CROMER CYCLE AIR CONDITIONER: A STUDY TO CONFIRM TARGET PERFORMANCE

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ABSTRACT

The Cromer cycle uses a desiccant wheel operating in conjunction with a typical air conditioning system. Simulations and laboratory prototypes demonstrate that the cycle has the potential for enhanced humidity control with sensible heat ratios as low as 40% and with far less energy use than other humidity control strategies. The research of this paper includes the purchase of “off the shelf” materials and the assembly of a working residential sized Cromer cycle desiccant air conditioning system. A desiccant wheel was retrofitted on an existing operational two and one half ton air conditioning system within an occupied residence in Cocoa Beach, Florida to validate the energy reduction targets and humidity control performance of this new technology. The unit was constructed and installed during the winter months of 2000. The monitoring for energy and dehumidification performance presented in this paper took place in the Spring of 2001. The unit was installed and removed so that performance data “with” and “without” the Cromer cycle was obtained for comparison. Performance data on the AC unit were acquired using the air-enthalpy method of ARI/ASHRAE test procedures with data recorded on Campbell Scientific CR-10 data acquisition system and downloaded to computer for analysis. The Cromer cycle system provided a three fold increase in moisture control capacity with a 4% reduction in energy use at the conditions tested.

WHAT IS THE CROMER CYCLE?

The Cromer cycle is a novel desiccant cycle working in conjunction with a cold coil that dramatically reduces the amount of energy needed to do moisture removal with the chilled water coil. This cycle uses a desiccant to transfer moisture between the supply and return air stream. The desiccant absorbs moisture from the high relative humidity air leaving the coil, wetting the desiccant and providing a much dryer duct system and conditioned space. The desiccant transfers its moisture to the air returning from the space drying the desiccant. The release of this moisture into the air before the cold coil increases the moisture removal of the coil enhancing its dehumidification. This cycle will provide additional drying (shift of sensible to latent work) with very little reduction in coil temperature. With a "colder coil" strategy such as heat pipes, or lower air flow, some additional moisture removal is achieved but with a decrease in efficiency and an increase in energy use. For this cycle to operate, a desiccant must be cycled back and forth between: a. the air returning to the air conditioner from the air conditioned space (return air), and b. the air being supplied to the space from the air conditioner (supply air). Any cycling mechanism can be used, however, an easy mechanical application of this cycle is a rotating wheel loaded with desiccant. Figure 1 provides a diagram of just such a wheel type system operating as a space conditioning device.

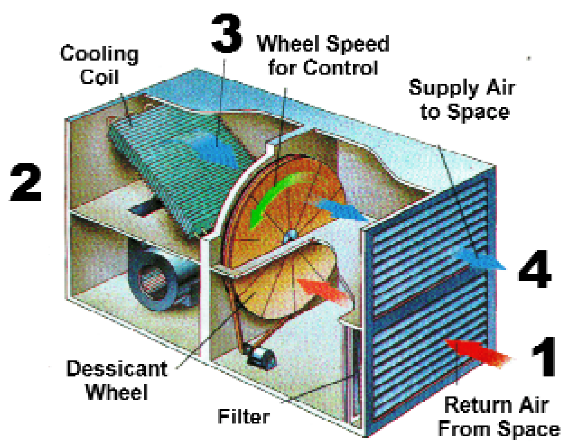

Figure 1. Cromer Cycle Schematic

Air conditioning and operational state points can be depicted on a psychrometric chart to describe a process and is shown in Figure 2. The following state points are provided to be representative and can be used to obtain a technical understanding of how the cycle operates. State point 1 is the air that returns from the space to the system (return air - 80F/51%RH). For a typical cooling coil, this air at state point 1 enters a cooling coil and leaves at state point 4’ after cooling and drying. State point 4’ represents the temperature and moisture content of the air that leaves the unit typically 45 F and 98%RH. The Cromer cycle is depicted with the dotted line. A loaded (wet) desiccant is presented with the return air at state point 1 and is desorbed. In doing so, the moisture evaporated from the desiccant cools the air to state point 2 (71F, 80%RH). Which goes to the cooling coil. The air exits the cooling coil at 3

(50F,98%RH). The dried desiccant is then used to remove moisture from the high humidity air exiting the cooling coil at 3. This sorption of moisture dries the supply air and it follows the line between state point 3 to state point 4, reentering the duct system for supply to the space at about 59F,52%RH. The moisture taken from the supply air by the desiccant is the same moisture that is re-evaporated for pre-cooling into the return air prior to it reaching the cooling coil, no extraneous water is added for this pre-cooling benefit.

