COOLERADO AND MODELING
AN APPLICATION
OF THE MAISOTSENKO CYCLE

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The National Snow and Ice Data Center (NSIDC) recently replaced its traditional cooling system with a new air conditioning system that utilizes an economizer and Coolerado air conditioning units. These units represent one of the first commercially available applications of the Maisotsenko Cycle (M-Cycle). A data logging system was installed that measured the data center’s power consumption before and after the cooling system was replaced. This data was organized and used to prove a 90% cooling energy reduction for the NSIDC. The data logging system also collected temperatures and humidities of inlet and outlet air of a Coolerado air conditioner. After using these data to validate a theoretical model developed by researchers at the National Renewable Energy Laboratory, the model was used to simulate slightly modified heat and mass exchanger designs of the Coolerado system to improve performance.

Sensitivity analysis was performed and found a few design parameters that are important to the thermodynamic performance of the Coolerado system, while others were proved insignificant. Overall, it was found that the current design performs reasonably well and with minor modifications could perform optimally, as suggested by the theoretical model.

KEY WORDS: data centers, indirect evaporative cooling, economizer, PUE, energy savings, heat exchanger

INTRODUCTION

Current expansion and the use of digital technology and communication are causing significant growth demands on the information technology (IT) industry. The worldwide capacity for data storage and processing is increasing rapidly as more and more industries and companies must provide or store data for clients, as well as for internal business purposes. Examples include: online financial services and mobile banking; the communication and entertainment industry as they expand into social websites and internet access markets; and digital healthcare records. Almost all shipping and mail services now offer online tracking of packages. GPS technology has become smaller and more affordable, and a standard feature on many new vehicles. Real-time data processing of these geospatial data will increase exponen-
ially. With all of these industries expanding their digital footprint, data center demand capacity for storage and processing will also grow accordingly.

Data centers provide a clean, secure and stable environment for online servers and must be maintained physically and virtually at all hours of the day. Such facilities across the U.S. typically house anywhere from a few dozen to a few thousand servers and are geographically dispersed. In 2010, over 11 million servers were installed in the U.S. (Koomey, 2011).

Many of these computers are meticulously maintained for peak performance, and this means their physical environment where they are stored must be kept very clean and maintained within specific temperature, humidity and air quality standards. So while improvements in energy efficiency are important, machine room environment and support systems’ reliability are also critical to these data centers.

The green data center project, funded by the National Science Foundation (NSF) and NASA, demonstrates how a data center can achieve energy efficiency yet maintain reliability, temperature, humidity and air quality standards.

BACKGROUND

The NSIDC provides data for studying the "cryosphere" — those areas of the planet where snow or ice exist. The two most extensive regions are Antarctica and the Arctic but high elevation regions are also important as they contain glaciers, and permanent ice and snow fields. The Center began operations at the University of Colorado in 1976.

Some of the data held at NSIDC indicates dramatic changes in the cryosphere over the past decade and many of these changes have been attributed to global warming. NSIDC has operated a Distributed Active Archive Data Center (DAAC) for NASA’s Earth Observing System since 1992. The data managed by the DAAC computer systems are distributed to scientists around the world. In order to deliver these cryospheric data to national and international clients, the data center must be online around the clock.

The NSIDC computers are housed in a secure facility within a mixed use office and research building located in Boulder, Colorado and operates continuously. In 2008, the Uptime Institute said that annually, global data center CO₂ emissions are poised to equal that of the airline industry if current trends persist, and it estimated that data center CO₂ emissions will quadruple between 2010 and 2020 (Carmichael, 2008). The irony of the NSIDC’s situation stems from the fact that the use of tools (the data center) necessary to study the problem (climate change) are actually contributing to the problem. When it became necessary to replace and upgrade the existing cooling infrastructure, all options were considered, and in particular, solutions that would reduce energy demand and dependency on the local utility provider were given high priority.
ASHRAE Standards

Computers consume a significant amount of electrical energy. Most of this energy is converted into heat within a computer that then must be exhausted to prevent damage to the electronic components. The American Society of Heating, Refrigeration, and Air Conditioning Engineers (ASHRAE) has established temperature and humidity standards that are suitable for data centers and computer rooms (ASHRAE, 2008). The limits on this psychrometric "envelope" are shown in Table 1.

Humidity control in a data center is critical to stable operation. Too much moisture in the air allows for condensation possibly leading to short circuits and damage to any part of the delicate integrated circuits, power supplies, and other hardware. Too little humidity allows for static charge accumulation and potential damage from a single, large discharge could be significant. Exceeding a certain ambient air temperature can cause thermal damage to equipment. Very low ambient temperatures can cause bearing failures on disk drives leading to premature failure.

It is a vital concern of data center managers to maintain an appropriate air state using heating, ventilation and air conditioning (HVAC) systems.

