Public Interest Energy Research (PIER) Program

FINAL PROJECT REPORT

Evaporative Cooling Retrofit Strategies for Retail Buildings

Testing Non-Chemical Water Treatment Methods

Prepared for: California Energy Commission

Prepared by: Western Cooling Efficiency Center, University of California at Davis

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PREFACE

The California Energy Commission Public Interest Energy Research (PIER) Program supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

The PIER Program conducts public interest research, development, and demonstration (RD&D) projects to benefit California.

The PIER Program strives to conduct the most promising public interest energy research by partnering with RD&D entities, including individuals, businesses, utilities, and public or private research institutions.

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- Buildings End-Use Energy Efficiency
- Energy Innovations Small Grants
- Energy-Related Environmental Research
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- Environmentally Preferred Advanced Generation
- Industrial/Agricultural/Water End-Use Energy Efficiency
- Renewable Energy Technologies
- Transportation

Evaporative Cooling Retrofit Strategies for Retail Buildings, Testing Non-Chemical Water Treatment Methods is a final report for the Retrofit Strategies for Retail Buildings project, contract number 500-99-013 and work authorization number POB224-D29, conducted by the Western Cooling Efficiency Center. The information from this project contributes to PIER's Buildings End-Use Energy Efficiency Program.

For more information about the PIER Program, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-654-4878.

ABSTRACT

Evaporative pre-cooling of condenser air improves the efficiency of compressor-based cooling, particularly in the hot, dry climate that exists in most of California. When implementing evaporative pre-cooling technologies, one of the main concerns for end-users is deposition of minerals contained in the water onto the surfaces of the heat exchanger, condenser coils, or cellulose based media. This project consisted of a laboratory investigation of the performance of two commercially-available non-chemical water treatment systems, as applied to evaporative pre-cooling of condenser air.

The major claim of most non-chemical water treatment technologies regarding the mechanism by which they work is that they precipitate minerals in bulk water and reduce the amount of calcium carbonate scale that forms on heat exchanger surfaces. An experimental method and apparatus was developed to evaluate non-chemical water treatment products applied to a spray-evaporative cooling application and to provide results on two specific products.

A physical treatment device manufactured by Watts OneFlow extended the life of the heat exchanger under test by 14% over the baseline when installed with a sufficient flow rate through the device. A permanent magnet device manufactured by GMX extended the life of the heat exchanger under test by 28% over the baseline. Execution of the tests and analysis of the results generated additional insight into how future testing can be improved to reduce uncertainty. However, the results obtained to date provide confidence that physical water treatment systems have the potential to reduce scale formation in evaporative cooling systems and should be further investigated.

Keywords: evaporative cooling, evaporative pre-cooling, hard water, water treatment, non-chemical water treatment, physical water treatment, scale formation

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TABLE OF CONTENTS

EXECUTIVE SUMMARY

Evaporative pre-cooling of condenser air improves the efficiency of compressor-based cooling, particularly in the hot, dry climate that exists in most of California. When implementing evaporative pre-cooling technologies, one of the main concerns for end-users is deposition of minerals contained in the water onto the surfaces of the heat exchanger, condenser coils, or cellulose based media. This project consisted of a laboratory investigation of the performance of two commercially-available non-chemical water treatment systems, as applied to evaporative pre-cooling of condenser air.

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The experimental apparatus was designed to simulate water-spray pre-cooling of outdoor air that is then blown through a condenser coil. The experimental apparatus substituted a hot-water coil for the condenser coil. More significantly, the key difference between the experimental apparatus and current commercial applications is that the experimental apparatus does not attempt to protect the coil from un-evaporated water-spray droplets in any way. Instrumentation located throughout the apparatus was used to monitor the performance of the system, including air flow rate, air conditions before and after the water spray, air pressure drop across the coil, water temperature at the inlet and outlet of the water coil, water flow rate through the water coil, and properties of the spray water. Digital on/off control of duct heaters was used to condition the incoming air to maintain a constant wet-bulb depression of 20°F, and the operation of the water spray was also managed by a digital on/off control. In order to better emulate the behavior of a spray evaporative system in a real application, the spray water was cycled on (60 minutes) and off (30 minutes). The fans and the circulation pump for the water coil remained on during the spray-off period. This allows the heat exchanger to dry out periodically, which occurs with actual condensers in the field.

Each experiment was run, starting with a new heat exchanger, until the air flow resistance across the heat exchanger reached a failure point due to scale build-up. The test was run under four conditions:

1. No water treatment (baseline)
2. Watts OneFlow water treatment (at 0.024 GPM)
3. GMX (magnetic water treatment)
4. Watts OneFlow water treatment (at 0.6 GPM through device, 0.024 GPM thru nozzle)


FIGURE 1 - EXAMPLE HEAT EXCHANGER AT FAILURE

After the first test of Watts OneFlow, communication with the manufacturer indicated that the minimum water flow required for the system to perform effectively and fluidize the treatment media is 0.5 GPM or higher. Thus, the test of this device was repeated at a higher flow rate, with the required flow rate supplied to the nozzle and excess water dumped to the drain.

For each test, the cumulative mineral mass sprayed on the heat exchanger in grams was calculated when the heat exchanger reached the defined failure point (an air-flow resistance of 0.03 $Pa^{0.6}$/CFM, which is an increase of 275% over the initial air-flow resistance). Changes in performance were calculated based upon the cumulative mineral mass sprayed on the heat exchanger associated with failure during treated-water tests, as compared to the cumulative mineral mass sprayed on the heat exchanger during the baseline test, with the following results (Table 1):

1. The OneFlow test result at 0.024 gpm shows a reduced lifetime compared to the baseline, but the result is not statistically significant.
2. The second OneFlow test result at 0.060 gpm shows an improved lifetime 14%greater than the baseline. The result appears to be statistically significant.
3. The GMX test result shows an improved lifetime 28%greater than the baseline. The result appears to be statistically significant.

TABLE 1 - CUMULATIVE MINERAL MASS SPRAYED ON HEAT EXCHANGER AT FAILURE FOR EACH TEST

	Cumulative mineral mass sprayed on heat exchanger (g)	Uncertainty of Cumulative mineral mass sprayed on heat exchanger (g)	% Difference from Baseline
No Treatment	360	16.1	
OneFlow (0.024 GPM)	335	10.6	-7%
GMX	460	11.8	28%
OneFlow (0.6 GPM)	410	14.9	14%

In analyzing the data, it became clear that many assumptions and corrections were needed to derive the end results. While these assumptions and corrections were consistent in all the analyses, the results should be reviewed within this context. The following items should be considered:

1. Incoming water quality was not controlled and was highly variable. The results were normalized by total hardness, but this may not be the only factor affecting scale formation. A review of the City of Davis water quality report[1] shows the presence of metallic cations such as iron, copper, and zinc, which could affect in scale formation. The variance of these metallic ions in the water supply over the course of the tests is unknown.
2. Each experiment used a new spray nozzle that was positioned manually using a marking on the wall of the duct. Confidence in this assumption was gained by

[1] City of Davis Water Quality Report 2009
http://cityofdavis.org/pw/water/pdfs/2009_waterqualityreport.pdf

reviewing the pressure at the nozzle and the evaporated water metric, which was consistent between experiments.

3. Each experiment used a new heat exchanger, all of which were purchased from the same manufacturer.
4. Measurements from ten instruments were required to obtain the final results. Confidence in the results was gained by performing an uncertainty analysis and assuring that the same instruments were used throughout all experiments.
5. It was assumed that evaporated water did not remove any minerals from the system and that the minerals concentrated equally in the remaining water that was partially sprayed on the heat exchanger and partially drained from the duct. A one-time experiment in which the conductivity of the drained water was measured showed this assumption to be reasonable.
6. An additional correction was needed to obtain spray flow rate for the second OneFlow test. Although post-test experiments showed that this correction is valid, this is an inconsistency in the test procedure that adds to the uncertainty of the second OneFlow test result.