By observation of the psychometric process, there are a number of improvements to the AC cycle that should be apparent. First, the end state point 4 for air from the wheel represents a significant latent ratio increase, to about 45% (SHR .55) as opposed to the 28% (SHR .72) of the typical coil shown. Secondly, the air quality delivered by this cycle is much drier, i.e. about 50% RH (state point 4) rather than 98% with the standard coil (state point 4'). Third, this is accomplished with a higher cold coil temperature. This is significant because other things being equal, the higher the cold coil temperature, the more efficient is the refrigeration cycle and the greater capacity any particular system can deliver.

This is how the Cromer cycle saves energy over a typical air conditioner cycle. The psychrometric chart demonstrates this win-win situation. Psychrometricly, this cycle should deliver a major increase in dehumidification control, higher EER, and greater capacity to efficiently handle humidity requirements than any of the alternative strategies. The moisture that is absorbed by the desiccant is moisture that would have gone back to the space. It is returned to the cold coil to remove it. This is a much more efficient moisture removal process than a stand alone dehumidifier and also provides energy cost savings over gas fired systems sold to take care of the latent load of the building.

This cycle is a new approach to the integration of active desiccant into the HVAC cycle. The desiccant is required to sorb moisture from near saturated air coming off of the coil that is colder (45 to 65 degrees F) and about 98% RH and desorb (evaporate) moisture to return air that has been heated by the space and is at a lower RH. This is a much different set of operating conditions than gas fired high temperature regenerated desiccants, and it is anticipated that desiccants with isotherms other than those used in gas fired units will provide optimum performance. Desiccants that have isotherms of the type shown in Figure 3 (Type III), are common.

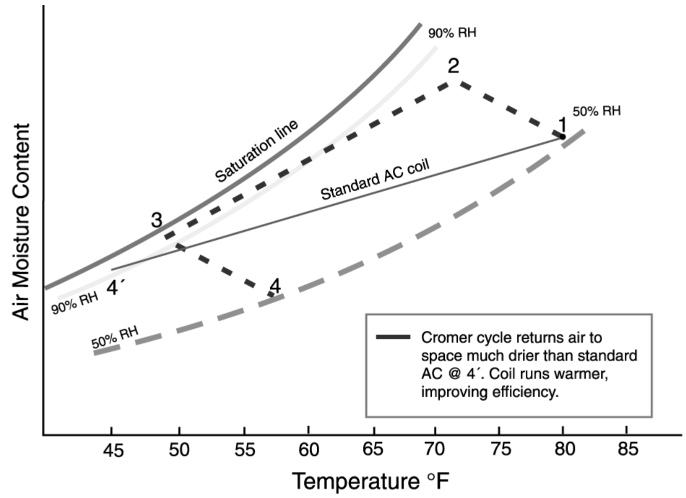

Figure 2. Psychrometric chart of Standard AC Cycle and Cromer cycle AC System

Type III desiccants absorb little moisture below 70% RH but many will take up more than their own weight in water from the air when presented with over 90% RH. The desorption isotherm is very steep between 90 to 100% RH. Low temperature desiccants of this type have plenty of potential for the cycling of moisture from the supply air stream to the return air stream. Where does the water adsorbed by the desiccant go? Persons familiar with gas fired desiccant systems may have difficulty in first understanding how this cycle works. Unlike these other desiccant systems that use the desiccant to transfer moisture only to gas heated air leaving the building, in the Cromer cycle any moisture captured by the desiccant is presented again to the cooling coil - for another shot at condensing it down the drain as condensate.