The NSIDC computer facility meets the ASHRAE Allowable Class 1 Computing Environment, as shown in Table 1. Internal review of manufacturer specifications revealed that all of the computer and IT equipment in the data center can operate safely (and under warrantee) under the ASHRAE Allowable and Recommended Computing Environments.

Traditionally, data centers have been cooled by standard (and often packaged or unitary) air conditioning systems that utilize a direct expansion (DX) heat removal process through a liquid refrigerant. This is the same type of cycle that runs the air conditioning in a car or refrigerator. Although a robust and mature technology, the DX cycle has been under scrutiny over the past few decades as other technologies capable of producing the same amount of cooling with less power requirement have entered the market and become economically viable. The DX cycle requires environmentally harmful, synthetic refrigerant (R-22) and substantial energy to run a compressor. Of all the parts in a DX air conditioner, the compressor requires the most energy. Up until the past few years, typical data centers have...
been cooled by these packaged DX systems, commonly referred to as computer room air conditioners (CRAC), because they are very reliable. These systems have been economical in the past due to their off-the-shelf packaging, compact size and simple controls. However, as the need for energy use reduction and environmentally friendly technology becomes more and more prevalent, the DX systems for data centers are becoming antiquated. The NSIDC operated 2 CRAC (50 tons in total) units full time until June 2011.

Most data centers are connected to a regional utility grid, as this has traditionally been the most reliable way available to power critical computer systems. Power is delivered to an uninterruptible power supply (UPS), which conditions and delivers power to the computers in a data center. The UPS in the NSIDC charges a large battery array for use during a power outage. (The NSIDC does not have any backup generators). This allows the data center to remain online for about two hours without power from the utility provider.

NEW COOLING INFRASTRUCTURE

The Green Data Center project was separated into two main components: server consolidation and virtualization and installing a more efficient cooling system. The latter is the focus of this paper, but it is important to realize that some energy reduction was achieved by simply reducing the IT load (which also reduces the cooling power required). The new cooling system design includes a unique cooling system that uses both airside economization and a new air conditioner that uses the efficient Maisotsenko Cycle.

Airside economization is not a new technology, but does add some complexity to the control system. Simply put, an airside economizer is a control mode that allows the air handling unit (AHU) to cool the space solely with outdoor air when the outdoor air is cooler than the air in the space. This is commonly referred to as "free cooling". In this mode, no air conditioning (DX or other process) is required, and recirculation of room air is reduced to a minimum. As stated previously, humidity is an issue for computers and electronic systems. And in many locations, particularly the Midwest and East Coast of the United States, airside economization may not be possible due to the hot and humid climate. However, the State of Colorado (and much of the western U.S.) is much drier year round and cool enough for about 6 months of the year, so an airside economizer is a viable option for data centers to maintain an air state that is within the ASHRAE limits.

Note that the AHU is completely responsible for the airside economization by regulating the amount of outdoor air into the space. When the outdoor air is cool enough (late fall, winter and early spring months), the AHU introduces a mixture of this cool air with some hot return air from the backside of the server racks. This mixed air is introduced into the room and allowed to flow out from beneath the Coolerado air conditioners to keep the "cool" areas of the room around 22.2°C (72°F).
The centerpiece of the new system revolves around a series of Coolerado® air conditioners that utilize the Maisotsenko Cycle (Wicker, 2003). Cool supply air is delivered directly to the computer room. Working air, warm and saturated, is ultimately exhausted to the outside, or directed into the space when room humidity is below the humidity setpoint. The room humidity can drop below 25% relative humidity if the outside humidity is very low (which happens often in Colorado), and the AHU is in economizer mode, providing a significant amount of outdoor air to the room. In the winter months, this happens often and working air is directed into the space most of the time through the working air dampers (see Figure 4). One CRAC unit (30-ton) was left in the room for redundancy and for dehumidification during occasional hot and humid periods in the summer. The new system (exclusive of the 30-ton CRAC) was not designed with a dehumidification mode because this is rarely necessary in Colorado’s climate.

Note that this cooling process does not have a compressor or condenser in its cycle (when the CRAC isn’t operating). Air from the AHU can now be cooled to a comparable cool temperature using an average of one tenth the energy that would be required by the CRAC system. Water is used in this cycle, and is only used once (single pass). Based on measurements, all eight of the Coolerado units consume an average of 1.7 liters per minute (0.5 gallons per minute) when in humidification mode, which is most of the winter. Unfortunately, similar measurements were not taken of CRAC water use, although it is expected that the CRACs use and waste more water due to an inefficient humidification process.

The new cooling system consists of a rooftop air handling unit (AHU) powered by a 7.5 kilowatt (10 horsepower) fan motor via a variable frequency drive (VFD), eight Coolerado air conditioners and hot aisle containment.