Further tests evaluating water treatment technologies should consider the following recommendations to reduce uncertainty in the results:

1. Incoming water quality should be consistent between tests. This could be done either by running tests in parallel (baseline versus treatment under test) with the same incoming city water or by mixing hard water in the laboratory.
2. The spray mechanism is quite complicated. A rig that re-circulates water over a hot coil may be more practical. This assures that all the water hits the coil and that the complication of drained water is removed.
3. For minerals to reach the heat exchanger and not contribute to flow resistance, the assumption is that the minerals are not "sticking". Mineral dust was clearly seen in the laboratory but was not caught or measured. There may be a way to filter this dust from the exiting air stream and collect it for measurement.

While the results of these experiments have provided evidence that physical water treatment systems have applications to evaporative cooling, further research is needed to understand the mechanism by which scale buildup is reduced. Subsequent research, which will focus on understanding the mechanisms by which physical water treatment systems reduce scale, will allow improved experimental designs to quantify the performance and application of various technologies to reduce water consumption and/or improve the performance of evaporative coolers.

1 Introduction

This project contributes to PIER's Buildings End-Use Energy Efficiency Program by conducting a laboratory investigation of two commercially available non-chemical water treatment systems as they may be applied to evaporative cooling products.

1.1 Background

When implementing evaporative pre-cooling technologies, one of the main concerns for end-users is deposition of minerals contained in the water onto the surfaces of the heat exchanger, condenser coils, or cellulose based media. The most common scale forming minerals in water are calcium and magnesium, which, in the presence of carbonate in water, form crystals of calcium carbonate ($CaCO_3$) and magnesium carbonate ($MgCO_3$). The total concentration of magnesium and calcium in water is called water hardness. Calcium and magnesium carbonate are inversely-soluble minerals, meaning that their solubility decreases with increasing temperature. This phenomenon, combined with the evaporation of water, causes the solution to become saturated and the minerals to precipitate on hot heat exchanging surfaces, causing crystal formation, commonly known as scale. Deposition of scale on the heat exchanging surfaces blocks air flow and reduces the capacity and energy efficiency of the system over time.

In order to reduce the concentration of scale-forming minerals in evaporative systems that circulate water, the water in the sump is either regularly bled off, or the sump is completely purged on a schedule, replaced with fresh make-up water in both cases. Treating water for hardness could reduce the water consumption for evaporative cooling systems, by reducing or eliminating the need for bleeding or purging. In once-through spray evaporative systems, where water is not re-circulated, treated water could reduce build up of scale on heat exchangers and maintain equipment performance over time.

Chemical water treatments change the chemistry of the water to inhibit the formation of scale. Most chemical water treatment systems need to be regenerated and their waste products require proper disposal. The costs for the chemicals and the regeneration of the system, environmental concerns regarding disposal of regeneration waste, and safety issues concerning handling the chemicals have led to the search for alternative options. This has generated significant interest in non-chemical or "physical" water treatment by electric utilities looking to promote the use of evaporative coolers, and by end-users looking to solve problems caused by water hardness.

Non-chemical water treatment systems use no chemicals and generally do not need regeneration. Some of the non-chemical water treatment technologies on the market include magnetic, electromagnetic, electrostatic, and AC induction technologies. The major claim of most of these technologies regarding the mechanism by which they work is that they precipitate minerals in bulk water and alter the crystal form of calcium carbonate from (the more difficult to remove) calcite to (the easy-to-remove) aragonite. The studies of these technologies often lack a baseline or control experiment, as well as comprehensive experimental designs and methods. The result is that there is no solid understanding of the mechanisms, and inconclusive results about the performance of non-chemical treatment technologies. The objective of this project was to evaluate the performance of two non-chemical treatment technologies in a once-though application of spray-evaporative condenser-air pre-cooling.

1.2 Project Objective

The objective of this project was to:

1. Conduct a laboratory investigation of the performance of two commercially-available non-chemical water treatment systems, as applied to flash-evaporative (i.e. spray) pre-cooling of condenser air (for both refrigeration and A/C), and recirculation water systems (e.g. sump system for direct and indirect evaporative cooling).

1.3 Report Organization

This report is organized as follows:

Section 1.0	Introduction
Section 2.0	Project Approach
Section 3.0	Project Outcome
Section 4.0	Conclusions
Section 5.0	Recommendations

2 Project Approach

An experimental apparatus was designed to test the application of non-chemical water treatment technologies for spray-evaporative pre-cooling of condenser inlet air. Direct evaporative pre-cooling can be accomplished using different methods to evaporate water prior to the condenser coil. In this setup, water was pressurized to 80-100 psi and sprayed into the air stream (often called “flash” evaporative). No water is re-circulated. A portion of the water evaporated and the rest was permitted to hit a simulated condenser coil to test if the water treatment technologies reduce scale buildup in this application. In practice, a cellulose media pad (often called a “drift eliminator”) would be used to protect the coil from droplets. However, in this experiment, no media pad was used in order to accelerate the impact on the coil.

Three experimental tests were conducted: 1) No water treatment, 2) Watts OneFlow water treatment, and 3) GMX water treatment.

The Watts OneFlow system consists of a chamber containing small beads that the manufacturer claims reduces scale formation due to hard water. According to the manufacturer of OneFlow, the beads use template assisted crystallization to attract the dissolved hardness minerals to the nucleation sites on the beads and convert them to micro-crystals that then break off from the sites and float freely though the system. The manufacturer claims the precipitated minerals are less likely to stick to heat exchanger surfaces and cause scale. The OneFlow system uses no salts or chemicals. The unit size was determined based on manufacturer recommendations considering the flow rate for the experiment. The single-cartridge Model OF110-1, which operates at up to 1 GPM, was selected.

GMX is a permanent-magnet unit placed around the water tube with the resulting magnetic flux lines perpendicular to the water flow path. The manufacturer claims that the magnetic field causes a small surface charge on the minerals, causing them to lose their bonding ability and preventing them from depositing on the surface of pipes. Other manufacturers of the magnetic water treatment technologies claim that the magnetic technology precipitates the calcium carbonate in the water stream which makes the minerals less likely to stick to the heat exchanger and cause scale. The strength and number of the GMX units were recommended by the manufacturer based on the pipe size, flow rate, and incoming water hardness. Three units of Model GMX-400 in series were recommended. According to the manufacturer, this results in a magnetic field with a magnitude of 0.4 Telsa for the distance of the magnets, approximately 0.6 ft in total length. Because the water is moving through the magnetic field at 0.3 ft/s, the water is exposed to the magnetic field for approximately 2 s.

2.1 Experimental Design

The experimental apparatus was designed to simulate water-spray pre-cooling of outdoor air that is then blown through a condenser coil (Figure 2). The experimental apparatus substituted a hot-water coil for the condenser coil. More significantly, the key difference between the experimental apparatus and current commercial applications is that the experimental apparatus does not attempt to protect the coil from un-evaporated water-spray droplets in any way. Instrumentation located throughout the apparatus was used to monitor the performance of the system, including air flow rate, air conditions before and after the water spray, air pressure drop across the coil, water temperature at the inlet and outlet of the water coil, water flow rate through the water coil, and properties of the spray water (Table 2). Digital on/off control of duct heaters was used to condition the incoming air to maintain a constant wet-bulb depression of 20°F, and the operation of the water spray was also managed by a digital on/off control. In order to better emulate the behavior of a spray evaporative system in a real application, the spray water was cycled on (60 minutes) and off (30 minutes). The fans and the circulation pump for the water coil remained on during the spray-off period. This allows the heat exchanger to dry out periodically, which occurs with actual condensers in the field.

