ARID-Dry®
Climate by Design International
2100 Park Drive • Owatonna, MN 55060 • 507.451.2198
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Meeting NHL Standards for Large Spectator Ice Arenas

The impact of water vapor on ice surfaces has been a definite target for the Recreational Arena (Less than 2500 seats). These venues have utilized desiccant dehumidification to maintain optimum humidity levels for the best ice quality while maintaining an energy effective environment. Spectator Arenas (2500 Seats or More) have generally used traditional mechanical systems that rely on refrigerant based cooling for dehumidification and a reheat system for temperature control. The design conditions were generally 70ºF at 50 % RH (Relative Humidity). Recently the NHL has issued revised design parameters to improve ice sheet and arena conditions for professional hockey venues. New modern arena construction has included active desiccant dehumidification to provide 65ºF at 35% to 40% RH (Relative Humidity) conditions to optimize the ice conditions.

The Prudential Center is conditioned with (4) ARID-Ice 48,500 SCFM, IAQ desiccant systems that maintain humidity while providing Indoor Air Quality to meet NHL standards.

The recreational market (Less than 2500 Seats) has used desiccant dehumidification exclusively to maintain low dew-points. The small rink market tends to allow the rink surface to cool the arena. Many arenas run with ambient temperatures of 55 ºF or lower. The Arid-Ice gas fired desiccant unit we can actively maintain rink dew-points of 35ºF. This benefits rink owners and operators by providing a fog and condensation free building. Eliminating condensation will enhance the building appearance and safety by eliminating mold growth on cold surfaces and providing dry, clean seats and stairs. Transferring the latent load to the gas fired dehumidifier can in many cases reduce the operating cost of the large electric rink refrigeration system.

The applications for the ARID-Ice dehumidification system revolve around the surface temperature of the ice sheet. The brine or refrigeration system which cools the ice must deliver refrigerant cold enough to provide an ice surface temperature of 20 to 24 º F. Generally this brine is circulated at 10 to 16 ºF to provide the required capacity. Ice sheet manufacturers provide capacity based on a heat load per square foot of ice sheet. The higher the load the lower required temperature of the supply brine and the larger the equipment and the energy cost. As the heat load rises it also takes longer to refreeze the ice surface after the resurfacing operation. In some professional or play off events, when the ambient outdoor moistures levels rise, play has had to be suspended or canceled due to areas of fog and poor ice conditions. Complaints from the players or media can put a negative spin on the level of play due to “slushy or wet” ice conditions.

In the recreational rink market the dew-point of the rink is maintained at or below 32 ºF, This will provide an indoor condition of 55 to 60ºF at 40 to 35% RH. The National Hockey League (NHL) has now issued requirements that are similar to the recreational market. The NHL has requested a 60ºF at 35% pre-game condition and a 65ºF at 45% post game condition. By maintaining a controlled humidity the ice surface can be optimized to provide the participants the best possible ice surface for the event. It is not surprising that these conditions are now recommended, most of the professional player practice in a recreational rink that is properly controlled.

If the humidity levels rise high enough the capacity of the rink refrigeration equipment can be overrun by the latent capacity of the air and areas of liquid water can be seen on the ice sheet. It is critical for the designer to design the HVAC/DH system to provide a proper humidity in the arena to allow the event to play on the best possible surface. Following is a comparison of energy and equipment necessary to properly maintain ice event conditions in accordance with the new requirements. The two systems will compare a refrigeration/reheat system with a desiccant system coupled to a refrigeration system.

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Application Description:

The cooling energy provided to the ice sheet is absorbed by three energy sources as described in the estimated flow diagram below. 15 % of the energy is going to the insulated under floor heating system, 10% to the edge of the slab, 40% is cooling the air boundary layer (dependent on radiant or lighting loads) above the ice sheet, and the remaining 35% energy is freezing water vapor out of the air.