Figure 1 shows a schematic of this system. On a design cooling day (hot summer day), the AHU pulls in outdoor air around 35°C (95°F) and introduces it to the front side of the Coolerado units. Then the Coolerado fans force the hot air through its HMX, splitting the airstream in half and cooling the supply air to about 12.7°C (55°F). This cool air is directed to the front side of the server racks. After being exhausted from the backside of the server racks at around 32°C (90°F), the air is pulled out of the hot aisles by suction pressure from the return air damper in Figure 1. This slight negative pressure is caused by the same AHU fan that initially pushed the air into the room. To ensure efficiency and prevent cool air from being pulled out of the room before cooling the servers, hot aisles are separated from the cool areas of the data center by plastic curtains (see Figure 1). This is referred to as hot aisle containment.

The Coolerado air conditioners are only used when the AHU can no longer supply cool enough air to the data center, presumably because the outdoor air temperature has climbed above the room temperature. The Coolerados are located in the room with the servers so that cool product air can be delivered directly to the front side of the servers (see 3 in Figure 1).
THEORY OF THE MAISOTSENKO CYCLE

M-Cycle Description

A company named Coolerado, based out of Denver, CO, has developed a high efficiency air conditioner that works well in many Western climates. Coolerado claims these units are theoretically capable of up to 80% energy savings over 2010 standards because of their use of a unique cooling cycle called the Maisotsenko cycle (Wicker, 2003). This cycle uses both direct and indirect evaporative cooling to produce a supply air state that is 16.7°C to 22.2°C (30°F to 40°F) below the incoming air temperature. The air is split into two separated air streams; a "working air stream" is used to remove heat from the "supply air stream." The working air is cooled evaporatively as it is directed through a channel with a wetted surface in a patented heat and mass exchanger (HMX). At the same time, the supply air is directed into an adjacent channel and is cooled through conduction and contact with a membrane that separates the two air streams. Working air is rejected to the atmosphere and supply air is retained. Theoretically, the cooling limit of the supply air after one iteration of this process is the wet bulb temperature of the moist, working air. However, if this splitting and adjacent flow process is applied again to the cooled supply air, it can be cooled even more, theoretically to the new wet bulb temperature of the once-cooled supply air. In the Coolerado units, this process is repeated 20 times and reduces the volume of supply air by about 38% (dependent on airflow). The theoretical limit of this entire process is the dew point temperature of the original supply air. See this process in Figure 2 on a psychrometric chart.
From Figure 2 above, it is evident that this use of the Maisotsenko cycle does not add or remove moisture from the supply air. Moisture is added to the working air, but that air is most often rejected to the atmosphere. For this reason, the Coolerado works very well for spaces in arid climates because latent loads (moisture introduced into space, usually by occupants) can be decreased by simply mixing less return air and more dry, outside air. For spaces with very little latent load (e.g., data centers), the Coolerado works well in even more climates because there is no latent load from electrical or IT equipment and moisture removal is of less concern. Introduction of some outdoor humidity is acceptable, as long as the room air (supply air to servers) remains within ASHRAE limits.

The HMX actually divides the incoming airstream into the final supply air and working air quantities at the entrance to the HMX. The current design uses a total of 8 channels: 3 for working air and 5 for supply air. Note that the "sheets" (pliable, polypropylene sheets) in Figure 3 alternate flow directions, and each sheet has a flow direction that is perpendicular to the flow direction on the sheets above and below it. Input air flows lengthwise as working air is directed sideways on the adjacent sheet below. This configuration classifies this HMX as a crossflow exchanger. The working air sheets have a wicking surface on them and are saturated with a constant trickle of water from the V-type trough in the middle of each sheet; as working air flows over this wet surface, direct evaporation cools the working air, the sheet surface, and therefore the supply air flowing on the other
side of the sheet. And as the working air travels sideways, across multiple supply air channels, it removes some heat from each supply airflow. The power of the Maisotsenko cycle, however, is realized when this process is incrementally repeated with the working air. Note that there are multiple holes in the working air channel, each corresponding to a different sideways channel. The current design uses 20 of these channels.

Coolerado’s M50 unit has 5 HMX’s stacked vertically and, at least in theory, the flow front to back of the HMX’s should be uniform, and pressure should decrease linearly in the straight sections of the channels (there may be a jump in pressure one way or the other as working air moves from the dry side to the wet side of the sheet through the hole).

**Heat Transfer Equations**

In many ways, the fundamental, heat transfer in the HMX is similar to a cooling coil. In a typical cooling coil, heat is removed from the supply air stream through convection with a very cold surface and condensation. The heat is transferred to the refrigerant (typically flowing in a small tube) and absorbed into condensed water in the heat of condensation (opposite of vaporization). The water drips downward to be collected at the bottom of the coil and the refrigerant is pumped and cycled through a condenser loop to reject the heat somewhere else. The supply air leaves the coil at a much colder and often drier condition.

