage potential. Due to economic reasons, L-DCS Technology chose standard polypropylene double plates as exchanger surface, which are covered by a wick-like polypropylene fleece. Polypropylene withstands the corrosion of the desiccant solution and is sufficiently temperature resistant. Sixteen plates at a time have been welded together to form a cassette providing fifteen air channels of about 4 mm width and 1m length. Polypropylene water circulation elements and solution circulation elements have been welded tightly to the cassettes to form the cooling water circuit and to connect the solution distributors. Eight cassettes are hanged in a frame which enables dilatation of the plates at higher temperatures. The cassettes are sealed against each other and against the frame. The whole package is hanging in a
polypropylene enclosure consisting of a welded trough and sealed walls up to a height of about 1.5 m. Package and enclosure are placed in a conventional housing of an air handling unit, see Figures 2 and 3.

The air flows from the top of the unit around the package into the enclosure and the trough. There it turns upward and flows through the channels formed by the plates in close contact with the cooled or heated desiccant solution. On top of the package is a hood, sealed to the frame, gathering the air and leading it to the downstream air duct. The desiccant solution is distributed by the distributors on top of the plates, trickles down the plates, drops into the trough and is gathered in a sump, from where it is pumped back to one of the tanks. There are no sprays, the desiccant solution is always in contact with a solid surface and the air velocity in the channels is low. By this means carryover of the desiccant is reliably avoided.

**Figure 2**: Housing of the absorber-regenerator unit

**Figure 3**: Absorber-regenerator-unit by L-DCS Technology for an air flow of 4000 m³/h

The two indirect evaporative coolers are based on standard products of HOWATHERM GmbH. The coolers consist of finned coils. The fins have been covered by the manufacturer with a hydrophilic coating. Water is sprayed parallel to the air flow onto the fins of the heat exchanger. The arrangement and operation of the standard sprays have been modified. A sump has been added to allow spray water recirculation and a higher spray water flow compared to the standard product. The cooler IEC1 in Figure 1 is built in three stages, the cooler IEC2 in two. Each stage consists of two injection tubes with three nozzles and a heat exchanger. In normal operation, the cooling water flows in series through all three stages. Each stage can be bypassed to measure the performance of a single stage and thereby achieve better measurement accuracy.

4. **SYSTEM DESIGN AND INSTALLATION**

In the design phase, the sizes of the components have been determined and the capacity of the system has been calculated for a design point using preliminary data provided by the manufacturers or known from former tests. The design point data are given in Figure 1. A total cooling capacity of 16 kW has been aimed at, using eight fan coil units, six installed in the jazz club and two in the kitchen. A regeneration capacity of 32 kW is intended, using hot water of about 75 °C from the district heating net. The desiccant storage capacity of 1.4 m³ of diluted solution should enable 9 hours of design point operation. The designed specific energy storage capacity related to the volume of the diluted desiccant is 120 kWh/m³ for cooling and 150 kWh/m³ for dehumidification. The theoretical maximum for the dehumidification storage capacity is about 200 kWh/m³ under the given temperature conditions.
The components have been delivered by the manufacturers and been assembled and installed on site. Figure 4 shows the fan coil units in the jazz club, Figure 5 the indirect evaporative coolers, packed in an air handling unit. Figure 6 and 7 show the desiccant handling equipment consisting of pumps, valves and filters, installed over a trough above the desiccant tanks. The system has been designed as a test facility. The cassettes of the absorber as well as the coils and sprays of the evaporative coolers can be removed and replaced by a second generation of components. Additional hydraulic installations allow direct heating of the indirect evaporative coolers for tests independent of activities in the jazz club. The measuring and data acquisition equipment enables energy and mass balances with an accuracy of about 5%. All hydraulics, controls and data acquisition have been set into operation and work well.

5. COMPONENT TESTS
The indirect evaporative cooler IEC1 combined with the recovery coils HX1 and HX2 and the fan coil units in the jazz club has been tested under experimental conditions. Only six fan coil units have been active, the temperature in the jazz club was only 21 °C and the air humidity ratio at the inlet of IEC1 was only 3 g/kg. The results are given in Table 1:

Table 1: Test of indirect evaporative cooler IEC 1,
M = measured, C = calculated

<table>
<thead>
<tr>
<th>Component / Parameter</th>
<th>M/C</th>
<th>Unit</th>
<th>Test 1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Indirect Evaporative Cooler IEC1</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air Volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>3000</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>C</td>
<td>°C</td>
<td>20.8</td>
</tr>
<tr>
<td>Inlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>3.0</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>16.2</td>
</tr>
<tr>
<td>Outlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>9.2</td>
</tr>
<tr>
<td>Heating Water</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>2.00</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>16.3</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>12.3</td>
</tr>
<tr>
<td>Heating load</td>
<td>C</td>
<td>kW</td>
<td>9.3</td>
</tr>
<tr>
<td>Cold Recovery HX1 HX2</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air Inlet temperature HX1</td>
<td>M</td>
<td>°C</td>
<td>23.4</td>
</tr>
<tr>
<td>Outlet temperature HX2</td>
<td>C</td>
<td>°C</td>
<td>19.7</td>
</tr>
<tr>
<td>Water Volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>0.838</td>
</tr>
<tr>
<td>Inlet temperature HX1</td>
<td>M</td>
<td>°C</td>
<td>18.3</td>
</tr>
<tr>
<td>Outlet temperature HX1</td>
<td>M</td>
<td>°C</td>
<td>21.9</td>
</tr>
<tr>
<td>Cold recovery load</td>
<td>kW</td>
<td></td>
<td>3.5</td>
</tr>
<tr>
<td>Fan Coil Unit Jazz Club</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Inlet air temperature</td>
<td>M</td>
<td>°C</td>
<td>21.0</td>
</tr>
<tr>
<td>Outlet air temperature</td>
<td>M</td>
<td>°C</td>
<td>16.3</td>
</tr>
</tbody>
</table>

Table 1 in column “Test 1” and shown in Figure 8. The line from point A2 to A3 has been calculated. The points P1 and P2 indicate the wetbulb temperatures before and after HX1 which determine the potential temperature to which the water can be cooled theoretically. The deviation of the energy balance of “Test1” is below 1 % and therefore well below the expected mean error of 6.2 %. These experimental data have been used to identify model parameters for the heat exchangers HX1 and HX2 and for IEC1 and to calculate the performance of the cooler under design point conditions, see Figure 1, using all eight fan coils. The result is plotted in Figure 9. The cooling load removed from the jazz club under design point conditions would be 12.8 kW. This is 80 % of what has been expected. The numerical analysis shows that the fan coil units are the bottleneck, the indirect evaporative cooler works as expected, the wetted part of the exchanger surface, however, is below 20 %.

The absorber-regenerator-unit has been tested in absorption and regeneration mode. In Table 2, the conditions and results of the first two absorption tests are given. Different solution mass flows have been tested. In both tests, only seven of the eight cassettes have been active. All mass flows, air, coolant and desiccant, of one cassette have been
blocked. Figure 10 shows the process of Test 1 in a psychometric chart. The deviation of the energy balance is 5.4 % (Test 1) and 5.7 % (Test 2), the expected mean error is 7.5 % (Test 1) and 8.6 % (Test 2). The air was dehumidified from 12 to 8.7 g/kg and from 12.5 to 9.7 g/kg corresponding to a dehumidification energy storage capacity of