Data acquisition equipment interfaced with LabVIEW 8.6 software, which took measurements at 2 Hz, saving the average values to a text file every 30 seconds. The data file was e-mailed to staff every 1 ½ hours so that the experiment could be monitored when the laboratory was closed.

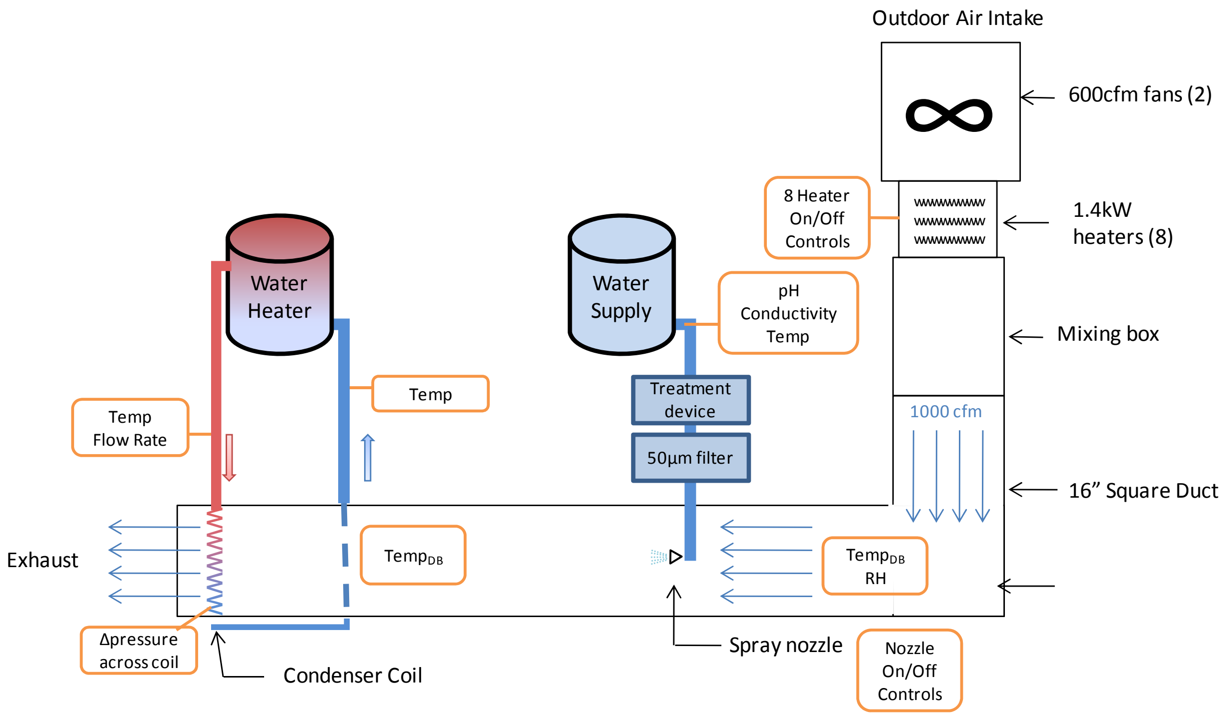

FIGURE 2 - EXPERIMENTAL APPARATUS AND MEASUREMENT LOCATIONS

TABLE 2 – LIST OF RECORDED DATA AND SENSORS/DATA ACQUISITION DEVICES

	Record	Units	Sensor Type	DAQ/Control Device
1	Date and Time			
2	Dry Bulb Temperature - Before Water Spray	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
3	Water Temperature - Output of Heat Exchanger	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
4	Water Temperature - Input of Heat Exchanger	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
5	Outdoor Air Temp	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
6	Dry Bulb Temperature - After Water Spray	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
7	Water Temperature - Spray	°F	T-Type Thermocouple	National Instruments 9213 Calibrated 09/09
8	Spray Water - Conductivity	µS/cm	Conductivity Sensor Omega CDTX-300 Calibrated Each Test	National Instruments 6321 Calibrated 09/09
9	Spray Water - pH	--	pH Sensor Omega PHE-7351-15 Calibrated Each Test	National Instruments 6321 Calibrated 09/09
10	Dry Bulb Temperature - Before Water Spray	°F	Resistive Thermometer Vaisala HMD50Y Calibrated 07/09	National Instruments 6321 Calibrated 09/09
11	Relative Humidity – Before Water Spray	% RH	Relative Humidity Sensor Vaisala HMD50Y Calibrated 07/09	National Instruments 6321 Calibrated 09/09
12	Wet Bulb Temperature – Before Water Spray	°F	Calculated using the Psychometric chart	
13	Pressure Drop Across Heat Exchanger	Pa	Differential Pressure	Energy Conservatory DG-500 Calibrated 09/08
14	Air Flow Rate	CFM	Differential Pressure	Energy Conservatory DG-500 Calibrated 09/08
15	Spray Water Pressure	psi	Absolute Pressure (Omegadyne PX309-200A5V) Calibrated 12/09	National Instruments 6321 Calibrated 09/09
16	Water Flow Rate Through Heat Exchanger	GPM	Turbine Flowmeter Omega FTB-4607	National Instruments 6321 Calibrated 09/09
17	# of Heaters On	--	Relay On/Off	National Instruments 6321 Calibrated 09/09
18	Spray ON/OFF Status	--	Relay On/Off	National Instruments 6321 Calibrated 09/09
19	Hot Water Recirculation ON/Off Status	--	Relay On/Off	National Instruments 6321 Calibrated 09/09

The apparatus consists of two axial centrifugal fans of variable speed with a range of 10 to 600 cubic feet per minute (cfm) each. The speed can be set by an analog dial on the side of the fan box. The fans were powered by a battery backup to stabilize the voltage in an effort to stabilize the flow rate. The fans were used to draw outdoor air into the box through filters and then push that air through the duct. Outdoor air temperature was measured by a thermocouple installed at the inlet to the fan boxes.

The inlet gates to the fan boxes can be easily varied in size by adjusting a sliding piece of sheet metal. Testing with a calibrated flow meter determined the relationship between pressure drop across the inlet and the volumetric flow rate in cfm of air for different fixed inlet sizes. The result showed that, for a given inlet size, the volumetric flow rate is the square root of the pressure differential multiplied by a constant associated with the size of the inlet (Equation 1),

$$\dot{V} = G\sqrt{\Delta P} \qquad 1$$

where G is the "gate coefficient" associated with each different size of inlet on the fan box. The largest gate with a coefficient of 72 CFM/(Pa)$^{0.5}$ was used in this experiment and the target flow for each fan was 500 CFM, for a total of 1000 CFM. The differential pressure reading across the gate was made using a differential pressure gauge from Energy Conservatory (DG-500).

The air leaving the fans passed through a two sets of heaters (one set for each fan box) and entered a mixing box (Figure 3). The heaters have eight stages, with each stage delivering an additional 1.4kW of heat. Because evaporation rate is primarily driven by the differential between the dry bulb temperature and the wet bulb temperature of the air, the heaters were controlled to maintain a target wet-bulb depression of 20°F prior to the water spray throughout the experiment. Turning on heaters increases the wet bulb depression of available outdoor air, but the experiment did not have equipment to reduce wet bulb depression. If the outdoor air wet bulb depression exceeded 22°F, the experiment held in the spray-off condition until the wet bulb depression of incoming outdoor air dropped below 22°F.

