

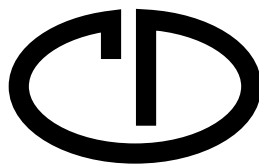
Ice and easy

Desiccant dehumidification systems



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A special reprint for



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Desiccant dehumidification systems

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Meeting the new NHL requirements to improve conditions in large spectator ice arenas should be simple using the right systems

The impact of water vapor on ice surfaces has been a definite target for the recreational arena, that is those with less than 2,500 seats. These venues have used desiccant dehumidification to maintain optimum humidity levels for the best ice quality, while maintaining an energy-effective environment. Spectator arenas, with 2,500 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 70°F at 50 per cent RH (relative humidity). Recently the National Hockey League (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 65°F at 35-40 per cent RH conditions to optimize the ice conditions.

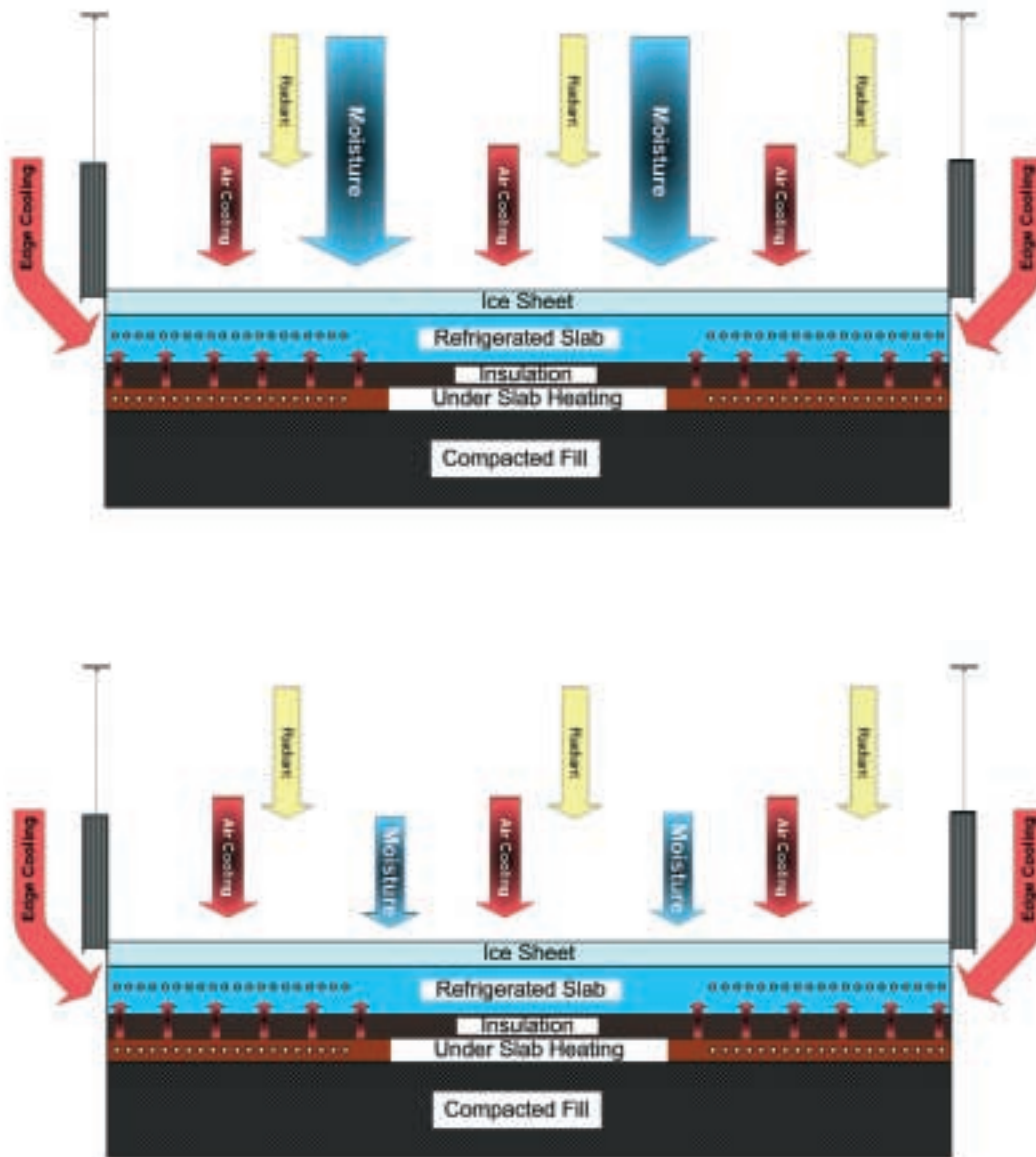
The recreational market, involving less than 2,500 seats, has used desiccant dehumidification exclusively to maintain low dewpoints. In the small rink market it is usual to allow the rink surface to cool the arena. Many arenas run with ambient temperatures of 55°F or lower. With the Arid-Ice gas-fired desiccant unit from Concepts & Designs, rink dewpoints of 35°F can be actively

maintained. 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

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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