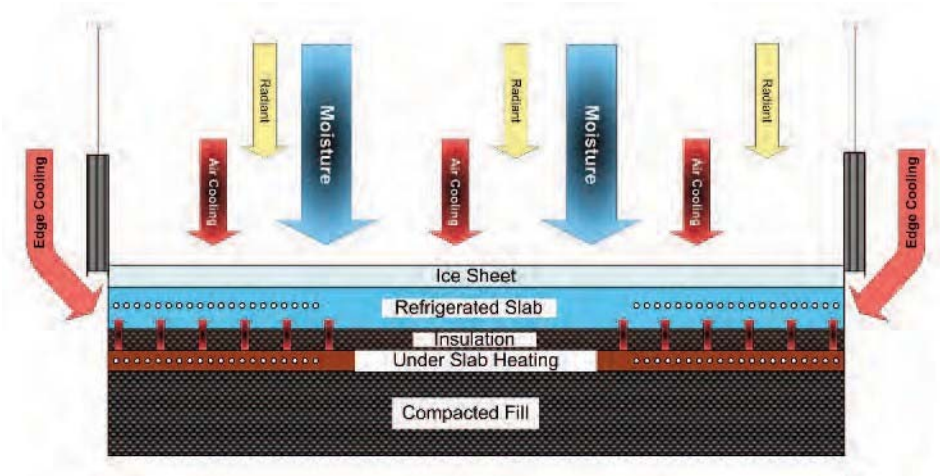

Figure 1 (70˚F at 50%) Large Latent Load

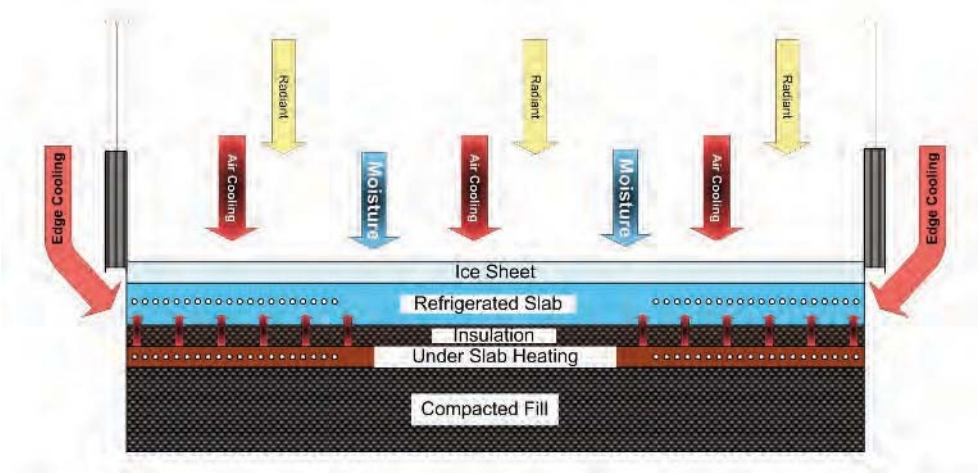

Figure 2 (65˚F at 40%) Reduced Latent Load

It is important to understand that humidity is driven by differential vapor pressure. Air with a higher vapor pressure (High Humidity) tries to equalize with air of low vapor pressure (Low Humidity). The boundary layer of air close to the ice sheet is cooled to a low temperature which will have a low vapor pressure. This differential pressure provides a strong motive force to draw moisture from other areas of the arena directly to the cold surface of the ice sheet. As the high humidity air is cooled as it approaches the ice sheet, fog and condensation result increasing the moisture (Latent) load on the ice sheet. By reducing the arena humidity we can reduce the energy required by the ice surface and allow the surface to maintain a better ice condition.