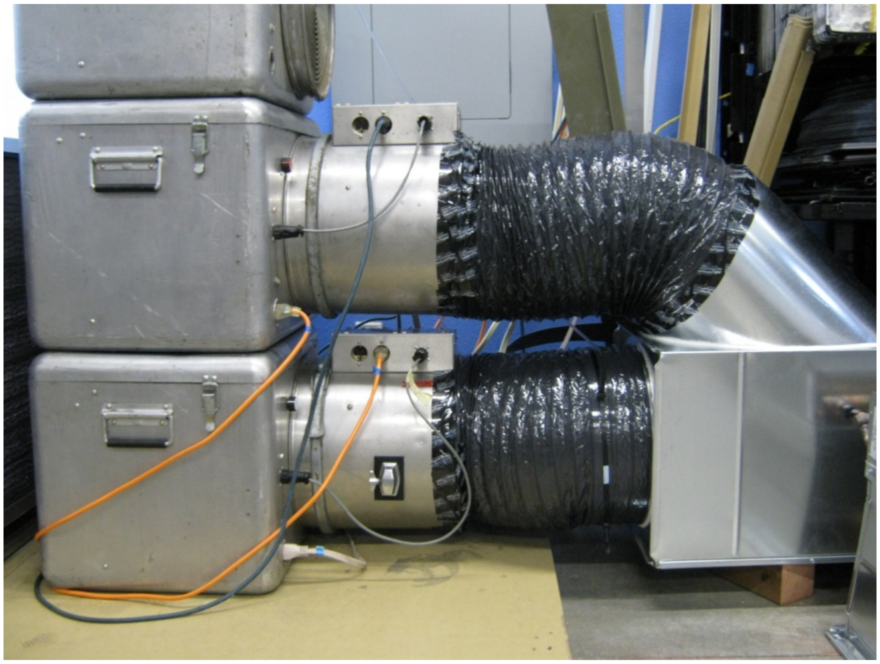

FIGURE 3 – PHOTO OF THE FANS, HEATERS, AND MIXING BOX

After exiting the mixing box, the duct turns 90º to fit within the laboratory space and then connects of a piece of 11′ long duct. Two transparent windows are located on the top of the long duct for access and viewing of the experiment. An averaging type-T thermocouple with 10 nodes was placed in the duct before the spray to obtain the average air temperature (Figure 4). The temperature and relative humidity (RH) of the supply air was measured in the center of duct with a temperature/RH transducer (Vaisala HMD50Y).

The water was transported from the city water supply to the spray nozzle using a peristaltic pump (Cole Parmer 7524-40 with high-pressure pump head 77250-62). The water was sprayed through a nozzle located 3.5′ from the beginning of the straight duct and 7.5′ from the face of the condenser coil (Figure 4). The brass nozzle (Bete PJ-10), which is marketed for evaporative cooling applications, has an orifice of 0.01 inch and a 90° spray pattern. The peristaltic pump operated at a flow rate of 0.024 GPM, which results in a spray pressure of 80-100 psi. The back pressure of the nozzle was monitored using a pressure transducer (Omegadyne PX309-200A5V) installed in line with the nozzle.

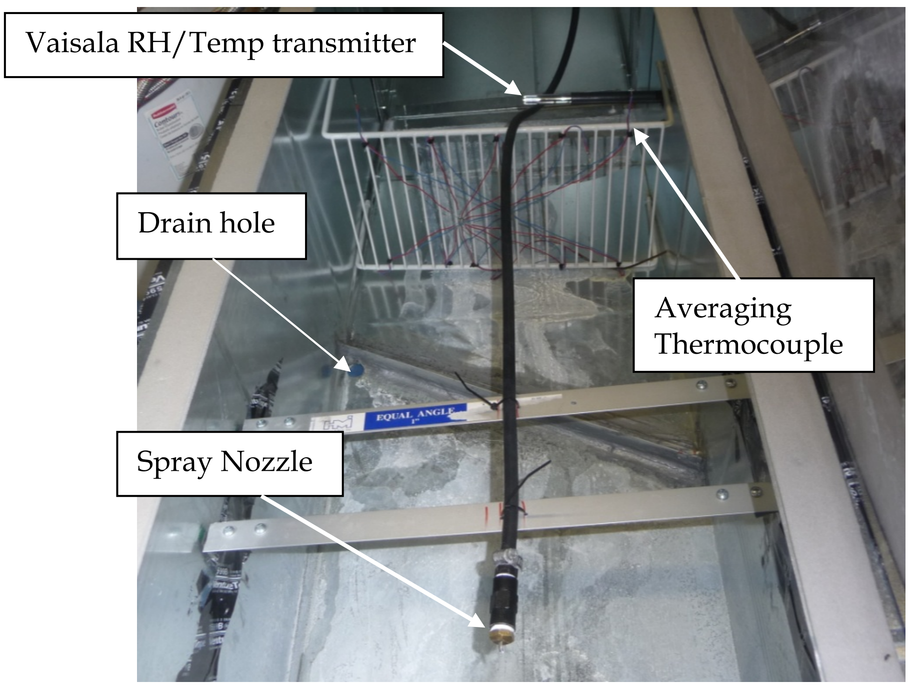

FIGURE 4 - PHOTO OF AVERAGING TEMPERATURE, SPRAY NOZZLE AND TEMPERATURE/RH TRANSDUCER

A sediment filter for particles >50μm was installed prior to the spray nozzle pump to capture any debris in the water line. In the OneFlow experiment, the sediment filter was installed after the treatment as recommended by the manufacturer because it was found that the OneFlow releases some debris. In the GMX experiment, the sediment filter was installed prior to the treatment as recommended by the manufacturer. In either case, the sediment filter does not remove minerals from the water, as the minerals are too small to be captured by the filter and therefore the filter does not affect the hardness.

The OneFlow treatment unit was applied as recommended by the manufacturer using the smallest unit available, Model OF110-1, with a maximum flow rate of 1 GPM. The unit was installed prior to the spray pump (Figure 5, left). In the first test of the OneFlow, the flow rate of water passing through the OneFlow unit was the same as the spray flow, 0.024 GPM. After completing this test and discussing the results with OneFlow, they recommended running the unit at a higher flow rate to assure that the media beads inside the cartridge were fluidized by the water flow. Thus, for the second OneFlow test, 0.6 GPM of water was run through the OneFlow. To regulate the flow rate at 0.6 GPM, another peristaltic pump (Cole Parmer 77410-10 with pump head 77601-10) was installed in parallel with the peristaltic pump serving the spray nozzle. The excess water from the added peristaltic pump was dumped to the drain.

The GMX treatment was applied as recommended by the manufacturer, using three GMX-400 units in series installed around the plastic tubing between the peristaltic pump and the spray nozzle (Figure 5, right). According to the manufacturer, this results in a magnetic field with a magnitude of 0.4 Telsa for the distance of the magnets, approximately 0.6′ in total length. Because the water is moving through the magnetic field at 0.3 ft/s, the water is exposed to the magnetic field for approximately 2 s. The GMX units began 1.5′ after the exit of the pump and ended 5′ before the beginning of the nozzle.