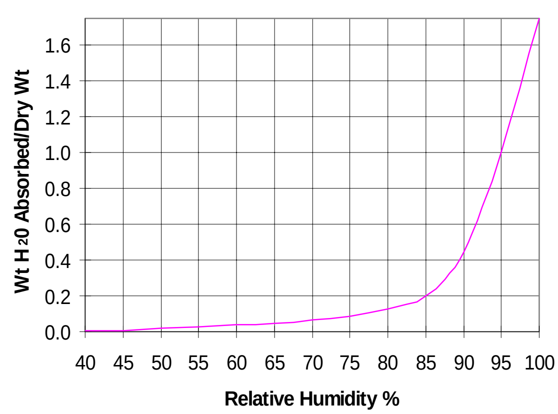

Figure 3. Silica Gel Type III Isotherm

SIMULATIONS VALIDATE THIS CYCLE

The Cromer cycle is a fresh new approach in combining HVAC and desiccant equipment. It requires the desiccant to respond to state points unlike high temperature desiccant desorption processes. A question of interest is, of course, will the desiccant respond to the state points - sorbing and desorbing moisture - as depicted on the psychrometric chart? The answer is a resounding YES - even without the high temperatures for desorption familiar in gas systems. Three independent simulations have been conducted to determine if this desiccant cycle concept is scientifically valid. The first, conducted by Dr. Cromer, used a wheel simulation model developed by Kirk Collier (DCSSMX1) which incorporates the finite difference algorithms for moisture sorption and desorption developed by Ian McClaire-Cross in Australia (MOSHMIX) into the DESSIM wheel model developed at NREL (then SERI). The simulation data from this work provided the state points of air before and after the wheel. The air response through the coil and performance data were provided by a set of equations developed from measured data on a 3.5 ton split system AC unit. Two desiccant types, three wheel sizes, and three wheel thicknesses were simulated. All showed excellent moisture transfer and increase of the AC system moisture removal. Significant latent ratios were predicted -- up to 50%. All three parameters, desiccant selection, wheel size, and wheel thickness, had an effect on optimal rotation speed. The simulations provided that at a LR of 40% the Cromer cycle would save 68%, 39% and 5% in energy over the alternatives: electric reheat, hot gas bypass reheat, and heat pipes respectively, with a 66% increase in total capacity above the reheat options [1].

The second set of simulations were completed by Dr. Bruce Nimmo at the Florida Solar Energy Center (FSEC) in 1993 and published in *ASHRAE Transactions*. [2] Dr. Nimmo's simulations found the Cromer cycle to provide better moisture removal capability than the alternatives simulated, and at a LR of 52%, showed an improvement in EER from 10.1 to 11.1 over the heat pipe application, a 10% energy savings. Simultaneously, the Cromer cycle increased capacity from 30.8 kBtu/hr to 34.0 kBtu/hr, a 10% increase in capacity over the state-of-the-art heat pipe system. Dr. Nimmo used an upgraded version of the Collier-Cross simulation developed by H. Henderson at FSEC, and the HVAC response was simulated by DOE-2. A Type III silica gel was used as the desiccant in the simulation. He writes in his conclusion, "The parametric study and the seasonal simulation results indicate that the DEAC (Cromer cycle) process is feasible and holds promise for maintaining a healthy and comfortable environment at a lower cost for residential and fast food restaurant applications. In addition, the (Cromer cycle) can save energy compared to current high-efficiency air conditioners if both systems are required to maintain the ASHRAE recommended comfort levels."[3]

The third set of simulations were conducted by Dr. Chant and Dr. Jeter while Dr. Chant was at the Georgia Institute of Technology, Atlanta and also published by ASHRAE. [4] Dr. Chant used a simulation developed at Georgia Tech. in 1991 using a parabolic concentration profile model (PCP) to model the desiccant moisture and sensible exchange. Chant's model predicted that the Cromer cycle, when providing excellent moisture removal, i.e. a LR of 52%, would perform better than the heat pipe system by providing an energy savings (improvement in COP) of 2.58 to 2.68 (4%) and also an improvement in total cooling capacity from 9.23 Kw with the heat pipe to 10.39 Kw with the Cromer cycle (12.6% improvement). It should be noted that Dr. Chant assumed the desiccant wheel would have a greater pressure drop than the double coil heat pipe system and consequently the simulation added a large fan energy penalty to the Cromer cycle (which was called DEC). Dr. Chant writes,