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Technical:

To properly maintain humidity levels we need to estimate moisture loads during occupied events and determine the outside air requirements. We will use an example of a 20,000 seat arena with a mechanical room in each quadrant serving approximately 25 % of the space. For the purpose of this example we will compare a refrigeration system designed to maintain 65 ºF at 40% RH (36.6 Grains of Moisture/Pound of Dry Air) with a desiccant system designed to maintain 65ºF at 40% (36.6 Grains of Moisture/Pound of Dry Air). Each system provides a 12ºF and 8 grain depression to accommodate internal loads. For normal application each quad will use 100,000 SCFM (Standard Cubic Feet Minute @ 70ºF) total air with 37,500 SCFM of outside air based on 7.5 SCFM per person. The total arena will use approximately 400,000 SCFM with 150,000 SCFM of outside air. The system below would be located in each quad. The diagrams below will show dew-points and Grains/Lb in lieu of relative humidity. Relative Humidity can be confusing with multiple temperature changes.

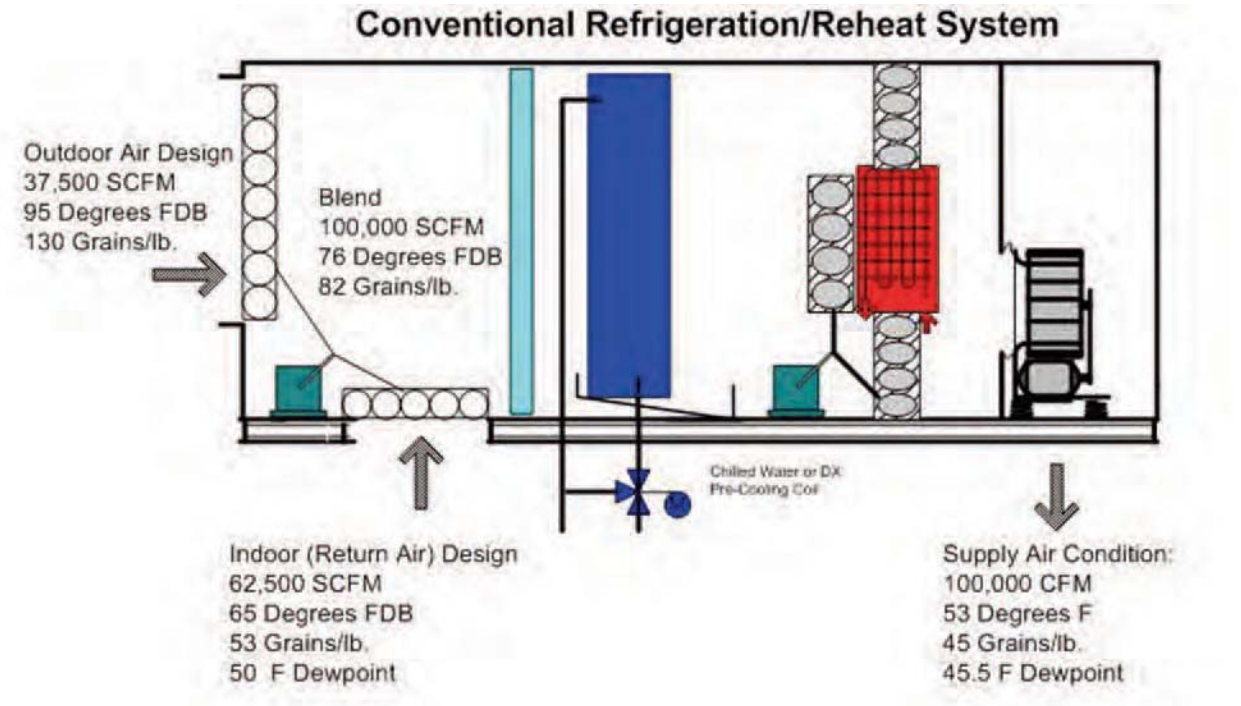

Figure 3

Heating and Cooling Energy Usage:

Refrigeration at a 33º F Dew-Point Supply Condition.** (Not practical for continuous supply)
- 7,670,839 BTUH, 639 Nominal Tons
- De-Rate Equipment 12% (EST) for 32º F Discharge
- De-rate Equipment 7.5 % for 20% E.G. freeze Protection, 769 Actual Tons