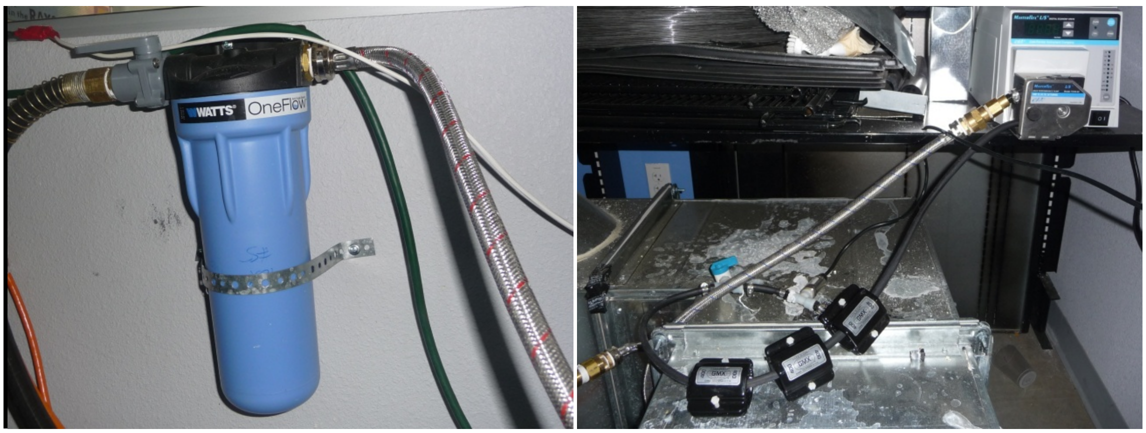

FIGURE 5 - INSTALLATION OF ONEFLOW (LEFT) AND GMX WATER TREATMENT (RIGHT)

A portion of the water droplets from the spray hit the walls of the duct and collected in the bottom of the duct. The duct was placed on a slight incline and a drain hole was made in the bottom surface of the duct, approximately one foot behind the spray location. The water droplets that did not evaporate in the air stream or on the heat exchanger drained through this hole into a container. The water in the container was measured and dumped on a daily basis.

It is extremely difficult to measure the condition of the air after the water spray because temperature and relative humidity sensors do not function well in wet conditions. Even for temperature sensors that can withstand water (such as thermocouples), water droplets evaporating from the sensor result in a reading of temperature somewhere between wet bulb and dry bulb. In order to ascertain the condition of the air after the spray, the wet-bulb temperature after the spray is assumed to be the same as before the spray, which is valid since no significant heat is added or removed from the system. In order to measure the dry bulb temperature, an improvised apparatus was used. This apparatus involved inserting a 1″ plastic tube with 3/16″ walls into the center of the duct (Figure 6, left). The thermocouple is placed near the outlet of the tube. Originally, a fan was operated at the outlet of the tube, but it was found to be unnecessary, as the elevated pressure inside the duct causes the air to flow through the tube to the room.

The air stream with water droplets passes through a water coil to air heat exchanger (which simulates a condenser coil) at the end of the duct. The heat exchanger was selected to have properties similar to a condenser coil with 10 fins per inch. The hot water was supplied to the heat exchanger at a typical condenser inlet temperature of 130°F. Water temperature was controlled using a gas-fired commercial tank water heater with a volume of 100 gallons and a maximum heating capacity of 199 kbtu/hr (Figure 6, right). Installation consisted of piping the hot water to the supply line of heat exchanger through a through a single speed hot water circulating pump with 1/40 horsepower (Bell & Gossett NRF-9F/LW), and piping the return water from the heat exchanger through a flow meter (Omega, FTB-4607) (Figure 7).

FIGURE 6 - DEVICE TO MEASURE DRY BULB TEMPERATURE AFTER THE SPRAY (LEFT) AND WATER HEATER (RIGHT)

Type-T thermocouples were attached to the pipes and insulated to measure the inlet and outlet temperatures of the heat exchanger. Lastly, the differential pressure across the heat exchanger was measured using a differential pressure gauge from Energy Conservatory (DG-500). Originally, the pressure measurement port upstream of the heat exchanger was on the side of the duct. During the first experiment, the port tended to get clogged with water. The measurement location was then moved to the top of the duct which solved the problem.

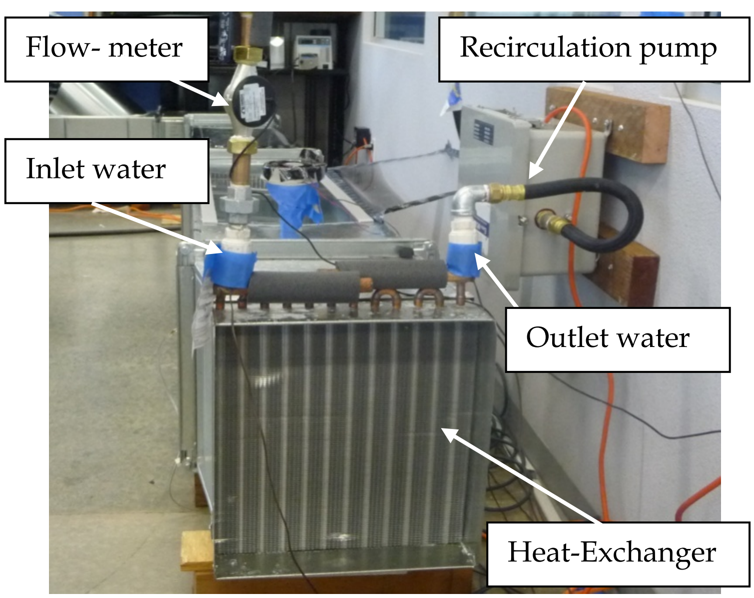

FIGURE 7 - INLET/OUTLET WATER LINES, FLOW-METER, HEAT EXCHANGER, AND CIRCULATION PUMP

As incoming supply water from the City of Davis is highly variable, the pH and conductivity of the sprayed water was continually monitored (Omega CDTX-300 and PHE-7351-15) (Figure 8). The conductivity and pH sensors were calibrated with standard solutions before each experiment. Water samples were taken at different points during the experiment for analysis in a commercial laboratory to correlate conductivity to total hardness. In addition, conductivity and pH were measured before and after the treatment technology to evaluate if a change in either as a result of the treatment could be measured.

FIGURE 8– PH AND CONDUCTIVITY SENSORS IN WATER LINE OF SPRAY WATER

The measured data was recorded and stored in a text file using a multifunction data acquisition system from National Instruments (Model PCI-6321) that provides up to 16 analog input channels for resistance thermometers, humidity sensors, and water quality instrumentation, 24 digital input/output channels to turn on and off heaters and pumps, and 4 counters for water flow sensors. The data acquisition equipment also included two 2-channel digital pressure gages from Energy Conservatory (Model DG-700) and an 8-channel thermocouple module from National Instruments (Model 9213) with integrated cold junction compensation (Figure 9). LabView 8.6 software was used to collect and record the data and issue the control commands. The data was sampled at 2Hz, and averaged and stored every 30 seconds. The graphical LabView interface has indicators to visually observe the status of the experiment. The data file was e-mailed to staff every 1 ½ hours so that the experiment could be monitored when the laboratory was closed.

FIGURE 9–THERMOCOUPLE MODULE, PRESSURE GAGES, AND MULTIFUNCTION MODULE CONNECTOR BLOCK

2.2 Test Protocols

The experiment was run under four conditions:

5. No water treatment (baseline)
6. Watts OneFlow water treatment (at 0.024 GPM)
7. GMX (magnetic water treatment)
8. Watts OneFlow water treatment (at 0.6 GPM through device, 0.024 GPM thru nozzle)

After the first test of Watts OneFlow, communication with the manufacturer indicated that the minimum water flow for the system to perform effectively and fluidize the treatment media is 0.5 GPM or higher. This value was not stated in their literature and was not known until after the first test was completed. Another test was conducted to evaluate the performance of the OneFlow at 0.6 GPM, with the same flow rate of water sprayed through the nozzle and the excess water discarded to the drain using a second peristaltic pump working in parallel with the peristaltic pump supplying the spray nozzle.

Test protocols before each experiment included:

1. Installing a new heat exchanger and water treatment technology, if applicable.
2. Measuring the air pressure drop across the heat exchanger as a function of the air flow to characterize the airflow resistance of the heat exchanger.
3. Calibrating the pH and conductivity sensors. A three-point calibration was completed for each sensor using standard solutions and following the manufacturer recommendations.
4. Calibrating the peristaltic pump supplying water at a flow rate of 0.024 GPM to the spray nozzle.