The Maisotsenko cycle and the Coolerado HMX differ from this traditional cooling method because the working air acts as the refrigerant (instead of a liquid substance) and the condensation is reversed to evaporation. This evaporation occurs on the working air (refrigerant) side, which leaves the product air humidity constant. See Figure 4.
\( \text{SA} = \text{Supply Air} \)
\( \text{WA} = \text{Working Air} \)
\( h = \text{enthalpy} \)
\( h_{\text{conv}} = \text{convection coefficient between wetted surface and working air} \)
\( h_{\text{DO}} = \text{evaporation mass transfer coefficient} \)
\( T = \text{dry-bulb temperature} \)
\( C_{p,a} = \text{heat capacity of air} \)
\( i, i+1 = \text{indirect side index} \)
\( j, j+1 = \text{direct side index} \)

Table 2 shows the heat transfer equations for a discrete portion of the HMX. Equation 1 is an "overview" equation simply labeling the terms in Equation 2;

**TABLE 2:** Heat transfer equations for HMX

\[
\text{SA}(\Delta_{\text{sensible}}) = \text{WA}(\Delta_{\text{latent}} + \Delta_{\text{sensible}}) + \text{water}(\Delta_{\text{sensible}}) \quad (1)
\]

\[
\dot{m}_{\text{SA}}(h_i - h_{i+1}) = x \dot{m}_{\text{water},j} h_{\text{evap}} + \dot{m}_{\text{WA}} C_{p,a}(T_{j+1} - T_j) + (1 - X) \dot{m}_{\text{WA}} (h_{\text{water},j+1} - h_{\text{water},j}) \quad (2)
\]

where \( X = \frac{\% h_{\text{DO}} dA W_j - W_{j+1}}{m_{\text{water},j}} \quad (3) \)

\[
dq = h_{\text{conv}} dA (T_i - T_{\text{water},j}) = \dot{m}_{\text{SA}} (h_i - h_{i+1}) \quad (4)
\]

\[
dq_{\text{sensible,WA}} = \frac{h_{\text{conv}} dA}{C_{p,a}} (h_{\text{water},j} - h_{j}) = \dot{m}_{\text{WA}} C_{p,a}(T_{j+1} - T_j) \quad (5)
\]
these equations are equal. Equation 3 defines the amount of water that is evaporated into the working air, as a fraction of total water flow. Equation 3 relates the perpendicular heat transfer from the product air surface to the top of the water film in the working air side (which must be sensible conduction only) to the sensible heat loss of the product air. Equation 5 defines the sensible heat gain to the working air, which would occur between the top of the water film and the working air flowing parallel. This is in addition to evaporation (mass transfer) that also occurs.

Methodology
To confirm Coolerado’s claimed energy savings, all major power consumption locations were equipped with power meters in the NSIDC. To develop a set of steady-state performance characteristics of the M50 and HMX system for calibration of a theoretical model, one of the units at the NSIDC was equipped with a power meter, temperature and humidity sensors and additional one-time measurements were taken.

DAQ Continuous Measurement Systems
A combination of data logging systems were used for this research. National Instruments USB data acquisition (DAQ) and a Campbell Scientific CR10X data logger were used at different times during the project. Table 3 defines these systems in more detail. The USB interface was used initially for just power measurements of the old system and logged measurements every 10 seconds for a period of almost 1 year (June 2010 – June 2011). The CR10X data logger was used for monitoring all power uses in the new cooling system as well as local temperature, humidities and airflows around one of the Coolerado units. This system logged data every 3 minutes for about 10 months (June 2011 – April 2012).

One-time Measurements
Some measurements can only be obtained at a single point in time due to expensive instrumentation or impractical permanent setup arrangement. Most of these measurements are only relevant to the calibration of the theoretical model and were not a requirement of nor relevant to the IT management team of the NSIDC. The one-time measurements needed for this project were pressure differentials across a Coolerado unit and airflow at the inlet. These on-time measurements were performed on February 17, 2012 at 11:00 AM. Measurements were taken at 4 different fan speeds, which were manually adjusted with a built-in potentiometer on the new system’s control board.

Due to the unique design of the HMX, the two airstream pressure differentials (supply and working) must be measured separately. The product air pressure drop is defined as the difference between the point just before the HMX (within the unit) and the point immediately after the HMX at the supply air outlet of the unit. The working air pressure drop is defined as the pressure difference between the
point just before the HMX and the point at the exhaust air exit of the HMX (within the unit). Both of these differentials are directly related to the fan speed of the unit, which is continuously variable. These measurements were performed with a self-averaging setup exactly similar to Coolerado’s in-house testing setup for comparison purposes.

A hot wire anemometer was used to determine the velocity profile for both of these measurements. Each velocity measurement must be made to represent an equal area of flow. When this condition is met, a simple average of all measurements will provide an accurate average velocity. This velocity can easily be converted to volumetric flow by multiplying by the total area of the inlet. Note that all flow measurements were taken perpendicularly to the inlet. And although there are probably some velocity vectors that are not perpendicular to the inlet plane (especially around the edge), these were not considered in this measurement due to time constraints and the number of measurements required. Nine measurements were taken at 4 different fan speeds, for a total of 36 measurements.