"The DEC (Cromer cycle) system uses mass transfer in a similar way that a heat pipe system uses heat transfer to enhance the latent capabilities of a cooling coil. The simulations found that the (Cromer cycle) system experiences a dramatic rise in latent capacity compared with a vapor compression (VC) unit alone. Heat pipes are currently considered the state of the art technology for controlling the latent load of a conditioned space. The (Cromer cycle) system compared favorably to the heat pipe system. A desiccant wheel installation is hardly more complicated than an auxiliary heat pipe heat exchanger but promises a higher coefficient of performance (COP) and increased capacity for a given sensible heat ratio."[5]

SIMULATIONS CHECKED BY LAB TESTS

A desiccant wheel by was tested at various state points to determine if the simulations were representative. The two foot diameter wheel was installed on a five (5) ton air conditioner, and was tested in the Appliance Laboratory environmental control chambers at the Florida Solar Energy Center. The laboratory chamber and duct set-up is depicted in Figure 4. The method of test utilized is defined by the Air Conditioning and Refrigeration Institute (ARI) and the American Society of Heating, Refrigeration and Air Conditioning Engineers (ASHRAE) for testing the performance characteristics of unitary air conditioning equipment. The ARI Standard 210/240 [6] references the ASHRAE Method of Testing for Seasonal Efficiency of Unitary Air Conditioners and Heat

Pumps, ANSI/ASHRAE 116-1983. [7] The test set-up was configured to use the Tunnel Air Enthalpy Test Method Arrangement of Standard 116 (section 6.1.1), with the addition of chamber bypass as a suitable means for maintaining tight control of chamber temperature and moisture. Control was provided by a pair of air handlers, fed by a 5 h.p. R-502 condensing unit, which conditioned the rooms to the tolerances prescribed in Standard 116 for the tests. These tests, conducted on a five (5) ton split system air conditioner, were presented in a paper at the Twelfth symposium on improving Building systems in Hot and Humid Climates, and confirmed that the desiccant wheel would sorb and desorb moisture as well or better than predicted by the previous simulations at the required state points and the integration concept of the desiccant wheel with the HVAC evaporator coil was a scientifically valid approach [8].

The lab test data showed not only a major increase in Latent Ratio (the Btu moisture removed divided by the total Btu cooling) to 50%, but also a 14 to 16% reduction in energy used by the AC system when providing the same combined (sensible and latent) cooling delivered. It was thus hoped that a field system would be able to achieve up to twelve percent energy savings while providing the increased latent control of the cycle. The twelve percent energy savings was set as the field target for the installed system.