[Top] [Figure 1] 70°F at 50 per cent. Large latent load

[Bottom] [Figure 2] 65°F at 40 per cent. Reduced latent load

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.

The cooling energy provided to the ice sheet is absorbed by three energy sources (see Figures 1 and 2, left): 15 per cent of the energy is going to the insulated under floor heating system; 10 per cent to the edge of the slab; 40 per cent is cooling the air boundary layer (dependent on radiant or lighting loads) above the ice sheet; and the remaining 35 per cent energy is freezing water vapor out of the air.

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 the energy required by the ice surface can be reduced to allow the surface to maintain a better ice condition.

Refrigeration vs desiccant

To properly maintain humidity levels it is necessary to estimate moisture loads during occupied events and determine the outside air requirements. The example (on p.188) is of a 20,000 seat arena with a mechanical room in each quadrant serving approximately 25 per cent of the space.

This example compares a refrigeration system designed to maintain 65°F at 40 per cent RH (36.6 grains of moisture/pound of dry air) with a desiccant system designed to maintain 65°F at 40 per cent (36.6 grains of moisture/pound of dry air). Each system provides a 12°F and 8-grain depression to accommodate internal loads. For normal application each quad will use 100,000 SCFM (standard cubic feet per minute) @ 70°F total air with 37,500 SCFM of outside air based on 7.5 SCFM per person. The total arena will use

or refrigeration system that cools the ice must deliver refrigerant cold enough to provide an ice surface temperature of 20-24°F. Generally this brine is circulated at 10-16°F 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 the required temperature of the supply brine and the larger the equipment and 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.

Best possible surface

In the recreational rink market the dewpoint of the rink is maintained at or below 32°F. This will provide an indoor condition of 55-60°F at 40-35 per cent RH. The NHL has now issued requirements that are similar to the recreational

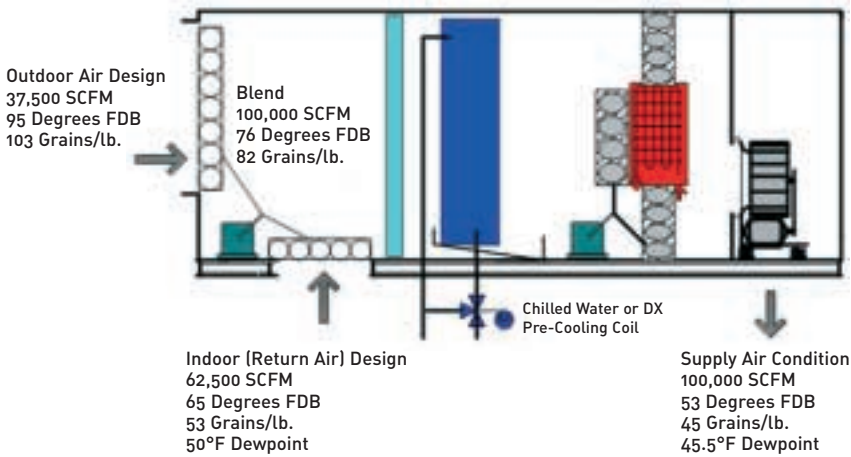
market. It has requested a 60°F at 35 per cent pre-game condition and a 65°F at 45 per cent post-game condition. By maintaining a controlled humidity the ice surface can be optimized to provide the participants the best possible ice surface for the event. It is not

In the small rink market it is usual to allow the rink surface to cool the arena

surprising that these conditions are now recommended, with most of the professional player practice taking place in a recreational rink that is properly controlled.