The pressure and airflow data that were collected were compared to the same data collected by Daniel Zube of Coolerado at their manufacturing facility on the

<table>
<thead>
<tr>
<th>TABLE 3: DAQ systems used for Coolerado HMX research</th>
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<tr>
<td><strong>Manufacturer:</strong> National Instruments</td>
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<tr>
<td><strong>Logging units:</strong> USB-6008</td>
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<tr>
<td><strong>Ports:</strong></td>
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<tr>
<td>4 differential analog inputs</td>
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<tr>
<td>12 digital I/O</td>
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<tr>
<td>1 USB port</td>
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<tr>
<td>1 serial I/O port</td>
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<td><strong>Data Points Monitored:</strong></td>
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<tr>
<td>30-ton CRAC Power WattNode</td>
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<td>UPS Input Power WattNode</td>
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<tr>
<td><strong>PC Interface:</strong> USB</td>
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<tr>
<td><strong>Power:</strong> 5 V DC (provided from USB)</td>
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<td>I/O = Inputs/outputs</td>
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same day. The similarity between the NSIDC (CU4) data and the Zube data can be seen in Figure 5. Power and airflow regressions were also developed, but are not included in this report.

Figure 5 also includes uncalibrated model simulations. The theoretical model is described in further detail below.

**Theoretical NREL Coolerado Model**

In 2009, an effort was made to model the Coolerado HMX cooling performance using purely theoretical relations and equations. Much of Coolerado’s in-house design work is currently based on mostly experimental data and curve fit relations. The National Renewable Energy Laboratory (NREL) developed a MatLAB model that can be used to develop more efficient HMX systems based on the M-Cycle. To date, this has not been done before, and using the model for this purpose is the objective of this research. MatLAB was the chosen platform because it is very flexible as a generic programming language and handles iterations and matrix calculations very efficiently; both of these functions are critical to solving the fundamental heat and mass transfer equations within a Coolerado HMX outlined in Table 2. The model is complete with a graphical user interface (GUI) that allows a user to change any number of inputs from their default values. This model makes several basic assumptions that may not be entirely realistic but are necessary to allow the discretized, theoretical computations to converge properly:

**Rigid sheet structure** — In reality, the polypropylene sheet material is very thin (like paper) and pliable. Most of the structure of the HMX blocks comes from the outer frame (rigid polypropylene/glass material) and the polyethylene separation walls between each channel. Because the actual sheets are pliable,
varying flows and pressures would likely have an effect on the geometry of each channel; they would not be perfectly rectangular as the model assumes.

**Airflow Knob Factor** — The developer of the model calibrated the airflows with experimental data that was taken at a limited number of performance points. The experiment was performed at NREL facilities. "Knob factors" were used to adjust the theoretical, calculated friction factors within the model to fit this data.

**Hole Placement** — The actual HMX design has holes in the first 2 inlet channels only (10 holes each); the 2nd and 3rd inlet channels are connected through 2 breaks in the separation wall. The concept behind this is to direct more air into the last 5 holes, which otherwise may be limited due to pressure resistance. The model does not have the capability to place breaks in the separation wall. The default model design simply has 10 holes in the 1st inlet channel and 5 holes in each of the next inlet channels.

**Number of Process Layers** — The model assumes that the heat and mass transfer phenomena within an HMX block are symmetrical; each sheet has two sides that are symmetrical across the trough in the middle. NSIDC HMX blocks are 39 inlet channels tall, and the M50 air conditioner has 5 HMX blocks in it. Therefore, the "number of process layers" input on the model should be 390. (39)(2 sides)(5 blocks) = 390. This also assumes that thermodynamic and fluid flow performance is the same for each HMX block stacked on top of each other.

**Steady-State Conditions** — The model can only simulate steady-state Coolerado conditions. It is not designed to simulate varying fan speeds or dynamic inlet conditions, although the actual M50 units are equipped with VFDs and are rarely maintained at a constant speed.

**Discretization** — The NREL model uses a basic discretization scheme. Although official documentation or information about the method used in this model is unavailable, inspecting the code reveals the simple method. The model selects the smaller of two dimensions as the length and width of each discretization square: minimum divider width or minimum channel width divided by 6. If the divider widths are larger than channel width divided by 6, the model forces each channel (product and working) to have at least 6 cells perpendicular to the flow path to account for a temperature and heat transfer gradient perpendicular to the airflow path. This is expected in a cross-flow heat exchanger design like the Coolerado HMX. For all of the simulations and analysis performed in this research, the channel divider width defined the discretization size because the minimum divider width (1.6mm) is significantly smaller than the channel width divided by 6 (which would be about 4.0 mm). Most of the simulations have about 15 cells across each channel and 320 cells along the length of the product (dry) channels. This results in a 320 by 128 grid, or 40960 cells.