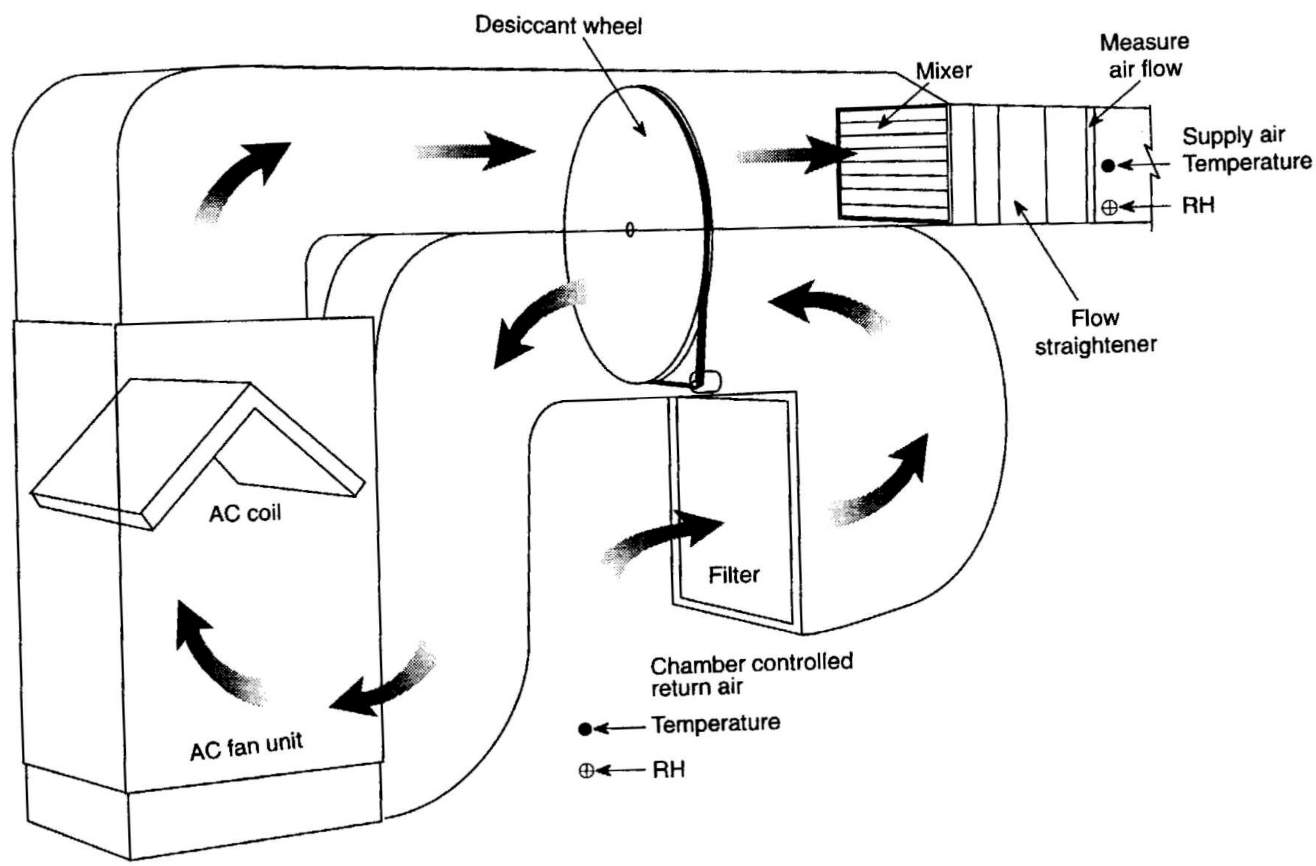

Figure 4. Lab Test Set-Up to Validate Desiccant Response (From Reference [8]).

INSTALLATION

A residence was identified in Central Florida that was suitable for the easy modification of the duct system such that the desiccant wheel could be installed in the flow of the return air and supply air. A 24 inch diameter wheel was used for the test. The AC unit was a 2 ½ ton split system, which has its inlet for return air at the bottom and blows its supply air out the top of the air handler which was connected to the duct system. The unit was located in an air conditioned workshop area and was moved over about 3 feet (1 m) such that the ducting could be modified similar to the diagram of Figure 4 and the air handler again attached to the duct system. This provided access to the desiccant wheel which could be easily slid out and the two paths of the ducting sealed back up. A photo of the ducting constructed on the air handler is shown in Figure 5.

Figure 5. Photo of Desiccant Wheel Ducting on 2 1/2 Ton Air Handler.

MONITORING

The monitoring system used a Campbell CR-10 datalogger with 10 second scans and averaging on ten minute intervals. Data includes: Julian date, time, outdoor temperature (F), Watt-hr used by the AC system and fans, Cfm air flow through the duct, return air temperature (F) and relative humidity (%), and supply air temperature (F) and relative humidity (%).

TEST PROCEDURE

The test procedure is to monitor the system during the Summer months and allow the AC system to cycle on its thermostat which is set at 80 degrees F. The desiccant wheel will be installed and removed every two weeks such that comparative operational data may be obtained for similar outdoor conditions.

At the time of this writing, April 2001, there has been little use of the AC system because of mild weather. Also, the time periods of operation have been of quite different outdoor load conditions. For the data of this paper, the thermostat was set to a high setting at 1300 hrs (1 PM) for two consecutive days. This maintained the AC system in a continuous run condition for the test. Both tests were run for an hour to stabilize and then one and one half hour (9 - ten minute data sets) as test data. The first day was run with the wheel installed. That evening, the wheel was removed and the duct resealed. The two days had quite similar weather and the data obtained and can provide some insight into the system operation.

COMPARATIVE DATA

Table 1 provides a data set for the system operating without the wheel installed. Table 2 provides the data set for the previous day with the wheel installed. For the times recorded, these two data sets were almost identical in ambient condition and in the return air condition from the space.

Table 1. Test Data on Standard AC System.