If the humidity levels rise high enough the capacity of the rink refrigeration equipment

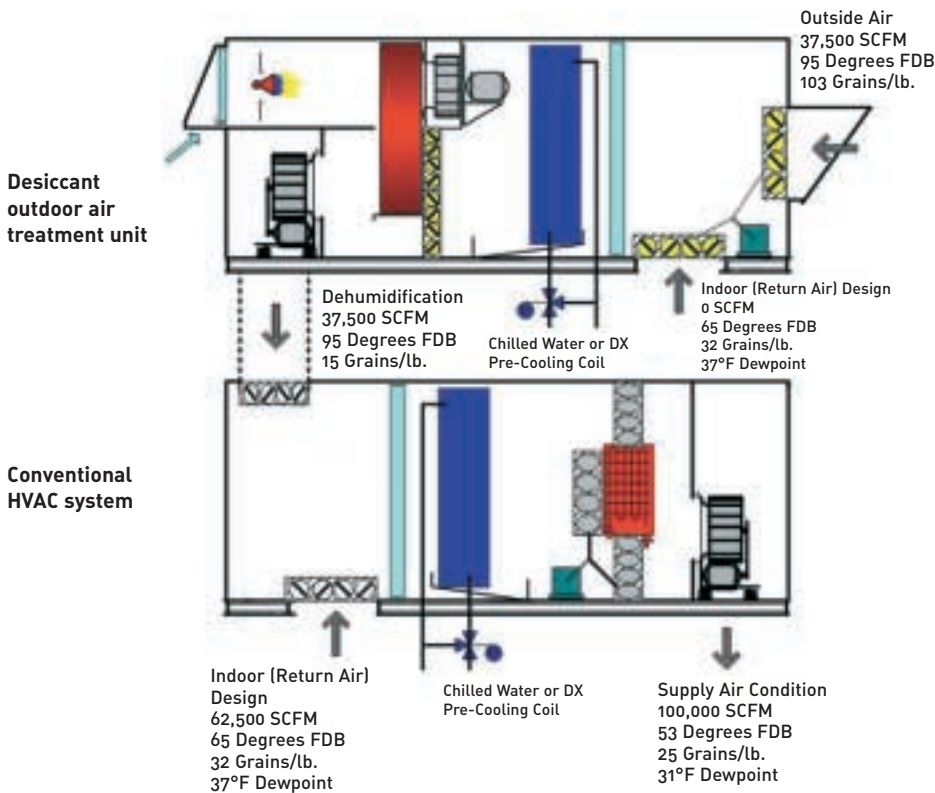
CONVENTIONAL REFRIGERATION/REHEAT



Heating and cooling energy usage
 Refrigeration at a 33°F dewpoint supply condition. (Not practical for continuous supply.)
 7,670,839 Btu/h – 639 nominal tons
 De-rate equipment: 12 per cent (EST) for 32°F discharge
 De-rate equipment: 7.5 per cent for 20 per cent EG freeze protection
 Total: 769 actual tons

Reheat Coil: 1,849,586 Btu/h
 De-rate 80 per cent boiler efficiency
 2,219,503 Btu/h

DESICCANT COMBINATION SYSTEM



Energy usage
 DH precooling
 3,737,000 Btu/h – 311 nominal tons
 AHU post cooling
 2,786,000 Btu/h – 232 nominal tons
 Total: 543 actual tons

Desiccant gas reactivation
 2,113,000 Btu/h at 99.5 per cent efficiency

approximately 400,000 SCFM with 150,000 SCFM of outside air. The system below would be located in each quad. The diagrams here show dewpoints and grains/lb in lieu of relative humidity, because relative humidity can be confusing with multiple temperature changes.

Energy saving

By comparing the two systems it becomes apparent the Arid-Ice 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 65°F at 35 per cent RH (32 grains/lb) is within the part load capability of the equipment, requiring only one or two of the quads to operate in 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.

It is also apparent that additional pumping horsepower will be used for the low temperature refrigeration. Designing for a 33°F leaving air temperature with 32°F chilled water requires special consideration. The chillers need to be de-rated to provide the

The humidity level can also be easily modulated by a simple set point adjustment for different events

colder-than-standard fluid temperatures. Pumping horsepower will rise with the small differential. A 33°F leaving air temperature from the cooling coil is impractical for continuous application. Frost or ice on a cooling coil will reduce air flow and cooling performance.

Using the desiccant dehumidifier we can use standard temperature water (42-45°F) 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 the close approach cooling air/inlet chilled water, provides significant additional savings.

Concepts and Designs have provided desiccant dehumidification solutions for over 300 recreational and spectator arenas throughout North America and continue to apply active desiccant dehumidification equipment for these and many other industrial projects. ■