PERFORMANCE OF DESICCANT/EVAPORATIVE COOLING IN CANADIAN OFFICE BUILDINGS USING THE FUNCTIONS OF DOE-2.1E

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ABSTRACT

The past 15 years has seen much research into the benefits of desiccant and evaporative cooling. Stand alone evaporative cooling systems are seen as a viable alternative to traditional vapor compression cooling systems in dry climates, however it is less effective in high humidity climates where low wet bulb depressions limit its performance. High latent system loads is also a known deficiency of evaporative cooling systems. The inclusion of a desiccant dehumidifier before the evaporative cooler enables the cooling system to control latent cooling. The desiccant dehumidification process may be idealized as isenthalpic. The latent heat of the removed moisture increases the dry bulb temperature of the dried air and the wet bulb temperature of the idealized process remains constant. This paper studies and evaluates the performance of desiccant/evaporative cooling systems in a commercial building in Canada from the point of view of comfort for the building occupants using hourly building energy simulation software.

The simulation of the building heating and cooling loads as well as the effects of the desiccant-evaporative cooling system is performed using DOE-2.1E building energy usage simulation software. The incorporation of the model for desiccant/evaporative cooling is accomplished by using the functions of DOE-2.1E. The desiccant evaporative cooling system involves a rotary desiccant dehumidification wheel, an indirect evaporative cooler and finally a direct evaporative cooler. The energy consumed by this system includes the electricity necessary to run the ventilation fans, and a natural gas heat source used for regeneration of the desiccant material. The system also consumes water in the indirect and direct evaporative cooling components.

The study reveals that under climatic conditions, typical of the region of Ottawa, increased levels of ventilation are necessary when compared to conventional vapor compression system as equivalent supply temperatures (typically 13°C) are not achievable. The comfort conditions in the building are found to be acceptable when using elevated amounts of supply air. Typically the circulation rate in the building is increased between 150 and 175% from normal direct expansion coil ventilation rates to achieve satisfactory comfort conditions.

1. INTRODUCTION

This paper investigates the feasibility of combined desiccant and evaporative cooling for commercial office buildings in Canada. Due to environmental concerns and the ever increasing demand for space cooling there is a great need to develop low energy cooling systems that do not rely on ozone depleting refrigerants. The low energy cooling made possible by desiccant-evaporative cooling is due to the separate treatment of latent and sensible cooling loads.

In a study conducted in the late eighties by Alberta Energy at the University of Lethbridge [1], direct evaporative cooling was used in a retrofit to reduce chiller operation time. It was concluded that direct evaporative cooling should be considered in commercial buildings in dry climates such as in southern Alberta. When 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 humidity levels do not allow acceptable control of indoor humidity levels.

The ability of desiccant and evaporative cooling to separately control humidity and air temperature is a key to its energy and comfort benefits. Conventional direct expansion coils must cool the process air to below its saturation temperature to remove any appreciable amounts of moisture. This process is inherently coupled to simultaneous sensible cooling which may reduce the dry bulb temperature to below comfort standards. Thus the air must be reheated using heating coils, an energy wasting process. Desiccant-evaporative cooling therefore has the benefit of maintaining comfortable humidity levels in the building space without reheat. As an added health benefit, the desiccant wheel is capable of removing airborne contaminants as noted by Meckler [2].

Waugaman et al. [3], outlined the major advantages of desiccant cooling as:

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, maintenance and construction are simplified.

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

The first patented desiccant cooling cycle was the Pennington cycle (cited by Waugaman et al. [3]) or ventilation cycle. It dates back to 1955 and is probably the most studied desiccant cooling cycle. An early variation of the Pennington cycle was the "recirculation cycle" [4]. The recirculation cycle varies from the ventilation cycle by recirculating the air from the conditioned space back through the dehumidifier and other cooling equipment. In this study the recirculation cycle is modified to include an indirect evaporative cooler in place of the sensible heat exchanger.

To this end, a computer simulation, DOE-2.1E, is used to evaluate the comfort performance and energy consumption of the desiccant-evaporative cooling system. The scope of the study is restricted to medium sized commercial buildings. A commercial building typical for Canada has been used for this study. The meteorological data used to calculate the building's energy usage were for Ottawa, Ontario The investigated cooling system combines desiccant, indirect and direct evaporative cooling.

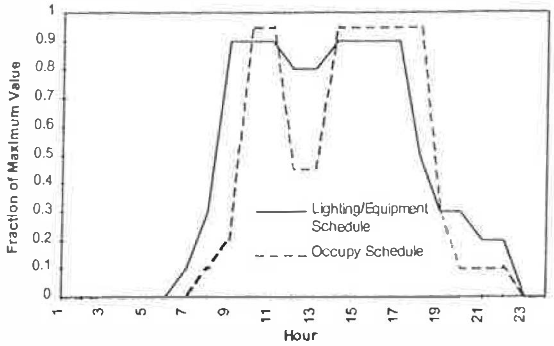


Figure 1. Building Occupancy and Lighting Schedules

The intent of this report is to determine if this system is capable of meeting the cooling loads of the building with respect to occupant comfort. The criteria of occupant comfort will include the temperature and relative humidity that will be maintained in the space.