Reheat Coil:
- 1,849,586 BTUH
- De-rate 80% Boiler Efficiency
- 2,219,503 BTUH

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Conventional Refrigeration/Reheat System

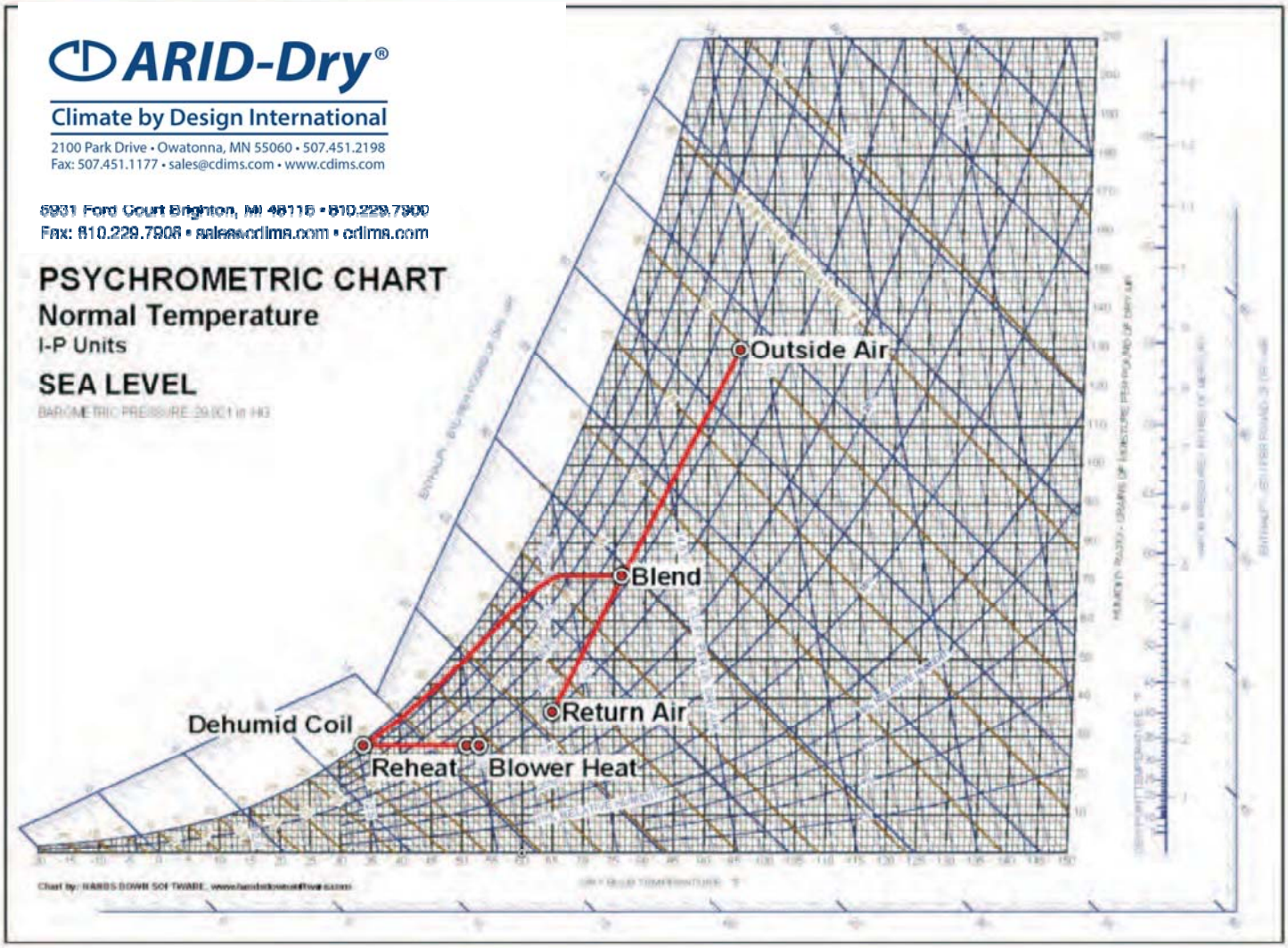

Figure 4

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Desiccant Combination System

