DEVELOPMENT OF A PROCEDURE FOR THE ASSESSMENT OF THE PERFORMANCE OF DESICCANT-EVAPORATIVE COOLING SYSTEMS

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ABSTRACT

Decreasing energy costs and reducing CO_2 emissions are presently major concerns. As an alternative to the conventional vapour compression technology for space cooling, evaporative cooling is an attractive method to reduce energy consumption. However evaporative cooling alone is unable to address latent cooling loads; the combination desiccant dehumidification/evaporative cooling is a promising solution for both sensible and latent cooling.

Previous work by the authors developed desiccant-evaporative and evaporative air cooling algorithms that were incorporated into the building energy simulation software DOE-2.1E. This work showed for a humid cooling season, represented by Ottawa, Canada; sufficient cooling capacity is available to maintain 23 °C and 70% relative humidity for most of the cooling season. New work is presented that expands on the previous in refining the desiccant-evaporative and evaporative cooling system models and extending the analysis to three distinct climate types. Furthermore, complete energy and economic analyses are presented that determine the potential of this low energy cooling technology.

The energy consumption for the combined cooling technologies desiccant dehumidification/evaporative cooling, comes from thermal energy for the desiccant material regeneration and electricity for the fan operation, particularly in indirect evaporative cooling. Evaporative cooling consumes also a lot of water. The capital and operating costs of desiccant/evaporative cooling technology are for each case calculated and compared to those of a conventional packaged rooftop direct expansion cooling system.

In drier climates, represented in this study by Calgary Canada, evaporative cooling only is capable of meeting the building cooling load at significant energy and cost savings. In more humid climates, represented by Halifax and Ottawa, desiccant dehumidification, applied upstream of the indirect and direct evaporative coolers, can maintain acceptable temperature and humidity conditions in the building. The energy required is thermal energy and may come from different sources (e.g. gas, thermal wastes, solar energy) while the direct expansion system works with electrical energy only. Thus, CO_2 emissions as well as operating costs for the desiccant/evaporative cooling system may be reduced depending on the thermal energy source. The cost analysis reveals that the weakness of the desiccant dehumidification technology is the high cost of desiccant wheels compared to conventional direct expansion cooling coils. Presently, several R&D projects are being performed in advanced desiccant materials to improve desiccant wheel performance and decrease their cost.

KEYWORDS

Evaporative cooling, desiccant dehumidification, desiccant cooling.

INTRODUCTION

In a study conducted in the late eighties by Alberta Energy at the University of Lethbridge (Safronek, 1988), direct evaporative cooling was used in a retrofit to reduce chiller operation time. It concluded that direct evaporative cooling should be considered in commercial buildings in dry climates such as southern Alberta. When direct evaporative cooling is combined with desiccant cooling the range of climates and cooling loads that could be satisfied is greatly expanded. Desiccant cooling has been under investigation as an alternative to conventional direct expansion coils and as a method to enhance evaporative cooling systems in climates where outdoor humidity does not allow acceptable control of indoor humidity levels.

Waugaman et al. (1993), outlined the following advantages of desiccant cooling:

1. Only air and water are required as working fluids. Fluorocarbons are not required; thus, there is no impact on the ozone layer.

2. Significant potential for energy savings and reduced consumption of fossil fuels. The electrical energy requirement can be less than 25 percent of conventional refrigeration systems. The source of thermal energy can be diverse (i.e. solar, waste heat, natural gas).

3. Indoor air quality is improved due to the higher ventilation rates and the capability of desiccants to remove airborne pollutants.

4. Since desiccant systems operate near atmospheric pressure, construction and maintenance are simplified.

5. Desiccant cooling systems can supply heating, thus eliminating the need for a separate furnace for space heating in the winter season.

This paper presents the energy and economic analysis performed by Kemp and Ben Abdallah [1998] in which the ability of evaporative and desiccant-evaporative cooling systems to create comfortable working conditions in medium sized office building was evaluated. This work continues that evaluation and determines the energy use and compares the economics to a conventional compression cooling system.

METHODOLOGY

Description of a simulated building

The modeled building is a two-storey open plan office building with a basement that is completely below grade and a total air-conditioned floor area of 4270 m². The total building UA-value is 1977 W/K. The building has been modeled with 41 thermal zones using DOE-2.1E computer simulation package. Detailed information on construction parameters, occupancy and lighting schedules of each of the zones is included in previous publications (Kemp et al. 1996, and Kemp and Ben Abdallah, 1998).