### Table 2: Absorber tests, 7 of 8 cassettes active.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>M/C</th>
<th>Unit</th>
<th>Test 1</th>
<th>Test 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>3188</td>
<td>3212</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>27.5</td>
<td>27.6</td>
</tr>
<tr>
<td>Inlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>12.0</td>
<td>12.5</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>26.9</td>
<td>27.1</td>
</tr>
<tr>
<td>Outlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>8.7</td>
<td>9.6</td>
</tr>
<tr>
<td>Dehumidification load</td>
<td>C</td>
<td>kW</td>
<td>8.4</td>
<td>7.1</td>
</tr>
<tr>
<td><strong>Cooling Water</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>2.6</td>
<td>2.5</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>23.4</td>
<td>23.5</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>26.7</td>
<td>26.4</td>
</tr>
<tr>
<td>Cooling load</td>
<td>C</td>
<td>kW</td>
<td>9.6</td>
<td>8.6</td>
</tr>
<tr>
<td><strong>Liquid Desiccant</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass ratio air/desiccant</td>
<td>C</td>
<td>kg/kg</td>
<td>25</td>
<td>41</td>
</tr>
<tr>
<td>Inlet volume flow</td>
<td>M</td>
<td>l/h</td>
<td>110</td>
<td>67</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>29.6</td>
<td>30.0</td>
</tr>
<tr>
<td>Inlet concentration</td>
<td>M</td>
<td>%</td>
<td>39.7</td>
<td>39.7</td>
</tr>
<tr>
<td>Outlet volume flow</td>
<td>C</td>
<td>l/h</td>
<td>121</td>
<td>77</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>29.6</td>
<td>30.2</td>
</tr>
<tr>
<td>Outlet concentration</td>
<td>M</td>
<td>%</td>
<td>36.6</td>
<td>35.7</td>
</tr>
<tr>
<td>Dehumid. storage capacity</td>
<td>C</td>
<td>kWh/m³</td>
<td>68</td>
<td>84</td>
</tr>
</tbody>
</table>

### Figure 10: Absorber Test 1, mass ratio air / desiccant = 25

### Table 3: Regenerator Test 1, 7 of 8 cassettes active. M = measured, C = calculated

<table>
<thead>
<tr>
<th>Parameter</th>
<th>M/C</th>
<th>Unit</th>
<th>Test 1</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Total volume flow</td>
<td>M</td>
<td>m³/h</td>
<td>2038</td>
</tr>
<tr>
<td>Volume flow regenerator</td>
<td>C</td>
<td>m³/h</td>
<td>794</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>23.9</td>
</tr>
<tr>
<td>Inlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>3.4</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>34.3</td>
</tr>
<tr>
<td>Outlet humidity ratio</td>
<td>M</td>
<td>g/kg</td>
<td>11.7</td>
</tr>
<tr>
<td>Humidification load</td>
<td>C</td>
<td>kW</td>
<td>12.7</td>
</tr>
<tr>
<td>Sensible heating load</td>
<td>C</td>
<td>kW</td>
<td>6.3</td>
</tr>
<tr>
<td><strong>Heating Water</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Volume flow</td>
<td>M</td>
<td>M³/h</td>
<td>2.2</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>M</td>
<td>°C</td>
<td>69.1</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>M</td>
<td>°C</td>
<td>60.5</td>
</tr>
<tr>
<td>Heating load</td>
<td>C</td>
<td>kW</td>
<td>21.4</td>
</tr>
<tr>
<td><strong>Liquid Desiccant</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mass ratio air/desiccant</td>
<td>C</td>
<td>kg/kg</td>
<td>1.50</td>
</tr>
<tr>
<td>Inlet volume flow</td>
<td>M</td>
<td>l/h</td>
<td>463</td>
</tr>
<tr>
<td>Outlet volume flow</td>
<td>C</td>
<td>l/h</td>
<td>445</td>
</tr>
<tr>
<td>Outlet concentration</td>
<td>M</td>
<td>%</td>
<td>41.1</td>
</tr>
</tbody>
</table>
68 kWh/m³ and 84 kWh/m³ related to the volume of the diluted solution. The cooling water inlet temperature (23.5 °C) was above design conditions, the desiccant inlet concentration (39.7 %) was below. So the design point dehumidification and storage capacity could not be reached in the test. The absorber parameters have not yet been identified exactly. Preliminary values for the wetted fraction of the surface are 44 % in Test 1 and 33 % in Test 2. Further tests are necessary, but nevertheless the absorber performance, dehumidification and energy storage capacity, is significantly below expectations.

In Table 3 the conditions and results of the first regeneration test are given. The solution mass flow has been set quite high to achieve an acceptable heat input to the regenerator, resulting in a very low concentration difference. Regeneration temperature and heating water mass flow were slightly below the design point values. The parameter identification is not yet finished, but the result is similar to that of the absorber test. Design point performance will not be reached under design point conditions.

4. CONCLUSION

The cooling system is in operation. The performance of the controls and the measurement accuracy is good. The plant has proved to be an excellent facility for performance tests of absorber and indirect evaporative cooler. The jazz club can be cooled. The performance of the indirect evaporative cooler and the fan coil units is close to what has been expected. In the first tests, the absorber-regenerator-unit did not yet achieve the intended performance. Probable reasons are insufficient wetting of the surface and insufficient cooling of the surface. This may be caused by remaining air in the cooling water circuit and in the desiccant distributors. A similar absorber-unit used in a parallel project showed significant better performance. Therefore the authors are confident to lift the absorber performance, dehumidification and energy storage, to an acceptable level.

The wetting of the exchange surfaces (20 - 40 %) of the tested, commercially available devices is far below what has been reached in laboratory (> 90 %). An improvement in surface wetting would be a major step towards favorable economics of this technology.

ACKNOWLEDGMENTS

We wish to express our deep appreciation to the German Ministry of Economics and Labor for funding this project. We are especially indebted to Dr. V. Lottner (PTJ, Forschungszentrum Juelich) and Dr. Kaup (Howatherm GmbH).

REFERENCES


APPENDIX F

SELECTED INFORMATION BY CANDIDATE COMPANIES

MENERGA APPARATEBAU, GMBH
System overview

Indoor swimming pool air conditioning | Comfort air conditioning | Process air conditioning
Menerga:
Minimal Energy Application

We supply air conditioning systems individually designed for your requirements. Our philosophy, "Creating a good indoor climate – through Minimal ENERGY Application", is something we have succeeded in every single day, since the company was founded over 30 years ago. We are proud to be part of the international successful Systemair group since 2013.

Our systems are first-class, intelligent works of engineering and handicraft. They remain reliable in operation for many, many years, significantly reducing operating costs. How is this possible? In the basic design stages we already integrate all the components for air conditioning, such as the ventilation, heating and refrigeration systems and equip everything with an intelligent control and regulation system. Every system is fully tested before delivery within the framework of a test run. The compact units are always delivered "ready for connection". At the building site, they are connected up and made operational in just a few work stages.

With over 40,000 systems installed worldwide, we cover almost every application. We not only sell the units, but also offer you our many years of experience. When looking for the best solution, we analyse the specific conditions at the location together with you. For the optimal solution we ask a lot of questions. Might it also be possible to use an alternative source of energy in order to reduce the operating costs even further? In this manner, we and our partners have jointly implemented countless projects which have received many awards for energy efficiency. We are proud of this. But what we really like about this is the know-how from jointly developed solutions, which allows operators and investors to save hard earned money – day after day, month after month and year after year. The investment costs are amortised within a short period.

We will be happy to produce reference lists for the building types in which you are interested in. And in the event that you surprise us with a totally new project: We are convinced to find the right solution for you. With our eyes sharpened by countless special projects, e.g. the „ALMA“ telescope facility in the Atacama desert or the “Princess Elisabeth Station” at the South Pole, we will be happy to accept the challenge.

Convincing arguments for Menerga

- Intelligent Technology
  - lasting low operating costs
- Use of regenerative energy sources
- Very compact design
- Integrated control and regulation systems
- Factory test run as standard
- Ready-for-connection delivery
- Excellent maintenance concepts
Experts at your service
Technical Service

Experts at your service, anytime, anywhere. With a comprehensive range of services and an extensive service network throughout Europe, the Menerga Technical Service guarantees the most economical and advanced services over the entire life cycle of your system, from the day of commissioning onwards.

More than 120 service technicians at various service centres and 40 service engineers at the Menerga locations provide a professional all-inclusive service with the objective of achieving high availability of the systems and maximum efficiency. The range of services offered by the Menerga Technical Service covers everything from the test run at the factory and on-site commissioning through periodic servicing, repairs, remote maintenance and remote diagnosis by means of direct dial-up options, to the refurbishment and optimisation of the systems. And this all not only for Menerga units!

We supply you with the right service, customer-specific and application-specific. In the event of an emergency, you can reach us 24 hours a day on the following telephone number:
+49 208 9981-199

Eurovent
Most of our ventilation units are as standard version Eurovent certified. This means all series that are equipped with our Menerga Air housing “MB 50” with 50 mm panels and filter classes up to F7/F9.

Passivhaus Institut
The complete Resolair 64 series and Adconair up to 76 16 01 are officially certified components of the Passive House Institute. They are ideally suitable for passive houses and all other low energy buildings.