The model also refers to a series of psychometric functions that were written in MatLAB the same time the model was developed. These functions were derived from the 2005 ASHRAE Handbook of Fundamentals.
Model Test Conditions (MTCs)

Based on the extensive amount of data collected by the new CR10X DAQ system, a series of 23 potential model calibration conditions were developed. These are referred to as Model Test Conditions (MTCs). For calibration of the model, it was important to choose operating conditions that were relatively steady-state because the Coolerado model is not capable of simulating dynamic conditions. Steady-state conditions were initially selected by visual inspection of plots of the data. The initial intent was to calibrate the model using each of the 23 MTCs to represent a wide range of operation modes and inlet temperatures.

Unfortunately, it was later discovered that the Coolerado units at the NSIDC were not operating as designed. This was evidenced by temperature gradients in both the product airstream and the working airstream. And because only 1 temperature and humidity sensor was placed in each location, even linear approximation of the gradient was not possible. The dynamic nature of this gradient in time added to the complications. After trial and error, it was determined that the water pressure and dispersion within the Coolerado unit was the problem. If either of these features differs from the as-designed specifications for the Coolerado, the temperature gradients will result due to the HMX drying out. Coolerado has largely addressed this issue in the most recent HMX design.

Fortunately, one of the MTCs was able to be taken at a time when the temperature gradient in the product airstream was non-existent and the gradient in the working airstream was known; the sensor in the working air duct was recording 3.3°C (5.9°F) less than the average temperature in the airstream. Using this single point in time MTC, the model was calibrated.

ANALYSIS AND RESULTS

Model Calibration

Calibration of the model to the data was performed by varying the ratio of product channel height to total inlet channel height (which is the sum of product and working channel heights). This is referred to as the product height ratio. Due to inconsistencies in the current manufacturing process, a direct measurement of the product channel height was not possible. A hard epoxy is applied to the front face of the HMX, which causing significant deformation of channel geometry at the inlet. Three different models were simulated and compared to the MTC data: 55/45 split, 60/40 split and 65/35 split. (The numbers refer to product channel and working channel percentages of total inlet channel height, respectively.) The results of the simulations can be seen in Figure 6.

Note that the MTC working air data point is very close to the same temperature as the 65/35 model (in fact the data is 16.16°C and the model predicts 16.18°C). The humidity level is not as close (93.5% data vs. 101.7% model). This is likely due to slight water deprivation in the CU4 unit, which would directly impact the
achievable humidity ratio in the working air stream. The model assumes that there is enough water in each working air channel to fully saturate the wicking surface. In reality, controlling the water flow with that degree of accuracy is very difficult, especially with other influencing factors like building water pressure, filter cleanliness (each Coolerado has a separate water filter), variable airflow and humidity ratio of incoming air. However, on the product air end, the CU4 data and 65/35 model are very close, both in temperature and humidity. Data and model temperatures are less than 0.1°C apart, while humidities are about 0.0003 kg water/kg dry air apart (10.85°C data vs. 10.76°C model and 0.0025 data vs. 0.0028 kg water/kg dry air model). Based on Figure 6, it is evident that the 65/35 split model best approximates the performance of the HMX’s in use at the NSIDC.

With a calibrated model, the performance impact of adjusting different variables could be analyzed. For this sensitivity analysis, an inlet air condition of 40°C (104°F) and 10.1% relative humidity (17.2°C or 31.0°F wetbulb temperature) was selected as an inlet air state. This air condition represents a typical design condition used by Coolerado in their in-house testing, which Coolerado has determined to be an ideal condition for the HMX system. It allows the working air stream plenty of capacity for humidity absorption, which in turn drives the product air temperature down. Ten percent RH is a relatively low humidity, but is common in the western U.S. Temperatures in this same region can reach 40°C in the heat of summer, particularly in the southwest, where cooling capacity is needed most.

The key performance indicator used was the coefficient of performance (COP) which is defined in Equation 6 below. The COP is a robust performance metric that by definition incorporates cooling power, product air temperature, product air

![FIG. 6: Calibration simulations compared to MTC data. Sensitivity Procedure](image)
flow rate and total air flow rate. If any of these variables change, the COP will reflect that change.

This analysis was subject to some basic constraints, as requested by Coolerado. Except for cases in which these variables are modified, they are held constant at the values defined in Table 4.

In total, 45 sensitivity simulations were performed, as outlined in Table 5. The channel height sensitivities were the most complex because the product and working air heights are interrelated when it comes to the thermal performance of the HMX. Because the current HMX frame design has an inlet height limit of 10.0”, changing the height of either (or both) channel changes the number of inlet channels available within this 10.0” limit. And although changing one channel height has an effect, changing both channel heights has a different effect, meaning that the product height ratio (percentage of total inlet channel height that the product channel takes up) may also be important. For instance, a 75/25 split performs differently than a 55/45 split.