In addition, when the duct system was first constructed and then re-sealed after the first test, a leak test was performed to determine air flow loss as a function of air pressure drop across the heat exchanger. For this test, the outlet of the heat exchanger was blocked and fan air flow was varied while the duct pressure was measured.

Test protocols during each experiment, which lasted up to four weeks each, included:

1. Sampling incoming water for hardness measurements (every 2-3 days)
2. Photographing the heat exchanger (every 2-3 days)
3. Measuring water loss (daily)
4. Reviewing data to check on experiment status (daily)

Test protocols at the end of each experiment included:

1. Measuring pressure drop across the heat exchanger as a function of the air flow to characterize the air flow resistance of the clogged heat exchanger.
2. Taking as-found measurements of the pH and conductivity sensors using standard solutions to check for drift of the sensors over the course of the experiment.

2.2.1 Deviation from Test Protocols

A deviation from the protocol inadvertently occurred during the second test of OneFlow (at 0.6 GPM flow rate). In analyzing the data at the end of the experiment, it was noticed that the average nozzle pressure was approximately 25% lower than in previous experiments. The reason for this was found to be that the two peristaltic pumps working in parallel (one for the spray nozzle, one to dump the excess water) caused the pumps to fight each other for the water supplied by the hose (for which the valve was only partially open) as the city line pressure varied. This invalidated the calibration of flow rate through the nozzle, which was used to calculate the cumulative mineral mass sprayed on the heat exchanger. At the end of the experiment, a test was conducted to find the correlation between nozzle pressure (measured continuously throughout the experiment) and nozzle flow rate to determine the actual spray rate that occurred during the experiment. In this test, the revolutions per minute (RPM) of the pump dumping the excess water was varied while holding the RPM of the spray pump constant. The spray flow rate through the nozzle was measured using a graduated cylinder and stop watch at each condition. The correlation of the spray flow rate and corresponding nozzle pressure (Figure 10) was used to determine the actual spray flow rate through the experiment. The result was linear with an excellent correlation and was used to correct the existing data.

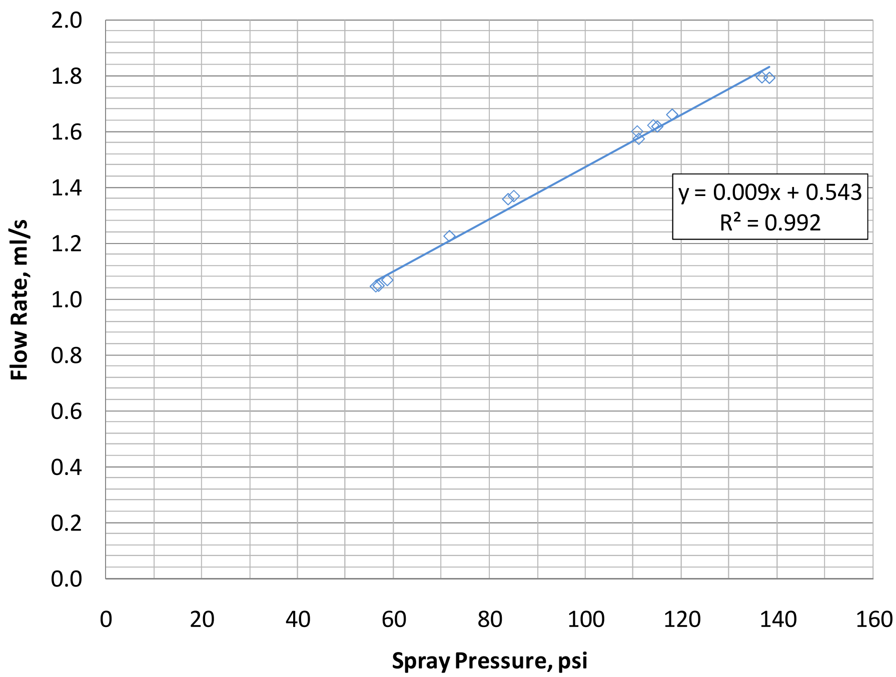

FIGURE 10 - CORROLATION BETWEEN SPRAY PRESSURE AND NOZZLE FLOW RATE FOR ONEFLOW TEST AT 0.6 GPM

Also, for a short time during the second OneFlow experiment (testing cycle 163-195), the water pressure transducer signal was not recorded because of a loose wire. This presented a problem because the pressure data is needed to calculate flow (Figure 10). However, a correlation was made using the measurement of evaporated water, which was determined from the temperature and relative humidity measurements in the system. A linear regression for the sprayed-water to evaporated-water ratio was used to populate the missing data for sprayed water (Figure 11). Note that the fit was only needed for 7% of the duration of the entire experiment.

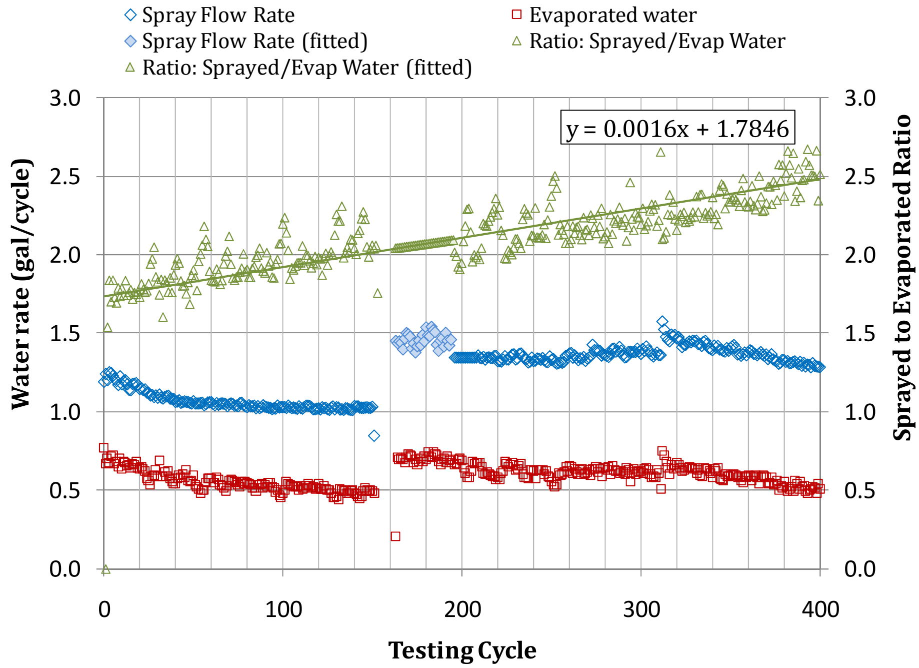

FIGURE 11 - SPRAY FLOW RATE, EVAPORATED WATER FLOW RATE, AND RESULTING RATIO FOR THE SECOND ONEFLOW EXPERIMENT AT 0.6 GPM. FIT WAS USED TO POPULATE MISSING SPRAY FLOW RATE DATA.

2.2.2 Conductivity of Treated Water

Conductivity was measured in a selection of samples before and after the OneFlow device at 0.6 GPM and before and after the GMX device. The same conductivity sensor was used to measure both sets of samples to eliminate any error from bias in the sensor.