Desiccant-evaporative cooling simulation

The mathematical model and the computer algorithm of the desiccant and evaporative cooling processes are well documented in Kemp and Ben Abdallah, 1998. The algorithm has been incorporated into DOE2.1E simulation model by making use of the built-in functions of DOE2.1E. The DOE2.1E subroutines which have been modified to simulate the desiccant-evaporative cooling equipment are the DKTEMP and SDSF subroutines. The details on modifications and utilization of these subroutines are given in Kemp et al. 1996 and Kemp and Ben Abdallah, 1998.

Description of Modeled Climates

The weather data from three Canadian locations are used to evaluate the performance of the desiccant-evaporative cooling systems. The selected locations represent three typical summer climates as follows:

The Halifax climate can be described as cool and humid. The summer months are characterized by daytime temperatures in the low twenty degrees Celsius and high humidity of over 60%. The high humidity in Halifax increases the latent heat load on cooling systems because of the need for fresh air ventilation. The low temperatures mean that the absolute humidity values are not

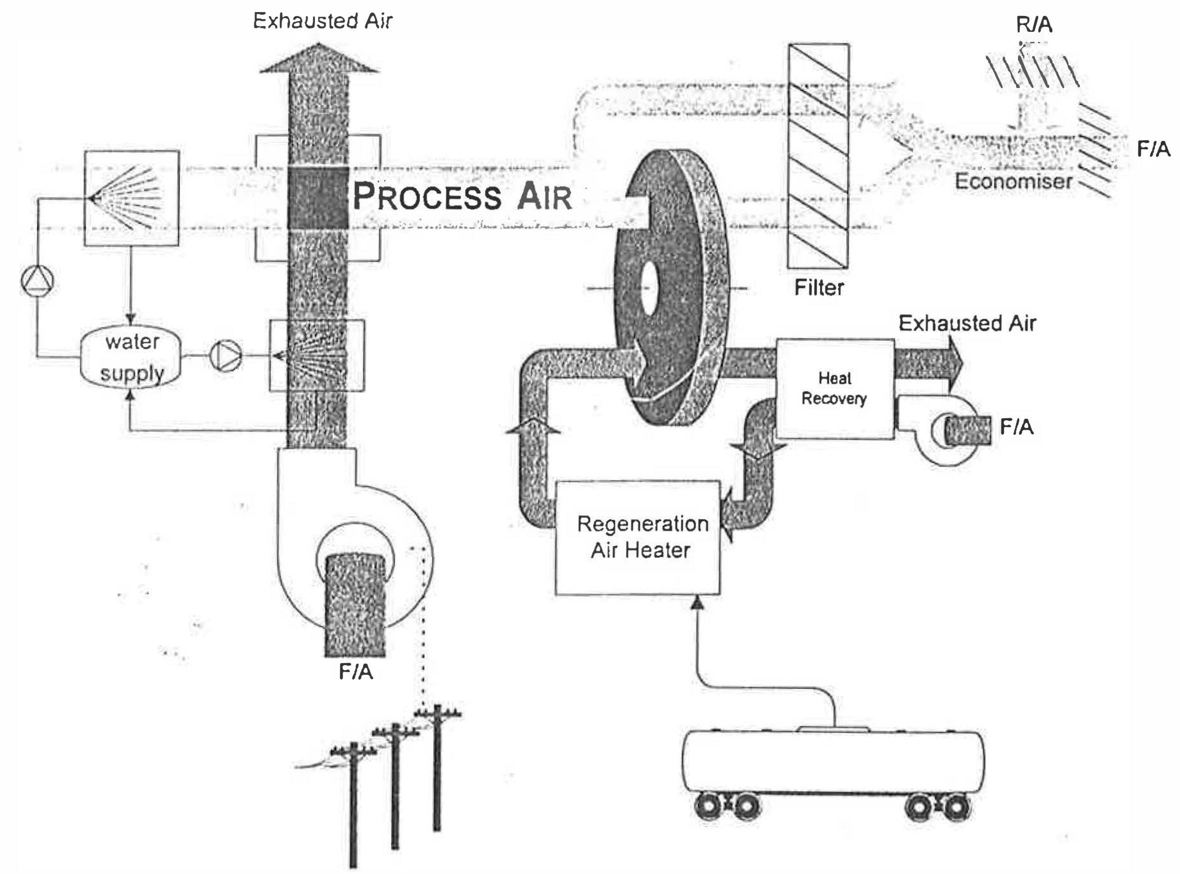

Figure 1 Desiccant-indirect-direct evaporative cooling system diagram (R/A = return air, F/A = fresh outdoor air)

excessively high despite the high values of relative humidity. The cooling season is prone to occasional very hot and humid days.