ATEX
The ATEX directive currently includes two directives in the field of explosion protection, the ATEX Directive 94/9/EC and the ATEX Workplace Directive 1999/92/EC. On request we produce your unit according the ATEX regulations for explosion-hazardous areas.

Manufacturers Association „RLT Hersteller Verband“
Menerga is a member of the german Manufacturers Association for AHU „Herstellerverband Raumlufttechnische Geräte e.V.”. Aim of this Association is to develop air handling units at the highest technical level as well as standardization work and technical recommendations.

Ecodesign directive
Most of our units fulfill the requirements of the Ecodesign directive from January 2018 already. With Menerga you can now already plan all the projects for the year 2018.

Of course we also have all common other certificates such as TÜV type examinations, hygiene certificates, ISO 9001 and more. Please contact us - we are happy to send you an overview or copies of the certificates you might require.
Menerga core competencies

Our areas of application

INDOOR SWIMMING POOL AIR CONDITIONING

Private swimming pools, public swimming pool halls, adventure pools, sports pools, saline baths, hotel pools, school pools, therapeutic pools and many more.
Last not least: heat recovery from waste water.
The air conditioning of swimming pool halls is one of the most challenging areas for air conditioning. Here we started 35 years ago, this is where we grew up and where we are now market leaders and innovation pioneers. Our special competency lies in the high heat recovery efficiency lowering operating costs.

COMFORT AIR CONDITIONING

Low-energy buildings, offices, museums, sports facilities, schools, clinics, hotels, banks, historical buildings and many more.
With comfort air conditioning, the focus is on people. Our technology is based on the respective requirements of a project, but we also always look for the most efficient method with the lowest consumption of energy. For example, we cool with water in order to save electrical energy or make use of sorption-based air conditioning, with which you dehumidify with heat, e.g. from solar thermal energy or process waste heat. It is even possible to store excess solar heat for an indefinite period without any losses for the purposes of dehumidification.

PROCESS AIR CONDITIONING AND CHILLED WATER

Air conditioning of data centres, industrial drying, process cooling, air conditioning for warehouses, cold water generation and much more.
Last but not least: heat recovery from waste water.
The process air conditioning system must ensure that defined air conditions prevail in a defined situation. Menerga systems guarantee reliable drying, cooling or heating. In the field of chilled water, our systems reliably provide the desired water conditions. Saving energy through the use of intelligent technology is our top priority in this sector as well.

SPECIAL SOLUTIONS

Research projects, special applications
Challenges and unusual projects are the milestones of Menergas company history. Since the foundation of our company, we have designed individual solutions for many of our customers. We enjoy taking on challenging projects, knowing that these are the projects that bring valuable experience and which also improve the quality of our “standard” systems.
Insight: Technology in detail

1. Quality: Menerga systems are developed in Germany and focus on highest quality.

2. Profiles and frames: the equipment design is based on a long-lasting, robust aluminum steel frame. Housing designs are available up to the highest thermal bridge class TB1.

3. Control and regulation: our systems are ready to connect upon delivery. The intelligent control & regulation equipment guarantees that the system always performs optimally.

4. Filters: all HVAC systems are equipped with an optimised filtration system to protect both people and equipment.

5. Heating or cooling coils: for covering the transmission heating or cooling requirement.

6. Fans: energy-efficient EC fan motor units.

7. Indirect adiabatic evaporative cooling: for cooling purposes, we use natural processes wherever possible, e.g. cooling with water.

8. Heat exchangers: we use polypropylene instead of aluminium with a big increase in efficiency and thus minimise both the weight of the system and CO₂ emissions during production.

9. Droplet eliminator: efficient mist collectors reliably eliminate aerosols from the air and prevent moisture from being carried into the air ducts.

10. Air damper systems: for precise control of the air flow.

11. Air distribution: intelligent bypass designs for efficient operation all year round.

12. Compressor refrigeration system / heat pump: meets the requirements of DIN EN 378 and is type-tested and certified in accordance with the pressure equipment directive. Individual acceptance is no longer necessary.
The ThermoCond 19, 23 and 29 series are multifunctional compact systems for air conditioning private swimming pool halls. The combination of first-class components with precise control and regulation systems guarantees economical operation at all times, while ensuring the highest degree of comfort air conditioning. ThermoCond systems dehumidify, heat and ventilate the swimming pool hall and simultaneously create a good climate while protecting the fabric of the building. Additional heat sources such as radiators or panel heating systems are generally not required.

**At a glance:**
- Dehumidifies, ventilates and heats
- Corrosion-free heat exchanger made from polypropylene
- Energy-saving unit design
- Compact design for minimal space requirements
- Integrated control and regulation system, compatible with all conventional building management systems

### ThermoCond 19
with cross-counterflow heat exchanger

<table>
<thead>
<tr>
<th>Unit Type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
<th>Dehumidification capacity (kg/h)</th>
</tr>
</thead>
<tbody>
<tr>
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<td>1,530</td>
<td>570</td>
<td>1,590</td>
<td>410</td>
<td>1,100</td>
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<td>730</td>
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<td>440</td>
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<td>730</td>
<td>1,910</td>
<td>540</td>
<td>2,000</td>
<td>12.9</td>
</tr>
<tr>
<td>19 25 01</td>
<td>1,690</td>
<td>890</td>
<td>1,910</td>
<td>610</td>
<td>2,500</td>
<td>16.2</td>
</tr>
<tr>
<td>19 35 01</td>
<td>1,690</td>
<td>1,210</td>
<td>1,910</td>
<td>720</td>
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</table>

### ThermoCond 23
with cross-counterflow-cross heat exchanger

<table>
<thead>
<tr>
<th>Unit Type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
<th>Dehumidification capacity (kg/h)</th>
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</thead>
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<tr>
<td>23 12 01</td>
<td>2,580</td>
<td>570</td>
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<td>450</td>
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<td>730</td>
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<td>600</td>
<td>2,500</td>
<td>16.2</td>
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<tr>
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<td>3,700</td>
<td>730</td>
<td>1,850</td>
<td>870</td>
<td>3,200</td>
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<td>1,850</td>
<td>1,100</td>
<td>5,000</td>
<td>30.2</td>
</tr>
</tbody>
</table>

### ThermoCond 29
with cross-counterflow heat exchanger and integrated heat pump

<table>
<thead>
<tr>
<th>Unit Type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
<th>Dehumidification capacity (kg/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>29 11 01</td>
<td>1,530</td>
<td>570</td>
<td>1,590</td>
<td>460</td>
<td>1,100</td>
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<td>730</td>
<td>1,590</td>
<td>500</td>
<td>1,500</td>
<td>9.7</td>
</tr>
<tr>
<td>29 20 01</td>
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<td>730</td>
<td>1,910</td>
<td>600</td>
<td>2,000</td>
<td>12.9</td>
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<td>890</td>
<td>1,910</td>
<td>680</td>
<td>2,500</td>
<td>16.2</td>
</tr>
<tr>
<td>29 35 01</td>
<td>1,690</td>
<td>1,210</td>
<td>1,910</td>
<td>830</td>
<td>3,500</td>
<td>22.6</td>
</tr>
</tbody>
</table>

1. Door fitting assembly increases unit width by 25 mm each operating side
2. incl. 100 mm unit feet and 120 mm duct connection (Series 19/29) incl. 100 mm unit feet and 60 mm cable duct (Series 23)
3. Dehumidification capacity according to VDI 2089
4. Switching cabinet arranged on top of unit, please add switching cabinet height (480 mm)
5. Different weight with optional pool water condenser

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.
Thanks to intelligent technology, the units in the ThermoCond 38 and 39 series achieve continuous comfort coupled with consistent energy efficiency. ThermoCond 38 is equipped with a full counterflow plate heat exchanger that achieves a heat recovery efficiency of over 95%*. The system can be equipped with an additional clean water heater that increases the efficiency of the entire system even further.