2.3 Analysis

For each test, the cumulative mineral mass sprayed on the heat exchanger in grams was calculated when the heat exchanger reached the defined failure point (an air-flow resistance of 0.03 $Pa^{0.6}$/CFM, which is an increase of 275% over the initial air-flow resistance). The air flow resistance through the heat exchanger was determined to be the best measurement for defining and comparing times to failure with the least amount of uncertainty. The collected data were averaged over each cycle (a cycle is 60 minutes water spray on/30 minutes spray off). The flow resistance was calculated by averaging measured air-flow resistance values during the dry cycles and the cumulative mineral mass sprayed on the heat exchanger was calculated by summing the mineral mass sprayed on the heat exchanger during each wet cycle, which was calculated from averaged measured values during that cycle. Detailed equations and the uncertainty analysis are described below.

2.3.1 Total air flow rate

Total air flow rate across the heat exchanger was measured by calculating the total air flow through the fans (based on the differential air pressure across the fan inlets and Equation 1) and subtracting flow that leaks out of the duct before the heat exchanger. As that leakage changes when the pressure in the duct changes, the nature of that leakage flow had to be characterized. To determine duct leakage as a function of the differential air pressure across the heat exchanger, a one-time test was performed, during which the duct outlet was completely blocked and the differential air pressure across the blocked exit was measured and correlated to the air flow rate into the duct as the fan speed was varied. Because the exit was blocked, any flow through the duct was leakage. The air flow was graphed as a function of the differential air pressure across the heat exchanger on a log-log scale and the power regression was determined (Figure 12). This test was repeated after the first experiment when the rig fasteners were improved to reduce the leakage. The resulting equations (Figure 12) were used to determine leakage at the actual differential air pressures measured over the course of the experiments. The first regression with higher leakage was used to calculate results for the first experiment and the subsequent regression was used for the rest of the experiments (OneFlow (0.024 GPM), GMX, and OneFlow (0.6 GPM)).

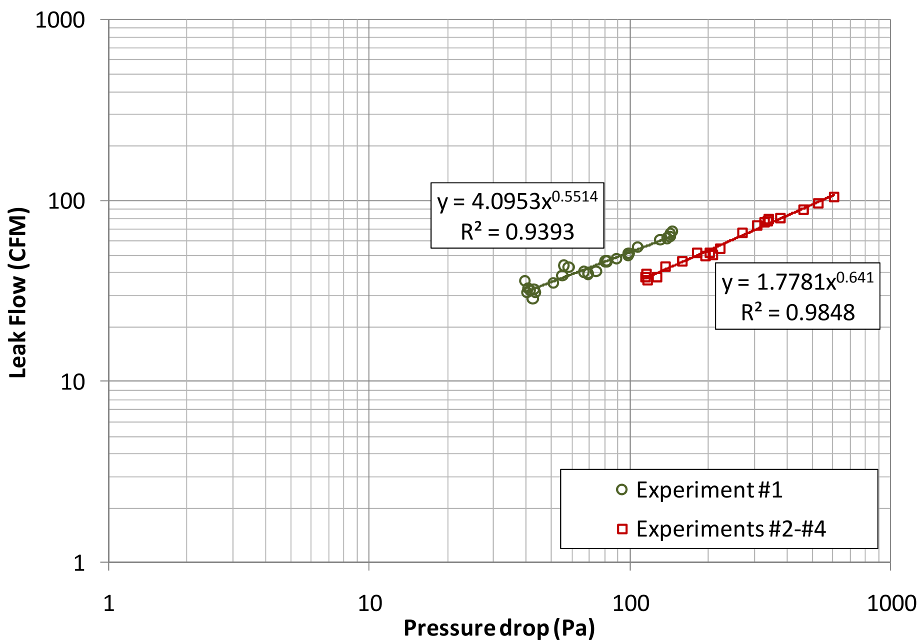

FIGURE 12– THE CORRELATION OF PRESSURE DROP AND LEAK FLOW FOR NO-TREATMENT EXPERIMENT (TEST 1), ONEFLOW (0.024 GPM), GMX, AND ONEFLOW (0.6 GPM) (TEST 2).

2.3.2 Flow resistance

The air flow resistance of the heat exchanger is expressed by a non-linear relationship between air-pressure differential and air flow rate, with the resistance defined as:

$$Resistance = 1/K = \Delta P^n/\dot{V} \qquad (2)$$

where $\dot{V}$ is the flow rate of air through the heat exchanger in CFM, ΔP is the pressure drop across the heat exchanger as a result of that flow in Pa, and n is an exponent determined by fitting a power law curve to the pressure drop versus flow rate data. For all heat exchangers at both the beginning and end of the experiments, the value n was close to 0.6 (actual range 0.56-0.61), which is expected in a non-fully developed laminar flow regime. For the analysis of all results, n was held constant at 0.6. The "failure" point for the heat exchanger was defined as reaching a flow resistance of 0.03 $Pa^{0.6}$/CFM.

2.3.3 Heat exchanger capacity

The heat exchanger capacity is generally expressed as a function of the hot water flow rate through the coil and the temperature difference between the water inlet and outlet of the coil, normalized by the temperature difference between inlet water and inlet air, and the air flow rate. Initial analysis of the coil capacity over time showed very noisy results. The reason for this is that it was difficult to measure the temperature differential between the water inlet and outlet of the coil, which was about 8°F, using the thermocouples installed exterior to the pipe. After reviewing the results and researching measurement methods further, resistive temperature devices that are inside the water flow will be used for future differential measurements of this type. The data for capacity are not presented because of the high level of noise and uncertainty in the data.

2.3.4 Volume of sprayed water hitting the heat exchanger per cycle

The volume of water hitting the exchanger per cycle is determined by subtracting, from the total amount sprayed, the volume of water droplets evaporated and the volume of water drained from the duct:

$$V_{HE} = V_{sprayed} - V_{evap} - V_{drained} \tag{3}$$

where V_{HE} is the resulting volume of water hitting the heat exchanger and all units are in gallons/cycle. The rate of evaporation is a function of the mass flow rate of the air stream and the difference of humidity ratio between supply air (after spray) and intake air (before spray). The volume of evaporated water in gallons/cycle is calculated from:

$$V_{evap} = \dot{m}_{air}\left(W_{air,supply} - W_{air,intake}\right) \times (\Delta t_{cycle})/\rho_{water} \tag{4}$$

where, $\dot{m}_{air}$ is the mass flow rate of intake air in lb/min, $W_{air,supply}$ is the humidity ratio of air stream after spray (supply air), $W_{air,intake}$ is the humidity ratio of air stream before spray (intake air), Δt_{cycle} is the length of the spray cycle (60 minutes), and ρ_{water} is the density of water (8.33lb/gal).

2.3.5 Total hardness of sprayed water (cycle average)

Total hardness is defined as the sum of calcium and magnesium hardness in mg/L as $CaCO_3$. Total hardness of the City of Davis water supply varies significantly as groundwater wells throughout the city are turned on and off in response to the demand for water. In order to monitor the fluctuation, electrical conductivity was measured continually throughout each experiment and correlated to total hardness by taking frequent samples of supply water, which were analyzed for total hardness by the Agriculture and Natural Resources (ANR) laboratory at UC Davis. The measured electrical conductivity of the water and the total hardness were plotted and a correlation was made for all of the data (Figure 13). Because the conductivity sensor was calibrated before each experiment using the same standard solutions, the sensor maintains the same correlation to total hardness throughout all of the experiments. The results show that all sets of data are in good agreement, and all data were combined to provide the most robust regression. The conductivity over the duration of the wet cycle was averaged and the regression was applied to determine the cycle average for the total hardness of the sprayed water.