The Ottawa climate is representative of the central industrial area of Canada. Unlike Halifax, the high humidity is often combined with high temperatures and the summer cooling season is both hot and humid. The ventilation air often has higher absolute humidity than desired, increasing the latent cooling load on the air conditioning equipment.

The Calgary climate is representative of the Prairies region of Canada. Unlike Halifax and Ottawa, the cooling season is hot and dry. This is due to the easterly direction of most major weather systems. This brings air over the Rocky Mountains where the air is cooled and the humidity is precipitated out of the air.

Description of selected baseline system

A conventional compression refrigeration cooling system is used as a baseline case to which desiccant-evaporative cooling systems are compared. The baseline direct expansion system is sized by DOE-2.1E for each respective climate. DOE-2.1E determines the airflow rate to each zone, and the size of the cooling coil needed to supply the 15°C supply air on the maximum cooling load day calculated from the load results. In Halifax, this resulted in a system supplying 62,000 m³/hr or 4.8 air changes per hour of supply air through a 264 kW cooling coil. In Ottawa, this resulted in a system supplying 65,000 m³/hr or 5.1 air changes per hour of supply air through a 378 kW cooling coil. Finally, for Calgary, this resulted in a system supplying 64,600 m³/hr standard air or five air changes per hour of supply air through a 270 kW cooling coil.

Description of the Desiccant/Indirect/Direct Evaporative Cooling System

A system diagram showing the major components of the combined desiccant-evaporative cooling system is shown in figure 1. The corresponding psychrometric processes involved are schematically indicated in figure 2. The desiccant/indirect/direct evaporative equipment includes a rotating desiccant wheel, a direct evaporative cooler as well as an indirect evaporative cooler. The circulation of supply air consists of a mixture of return and outdoor air. The mixing ratio of fresh air and recirculated air is selected by the economiser. The ratio is dependent on the relative specific enthalpy of the outdoor air to the recirculated air as well as the fresh air requirement of the occupants. The utilities requirements of the system include gas for the regeneration of the desiccant wheel and water supplied to the indirect and direct evaporative cooling equipment. The various fans in the air handling system consume electrical energy.

The control logic of the model has been oriented to only perform desiccant dehumidification when it is needed to maintain the space relative humidity levels to below 60%. This is to limit the gas consumption needed for regenerationl. The need for desiccant dehumidification is determined by the humidity content of the process air after the economiser. The model attempts to reduce the process air temperature to at least 15°C and no cooler than 12.8°C to prevent the occupants' perception of drafts. If any particular cooling apparatus attains this goal then no further cooling will be done. For example, if the indirect evaporative cooler is able to cool the process air to 14°C then the direct evaporative cooler will remain inactive. However, if the humidity level of the building has been increased to above the allowed