2. THE BUILDING

The building of interest is a two-story open plan office building with a basement that is completely below grade and a total volume of 17432 m^3. The total building floor area is 4417 m^2 of which 4271 m^2 are serviced by the desiccant system, the remaining 146 m^2 are either uncooled, e.g. the mechanical rooms, or controlled by the independent air-conditioning systems of the entranceways. Of this area, the office spaces were modeled with an occupant density of one person per 23 m^2 plus suitable occupancy values for other

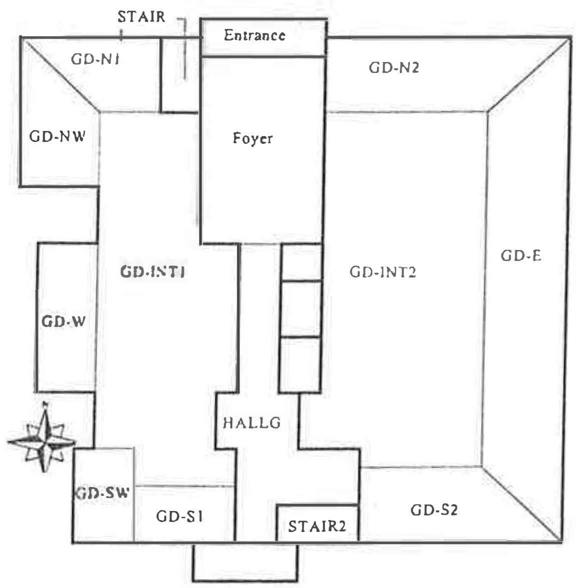


Figure 2. Simulation Building Ground Floor Zoning

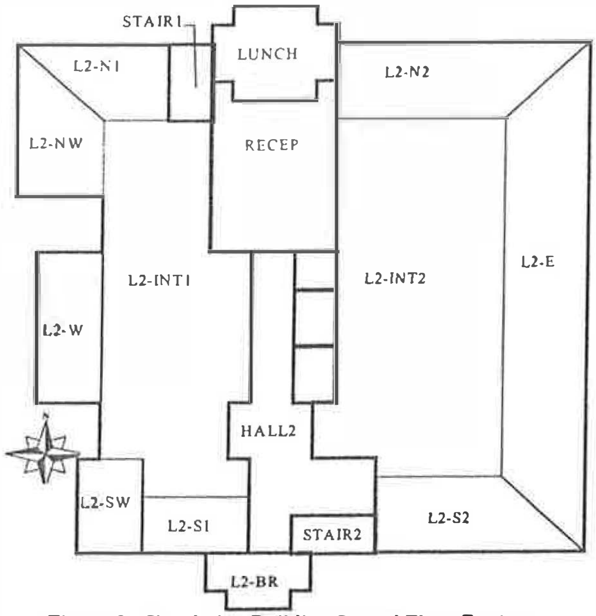


Figure 3. Simulation Building Second Floor Zoning

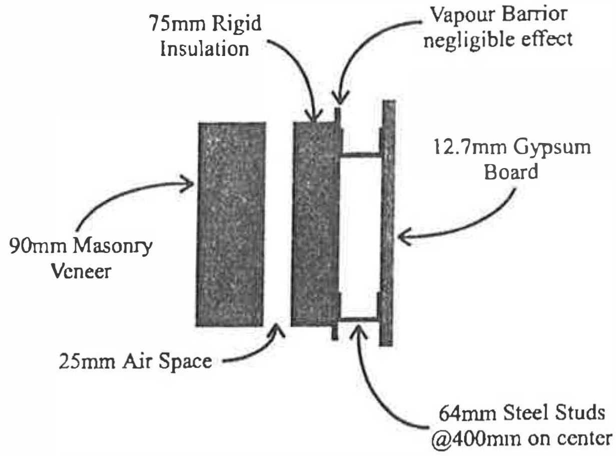
miscellaneous spaces such as foyers, stairwell etc. [5] The total building occupancy is 202 and is simulated according to the occupancy schedule shown in Figure 1. The office space is open plan and it was assumed to use 2 m high partitions to divide the space into individual workstation areas.

The building has been modeled with 41 zones; 36 zones are cooled by the central air handling system. Of these, 30 utilize thermostatic control while the remaining six zones do not. The zones without thermostatic control for cooling include those in the

Table 1. Building Envelope Insulation and UA values

	R-Value (m^2K/W)	Area (m^2)	UA Value (W/K)
Ext Wall			
Type I	3.26	1183.3	363.0
Type II	3.31	12.4	3.8
Type III	2.65	4.4	1.7
Window			
Type I	0.59	348.6	592.9
Type II	0.57	11.3	19.8
Roof			
Type I	3.15	1394.3	442.6
Type II	3.14	199.7	63.6
Type III	3.09	34.5	11.2
Below Grade Wall			
Type I	2.58	616.3	238.9
Type II	2.35	93.9	39.9
Slab	6.67	1335.2	200.2
Total Building UA Value			**1977.5**

Outside Walls



$$U-\text{value} = 0.307\ W/(m^2 \cdot K)$$

Figure 4. Typical Wall Construction for Simulation Building

below grade basement as well as the two stairwells. These zones receive ventilation air from the system at a constant air flow rate. The reception area on the top floor is open to the foyer below and to an atrium above. This open atrium extends to cover the lunch area as well. The zoning for the ground and second floors may be seen in Figure 2 and Figure 3.

The building total envelope area (including the underground slab and roof) is 5234 m^2 with 1200 m^2 exterior wall area and 360 m^2 (30%) fenestration. There is 1628 m^2 of roof surface area, most of which is flat with the exception of the vaulted section above the atrium space. The envelope UA value is 1978 W/K. A typical wall cutaway is shown in Figure 4. The building was assumed to be in a business district and therefore predominantly surrounded by asphalt. This theoretical building was modeled without any exterior shading from neighboring buildings.