At a glance:
- Dehumidifies, ventilates and heats
- Corrosion-free heat exchanger made from polypropylene
- Two-stage supply air filtration
- Heat recovery efficiency up to > 95%*

ThermoCond 38
with full counterflow plate heat exchanger and load-independent volume flow rate adjustment

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 1 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
<th>Max. flow rate 1 (m³/h)</th>
<th>Dehumidification capacity 1 (kg/h)</th>
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<td>790</td>
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</tr>
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<td>790</td>
<td>2,340</td>
<td>1,600</td>
<td>4,000</td>
<td>6,000</td>
<td>25.8</td>
</tr>
<tr>
<td>38 10 01</td>
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<td>1,110</td>
<td>2,340</td>
<td>1,900</td>
<td>6,000</td>
<td>9,500</td>
<td>38.8</td>
</tr>
<tr>
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<td>1,430</td>
<td>2,340</td>
<td>2,350</td>
<td>7,900</td>
<td>10,500</td>
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<td>38 16 01</td>
<td>5,770</td>
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</tr>
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<td>38 19 01</td>
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<td>2,070</td>
<td>2,340</td>
<td>3,000</td>
<td>11,800</td>
<td>18,000</td>
<td>76.2</td>
</tr>
<tr>
<td>38 25 01</td>
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<td>2,070</td>
<td>2,980</td>
<td>3,900</td>
<td>15,800</td>
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<td>101.2</td>
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<tr>
<td>38 29 01</td>
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<td>2,390</td>
<td>2,980</td>
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<tr>
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<td>3,030</td>
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<td>5,700</td>
<td>23,600</td>
<td>35,900</td>
<td>152.5</td>
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</tbody>
</table>

ThermoCond 39
with asymmetric high-capacity heat exchanger, integrated output-regulated heat pump and efficient volume flow control and integrated clean water heater

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 1 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
<th>Max. flow rate (m³/h)</th>
<th>Dehumidification capacity (kg/h)</th>
</tr>
</thead>
<tbody>
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<td>1,700</td>
<td>1,050</td>
<td>2,600</td>
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</tr>
<tr>
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<td>4,100</td>
<td>1,110</td>
<td>1,700</td>
<td>1,300</td>
<td>3,900</td>
<td>5,300</td>
<td>25.2</td>
</tr>
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<td>790</td>
<td>2,340</td>
<td>1,350</td>
<td>4,000</td>
<td>6,300</td>
<td>25.8</td>
</tr>
<tr>
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<td>4,740</td>
<td>1,110</td>
<td>2,340</td>
<td>1,650</td>
<td>6,000</td>
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<tr>
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<td>4,900</td>
<td>1,430</td>
<td>2,340</td>
<td>2,050</td>
<td>7,900</td>
<td>12,600</td>
<td>51.0</td>
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<tr>
<td>39 16 01</td>
<td>4,900</td>
<td>1,750</td>
<td>2,340</td>
<td>2,250</td>
<td>9,800</td>
<td>15,800</td>
<td>63.3</td>
</tr>
<tr>
<td>39 19 01</td>
<td>4,900</td>
<td>2,070</td>
<td>2,340</td>
<td>2,500</td>
<td>11,800</td>
<td>19,000</td>
<td>76.2</td>
</tr>
<tr>
<td>39 25 01</td>
<td>5,700</td>
<td>2,070</td>
<td>2,980</td>
<td>3,250</td>
<td>15,800</td>
<td>25,000</td>
<td>102.1</td>
</tr>
<tr>
<td>39 32 01</td>
<td>6,180</td>
<td>2,070</td>
<td>3,620</td>
<td>3,900</td>
<td>19,900</td>
<td>30,000</td>
<td>128.6</td>
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<tr>
<td>39 36 01</td>
<td>6,180</td>
<td>2,390</td>
<td>3,620</td>
<td>4,450</td>
<td>23,100</td>
<td>33,500</td>
<td>149.2</td>
</tr>
</tbody>
</table>

1 May change depending on chosen option, e.g. recuperator in short version (- 960 mm)
2 Door fitting assembly increases unit width by 65 mm each operating side
3 incl. 120 mm base frame, incl. 60 mm cable duct
4 May require alteration of the technical equipment
5 Dehumidification capacity according to VDI 2089 at opt. flow rate

At series 39 different weight with optional pool water condenser

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.

* 2 at RA = 30°C / 54% r.h., OA = -12°C / 90% r.h., 1/3 OA rate

ThermoCond 38/39
A GOOD CLIMATE FOR PUBLIC INDOOR SWIMMING POOLS
Trisolair
THREE-STAGE RECUPERATIVE HEAT RECOVERY

Units in the Trisolair 52 and 59 series achieve the highest heat recovery efficiency at low to medium air volume flow rates and can be used in a wide range of comfort air conditioning applications. Thanks to their compact design, the systems are ideally suited for refurbishment projects. A compressor refrigeration system integrated into the 59 series increases the cooling capacity of the overall system at higher temperatures and additionally allows the dehumidification of outside air.

Trisolair 52
with cross-counterflow-cross heat exchanger

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. fl. flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>52 12 01</td>
<td>2,580</td>
<td>570</td>
<td>1,210*</td>
<td>420</td>
<td>1,200</td>
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<tr>
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<td>730</td>
<td>1,530</td>
<td>560</td>
<td>1,800</td>
</tr>
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<td>3,700</td>
<td>1,050</td>
<td>1,850</td>
<td>1,050</td>
<td>3,600</td>
</tr>
</tbody>
</table>

Trisolair 59
with cross-counterflow-cross heat exchanger and Integrated compressor refrigeration system

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
<th>Opt. fl. flow rate (m³/h)</th>
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</thead>
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<tr>
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<tr>
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<td>1,050</td>
<td>1,850</td>
<td>1,280</td>
<td>3,600</td>
</tr>
</tbody>
</table>

1 Door fitting assembly increases unit width by 25 mm each operating side
2 Height incl. 100 mm unit feet and 60 mm cable duct
3 May require alteration of the technical equipment

* Switching cabinet arranged on top of unit, please add switching cabinet height (480 mm).

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process. With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.

Specifications of technical data relate to the optimum flow rate and return air condition 22° C / 40% r.h., outside air condition -12° C / 90% r.h. and standard density (1.204 kg/m³), unless otherwise specified.

At a glance:

- Over 80% temperature efficiency through three-stage recuperative heat recovery
- Energy efficiency class H1 according to EN 13053:2012
- Energy-saving EC fan motors
- Integrated compressor refrigeration system (59 series)
- Fulfils the requirements of VDI 6022

Solvus „Zero Emission Factory“, Braunschweig

Hotel Dollenberg
Dosolair
TWO-STAGE RECUPERATIVE HEAT RECOVERY

Units in the Dosolair 54 series achieve high heat recovery efficiency at medium to high air volume flow rates and can be used in a wide range of comfort air conditioning applications. The combination of first-class components with precise control and regulation systems guarantees economical operation at all times, while ensuring the highest degree of comfort air conditioning.

At a glance:
- For heat and cooling recovery
- Intelligent air bypass duct
- Two-stage supply air filtration
- Integrated defrost function
- Freely configurable HVAC system

Dosolair 54
with two-stage heat recovery

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>54 06 01</td>
<td>5,630</td>
<td>790</td>
<td>2,340</td>
<td>1,500</td>
<td>4,000</td>
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<td>5,630</td>
<td>1,110</td>
<td>2,340</td>
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</tr>
<tr>
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<td>7,900</td>
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<td>5,790</td>
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<td>2,750</td>
<td>11,800</td>
</tr>
<tr>
<td>54 25 01</td>
<td>6,430</td>
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<tr>
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<td>2,390</td>
<td>3,620</td>
<td>5,400</td>
<td>23,100</td>
</tr>
</tbody>
</table>

Units with opt. volume flow 40 000 and special units on request.

1 Door fitting assembly increase unit width by 65 mm each operating side
2 May require alteration of the technical equipment

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process. With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.

Specifications of technical data relate to the optimum flow rate and return air condition 22 °C / 40% r.h., outside air condition -12 °C / 90% r.h. and standard density (1.204 kg/m³), unless otherwise specified.
Adsolair

COOLING WITHOUT POWER CONSUMPTION

Units in the Adsolair series achieve high heat recovery efficiencies and can be used in a wide variety of comfort air conditioning applications. The integrated adiabatic evaporative cooling system allows temperature reductions of over 12 K*.

Adsolair 56
with double plate heat exchanger and adiabatic evaporative cooling system

A compressor refrigeration system integrated into the 58 series increases the cooling capacity of the overall system at high temperatures and allows the dehumidification of outside air. The combination of first-class components with precise control and regulation systems guarantees economical operation at all times, ensuring the highest degree of comfort air conditioning.