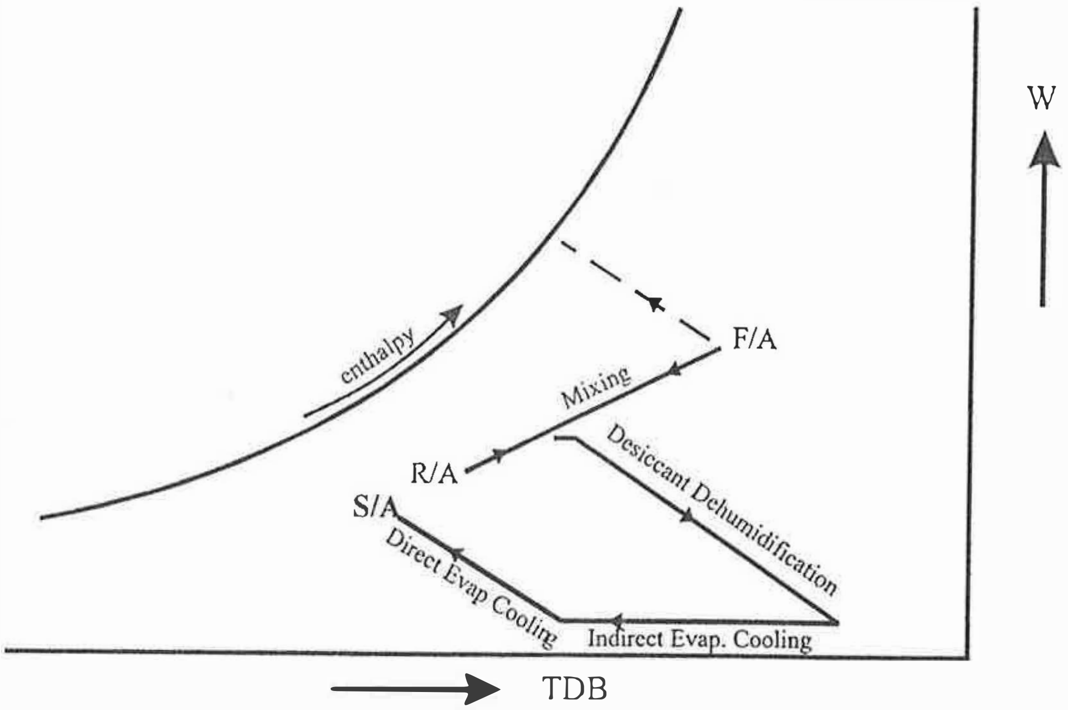

Figure 2 Psychrometric process for the Desiccant-indirect-direct evaporative system. (F/A = fresh air, R/A = return air, S/A = supply air, W = specific humidity, TDB = dry-bulb temperature)
Note: the dashed line represents the wet-side of the indirect-evaporative cooling process.

limits and desiccant dehumidification has not occurred, the cooling process will be remodelled to include dehumidification.

Cooling systems simulated

The performance of four evaporative and desiccant cooling systems has been investigated in the three above mentioned climatic regions. The investigated systems are:

- direct evaporative cooling,
- indirect evaporative cooling,
- combined indirect-direct evaporative cooling
- combined desiccant-indirect-direct evaporative cooling

The performance of these systems is compared to a baseline vapour compression cooling system. The capability of meeting the cooling loads of the building, the energy costs and the economic benefits are used as criteria of performance.

System Design Parameters Studied

Process air flow rate has a significant effect on the performance of evaporative and desiccant-evaporative cooling systems. For desiccant systems, the fraction of supply air dehumidified has also a direct impact on the performance and the economic viability of these systems. Dehumidification of a large fraction of the total supply air results in an increase in the wet-bulb depression. This increase makes the evaporative coolers more effective at reducing the air temperature thus increasing the cooling capacity of the system. In this study, the supply air flow rates are varied from five air changes per hour to a maximum of nine air changes per hour. For the desiccant systems, desiccant fractions of 0.25, 0.5 and 0.75 are tested.

RESULTS AND DISCUSSION

Comfort Performance

The five systems were tested under different flow rates and desiccant ratios for the desiccant systems. A decision was then made as to which systems deserved further analysis for use in the building for the respective climate. The following selected tables show the cooling load results for the systems modelled. The fraction of cooling hours that the relative humidity is above 60% and the temperature is above 25°C is tabulated for each system and climate. A more detailed analysis is presented in Kemp and Ben-Adallah [1998]. Cooling hours are defined as those hours in which cooling is required from the system. The temperature results are an average of the number of hours, above 25°C, for each zone in the model. The averaging gives equal weighting to all the zones.

The direct expansion baseline system provides the benchmark for occupant's comfort, as can be depicted in Table 1. In Halifax, the baseline system simulation reports 6.3% of the cooling hours with relative humidity above 60% and 0.4% of the cooling hours with a temperature above 25°C. At the same time, no zone shows temperatures above 27°C and the relative humidity is never above 70%. The Ottawa baseline system simulation reports 5.1% of the cooling hours with relative humidity above 60% and 2.1% of the cooling hours with a temperature above 25°C. At the same time, the zones show a negligible number of hours above 27°C (less than 0.8%) and the relative humidity is never above 70%. In Calgary, the direct expansion system simulation reports 0% of the cooling hours with humidity above 60% and 0.3% of the cooling hours with a temperature above 25°C.

In Halifax and Ottawa, the commercial building studied indicated that the combined effects of the building internal latent cooling load and the humid outdoor conditions combine to rule out the use of the non-desiccant evaporative cooling systems. The non-desiccant evaporative cooling systems are inherently incapable of providing latent cooling to the process air. Neither the direct nor the indirect-direct evaporative cooling systems are capable of providing acceptable relative humidity conditions in the building for more than 80% of the cooling hours. The building relative humidity exceeded 80% in the direct system and 70% in the indirect-direct system. The results for Ottawa are significantly less satisfactory with respect to the temperature.