3. PSYCHROMETRIC ANALYSIS OF DESICCANT COOLING CYCLE

The desiccant indirect-direct evaporative cooling cycle is driven primarily by thermal energy. Figure 6 shows a diagram of the Desiccant-Indirect-Direct evaporative cooling system and the psychrometric processes of the complete cycle. The process air entering the cooling equipment is either a mix of recirculated return air from the building and outside ventilation air, or only outside air. In the first case, the minimum amount of ventilation air is determined by the ventilation requirements of the building. In the second case, 100 percent outside air is supplied when the outdoor air value for specific enthalpy is below that of the return air leaving the conditioned zones. This ensures that the supply air upstream of the cooling equipment is at the lowest possible enthalpy value, thus minimizing the cooling required from the system.

If it is determined that dehumidification is needed, the desiccant wheel is used to reduce the absolute humidity level of the air. After dehumidification indirect evaporative cooling is used to sensibly cool the air stream. The final step is direct evaporative cooling. Direct evaporative cooling will only be performed if the air temperature is above the desired supply temperature and if the humidity level is lower than the maximum humidity level required to maintain comfort in the space.

4. THE DOE-2.1E ALGORITHM AND THE FUNCTION MODIFICATIONS

The DOE-2.1E algorithm used to calculate the supply air conditions is depicted in flow chart form in Figure 5 and definition of the variables are shown in Table 2. The DOE-2.1E subroutines modified to simulate the desiccant-evaporative equipment are the DKTEMP and SDSF subroutines. The DKTEMP subroutine is responsible for selecting the supply air deck temperatures to be supplied to the zones. The SDSF subroutine is responsible for calculating the energy used by the equipment as well as the final return air values of temperature and absolute humidity.

DOE-2.1E use of the hourly simulation format leads to the inconsistency of having to recalculate the systems cooling equipment performance. The return air temperature and humidity values are used to determine the mixed air properties entering the cooling equipment. The mixed air properties are in turn a function of the supply air properties and zone loading. Thus to accurately determine the air properties through out the system all variables need to be simultaneously evaluated. To limit the computation time, DOE-2.1E uses only a single iteration and uses the previous hour's values for the return air conditions to model the current hour. The error introduced is usually small. However, in some instances the DKTEMP-2 routine will select a lower air temperature than may be provided with the more accurately calculated air temperature received by the SDSF function. Conversely the SDSF routine may be able to provide a lower air temperature than was possible in the DKTEMP routine. The impact of this is relatively small and results in errors of up to 1°C in approximately five to seven percent of the simulated hours depending on the supply air flow rate.

The DOE-2.1E system specified is a modified Variable Air Volume (VAV) system. Before air is introduced to the spaces the simulation attempts to determine the supply air temperature which may be attained with the desiccant-evaporative cooling equipment. This is accomplished by inserting the function, DKTEMPDES, into the DOE-2.1E sub-routine at the DKTEMP-1 function call. The implemented desiccant and evaporative cooling routine determines the supply air temperature based on the entering air temperature and humidity values and the supply air conditions required to satisfy occupant comfort. The achievable supply air condition is dependent on the desiccant and evaporative equipment performance.

The empirical dehumidification equations used in both the DKTEMPDES and SDSFDES functions were derived for a 20,400 m^3/hr desiccant wheel with a face velocity of 4 m/s using performance data from Seasons•4[i]. Second order regression analysis of the Seasons•4 S4SG-60 performance curves yielded the amount of moisture removed from the process air. The indirect and direct evaporative coolers were modeled with effectiveness values of 0.6 and 0.9, respectively.

The return air from the space is a function of the supply air temperature, air flow rate, and the zone cooling load. The DKTEMP algorithm therefore uses the previous hour's return air value. The subroutine SDSF determines the final values of the mixed air more precisely to calculate the supply air again.

The DKTEMPDES function implemented at the DKTEMP-2 function call receives the values for the mixed air temperature and absolute humidity from the DKTEMP subroutine. Knowing the present hour's latent heat load and the previous hour's return air humidity value and supply air flow rate a maximum supply air humidity is calculated. Using the mixed air temperature and absolute humidity value the function decides whether the equipment will perform no cooling, use evaporative cooling only, or desiccant and evaporative cooling according to the following criteria:

- If the mixed air temperature leaving the economizer is below the minimum supply air temperature then no cooling will be performed.

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[i] Seasons•4, 4500 Industrial Access Road, Douglasville, Georgia 30130, (404)-489-0716, Fax (404)-489-2938

- If cooling is determined to be needed and if the mixed air absolute humidity leaving the economizer is greater than the calculated maximum supply air humidity desiccant dehumidification will be performed.

- Indirect evaporative cooling is performed if the air entering the indirect evaporative cooler from the desiccant wheel is not below 15°C

- Direct evaporative cooling is performed only after, and if indirect evaporative cooling has been performed. If the air temperature exiting the evaporative cooler is below the minimum supply air temperature (again 15°C) then cooling will not be performed below this temperature.

- Should the exiting humidity be above the maximum supply air humidity after the direct evaporative cooler and the desiccant wheel not be active, then the algorithm will return to the beginning of the loop with the stipulation that dehumidification must be performed.