Adsolair 58
with double plate heat exchanger, adiabatic evaporative cooling system and compressor refrigeration system

At a glance:

- Over 75% temperature efficiency
- Energy-saving EC fan motors
- Intelligent air bypass duct
- Two-stage supply air filtration
- Meets the requirements of VDI 6022

Specifications of technical data relate to the optimum flow rate and return air condition 22° C / 40% r.h., outside air condition -12° C / 90% r.h. and standard density (1.204 kg/m³), unless otherwise specified.

---

Units with max. volume flow 52,800 m³/h and special units on request.

1 Door fitting assembly increases unit width by 65 mm each operating side
2 incl. 120 mm base frame, plus 60 mm cable duct
3 May require alteration of the technical equipment

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.

With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.
Units in the Resolair 62 and 66 series use regenerative heat recovery technology to achieve the highest heat recovery efficiency with low internal pressure losses. These are characterised by both high thermal and high electrical efficiency. A compressor refrigeration system integrated into 66 and 68 series increases the cooling capacity of the overall system at high temperatures.

**At a glance:**

- For heat and cooling recovery
- Over 90% temperature efficiency
- Energy efficiency class H1 according to EN 13053:2012
- Humidity recovery up to 70%
- Fulfils the requirements of VDI 6022

**Resolair 62/64 with highly efficient regenerative heat storage accumulators**

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>62 12 01</td>
<td>2,010</td>
<td>570</td>
<td>1,210*</td>
<td>410</td>
<td>1,200</td>
</tr>
<tr>
<td>62 18 01</td>
<td>2,170</td>
<td>730</td>
<td>1,530*</td>
<td>550</td>
<td>1,800</td>
</tr>
<tr>
<td>62 26 01</td>
<td>2,330</td>
<td>730</td>
<td>1,850</td>
<td>600</td>
<td>2,600</td>
</tr>
<tr>
<td>62 36 01</td>
<td>2,330</td>
<td>1,050</td>
<td>1,850</td>
<td>810</td>
<td>3,600</td>
</tr>
<tr>
<td>64 05 01</td>
<td>4,330</td>
<td>1,110</td>
<td>1,700</td>
<td>1,300</td>
<td>3,900</td>
</tr>
<tr>
<td>64 07 01</td>
<td>4,650</td>
<td>1,110</td>
<td>2,340</td>
<td>1,650</td>
<td>6,000</td>
</tr>
<tr>
<td>64 10 01</td>
<td>4,810</td>
<td>1,430</td>
<td>2,340</td>
<td>2,050</td>
<td>7,900</td>
</tr>
<tr>
<td>64 12 01</td>
<td>4,810</td>
<td>1,750</td>
<td>2,340</td>
<td>2,350</td>
<td>9,800</td>
</tr>
<tr>
<td>64 15 01</td>
<td>4,970</td>
<td>2,070</td>
<td>2,340</td>
<td>2,600</td>
<td>11,800</td>
</tr>
<tr>
<td>64 21 01</td>
<td>5,610</td>
<td>2,070</td>
<td>2,980</td>
<td>3,550</td>
<td>15,800</td>
</tr>
<tr>
<td>64 26 01</td>
<td>5,930</td>
<td>2,070</td>
<td>3,620</td>
<td>4,000</td>
<td>19,900</td>
</tr>
<tr>
<td>64 32 01</td>
<td>5,930</td>
<td>2,390</td>
<td>3,620</td>
<td>4,400</td>
<td>23,100</td>
</tr>
</tbody>
</table>

Units with max. volume flow 51,000 m³/h and special units on request.

**Resolair 66/68 with highly efficient regenerative heat storage accumulators and compressor refrigeration system**

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Opt. flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>66 18 01</td>
<td>3,310</td>
<td>930</td>
<td>1,530</td>
<td>790</td>
<td>1,800</td>
</tr>
<tr>
<td>66 26 01</td>
<td>3,470</td>
<td>930</td>
<td>1,850</td>
<td>850</td>
<td>2,600</td>
</tr>
<tr>
<td>66 36 01</td>
<td>3,470</td>
<td>1,050</td>
<td>1,850</td>
<td>1,100</td>
<td>3,600</td>
</tr>
<tr>
<td>68 05 01</td>
<td>5,380</td>
<td>1,110</td>
<td>1,700</td>
<td>1,750</td>
<td>3,900</td>
</tr>
<tr>
<td>68 07 01</td>
<td>5,700</td>
<td>1,110</td>
<td>2,340</td>
<td>2,150</td>
<td>6,000</td>
</tr>
<tr>
<td>68 10 01</td>
<td>5,860</td>
<td>1,400</td>
<td>2,340</td>
<td>2,700</td>
<td>7,900</td>
</tr>
<tr>
<td>68 12 01</td>
<td>6,020</td>
<td>1,750</td>
<td>2,340</td>
<td>3,050</td>
<td>9,800</td>
</tr>
<tr>
<td>68 15 01</td>
<td>6,180</td>
<td>2,070</td>
<td>2,340</td>
<td>3,500</td>
<td>11,800</td>
</tr>
<tr>
<td>68 21 01</td>
<td>6,810</td>
<td>2,070</td>
<td>2,980</td>
<td>4,450</td>
<td>15,800</td>
</tr>
<tr>
<td>68 26 01</td>
<td>7,300</td>
<td>2,070</td>
<td>3,620</td>
<td>5,100</td>
<td>19,900</td>
</tr>
<tr>
<td>68 32 01</td>
<td>7,300</td>
<td>2,390</td>
<td>3,620</td>
<td>5,500</td>
<td>23,100</td>
</tr>
</tbody>
</table>

Units with max. volume flow 51,000 m³/h and special units on request.

1. Door fitting assembly increases unit width by 25 mm each operating side (series 62 und 66) respectively 65 mm (series 64 und 68). Refrigerant pipe duct on backside of units 66 increases unit width by 80 mm.
2. Height incl. 100 mm unit feet and 60 mm cable duct (series 62 und 66) respectively incl. 120 mm base frame, incl. 60 mm cable duct (series 64 und 68)
3. May require alteration of the technical equipment

Etrium Cologne, DGNB seal in gold
City Hall Stralsund

**Industry-Resolair 65**

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Max. volume flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>65 07 91</td>
<td>4,110</td>
<td>3,700</td>
<td>1,170</td>
<td>2,300</td>
<td>10,000</td>
</tr>
<tr>
<td>65 17 91</td>
<td>5,390</td>
<td>4,340</td>
<td>1,490</td>
<td>4,550</td>
<td>20,000</td>
</tr>
<tr>
<td>65 26 91</td>
<td>6,030</td>
<td>4,660</td>
<td>1,810</td>
<td>6,100</td>
<td>30,000</td>
</tr>
<tr>
<td>65 36 91</td>
<td>6,030</td>
<td>4,980</td>
<td>2,130</td>
<td>8,050</td>
<td>40,000</td>
</tr>
</tbody>
</table>

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process. With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.

Specifications of technical data relate to the optimum Flow rate and return air condition 22°C / 40% r.h., outside air condition -12°C / 90% r.h. and standard density (1.204 kg/m³), unless otherwise specified.
Sorpsolair

COOLING WITH THE SUN

Units in the Sorpsolair 72 and 73 series were developed especially to use regenerative energy. The innovative air conditioning concept combines sorption-based dehumidification, adiabatic evaporative cooling and an efficient heat recovery system in a compact comfort air conditioning unit. The 72 series, without a brine tank, is suitable for directly using the waste heat, e.g. from combined heat and power system (CHPS), while the brine tank integrated into the 73 series allows the storage of solar thermal energy and hence increases the total efficiency of your installations. The combination of first-class components with precise control and regulation systems guarantees economical operation at all times, while ensuring the highest degree of comfort air conditioning. Sorpsolair systems are designed for all office and business buildings, as well as many other building types.

At a glance:

- Over 75% temperature efficiency
- Thermal coefficient of efficiency COPth from 1.5
- Brine regeneration through the use of solar thermal energy, district heat or existing process heat at a low-temperature level (from 65°C flow)
- Intelligent air bypass duct
- Integrated defrost function

Specifications of technical data relate to the optimum flow rate and return air condition 22°C / 40% r.h., outside air condition -12°C / 90% r.h. and standard density (1.204 kg/m³), unless otherwise specified.