	Location		
	Halifax	Ottawa	Calgary
Air Changes per Hour (standard air)	4.8	5.1	5.0
RH >60%	6.3%	5.1%	0.0%
Temp >25°C	0.4%	2.1%	0.3%

Table 1 Results for Baseline Direct Expansion System

In Calgary, all three of the non-desiccant evaporative cooling systems are capable of providing satisfactory comfort control. Therefore, the added expense and complexity of the desiccant systems cannot be justified. The indirect and indirect-direct evaporative cooling systems are capable of providing satisfactory sensible and latent cooling for the building in Calgary. The direct evaporative cooling system is marginal in its ability to provide latent cooling capacity in the building. This is due to a significant percentage of the cooling hours are spent above 60% relative humidity. In general, the relative humidity control decreases as the airflow rate increases.

In the buildings located in Halifax and Ottawa climates, the desiccant systems offer a better ability to provide comfortable conditions in the building. The desiccant-indirect evaporative cooling system provides exceptional relative humidity control, it needs however relatively high airflow to achieve acceptable temperature control. In general, for the desiccant-indirect evaporative cooling system, increasing the airflow rate increases the temperature control, while increasing the desiccant fraction or R_d increases the humidity control. The Ottawa system requires higher airflow rates and R_d values to provide similar comfort conditions as those in Halifax. In Halifax, the desiccant-indirect evaporative cooling system achieves acceptable comfort control at seven air changes per hour and an R_d value of 0.25. In Ottawa, the desiccant-indirect evaporative system cannot effectively control the building temperature. The desiccant-indirect-direct evaporative cooling system is required.

When a evaporative cooler is added to the Ottawa system, the cooling capacity increases further with marked improvement in systems. This airflow rate achieves the lowest energy consumpti on while achieving comfort for over 90% of the cooling hours for all systems. Direct evaporative cooling demonstrates the best energy performance by consuming approximately 61% less energy than baseline direct expansion system. The energy performance is however achieved at the expense of reduced cooling capacity or comfort performance. The relative humidity is often above 60%. The indirect and direct evaporative cooling systems achieve similar comfort levels to the baseline direct expansion system, however the energy consumption of the latter is nearly twice that of the direct evaporative cooling system.

The indirect and indirect-direct evaporative cooling system's increase in energy consumption over the direct evaporative system is mainly due to the electric humidifier. The indirect evaporative system is capable of meeting most of the sensible cooling needs of the building. It cannot however humidify the dry supply air. The additional direct evaporative cooler in the indirect-direct evaporative cooling system is therefore seldom used. This can be seen in the water consumption comparison for the indirect and direct evaporative coolers of the indirect-direct system. The lack of direct evaporative cooling also means that little moisture is added to the supply air through indirect-direct evaporative cooling. In contrast the direct evaporative system is unable to maintain the building relative humidity below 60%. This suggests that a better control strategy could both reduce the use of the electric humidifier for the indirect-direct system, while at the same time reduce the occurrences of high humidity for the direct evaporative cooling system.

Energy Category	Cooling System			
	Baseline DX	Direct Evaporative 5 AC/H	Indirect Evaporative 5 AC/H	Indirect -Direct Evaporative 5 AC/H
Electric Cool (MWh)	22.7	0	0	0
Vent Electric (MWh)	12.7	11.4	13.3	12.8
Gas Cooling (MWh)	N/A	N/A	N/A	N/A
Water Consumed (m³)	N/A	67.4	115.3	121.1
System Site Energy (MWh)	74.4	29.0	56.6	51.5
System Source Energy (MWh)	198.8	69.9	149.9	135.1

Table 5 Energy and Water Consumption in Calgary (Selected Systems)

In Halifax, as noted in the previous section, the cooling load/comfort performance of the indirect and direct evaporative cooling systems were not acceptable for the office building application studied, therefore an energy analysis was not under-taken. The combined indirect-direct cooling system was margin-ally acceptable, depending on the comfort penalties that the designer is willing to accept. As may be expected the energy consumption of the indirect-direct evaporative cooling system in Halifax is very low,(Table 6) as there is neither desiccant dehumidifier nor compressor present, neither gas nor electricity is consumed. The site energy values are between 32% and 50% of the baseline system depending on the airflow rate used.

In Halifax, the desiccant-indirect and desiccant-indirect-direct evaporative cooling systems have nearly identical energy consumption for equivalent cooling effect. However, due to its lower supply air temperatures, the desiccant-indirect-direct system is capable of meeting the cooling loads of the building with reduced airflow rates. The energy consumption of the desiccant systems in Halifax, show an improvement in source energy consumption over the baseline direct expansion system for nearly all parameters tested. The source energy savings for the desiccant-indirect-direct evaporative cooling system range from 28% savings for five air changes per hour and R_d of 0.25 to 9% savings for six air changes per hour and R_d of 0.5.