With the supply air temperature determined by the DKTEMP subroutine and the implemented DKTEMPDES function the DOE-2.1E algorithm

Table 2. Description of variable used in Figure 5

DBT	Outdoor dry bulb temperature
WBT	Outdoor wet bulb temperature
HUMRAT	Outdoor absolute humidity ratio
PO	Economizer setting for fresh air ratio
PASTWR	Previous hours absolute humidity ratio
TRPAST	Previous hours dry bulb temperature
TMMIN	Minimum supply air temperature
TC	Supply air temperature leaving cooling equipment
TR	Return air temperature
WR	Return air absolute humidity ratio
TM	Mixed air temperature
WM	Mixed air absolute humidity ratio
WCOIL	Supply air absolute humidity ratio leaving cooling equipment

determines the VAV air flow setting for each zone and therefore the supply air flow rate as well as the resulting temperature change in the zone, if any. The sum of the air flows as well as the weighted return air temperature is then passed to the SDSF subroutine.

In the SDSF subroutine the energy requirement of the air handling equipment is calculated and the return air

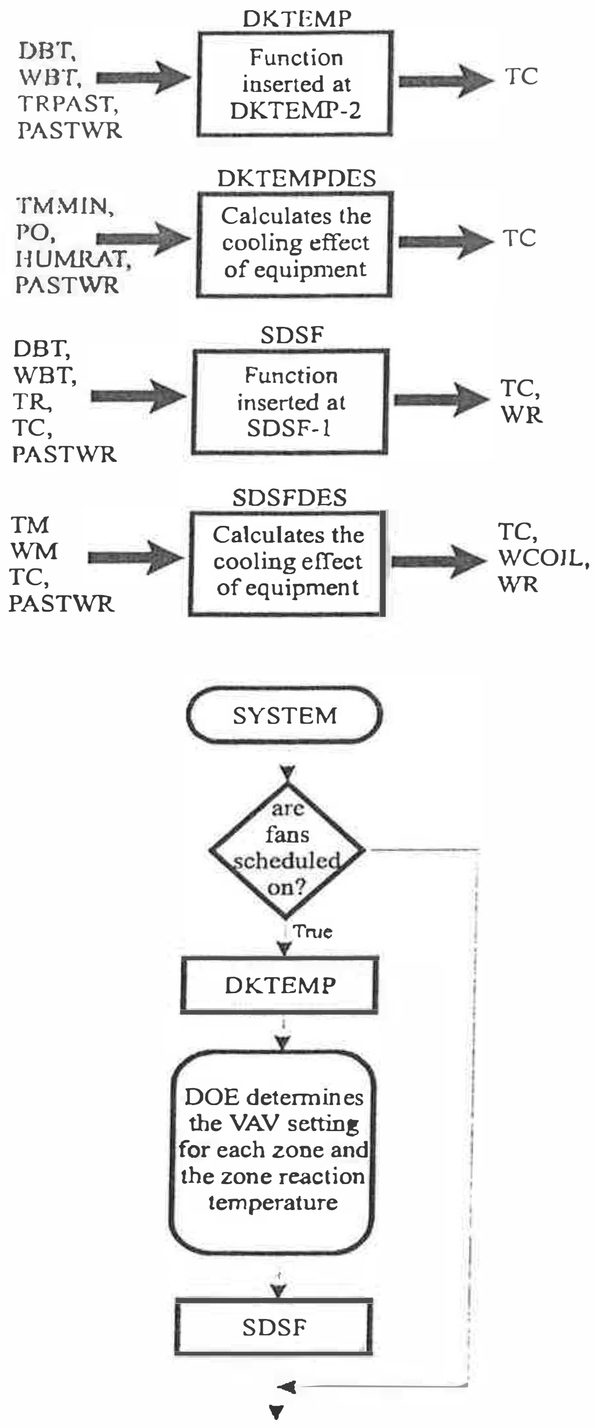

Figure 5. DOE-2.1EAlgorithm Flow Chart Diagram

humidity and temperature are determined. The updated values of the return air temperature and humidity may differ slightly from those used in the DKTEMP subroutine which uses the previous hour's values. As the building spaces have already been simulated as cooled with supply air conditions determined from the previous hour's return air properties, the recalculated

Table 3. Undercooled hours using Nominal Supply Air Flow Rate by Zone

Zone	May	June	July	Aug	Sept
			Month		
GD-N1	0	1	43	18	0
GD-WNW	0	5	47	21	3
GD-W	0	5	44	19	4
GD-SW	0	1	35	2	0
GD-S1	0	0	26	1	0
GD-INT1	0	1	40	11	0
GD-N2	0	0	40	15	0
GD-E	0	1	40	22	0
GD-S2	0	0	35	10	1
GD-INT2	0	0	33	6	0
HALLG	0	0	33	6	0
FOYER	0	0	29	1	0
L2-N1	0	1	43	13	0
L2-WNW	0	4	46	18	4
L2-W	0	4	45	16	4
L2-SW	0	1	36	2	0
L2-S1	0	0	28	1	0
L2-INT1	0	1	43	9	0
L2-N2	0	0	39	7	0
L2-E	0	1	40	16	0
L2-S2	0	0	36	7	0
L2-INT2	0	0	33	5	0
HALL2	0	0	35	6	0
L2-BR	0	19	74	83	17
LUNCH	0	0	36	9	0
Max	0	19	74	83	17
Min	0	0	26	1	0
Median	0	1	39	9	0
Mean	0.0	1.8	39.2	13.0	1.3

supply air conditions must obtain the same sensible cooling effect in the building. The fresh air ratio and the resulting mixed air temperature is determined using the present hour's value for the return temperature. The SDSFDES function inserted at the SDSF-1 function call uses the same logic as the DKTEMP-2. The supply air absolute humidity resulting from this calculation is then added to the moisture gain from the space to obtain the return air humidity.