Other interesting sensitivities included hole locations, adding product channels and elevation (ambient pressure). The current hole placement layout (10 holes in the first channel, 10 in each of the next 2 channels) is actually very close to the optimal design in terms of COP and modifying this has very little effect on the cooling power. The COP begins to drop off if more holes are included in the first channel and less holes in the next channels because the increased pressure drop reduces the efficiency of the HMX. Adding between 2 and 4 product channels, or increasing the size of the HMX may increase the COP slightly and the cooling power significantly. The elevation (ambient pressure) sensitivities showed that the

\[
COP = \frac{\text{Cooling Power}}{\text{Hydraulic Power}} = \frac{\dot{m}_{PA} C_p (T_{DB,PA,\text{out}} - T_{DB,PA,\text{in}})}{\dot{m}_{\text{Real}} (\Delta P)}
\]

TABLE 4: Sensitivity constraints

<table>
<thead>
<tr>
<th>Constraint</th>
<th>Reasoning</th>
</tr>
</thead>
<tbody>
<tr>
<td>10.0” Tall, 8” Wide, 20” Deep</td>
<td>Current HMX frame size; Changing this would mean extensive modification to Coolerado’s manufacturing line and Coolerado design.</td>
</tr>
<tr>
<td>25.4 mm (1.0”) Channel Width, O.C. divider to divider</td>
<td>Reducing this value while maintaining the HMX inlet face area would mean adding channels and/or widening dividers. This would decrease the effective area available for heat transfer on product and working air sheets. Increasing the channel width would improve performance by increasing the heat transfer area, however 1.0&quot; is a structural maximum according to Coolerado.</td>
</tr>
<tr>
<td>Constant 170 Pa (0.68”) Fan Pressure</td>
<td>Current maximum pressure differential achievable with the fan, at least the types installed in the NSIDC.</td>
</tr>
<tr>
<td>Constant 83.3 kPa (12.8 psi) Atmospheric Pressure</td>
<td>Elevation of Boulder (the location of the NSIDC) is 5300 ft. and atmospheric pressure should be taken into account.</td>
</tr>
</tbody>
</table>
COP can be increased by over 20% by using Coolerados at sea level as opposed to 5,300 ft. in Boulder, CO. It should be noted, however, that COP used does not include fan input power; and fan input power would increase with ambient pressure because the fan would have to work harder to push the denser air.

Varying the inlet air condition (temperature and humidity) also has an effect on the HMX performance, as expected. Hot and dry air states are ideal and result in the highest COPs. This can be seen in Figure 9.

**NSIDC Coolerado Operation**

The results of the inlet condition sensitivity simulations were applied to the NSIDC to determine when the Coolerado system can be used to cool the data center. As discovered by the sensitivity analysis, the performance of the HMX degrades significantly in cooler and very humid conditions. This is largely due to decreased cooling power. Although the wet-bulb effectiveness is about the same, at

**TABLE 5: Sensitivity simulation details**

<table>
<thead>
<tr>
<th>Parameter changed</th>
<th>Description</th>
<th>Values</th>
<th># Simulations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Channel Heights</td>
<td>±10%, ±5% Height Ratio &amp; ±4 Inlet Channels</td>
<td>55, 60, 70, 75% Product Channel Height to Total Inlet Height</td>
<td>15</td>
</tr>
<tr>
<td>Divider Width</td>
<td>±1.0 mm</td>
<td>1.0, 1.5, 2.5, 3.0 mm</td>
<td>4</td>
</tr>
<tr>
<td>Hole Diameter</td>
<td>±1.75 mm, +2.25 mm</td>
<td>4.0, 5.5, 6.25, 7.0 mm</td>
<td>4</td>
</tr>
<tr>
<td>Hole Location</td>
<td>±1 hole, ±2 holes, −3 holes</td>
<td>7, 8, 9, 11, 12 holes in first row</td>
<td>5</td>
</tr>
<tr>
<td>Waterfilm Thickness</td>
<td>±30% of Base Case</td>
<td>0.34, 0.64 mm</td>
<td>2</td>
</tr>
<tr>
<td>Water Conductivity</td>
<td>±50%, ±25% of Base Case</td>
<td>0.3, 0.45, 0.75, 0.9 W/(mK)</td>
<td>4</td>
</tr>
<tr>
<td>HMX Sheet Size</td>
<td>Adding Product and Working Channels</td>
<td>+2,4,6 Working Channels +2,4,6,8 Product Channels</td>
<td>11</td>
</tr>
</tbody>
</table>

![FIG. 7: 3D plot of COP for product height ratio sensitivities.](image)

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higher humidity ratios, the dew-point is also higher. And because the Maistosenko cycle is driven by the dew-point of inlet air, this results in less available heat transfer and difference between product and working air enthalpies. Because of this effect, the cooling capacity of the Coolerados at the NSIDC is not enough to cool the constant heat load (servers) for some inlet air conditions. In these scenarios, the backup CRAC should be turned on to supplement the Coolerados and help maintain the ASHRAE air state as defined in Table 1.