The desiccant-indirect evaporative system in Halifax achieves similar comfort levels at seven and eight air changes per hour. The energy savings come at the expense of reduced temperature and humidity control in the building when compared to the baseline system. Climate conditions are reasonable for human sedentary comfort. Table 6 shows the energy consumption of selected systems.

Energy Category	Cooling System			
	Baseline DX	Indirect-Direct Evaporative 5 AC/H	Desiccant-Indirect Evaporative 7 AC/H R_d=0.25	Desiccant -Indirect -Direct Evaporative 5 AC/H R_d=0.25
Electric Cool (MWh)	22.9	0	0.1	0.1
Vent Elec. (MWh)	12.5	6.5	19	16.3
Gas Cool (Mwh)	N/A	0	15.5	23.4
Water Consumed (m³)	N/A	64.2	66.7	74.3
Sys Site Energy (MWh)	44.4	14.3	44.3	47.6
Sys Source Energy (MWh)	122.6	34.5	90.5	88.4

Table 6 Energy Consumption Summaries of Selected Systems for Halifax

When tested under the hot and humid cooling season of Ottawa, the non-desiccant evaporative cooling systems were not able to provide adequate comfort level in the building. As shown in Table 7, the desiccant-indirect evaporative cooling system is capable of saving significant amounts of energy when compared to the direct expansion system. However, the airflow rates required to provide adequate temperature control penalize the energy savings. For the desiccant-indirect system at nine air changes per hour and a desiccant fraction of 0.25, the energy consumed is 9% greater than for the direct expansion system and the source energy consumed is 25% less. The desiccant-indirect-direct evaporative cooling system show a 65% increase in site energy, or a 11% decrease in source energy consumption for acceptable comfort performance using six air changes per hour and an R_d of 0.5. The greater latent and sensible cooling loads placed on the system by the Ottawa climate put greater demands on the regeneration requirements of the desiccant wheel. It is therefore, left to the economic analysis to determine if the lower cost of energy for natural gas is sufficient enough to create an economic advantage for the desiccant-indirect-direct evaporative cooling system over a more conventional direct expansion system.

When the desiccant-indirect evaporative cooling system is compared to the desiccant-indirect-direct evaporative cooling system the gas consumption of the system is greater than the desiccant-indirect system. The increased gas consumption is the result of two factors.

1. The direct evaporative cooler and the desiccant dehumidifier are in conflict. The humidity that is added to the air stream by the evaporative cooler might have to be removed by the desiccant wheel depending on the latent load in the building. Thus, the desiccant wheel is on for longer periods.

2. The maximum temperature parameter, $Tmax$ allows the use of dehumidification to increase the wet bulb depression of the process. This benefits the direct evaporative cooler, allowing it to achieve lower air temperatures at the expense of increased gas consumption.

Category	Cooling System		
	Baseline DX	Desiccant -Indirect Evaporative 9 AC/H R_d=0.25	Desiccant -Indirect -Direct Evaporative 6 AC/H R_d=0.50
Electric Cool (MWh)	37.4	0.2	0.6
Vent Elec. (MWh)	14.6	24.7	19.1
Gas Cool (MWh)	N/A	30.4	75.6
Water Consumed (m^3)	N/A	146.4	159.9
Sys Site Energy (MWh)	65.5	71.3	108.4
Sys Source Energy (MWh)	191.5	144.3	170.2

Table 7 Energy Consumption Summaries of Selected Systems for Ottawa

The conflict between the direct evaporative cooler and the desiccant dehumidifier cannot be easily resolved. One approach that was tried was to increase the minimum supply temperature. If the minimum supply temperature is increased from 15°C to 17°C, the air leaving the direct evaporative cooler will be correspondingly less humid. The drawback to this approach is increased fan power consumption, as higher supply air temperatures requires more airflow to accomplish equivalent cooling. In addition, increased airflow has been shown to adversely effect the humidity control. This increased fan power cancels out any benefits from the use of the desiccant wheel.

Economic Analysis

The energy costs were evaluated at each location. Local utilities supplying gas, electricity and water were contacted and their rates for 1996 were obtained. The economic analysis of the direct expansion cooling system is given in Table 8. All costs in the economic analysis are in Canadian dollars. In Calgary the evaporative only cooling systems achieve satisfactory comfort conditions, meeting both the latent and sensible cooling loads of the building, without the added cost of the desiccant dehumidifier. In particular the direct evaporative cooling system, using five air changes per hour provides the best economic performance. Its advantage over the direct expansion system is due to the elimination of the electrical power needed for the compressor and reduced electrical power required for the ventilation fans. The indirect and indirect-direct evaporative cooling systems are unable to provide any appreciable long-term energy cost benefits despite energy savings of approximately 25% over the baseline. The significant cost of the large heat exchanger required by the indirect evaporative cooler increases the total present worth cost when compared to the direct evaporative cooling system. The economic summaries for Calgary are presented in Table 9.