5. TEST METHODOLOGY

The building was tested for four different air flow rates which were multiples of a nominal design air flow. The air flow multiples were 100, 125, 150 and 175 percent of the nominal air flow. The nominal design air flow was defined as the building design air flow rate using direct expansion coils and the standard supply air temperature of 13°C [6]. The size of the desiccant wheel was adjusted accordingly to maintain a face velocity of 4 m/s. The resulting supply air flows were:

Nominal: 84,494 m^3/h – 4.8 air changes/hr

Table 4. Undercooled hours using 150% Nominal Supply Air Flow Rate by Zone

Zone	May	June	July	Aug	Sept
			Month		
GD-N1	0	0	16	0	0
GD-WNW	0	0	18	0	0
GD-W	0	0	16	0	0
GD-SW	0	0	9	0	0
GD-S1	0	0	8	0	0
GD-INT1	0	0	15	0	0
GD-N2	0	0	15	0	0
GD-E	0	0	16	0	0
GD-S2	0	0	12	0	0
GD-INT2	0	0	12	0	0
HALLG	0	0	14	0	0
FOYER	0	0	10	0	0
L2-N1	0	0	16	0	0
L2-WNW	0	0	19	0	0
L2-W	0	0	18	0	0
L2-SW	0	0	10	0	0
L2-S1	0	0	8	0	0
L2-INT1	0	0	16	0	0
L2-N2	0	0	16	0	0
L2-E	0	0	16	0	0
L2-S2	0	0	12	0	0
L2-INT2	0	0	12	0	0
HALL2	0	0	16	0	0
L2-BR	0	0	30	0	0
LUNCH	0	0	15	0	0
Max	0	0	30	0	0
Min	0	0	8	0	0
Median	0	0	15	0	0
Mean	0.0	0.0	14.6	0.0	0.0

125%: 105,504 m^3/h – 6.0 air changes/hr

150%: 126,587 m^3/h – 7.3 air changes/hr

175%: 147,524 m^3/h – 8.5 air changes/hr

6. COMFORT LEVEL RESULTS

DOE 2.1E is able to establish temperature reports on two levels. First the temperature values of each zone are binned by temperature ranges and time of day, then the number of hours, by month, that each zone spends out of the throttling range of the system is reported as undercooled. In this simulation the thermostats are set at 23°C with a 2°C throttling range which would make reported zone temperatures that are above 25°C considered as undercooled. The results in Table 3 show that significant number of hours that are undercooled when using nominal air flow rates. Table 4 shows a significant improvement in the number of undercooled hours using 150 percent nominal over the nominal supply air flow rates.

In the DOE-2.1E simulation the maximum percentage of hours which any zone spends outside the throttling range is reported. The results with respect to the supply air flow rate are as follows:

Nominal Air Flow: 17%

125% Nominal Air Flow: 12%

150% Nominal Air Flow: 8%

175% Nominal Air Flow: 6%

These numbers show a significant increase in temperature control from nominal to 150% nominal air flow rates with a marginal increase as the air flow is increased from 150% to 175% of nominal. A typical zone from Table 4 (GD-INT1) is selected to show the effect of the four air flow rates on the temperature performance. The results for this zone are shown in Figure 7, which indicates that as the air flow rate increases the temperature control in the space also increases.

The temperature profile for all zones is depicted in Figure 8. This chart shows the strong grouping of zone temperatures in the 20°C to 24°C range when using 150% nominal supply air. The exceptions are the low temperatures in the LUNCH and RECEP zones that are overcooled. Using 150% nominal air flow rate the maximum temperature seen in the building is 29.1°C in zone GD-W in the month of July.

The system's ability to control humidity is acceptable at all the tested air flow rates as is shown in Figure 9. There is a trend of increasing high humidity (above 70% relative humidity) as the air flow rate increases. During hours when the cooling equipment is not able to supply air at an absolute humidity at or below the value required to maintain the indoor humidity level, the supply air itself increases the humidity levels in the building. The humidity in the supply air will push the relative humidity in the building higher then would be experienced with the lower air flow rates.

The increased air movement due to increases in air flow rates may be perceived as uncomfortable to occupants. The ASHRAE recommendation for building air flow rates is 4 to 10 $L/s{\cdot}m^2$. The threshold upon which the air movement will feel drafty is dependent on the individual, however warmer supply air temperatures such those being delivered to the space by the desiccant-evaporative cooling system allow for higher air flow rates that are less likely to be perceived as drafts. The air flow rate tested here range from 5.5 to 9.6 $L/s{\cdot}m^2$ and are within the ASHRAE specifications [7].

7. ENERGY RELATED RESULTS

The major energy consuming equipment of the desiccant system include the fans used in providing the space with ventilation air as well as the gas used in regenerating the desiccant wheel. The amount of gas consumed in wheel regeneration has yet to be implemented into the inserted DOE-2.1E functions, however a record was kept of the number of hours in which the desiccant unit was active as an indication of comparative fuel usage results. As well DOE-2.1E makes its own calculation of the power consumed by both the supply and return fans. These results are plotted with respect to the supply air in Figure 10.