The known cooling load of the NSIDC is about 40 kW (derived from the power measurements). And by interpolating linearly between the inlet condition sensitivity simulations (and a few other simulations run just for this purpose), a line can be drawn on a psychrometric chart showing the limits of where the 8 Coolerados can be exclusively used to cool the data center. Outside of this range, the CRAC must be used to dehumidify the space. Figure 10 shows the operation modes. Note that

![Optimal Product Height Ratio](image)

**FIG. 8:** Optimal product height ratio.

![3D Psychrometric Performance](image)

**Fig. 9:** 3D HMX performance sensitivity to varying inlet condition.
the curved line between the CRAC and Coolerado operations is steeper than the relative humidity lines. This is a similar trend as was seen in the inlet air state sensitivities (see Figure 9) and is reasonable because cooling capacity has a direct impact on COP. The ASHRAE allowable upper humidity limit defines the limit of the Coolerado use above about 27.2°C (81°F); if there is more moisture in the air than this limit at the inlets to the Coolerados, the CRAC must be used to dehumidify the air. The Coolerados cannot remove moisture from the air. If the recommended limit is the target operation for the data center, then the CRAC may need to be run more often. Below an Coolerado inlet air temperature of 18.3°C (65°F), the economizer mode can be used exclusively to meet the cooling load.

CONCLUSION

NSIDC Energy Use

Since the installation of the new cooling system in June 2011, the Coolerado units and the AHU have reduced the energy required to cool the data center by almost 90% in winter months and over 70% in hot summer months. One of Coolerado’s early projections was 80% savings in energy (Modera, 2009); the NSIDC has certainly demonstrated that this is achievable and even beatable.

The new cooling system is relatively simple and has fewer extensive maintenance requirements. The lower maintenance and energy savings translates to significant operational cost reduction for the NSIDC. This results in a very reasonable payback period. The Coolerado air conditioner uses a unique technology that has demonstrated savings potential, and is a great addition to the NSIDC’s facility to further their vision of reduced environmental impact as the center continues its research on climate related issues.
Suggested HMX Design Improvements

It is evident that the Coolerado system is very efficient at delivering cooling power to an environment with a high heat load. However, there is always room for improvement and a portion of this research was devoted to sensitivity analysis to better understand how the HMX performance relates to each of the design variables. It was found that channel heights and the ratio between them have a significant impact on HMX performance; the optimal channel height chart (Figure 8) could be used to fine tune the HMX for optimal performance with a given inlet condition. Divider width was the parameter with the next biggest practical impact, but the COP increase associated with decreasing divider width is very small. Hole diameter reduction had a greater impact, however, this design variable may not be a practical way to change the performance because current manufacturing tolerances limit the precision needed to adjust this dimension accurately. Other parameters, hole location, water conductivity and waterfilm thickness, were found to have insignificant effects on the performance.

It is understood that some of these issues or changes may have already been addressed in newer versions of the HMX design. The theoretical MatLAB model, developed by NREL, has been validated at one point in time through this research, and can be applied to the actual design process in predicting the performance of the Coolerado units under a specified set of conditions. Sensitivity analysis was performed to determine which design parameters have the most impact on Coolerado performance.

Future Work

An extensive amount of time was put into this project from installing the datalogging system, collecting and consolidating data, validating that the model has enough theoretical detail and proving its relative accuracy, and using the model to simulate improved design strategies. However, there is still more work that can be done to further the understanding of the applicability of the Maisotsenko Cycle and the Coolerado system to the built environment. First of all, although good agreement was found between simulated and actual performance at one point in time, the original intent of this research was to validate the model over a range of actual operating conditions (all MTCs). Due to limited datalogging system capabilities, not enough measurements were taken at every point in time to properly characterize the thermal performance of the Coolerado units. More data that includes multiple sensor locations in both outgoing airstreams should be collected for a range of inlet conditions. Once the model is more thoroughly validated, there would be more confidence in its results.

A formal optimization simulation was not developed for the model. Although the sensitivity simulations show the effects that each individual parameter has the performance of the HMX, changes in more than one parameter require many more
simulations to be performed. To find the optimum values for each parameter requires an optimization algorithm. Because the model is completely based in MATLAB, developing an optimization engine should be possible and within the scope of further graduate student work. One challenge to this will be determining how to interface with the complex and robust GUI that has already been developed for this model.

Once more completely validated, the model could be used to simulate annual performance, based on a given set of inlet conditions (from a weather file like TMY3) and operating times (occupancy schedule). Ultimately, this research may prove useful in populating equipment libraries of whole-building energy simulation tools with theoretically accurate and experimentally validated components, namely the Coolerado air conditioner. As sustainable and integrated building design becomes more prevalent, tools to validate low-energy systems like Coolerados are necessary for proof of concept.

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