The desiccant systems are not cost effective in Halifax or Ottawa, despite their lower energy consumption and the lower cost of natural gas. This is due to the high capital costs of the system which may be reduced with widespread use of desiccant-evaporative cooling systems. For Halifax, the cost of gas may also be reduced.

	Location		
	Halifax	Ottawa	Calgary
First Year Operating Costs			
System Electrical	4,538	3,945	5,720
System Gas	0	0	N/A
Total Cost (includes water)	**4,692**	**4,018**	**5,979**
Capital Costs			
Direct Expansion Equipment	102,000	130,500	102,000
Total Capital Costs	**176,797**	**206,148**	**181,380**
Amortized Costs			
Yearly Payment on Capital	18,690	21,763	19,174
First Year Fuel Savings*	0	0	0
Total Present Worth Cost	**244,167**	**263,845**	**267,232**

Table 8 Economic Analysis of Basline Direct Expansion System for Halifax, Calgary and Ottawa

Category	Cooling System			
	Baseline DX	**Direct Evaporative 5 AC/H**	**Indirect Evaporative 5 AC/H**	**Indirect-Direct Evaporative 5 AC/H**
Relative Humidity >60%	6.3%	20.4%	1.4%	6.6%
Temperature >25°C	0.4%	8.1%	9.7%	9.4%
First Year Operating Cost	5,979	2,111	4,539	4,082
Capital Cost	181,380	94,010	179,437	193,278
Yearly Payment on Capital	19,174	9,938	18,969	20,432
First Year Fuel Savings	N/A	3,868	1,440	1,898
Total Present Worth Cost	267,232	124325	244,612	251,883
Total Present Worth Savings	N/A	142,906	22,620	15,348

Table 9 Economic Summaries of Selected Systems for ~~Calgary~~. Present Economic Conditions

Impacts of Components Costs on Economic of Desiccant Cooling Technology

The capital costs of the desiccant systems can be broken down into their major components. Table 10 shows the major capital cost categories for the equipment and the likelihood of decreasing the cost of the equipment.

Equipment	Economic Status
Supply Fan	Not likely to reduce in price
Return Fan	Not likely to reduce in price
Indirect Evaporative Cooling Fan	Not likely to reduce in price
Desiccant Wheel	May decrease in price
Evaporative Cooler	Not likely to reduce in price
Indirect Heat Exchanger	May decrease in price
Heat Recovery Unit	May decrease in price

Table 10 Major capital cost elements for the Desiccant System

The desiccant-evaporative cooling systems evaluated utilize fans that are larger than the baseline system, as well as using three fans instead of two. Therefore, their cost is a significant portion of the total system investment. Thus, lower fan costs will improve the capital cost comparison with the baseline system. Fan technology is considered mature and major improvements to reduce the capital costs are unlikely.

The desiccant dehumidification wheel, while not an entirely new technology, is not a high volume product. Generally, those sold are for special applications where humidity control is of primary importance. Larger scale production and new development in desiccant materials and supports would result in a decrease in the cost of desiccant wheels.

The air-to-air heat exchanger, the indirect heat exchanger and heat recovery unit, are large-scale heat transfer devices. Heat transfer for two air streams on such a large scale, 38,000 to 68,000 m³/hr is not particularly common. Again, to accomplish this the air-to-air heat exchanger used for the indirect evaporative cooler was actually two smaller units. Higher levels of production for heat exchangers of this capacity may reduce their capital costs.

If for Halifax, we assume that the capital costs of the desiccant wheel and heat exchangers will decrease, and assume that gas prices will approach the Alberta and Ontario prices, (of approximately $5.00/GJ) a new economic calculation can be made. Through iterative calculations it was found that the break-even points when compared to the baseline system is a 77% reduction in capital cost for the desiccant-indirect system and 61% for the desiccant-indirect-direct system. The break-even point was calculated for the minimum acceptable comfort performance of the systems. For the desiccant-indirect system, this is seven air changes per hour and R_d of 0.25, for the desiccant-indirect-direct system, five air changes per hour and R_d of 0.25 as shown in Table 11 and Table 12.