The graph shows a steady increase in the electrical energy load from the fan as would be expected. However, the increase in number of hours in which the desiccant dehumidifier is active is more significant. The gas consumed by the desiccant wheel increases with both the number of hours in which it is active and the air flow through the wheel. The approximately linear increase of hours in which the desiccant wheel is active will likely produce a second order increase of energy usage with increasing air flow. This is due to the fact more fuel will be needed as greater air flow passes though the desiccant wheel in addition to the fuel burned during the increase use of the desiccant wheel. There are 1420 hours simulated in which the system supplies air to the building therefore the desiccant dehumidifier is called on from 28.6% to 32.8% (Figure 10) of the cooling hours depending on the supply air flow rate.

8. DISCUSSION

The low values for indoor humidity, shown in Figure 9 are due to cool, dry air being introduced to the space. These conditions occur when the mixed air condition after the economizer is dry and cool enough not to be dehumidified or evaporatively cooled. The dry air lowers the indoor humidity to uncomfortable levels below 30%. The developed algorithm will be modified to provide a control option that will bypass the indirect evaporative cooler and using the direct evaporative cooling unit only when humidification of the air is needed.

The incorporation of desiccant-evaporative cooling routines into DOE-2.1E has been accomplished. The system is able to control the building comfort conditions on an hourly basis using the function routines DKTEMPDES and SDSFDES. High humidity control is acceptable for all air flow rates while low humidity controls need to be addressed. Several parameters have been identified for future parametric studies. These include:

1. Increasing the range over which the VAV boxes operate. The current industry norm for VAV systems is to allow the terminals to operate over a range of 40-100% air flow. By allowing a greater range of variability the fan power may be decreased during times when the supply air temperature is near 13°C. This will be particularly helpful for the increased air flow conditions where the increase in air flow capacity is only needed to satisfy the peak cooling loads in July when supply air temperatures as high as 17°C are being delivered to the space.

2. Allowing higher space humidity values to reduce the gas consumption of the desiccant wheel. This is subject to comfort and may only be allowed during hours of low temperature (20 to 23°C) to limit the effective temperature to the occupants.

3. Including a maximum air supply temperature value. Should the desiccant-evaporative cooling algorithm encounter that the supply air temperature is above a specified limit when no dehumidification is present (i.e. only indirect-direct evaporative cooling is performed) and a lower supply temperature may be achieved dehumidifying the supply air before performing evaporative cooling then to choose this method. This will have the effect of satisfying the cooling load which may have otherwise unsatisfied. The energy trade off will be between increasing the fuel usage by the desiccant wheel in favour of decreased electrical consumption by the fans.

4. Improving the error introduced between the DKTEMP and SDSF routines in regards to the supply air temperature and humidity supplied to the space. This may be accomplished by adjusting using the more accurate supply air condition as calculated in the SDSF-1 function call to adjust the air flow rate so that an equivalent amount of cooling is supplied to the space. This will ensure that the zone temperature reaction to the load and supply air is unaltered.

9. CONCLUSION

The use of DOE-2.1E functions has allowed the performance evaluation of a desiccant-evaporative cooling system. The results show that this system is capable of providing comfortable working conditions in this medium sized commercial office building for the hot and humid summer climate conditions of the simulated Ottawa area. The simulation results indicate that for this cooling system to be effective, the supply air flow rates will have to be increased to compensate for the higher temperatures that the system can deliver. Typical North American cooling design allows for supply air temperatures to be at 13°C. The supply air ducts and fans are designed to handle air flow rates for the building maximum cooling load conditions using cooling air at this temperature. These results also show that the air flow rates in buildings using desiccant-evaporative cooling systems will need to supply between 25 to 75% more air flow depending on the designers criteria of the minimum amount of undercooled hours. Future work will be done concerning the energy consumption of the desiccant unit before the trade off between energy usage and comfort may be evaluated. Low indoor humidity results will be addressed through better modeling of the direct evaporative cooler.