Day	Time	Amb F	Watt-hr	Cfm	InRH	OutRH	In F	Out F
52	1410	95.227	3230.0	943.20	45.600	99.934	80.925	56.243
52	1420	95.128	3230.0	944.83	45.868	99.930	80.846	56.213
52	1430	95.238	3230.0	943.20	46.086	99.926	80.658	56.117
52	1440	95.151	3251.4	945.24	45.187	99.921	80.875	56.180
52	1450	95.135	3230.0	947.29	45.104	99.919	80.852	56.106
52	1500	94.984	3251.4	943.20	45.341	99.915	80.728	56.043
52	1510	95.036	3230.0	946.06	45.201	99.908	80.920	56.248
52	1520	95.041	3251.4	948.92	45.751	99.907	80.696	56.317
52	1530	94.884	3251.4	945.24	45.125	99.900	80.944	56.334
Average:		95.092	3239.5	945.24	45.474	99.918	80.827	56.200

Table 2. Test Data on AC System with Cromer Cycle

Day	Time	Amb F	Watt/hr	Cfm	InRH	OutRh	In F	Out F
51	1410	95.269	3123.0	799.46	45.815	68.179	80.550	60.247
51	1420	95.252	3101.6	794.56	45.288	68.628	80.842	60.223
51	1430	95.285	3101.6	790.07	45.224	68.809	80.675	60.084
51	1440	95.221	3123.0	788.02	45.494	68.874	80.350	59.958
51	1450	95.207	3101.6	796.60	45.295	68.505	80.435	59.907
51	1500	95.053	3101.6	792.93	45.056	78.048	80.539	58.301
51	1510	95.147	3123.0	790.07	45.708	71.438	80.689	61.826
51	1520	95.109	3101.6	793.33	45.357	67.807	80.514	60.412
51	1530	95.090	3101.6	798.64	45.524	67.958	80.582	60.274
Average:		95.181	3108.7	793.74	45.418	69.805	80.575	60.137

PSYCHROMETRIC ANALYSIS

The average state points for the data in Table 1 and Table 2 can be evaluated by psychrometrics to determine the sensible and latent Btu delivered by the system during the test interval (1 ½ hr. of operation) on the sequential days. To do this analysis, the author used the software package “ASA PsyChart” [8]. Table 3 provides a summary of the results.

Table 3. Summary of Field Test Data.

	Btu/hr	Sensible	Latent	SHR	LR	EER
Standard	28,444	25,646	2,798	0.90	10%	8.78
CroCyc	27,303	17,869	9,433	0.65	35%	8.78

DISCUSSION OF RESULTS

The operation of the Cromer cycle did indeed provide a three fold increase in moisture removal and also ran with a slightly warmer evaporator coil which resulted in about 4% reduction in power consumption, even though there was a slight increase in ambient temperature for the comparison times. The 4% energy savings is short of the target 12% desired for the system and the operational total capacity was reduced slightly. However, it should be noted that this Spring time load is much lighter on humidity than the expected Summer load. For this test, the ambient humidity was not measured at the site, but local TV weather data indicated it was in the 30 to 45% humidity level during midday and rising to over 90% during the night. An RH sensor will be added to the test to measure ambient RH at the site for the summer period. Even though operational times of the air conditioner are limited, it has been noted that the standard equipment has allowed the humidity to rise over time as it cycled on the thermostat, while the Cromer cycle has reduced the indoor humidity level cycling on the same thermostat and set point. Also, the change in indoor humidity was a slow process, taking place over several days - but this may not be the case with the hot-humid conditions of summer. If the Cromer cycle provides a lower indoor humidity condition, this should be more comfortable.

It is assumed that the identical EER values for the test periods are serendipitous. It is expected from the lab work, that the Standard AC cycle will drop in EER as the moisture load increases on the system as it enters the Summer months, and the lab work indicates that higher moisture load improves EER for the Cromer cycle.

CONCLUSION

A standard two and one half ton air conditioner in a residence in Florida has been instrumented and modified to operate under the Cromer cycle such that the desiccant wheel can be easily installed and removed. Preliminary comparative results with the wheel installed and the wheel removed indicate that the Cromer cycle system provided a three fold increase in moisture control capacity with a 4% reduction in energy use at the conditions tested. Though these results are preliminary and inconclusive, they provide “real world” data that justify further field testing and validation of the cycle.

ACKNOWLEDGEMENTS

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