### Dimensions of brine tank

<table>
<thead>
<tr>
<th>Unit Type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>73 04 01</td>
<td>4,180</td>
<td>1,050</td>
<td>2,010</td>
<td>430</td>
</tr>
<tr>
<td>73 05 01</td>
<td>4,180</td>
<td>1,050</td>
<td>2,010</td>
<td>430</td>
</tr>
<tr>
<td>73 06 01</td>
<td>4,180</td>
<td>1,050</td>
<td>2,010</td>
<td>430</td>
</tr>
<tr>
<td>73 10 01</td>
<td>4,500</td>
<td>1,050</td>
<td>2,330</td>
<td>535</td>
</tr>
<tr>
<td>73 13 01</td>
<td>4,500</td>
<td>1,050</td>
<td>2,330</td>
<td>535</td>
</tr>
<tr>
<td>73 16 01</td>
<td>4,500</td>
<td>1,050</td>
<td>2,330</td>
<td>535</td>
</tr>
<tr>
<td>73 19 01</td>
<td>5,460</td>
<td>1,050</td>
<td>2,330</td>
<td>650</td>
</tr>
<tr>
<td>73 22 01</td>
<td>5,460</td>
<td>1,050</td>
<td>2,330</td>
<td>650</td>
</tr>
</tbody>
</table>

1. Door fitting assembly increases unit width by 25 mm each operating side
2. incl. 120 mm base frame, plus 60 mm cable duct
3. Empty weight, not operation weight

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process. With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.
Adconair
FULL COUNTERFLOW HEAT RECOVERY

With its full counterflow plate heat exchanger, the Adconair 76 series is setting new standards in the ventilation industry. The new heat exchanger works with a real counterflow proportion of over 80%. The internal pressure losses of the recuperator is just 150 Pa. Adconair units are optimally adapted for use in comfort air conditioning. The unit series is designed to comply with the requirements of the highest energy efficiency classes. Ideal areas of application include all residential and non-residential buildings. Thanks to their high capacity and intelligent regulation system, the units always create an excellent indoor climate.

Available options:
- Adiabatic evaporative cooling
- AdiabaticPro
- Compressor refrigeration system, also available as reversible system

At a glance:

- Designed for the requirements of the highest energy efficiency classes
- HRC class H1, even at high air velocities
- Thermal bridge factor \( k_b = 0.78 \) - class TB1
- Two-stage supply air filtration
- Meets the requirements of the German Energy Saving Ordinance (EnEV) and the German Renewable Energies Heating Law (EEWärmeG)

Adconair with counterflow plate heat exchanger

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length ( ^1 ) (mm)</th>
<th>Width ( ^2 ) (mm)</th>
<th>Height ( ^3 ) (mm)</th>
<th>Weight ( ^4 ) (kg)</th>
<th>Opt. flow rate ( ^5 ) (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>76 03 01</td>
<td>4,810</td>
<td>790</td>
<td>1,700</td>
<td>1,220</td>
<td>2,600</td>
</tr>
<tr>
<td>76 05 01</td>
<td>4,970</td>
<td>1,110</td>
<td>1,700</td>
<td>1,500</td>
<td>3,900</td>
</tr>
<tr>
<td>76 06 01</td>
<td>5,610</td>
<td>790</td>
<td>2,340</td>
<td>1,650</td>
<td>4,000</td>
</tr>
<tr>
<td>76 10 01</td>
<td>5,610</td>
<td>1,110</td>
<td>2,340</td>
<td>1,900</td>
<td>6,000</td>
</tr>
<tr>
<td>76 13 01</td>
<td>5,770</td>
<td>1,430</td>
<td>2,340</td>
<td>2,350</td>
<td>7,900</td>
</tr>
<tr>
<td>76 16 01</td>
<td>5,770</td>
<td>1,750</td>
<td>2,340</td>
<td>2,650</td>
<td>9,800</td>
</tr>
<tr>
<td>76 19 01</td>
<td>5,770</td>
<td>2,070</td>
<td>2,340</td>
<td>3,000</td>
<td>11,800</td>
</tr>
<tr>
<td>76 25 01</td>
<td>6,250</td>
<td>2,070</td>
<td>2,980</td>
<td>3,900</td>
<td>15,800</td>
</tr>
<tr>
<td>76 29 01</td>
<td>6,250</td>
<td>2,390</td>
<td>2,980</td>
<td>4,300</td>
<td>18,400</td>
</tr>
<tr>
<td>76 37 01</td>
<td>6,250</td>
<td>3,030</td>
<td>2,980</td>
<td>5,700</td>
<td>23,600</td>
</tr>
</tbody>
</table>

1 May change depending on chosen option, e.g. AdiabaticPro, compressor refrigeration system, recuperator in short version (- 960 mm)
2 Door fitting assembly increases unit width by 65 mm each operating side
3 incl. 120 mm base frame, incl. 60 mm cable duct
4 If option Adiabatic or AdiabaticPro is chosen, please affirm possible additional weight!
5 May require alteration of the technical equipment. If option Adiabatic or AdiabaticPro is chosen, we recommend optimum flow rate as maximum.

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. For service work above the unit, please allow 50 mm working height clearance above the cable duct. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process. With every single selection we do to your individual requirements our certified selection software automatically checks the Ecodesign compliance level 1 and 2.

Specifications of technical data relate to the optimum flow rate and return air condition 22°C / 40% r.h., outside air condition -12°C / 90% r.h. and standard density (1,204 kg/m³), unless otherwise specified.
Far too often, warm waste water is discharged into the sewer system, together with all the energy it contains. Units in the AquaCond series recover the majority of this heat energy and transfer it to the clean water. The combination of recuperator and heat pump means that only approx. 10% of the energy is required that would be needed by a conventional heating system. The heat exchanger cleaning system integrated in this series even allows the units to be used where the waste water is contaminated with dirt. Recover valuable energy, anywhere that warm waste water is produced and simultaneously warm clean water has to be provided, e.g. in the shower areas of swimming pools, hospitals or residential homes, in laundries and in many industrial processes.

AquaCond

HEAT RECOVERY FROM WASTE WATER

At a glance:

- Heat recovery from clean or contaminated waste water for heating clean water
- Automatic heat exchanger cleaning
- Flow rate regulation

### AquaCond 44 with automatic heat exchanger cleaning

<table>
<thead>
<tr>
<th>Unit Type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height 1 (mm)</th>
<th>Weight 1 (kg)</th>
<th>Max. Quantity of Flow m³/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>44 08 x1</td>
<td>1,050</td>
<td>730</td>
<td>1,370</td>
<td>430</td>
<td>0.8</td>
</tr>
<tr>
<td>44 12 x1</td>
<td>1,210</td>
<td>890</td>
<td>1,530</td>
<td>450</td>
<td>1.2</td>
</tr>
<tr>
<td>44 18 x1</td>
<td>1,370</td>
<td>890</td>
<td>1,690</td>
<td>650</td>
<td>1.8</td>
</tr>
<tr>
<td>44 24 x2</td>
<td>2,420</td>
<td>890</td>
<td>1,530</td>
<td>860</td>
<td>2.4</td>
</tr>
<tr>
<td>44 36 x2</td>
<td>2,740</td>
<td>890</td>
<td>1,690</td>
<td>1,260</td>
<td>3.6</td>
</tr>
<tr>
<td>44 54 x3</td>
<td>4,110</td>
<td>890</td>
<td>1,690</td>
<td>1,900</td>
<td>5.4</td>
</tr>
</tbody>
</table>

1 Door fitting assembly increases unit width by 25 mm each operating side
2 plus unit feet
3 Empty weight, no operation weight

Please comply with the dimensions for body size and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.

Technical data specified refer to max. volume flow rate and waste water temperature 31°C / clean water temperature 10°C
Units in the Drysolair series were developed especially for removing high levels of moisture from inside a building. Through the precooling in the recuperator of the air to be dried, the unit works with considerably lower compressor performance than a simple heat pump system and creates a consistently good climate in ice rinks. It is also suited to the drying of buildings or industrial drying processes. The combination of first-class components with precise control and regulation guarantees economical operation at all times and adjusts the temperature and humidity to requirements.