Acknowledgments

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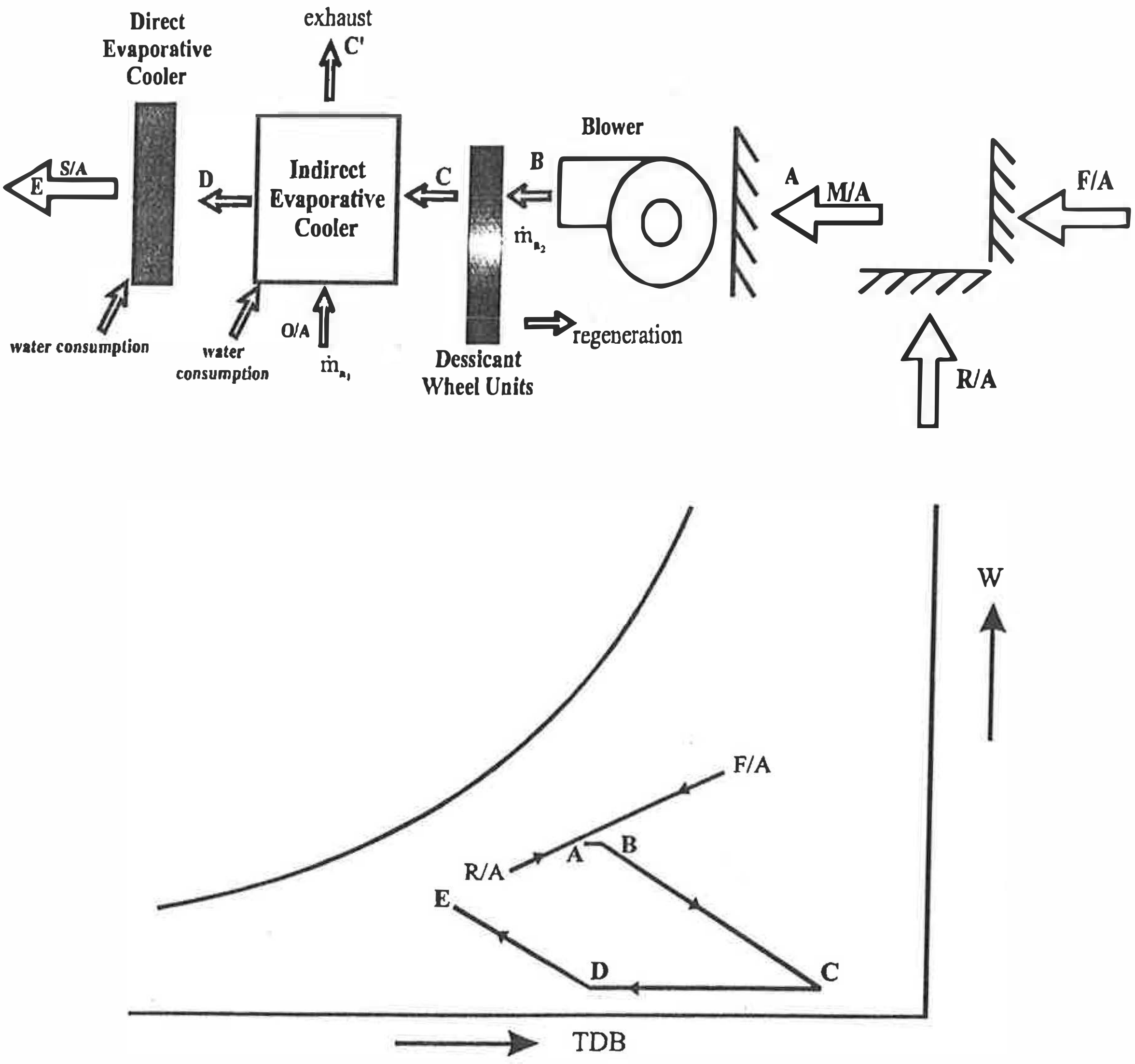

Figure 6. System Diagram and Psychrometric Process

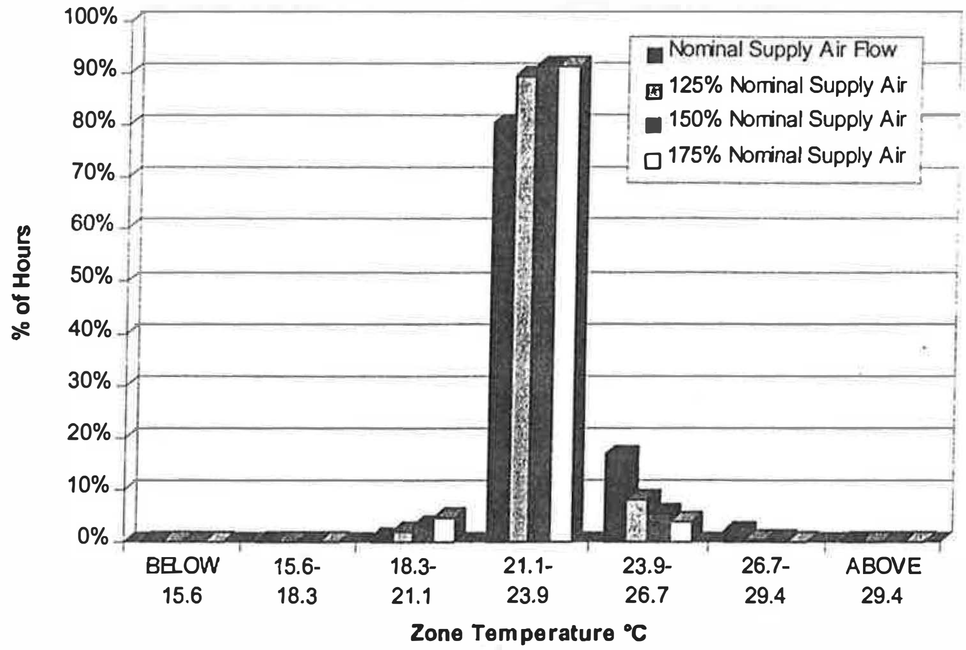

Figure 7. Temperature Chart for Zone GD-INT1. Percentage of Cooling Hours and Supply Flow Rate