Similarly, the results for Ottawa show the break-even point when compared to the baseline system is an 82% reduction in capital cost for the desiccant-indirect system and 78% reduction for the desiccant-indirect-direct system. The break-even point was calculated for the minimum acceptable comfort performance of the systems. This is nine air changes per hour with an R_d of 0.25 for the desiccant-indirect evaporative system and six air changes per hour and an R_d of 0.5 for the desiccant-indirect-direct evaporative system.

	Air Changes per Hour				
Category	5	6	7	8	9
Comfort/Cool Capacity					
RH >60%	0.0%	0.7%	1.4%	3.0%	3.4%
Temp >25°C	21.0%	13.5%	9.7%	7.5%	6.3%
Capital Costs					
Desiccant Wheel	24,730	26,934	29,137	31,341	33,544
Heat Exchanger	12,229	13,816	15,402	16,989	18,575
Evaporative Cooler	13,389	14,084	14,780	15,475	16,171
Total Capital Costs	**162,282**	**180,705**	**198,925**	**216,231**	**231,506**
Amortised Costs					
Yearly Payment on Capital	17,155	19,103	21,029	22,859	24,473
First Year Fuel Savings	1,869	1,713	1,541	1,385	1,129
Total Present Worth Cost	**202,815**	**223,477**	**244,162**	**263,712**	**282,660**
Present Worth Savings	**41,352**	**20,690**	**4**	**-19,546**	**-38,493**

Table 11 Economic Analysis of Desiccant-Indirect Evaporative Cooling System. R_d=0.25 Capital Cost Reduction of 77% and Gas Prices of $5.00/GJ for Halifax

	Air Changes per Hour			
	R_d=0.25		R_d=0.50	
Category	5	6	6	8
Comfort/Cooling Capacity				
RH >60%	6.6%	11.3%	1.4%	5.6%
Temp >25°C	9.4%	4.7%	7.2%	2.4%
Capital Costs				
Desiccant Wheel	42,114	45,866	60,875	80,056
Heat Exchanger	20,825	23,527	20,825	23,527
Evaporative Cooler	26,778	28,169	26,778	28,169
Total Capital Costs	**202,264**	**224,168**	**222,511**	**260,141**
Amortized Costs				
Yearly Payment on Capital	21,382	23,698	23,522	27,500
First Year Fuel Savings	1,774	1,561	1,540	1,344
Total Present Worth Cost	**244,167**	**269,121**	**267,777**	**308,212**
Present Worth Savings	**0**	**-24,954**	**-23,610**	**-64,045**

Table 12 Economic Analysis of Desiccant-Indirect-Direct Evaporative Cooling System. Capital Cost Reduction of 61% and Gas Prices of $5.00/GJ for Halifax

CONCLUSIONS AND RECOMMENDATIONS

In the dry cooling season of Calgary, the indirect-direct evaporative cooling systems can provide indoor climate control comparable to that obtained from the use of conventional compression refrigeration cooling systems. However, a more appropriate control strategy should be developed to provide better humidity control without resorting to supplemental air humidification during extreme dry conditions. Results of the energy consumption and economic analyses show that the application of the evaporative cooling technology results in excellent economic and environmental benefits to regions characterized by hot and dry summer weather conditions.

In the hot and humid cooling season of Ottawa, desiccant dehumidification is required prior to the evaporative cooling process to provide comfortable indoor conditions. Simulation results indicate that combined desiccant-evaporative cooling systems consume an equal amount of site energy than the compression refrigeration cooling systems (i.e. displacement of electric energy by thermal energy for desiccant regeneration). The use of desiccant-evaporative cooling systems can provide significant energy savings at the source. The reduction of energy consumption of the source should result in environmental benefits and lower fuel costs.

The current high capital cost of desiccant wheels represents a major constraint to application of desiccant-evaporative cooling in buildings. Future development in advanced desiccant materials, and heat and mass transfer equipment, should make this new HVAC technology economically viable.

In the warm and humid summer climate of Halifax, the indirect-direct evaporative cooling provides satisfactory environmental control in buildings. Poor environmental conditions are only experienced during peak load periods. The techno-economic feasibility of coupling desiccant-cooling with a compression refrigeration system to handle peak loads should be investigated. Desiccant-evaporative cooling systems can provide excellent indoor environment control; but, the present high installed costs of these systems make them economically non-viable.

ACKNOWLEDGEMENTS

This work has been supported by the Department of Natural Resrouces Canada (RNCan) and the Canadian Program on Energy Research and Development (PERD).

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