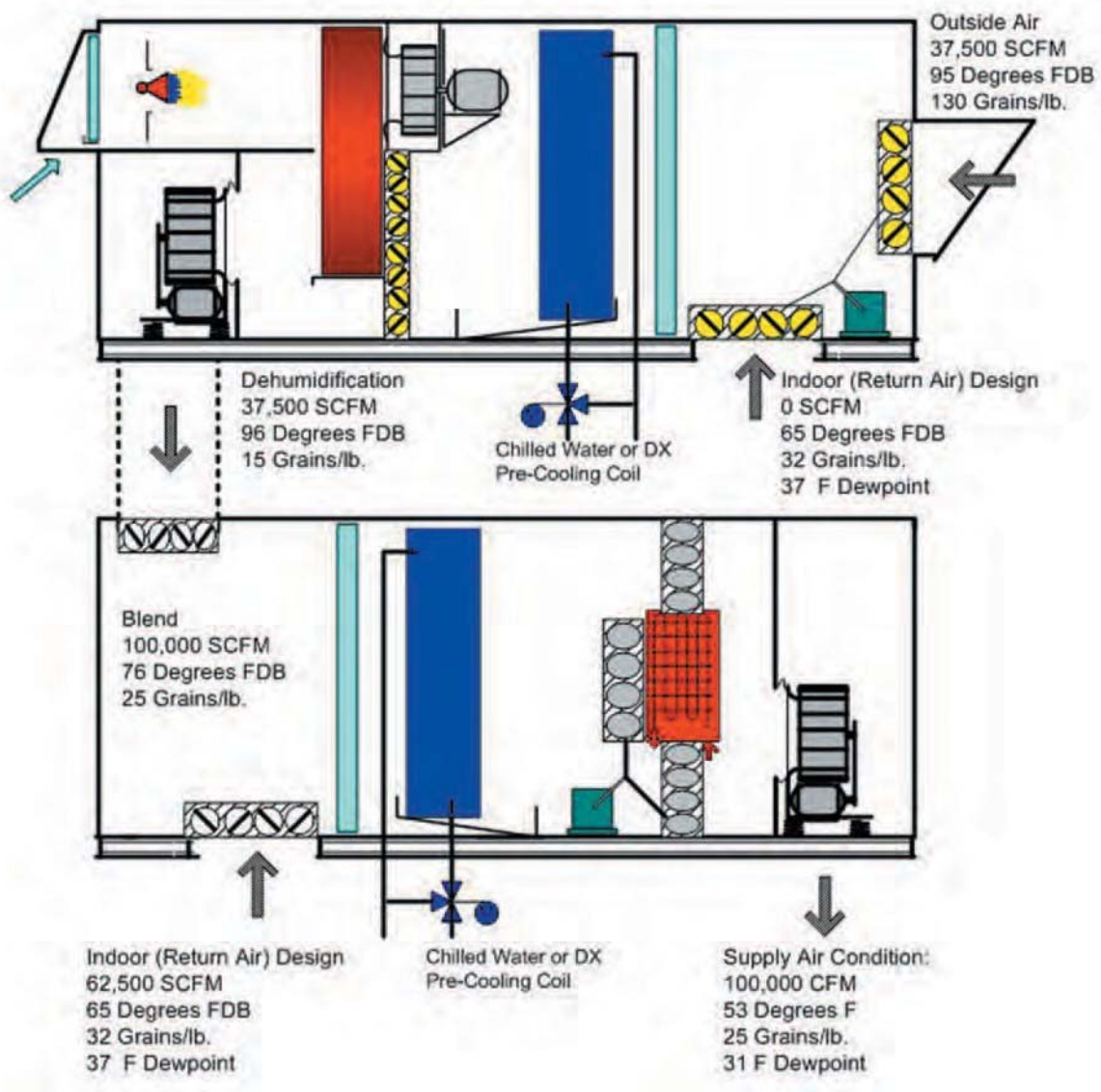

Figure 5 - Desiccant Outdour Air Unit

Energy Usage:

DH Precooling	3,737,000 BTUH 311 Nominal Tons
AHU Post Cooling	2,786,000 BTUH 232 Nominal Tons
Total	543 Actual Tons

Desiccant Gas Reactivation 2,113,000 BTUH at 99.5 % Efficiency

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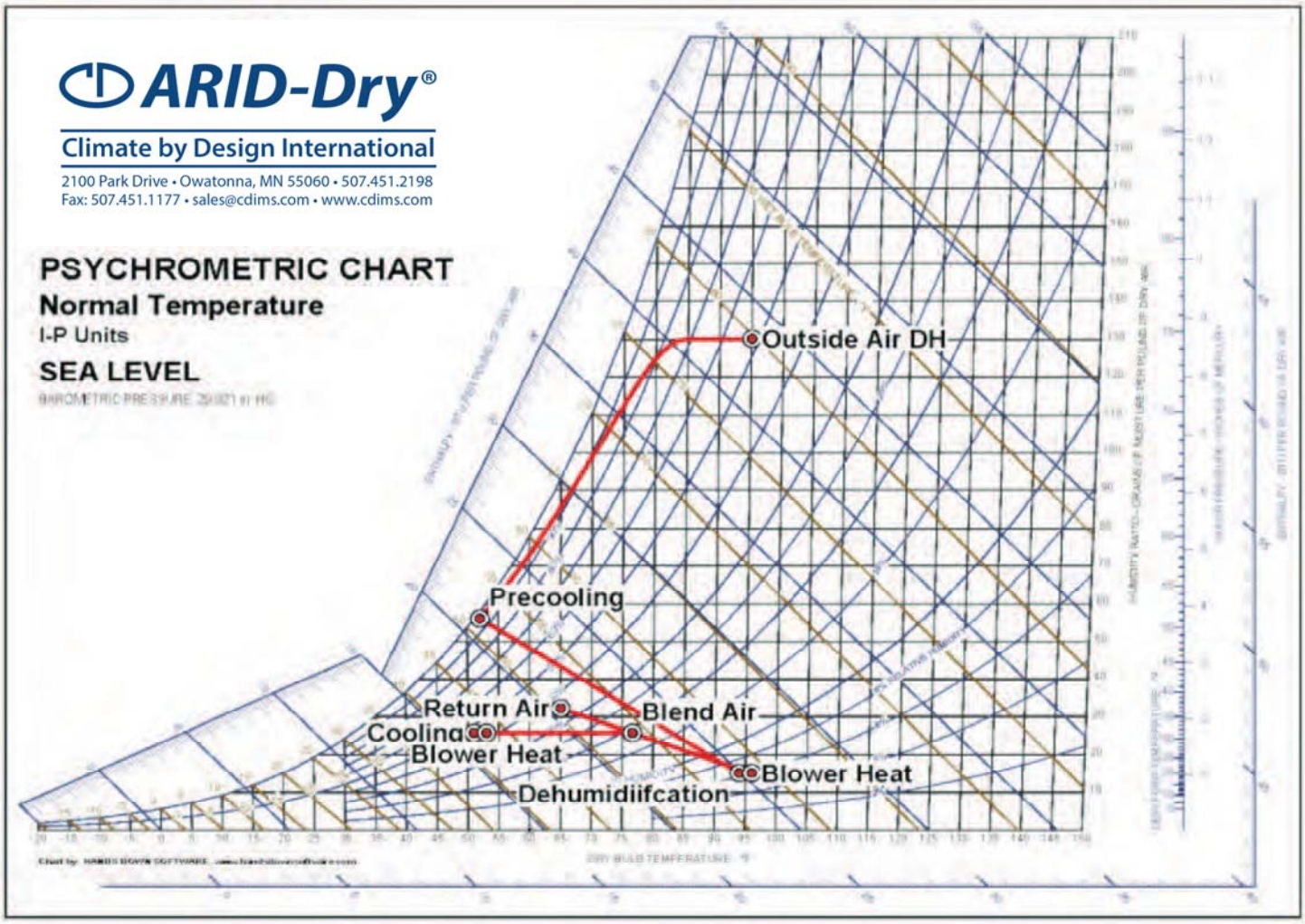