**At a glance:**

- For all drying applications
- Low connection capacity through the upstream installation of a recuperator
- Corrosion-free cross counterflow plate heat exchanger
- Intelligent air bypass duct
- Compact design

### Drysolair 11

<table>
<thead>
<tr>
<th>Unit</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Dehumidification capacity (kg)</th>
<th>Opt. flow rate (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>11 10 01</td>
<td>730</td>
<td>2,245</td>
<td>450</td>
<td>4.5</td>
<td>1,000</td>
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<tr>
<td>11 15 01</td>
<td>730</td>
<td>2,245</td>
<td>450</td>
<td>6.8</td>
<td>1,500</td>
<td></td>
</tr>
<tr>
<td>11 40 01</td>
<td>1,050</td>
<td>2,725</td>
<td>660</td>
<td>17.6</td>
<td>4,000</td>
<td></td>
</tr>
<tr>
<td>11 60 01</td>
<td>1,050</td>
<td>2,725</td>
<td>680</td>
<td>21.6</td>
<td>6,000</td>
<td></td>
</tr>
</tbody>
</table>

All technical data relate to optimum flow rate through heat recovery system and the air inlet conditions specified below:

1 Door fitting assembly increases unit width by 25 mm each operating side
2 incl. 100 mm unit feet
3 Air inlet 20°C / 70% r.h., other designs available upon request

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.
Frecolair
FREE COOLING FOR ROOMS WITH HIGH THERMAL LOADS

Units in the Frecolair 14 series were developed especially for discharging high internal heat loads into the atmosphere from buildings without humidity requirements. In data processing centres and technical facilities, these units ensure reliable operation and precisely regulate the supply air temperature down to the degree. The variability of the operating modes, in combination with first-class components and precise control and regulation systems, guarantees economical operation at all times.

At a glance:

- For discharging high heat loads
- Advantages of free cooling and recirculation mode in a single unit
- High electrical efficiency thanks to the lowest possible internal pressure losses
- Low space requirement, no additional construction measures for cooling are required

Frecolair 14

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width (mm)</th>
<th>Height (mm)</th>
<th>Weight (kg)</th>
<th>Cooling capacity 1 (kW)</th>
<th>effect. cooling capacity 1 (kW)</th>
<th>Optimum flow rate Return/Supply air (m³/h)</th>
<th>Optimum flow rate Outside/Exhaust air (m³/h)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14 03 01</td>
<td>2,330</td>
<td>730</td>
<td>1,490</td>
<td>660</td>
<td>11.3</td>
<td>10.5</td>
<td>2,600</td>
<td>3,500</td>
</tr>
<tr>
<td>14 04 01</td>
<td>2,490</td>
<td>890</td>
<td>1,490</td>
<td>700</td>
<td>14.2</td>
<td>13.1</td>
<td>3,300</td>
<td>4,600</td>
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<tr>
<td>14 05 01</td>
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<td>1,490</td>
<td>800</td>
<td>17.5</td>
<td>16.2</td>
<td>4,000</td>
<td>5,300</td>
</tr>
<tr>
<td>14 06 01</td>
<td>2,490</td>
<td>730</td>
<td>2,130</td>
<td>850</td>
<td>19.9</td>
<td>18.2</td>
<td>4,700</td>
<td>6,300</td>
</tr>
<tr>
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<td>1,050</td>
<td>2,130</td>
<td>1,210</td>
<td>30.8</td>
<td>28.1</td>
<td>7,100</td>
<td>9,500</td>
</tr>
<tr>
<td>14 13 01</td>
<td>2,810</td>
<td>1,370</td>
<td>2,130</td>
<td>1,450</td>
<td>38.7</td>
<td>35.2</td>
<td>9,500</td>
<td>12,600</td>
</tr>
<tr>
<td>14 16 01</td>
<td>2,970</td>
<td>1,690</td>
<td>2,130</td>
<td>1,670</td>
<td>47.5</td>
<td>43.4</td>
<td>11,800</td>
<td>15,800</td>
</tr>
<tr>
<td>14 19 01</td>
<td>2,970</td>
<td>2,010</td>
<td>2,130</td>
<td>1,850</td>
<td>58.1</td>
<td>52.7</td>
<td>14,200</td>
<td>19,000</td>
</tr>
<tr>
<td>14 25 01</td>
<td>3,220</td>
<td>2,010</td>
<td>2,860</td>
<td>2,150</td>
<td>72.6</td>
<td>65.7</td>
<td>18,700</td>
<td>25,000</td>
</tr>
<tr>
<td>14 32 01</td>
<td>3,540</td>
<td>3,500</td>
<td>2,350</td>
<td>85.4</td>
<td>76.7</td>
<td>76.7</td>
<td>24,000</td>
<td>32,000</td>
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<tr>
<td>14 36 01</td>
<td>3,540</td>
<td>2,330</td>
<td>3,500</td>
<td>2,550</td>
<td>99.0</td>
<td>88.8</td>
<td>27,000</td>
<td>36,000</td>
</tr>
</tbody>
</table>

All technical data relate to the optimum flow rate through heat recovery system and outside air conditions 32°C / 40% r.h., return air conditions 28°C / 40% r.h.

1 Door fitting assembly increases unit width by 25 mm each operating side
2 incl. 120 mm base frame
3 Recirculation air cooling mode, supply air temperature approx. 17°C

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.
Adcoolair

GREEN IT

Thanks to the combination of indirect free cooling, adiabatic evaporative cooling and the integrated output-regulated compressor refrigeration system, each of which supports the effectiveness of the others, the Adcoolair 75 unit series allows heat dissipation in recirculation mode from data processing centres and other rooms with high thermal loads, with minimal space requirements, low air pressure losses within the unit and very little energy consumption. The use of energy-efficient EC fan motors in combination with a demand-based flow rate control system, additionally contributes to the reduction of operating costs.

Adcoolair 75

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 2 (mm)</th>
<th>Weight (kg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>75 02 01</td>
<td>2,900</td>
<td>730</td>
<td>2,130</td>
<td>1,020</td>
</tr>
<tr>
<td>75 04 01</td>
<td>2,900</td>
<td>1,050</td>
<td>2,130</td>
<td>1,240</td>
</tr>
<tr>
<td>75 06 01</td>
<td>2,900</td>
<td>1,370</td>
<td>2,130</td>
<td>1,430</td>
</tr>
<tr>
<td>75 08 01</td>
<td>3,380</td>
<td>1,050</td>
<td>2,770</td>
<td>1,800</td>
</tr>
<tr>
<td>75 10 01</td>
<td>3,380</td>
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<td>2,770</td>
<td>2,660</td>
</tr>
<tr>
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<td>3,060</td>
<td>3,250</td>
<td>4,180</td>
</tr>
<tr>
<td>75 14 01</td>
<td>4,020</td>
<td>4,020</td>
<td>3,250</td>
<td>5,360</td>
</tr>
<tr>
<td>75 16 01</td>
<td>4,020</td>
<td>4,660</td>
<td>3,250</td>
<td>6,170</td>
</tr>
</tbody>
</table>

1 Door fitting assembly increases unit width by 25 mm each operating side
2 incl. 120 mm base frame

For service work, a clearance corresponding to dimension width is required on the operating side of the unit. If the width is smaller than one metre, please leave a clearance of one metre. Please comply with the dimensions for body size, air duct connections and electrical switch cabinet. Please seek approval of technical data and specifications prior to start of the planning process.

Specifications of technical data relate to the return air conditions 34° C / 20% r.h., outside air conditions 35° C / 40% r.h.