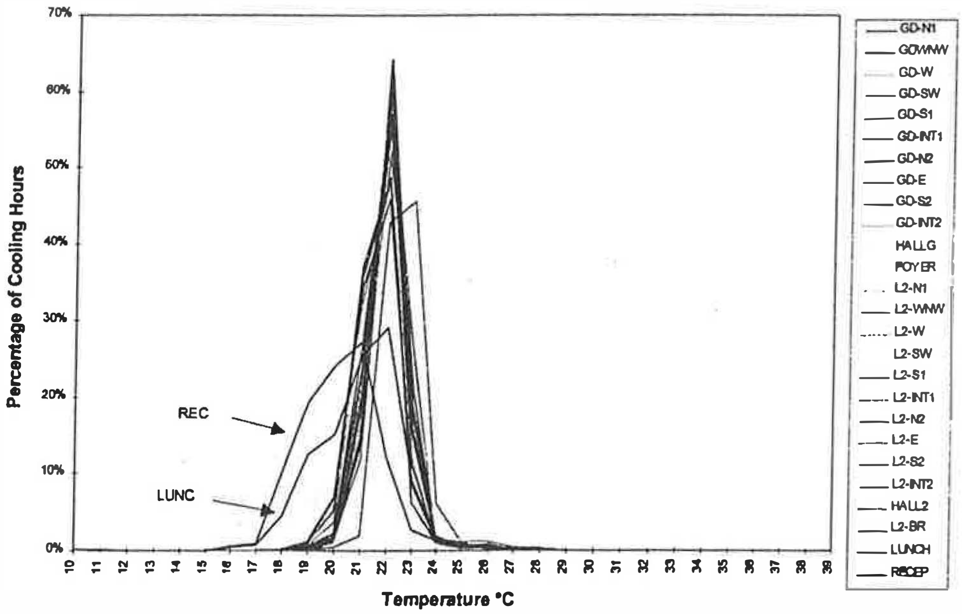

Figure 8. Line Plot of Zone Temperatures, percentage of cooling hours at each temperature for the 150% nominal supply air condition.

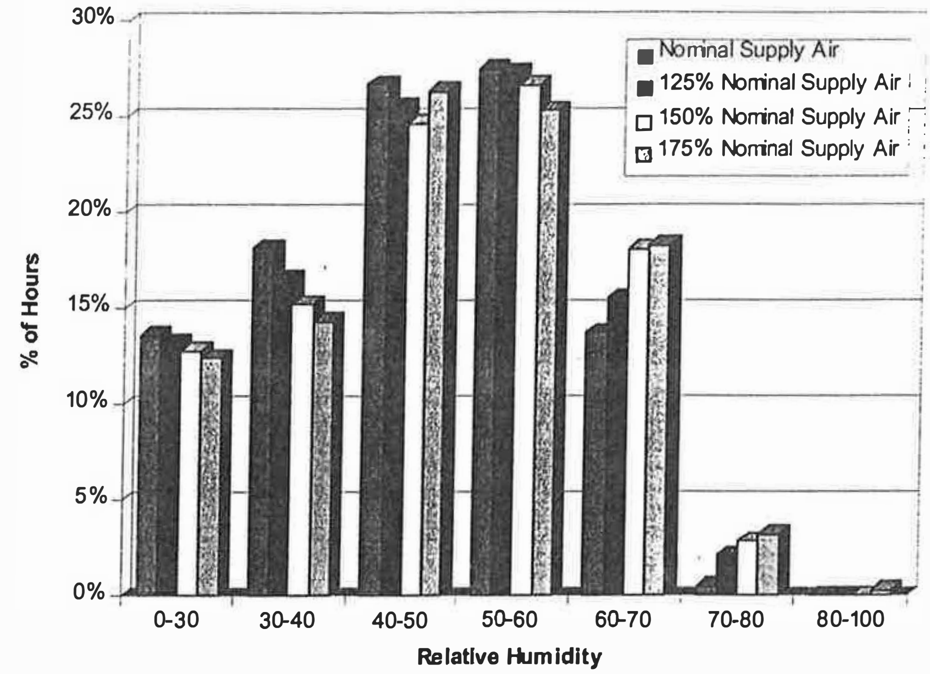

Figure 9. **Relative Humidity Chart for Building plotted against the percentage of cooling hours and the supply air flow rate**

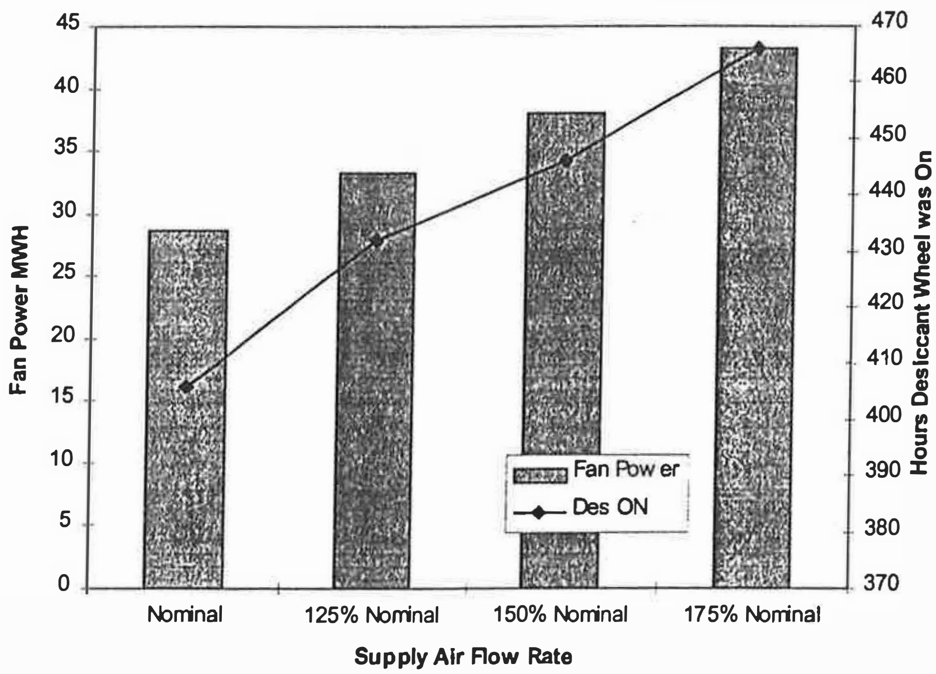

Figure 10. **Energy Related Results: Fan Energy Consumption and hours during which the Desiccant Wheel is Active**