Figure 6

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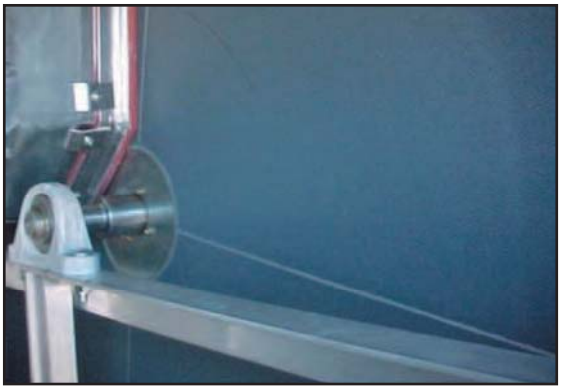
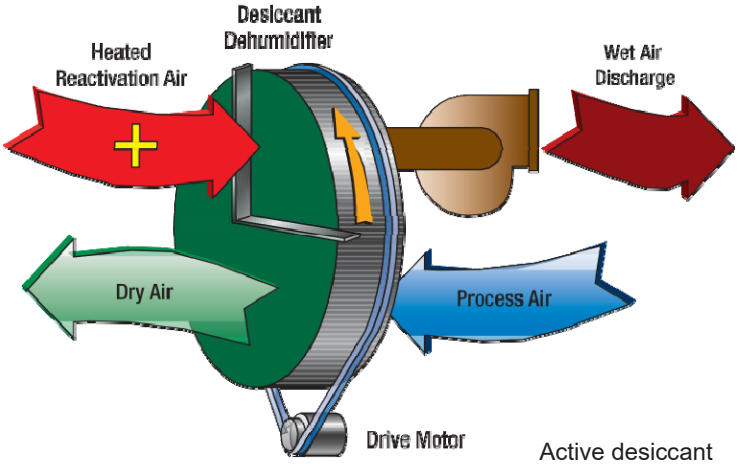
Summary:

By comparing the two systems it becomes apparent the ARID-Ice by Climate by Design International (CDI) desiccant system can maintain a lower humidity level while using less energy. Because the capacity is based on a fully occupied arena the pre-game condition of 65º F at 35 % RH (32 Grains/Lb) is within the part load capability of the equipment requiring only one or two of the quads to operate in a dehumidification mode. With the ability to limit outdoor air with partial spectator loading the capacity of the desiccants can easily be modulated to meet the exact requirements of the arena.



Enhanced silica gel desiccant dehumidification rotor



Active desiccant

It is also apparent that additional pumping HP will be used for the low temperature refrigeration. Designing for a 33 ºF leaving air temperature with 32ºF chilled water requires special consideration. The chillers need to be de-rated to provide the colder than standard fluid temperatures. Pumping horsepower will rise with the small differential. A 33ºF leaving air temperature form the cooling coil is impractical for continuous application. Frost or ice on a cooling coil will reduce air flow and cooling performance.

Utilizing the desiccant dehumidifier we can use standard temperature water (42ºF to 45 ºF) with no glycol required in most southern applications. The humidity level can also be easily modulated by a simple set point adjustment for different events using a face and bypass capacity control on the active desiccant rotor section. The main savings come from eliminating approximately 260 Tons per Quad for a total of 904 Tons. This coupled with the high pumping capacity for close the close approach cooling air / inlet chilled water, provides significant additional savings. Climate by Design International (CDI) have provided desiccant dehumidification solutions for over 300 recreational and spectator arenas through North America. We continue to apply active desiccant dehumidification equipment for this and many other industrial projects.

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