At a glance:

- Compact dimensions, optimised for installation in technology centres without cooling tower
- No contamination of the process airflow with dust or corrosive pollutants
- Moisture content of the process air remains unaffected
- Low airflow rate required for heat dissipation
- Excellent PUE values of up to 1.1

Adcoolair 751301 - simplified illustration
At a glance:

- Efficient cooling through the use of natural resources
- Very high power density while simultaneously having high EER and ESEER values
- Compressor refrigeration system and free cooler optimally adapted to the respective application
- Compact design thanks to integrated recoupling system, removing the need for cooling system components on the facade or on the roof

**Hybriteam 97 – efficiency-optimised**

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 1 (mm)</th>
<th>Weight 1 (kg)</th>
<th>Cooling capacity 4 (kW)</th>
<th>ESEER 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>97 04 01</td>
<td>3,700</td>
<td>890</td>
<td>1,650</td>
<td>1,470</td>
<td>33 - 48</td>
<td>5.5</td>
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<td>97 05 01</td>
<td>3,700</td>
<td>1,050</td>
<td>1,650</td>
<td>2,070</td>
<td>45 - 64</td>
<td>5.5</td>
</tr>
<tr>
<td>97 06 01</td>
<td>4,340</td>
<td>730</td>
<td>2,130</td>
<td>2,490</td>
<td>56 - 81</td>
<td>5.5</td>
</tr>
<tr>
<td>97 10 01</td>
<td>4,500</td>
<td>1,050</td>
<td>2,130</td>
<td>3,250</td>
<td>74 - 106</td>
<td>5.4</td>
</tr>
<tr>
<td>97 13 01</td>
<td>4,660</td>
<td>1,370</td>
<td>2,130</td>
<td>4,390</td>
<td>118 - 168</td>
<td>5.5</td>
</tr>
<tr>
<td>97 16 01</td>
<td>4,820</td>
<td>1,690</td>
<td>2,130</td>
<td>5,240</td>
<td>148 - 217</td>
<td>5.5</td>
</tr>
<tr>
<td>97 19 01</td>
<td>4,820</td>
<td>2,010</td>
<td>2,130</td>
<td>6,110</td>
<td>172 - 247</td>
<td>5.2</td>
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</table>

**Hybriteam 98 – performance-optimised**

<table>
<thead>
<tr>
<th>Unit type</th>
<th>Length (mm)</th>
<th>Width 1 (mm)</th>
<th>Height 1 (mm)</th>
<th>Weight 1 (kg)</th>
<th>Cooling capacity 4 (kW)</th>
<th>ESEER 5</th>
</tr>
</thead>
<tbody>
<tr>
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<td>2,070</td>
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<tr>
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<td>2,800</td>
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<td>4.7</td>
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<td>2,450</td>
<td>3,220</td>
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<td>2,450</td>
<td>4,830</td>
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<td>4.9</td>
</tr>
<tr>
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<td>2,450</td>
<td>5,700</td>
<td>244 - 350</td>
<td>5.1</td>
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<tr>
<td>98 19 01</td>
<td>4,820</td>
<td>2,010</td>
<td>2,450</td>
<td>7,170</td>
<td>319 - 455</td>
<td>4.9</td>
</tr>
</tbody>
</table>

1. Door fitting assembly increases unit width by 25 mm each operating side
2. incl. 120 mm base frame
3. Empty weight, no operation weight
4. dependent on flow/return-temperature and water flow rate, at OA = 32°C, 40%
r.h.
5. at flow = 6°C

For service work, a clearance corresponding to dimension width is required on the
operating side of the unit. If the width is smaller than one metre, please leave a
clearance of one metre. Please comply with the dimensions for body size, air duct
connections and electrical switch cabinet. Please seek approval of technical data and
specifications prior to start of the planning process.
## THE MENERGA UNIT KEY

<table>
<thead>
<tr>
<th>Series</th>
<th>Name</th>
<th>Function</th>
<th>Equipment</th>
<th>Design</th>
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</thead>
<tbody>
<tr>
<td>11</td>
<td>Drysolair</td>
<td>Air drying</td>
<td>Heat pump, recuperator</td>
<td></td>
</tr>
<tr>
<td>14</td>
<td>Fresolair</td>
<td>Ventilation/cooling</td>
<td>Free cooling, compressor refrigeration system</td>
<td></td>
</tr>
<tr>
<td>19</td>
<td>ThermoCond</td>
<td>Indoor swimming pool air conditioning</td>
<td>Cross-counterflow heat exchanger</td>
<td></td>
</tr>
<tr>
<td>23</td>
<td>ThermoCond</td>
<td>Heat recovery</td>
<td>Cross-counterflow-cross heat exchanger</td>
<td>01 Indoor installation, 91 Outdoor installation</td>
</tr>
<tr>
<td>29</td>
<td>ThermoCond</td>
<td>Heat recovery</td>
<td>Cross-counterflow heat exchanger, heat pump</td>
<td></td>
</tr>
<tr>
<td>38</td>
<td>ThermoCond</td>
<td>Heat recovery</td>
<td>Full countercurrent plate heat exchanger, volume flow reduction as required</td>
<td></td>
</tr>
<tr>
<td>39</td>
<td>ThermoCond</td>
<td>Heat recovery</td>
<td>Asymmetrical high-capacity heat exchanger, output-controlled heat pump, fresh water heater, volume flow reduction as required</td>
<td></td>
</tr>
<tr>
<td>44</td>
<td>AquaCond</td>
<td>Heat recovery from waste water</td>
<td>Heat pump, counterflow coaxial recuperator, heat pump, automatic heat exchanger cleaning</td>
<td></td>
</tr>
<tr>
<td>52</td>
<td>Trisolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Cross-counterflow-cross heat exchanger, air volume flow rate up to 5,000 m³/h</td>
<td></td>
</tr>
<tr>
<td>54</td>
<td>Dosolair</td>
<td>Comfort air conditioning, recuperative heat recovery</td>
<td>Double plate heat exchanger, maximum flow rates up to 52,200 m³/h</td>
<td></td>
</tr>
<tr>
<td>56</td>
<td>Adsolair</td>
<td>Comfort air conditioning, recuperative heat recovery</td>
<td>Double plate heat exchanger, adiabatic evaporative cooling, optimum flow rates up to 52,200 m³/h</td>
<td></td>
</tr>
<tr>
<td>58</td>
<td>Adsolair</td>
<td>Comfort air conditioning, recuperative heat recovery</td>
<td>Double plate heat exchanger, adiabatic evaporative cooling, compressor refrigeration system, maximum flow rates up to 52,800 m³/h</td>
<td></td>
</tr>
<tr>
<td>59</td>
<td>Trisolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Cross-counterflow-cross heat exchanger, compressor refrigeration system, air volume flow rate up to 4,800 m³/h</td>
<td></td>
</tr>
<tr>
<td>62</td>
<td>Resolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Heat accumulator module, maximum flow rates up to 4,300 m³/h</td>
<td></td>
</tr>
<tr>
<td>64</td>
<td>Resolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Heat accumulator module, maximum flow rates up to 51,000 m³/h</td>
<td></td>
</tr>
<tr>
<td>65</td>
<td>Resolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Heat accumulator module, air volume flow rates up to 40,000 m³/h</td>
<td></td>
</tr>
<tr>
<td>66</td>
<td>Resolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Heat accumulator module, compressor refrigeration system, maximum flow rates up to 4,300 m³/h</td>
<td></td>
</tr>
<tr>
<td>68</td>
<td>Resolair</td>
<td>Comfort and process air conditioning, regenerative heat recovery</td>
<td>Heat accumulator module, compressor refrigeration system, maximum flow rates up to 51,000 m³/h</td>
<td></td>
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<tr>
<td>72</td>
<td>Sorpsolair</td>
<td>Sorption-based air conditioning</td>
<td>Double plate heat exchanger, adiabatic evaporative cooling, sorptive dehumidification, maximum flow rates up to 14,900 m³/h</td>
<td></td>
</tr>
<tr>
<td>73</td>
<td>Sorpsolair</td>
<td>Sorption-based air conditioning</td>
<td>Double plate heat exchanger, adiabatic evaporative cooling, sorptive dehumidification, brine accumulator, maximum flow rates up to 14,900 m³/h</td>
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</tr>
<tr>
<td>75</td>
<td>Adcoolair</td>
<td>Recirculating air cooling</td>
<td>Free cooling, adiabatic evaporative cooling, compressor refrigeration system</td>
<td></td>
</tr>
<tr>
<td>76</td>
<td>Adconair</td>
<td>Comfort air conditioning, recuperative heat recovery</td>
<td>Counterflow-plate heat exchanger, air volume flow up to 35,600 m³/h, with adiabatic evaporative cooling, AdiabaticPro or refrigeration system</td>
<td></td>
</tr>
<tr>
<td>97</td>
<td>Hybritemp</td>
<td>Cold water set</td>
<td>Indirect free cooling, adiabatic evaporative cooling, efficiency-optimised compressor refrigeration system</td>
<td></td>
</tr>
<tr>
<td>98</td>
<td>Hybritemp</td>
<td>Cold water set</td>
<td>Free cooling, adiabatic evaporative cooling, efficiency-optimised compressor refrigeration system</td>
<td></td>
</tr>
</tbody>
</table>
Creating a good climate.
For over 35 years. Worldwide.

Menerga GmbH
Alexanderstraße 69
Muelheim an der Ruhr
Germany

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info@menerga.com

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