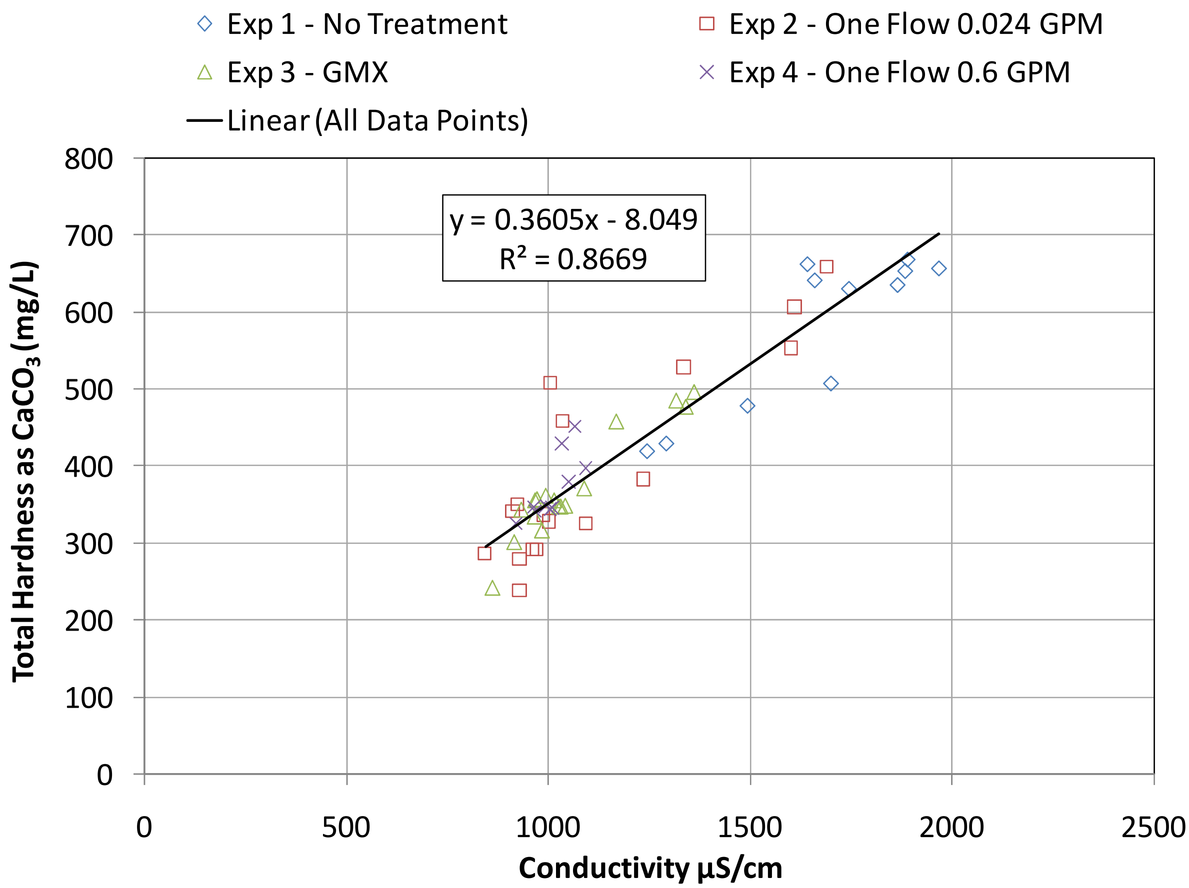

FIGURE 13 - CORRELATION BETWEEN MEASURED CONDUCTIVITY AND TOTAL HARDNESS

2.3.6 Cumulative mineral mass sprayed on the heat exchanger

Cumulative mineral mass sprayed on the heat exchanger is a measure of the volume of water reaching the heat exchanger and the total hardness of this water. As water was sprayed into the duct, it partially evaporated, part of it reached the heat exchanger and the rest was drained. Partial evaporation of the water increased concentration of the hardness minerals in the remaining water. The hardness of the water reaching the heat exchanger was calculated using the mass conservation of mineral concentration and water as follows:

$$V_{sprayed} = V_{drained} + V_{evap} + V_{HE} \tag{5}$$

$$C_{sprayed} \times V_{sprayed} = C_{drained} \times V_{drained} + C_{evap} \times V_{evap} + C_{HE} \times V_{HE} \tag{6}$$

Where,

V = volume per cycle (L/cycle)

C = Total Hardness (mg as $CaCO_3$/L)

It was assumed that the evaporated water does not remove minerals (C_{evap} = 0) and that the minerals from the evaporated water transfer to the remaining water proportionally such that that the hardness of the drained water was assumed to be equal to the hardness of the water droplets reaching the heat exchanger. Applying these assumptions, the hardness of the water sprayed on the heat exchanger per cycle is:

$$C_{HE} = C_{sprayed} \times V_{sprayed}/(V_{sprayed} - V_{evap}) \qquad 7$$

The cumulative mineral mass sprayed on the heat exchanger over the course of the experiment in milligrams is:

$$m_{HE,cumulative} = \sum_{i=1}^{n} C_{HE,i} \times (V_{sprayed,i} - V_{evap,i} - V_{drained,i}) \qquad 8$$

Where i is the number of the cycle and n is the number of cycles for which the cumulative result is calculated. The cumulative mineral mass sprayed on the heat exchanger was calculated for each test at the "failure" point for the heat exchanger, which was defined as a flow resistance of 0.03 $Pa^{0.6}$/CFM.

3 Project Outcomes

3.1 Results and Discussion

Objective: Conduct a laboratory investigation of the performance of two commercially-available non-chemical water treatment systems, as applied to flash-evaporative (i.e. spray) pre-cooling of condenser air (for both refrigeration and A/C), and recirculation water systems (e.g. sump system for direct and indirect evaporative cooling).

Outcome 1: The main result generated is the air flow resistance across the heat exchanger as a function of cumulative mineral mass sprayed on the heat exchanger (Figure 14). Air flow resistance across the heat exchanger increased as the mineral mass sprayed on the heat exchanger accumulated throughout the course of the experiment. The relative performance of each test at the defined failure point of 0.03 $Pa^{0.6}$/CFM is presented in a table and bar chart (Table 3 and Figure 15). At this flow resistance, the heat exchanger is visibly caked with scale (Figure 16).

Compared to the no-treatment condition, the first test of OneFlow at 0.024 gpm decreased the life of the heat exchanger by 7%, the second test of OneFlow at 0.60 gpm increased the life of the heat exchanger by 14%, and the test of GMX increased the life of the heat exchanger by 28% (Table 2).

An uncertainty analysis of the calculated values was completed using error propagation analysis[2] and instrumentation accuracy, which was determined from manufacturer data sheets and observed behavior in the laboratory. The average values for each test from the beginning of the test to the failure point are required in order to complete the analysis (Table 4). An example uncertainty analysis for the no treatment experiment shows that the uncertainty of the mineral mass sprayed on the heat exchanger over one cycle is quite high at 45% (Table 5). The first OneFlow test and GMX test had similar uncertainties (~44% for one cycle). The uncertainty of the second OneFlow test was higher (~70% for one cycle) because of the slightly reduced water spray rate (the analysis is highly sensitive to this value). However, over the course of the experiment, error propagation methods show that uncertainty of the *cumulative* mineral mass sprayed on the heat exchanger is significantly less than the uncertainty for one cycle. The cumulative hardness error bars are calculated (Table 3) and added to the results (Figure 15). The results are that:

1. The OneFlow test result at 0.024 gpm shows a reduced lifetime compared to the baseline, but the result is not statistically significant.
2. The second OneFlow test result at 0.060 gpm shows an improved lifetime 14%greater than the baseline. The result appears to be statistically significant.
3. The GMX test result shows an improved lifetime 28%greater than the baseline. The result appears to be statistically significant.

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[2] Figliola R.S. and Beasley D.E. *Propagation of Error,* Theory and Design for Mechanical Measurements 3rd edition. 2000. Pg. 161.

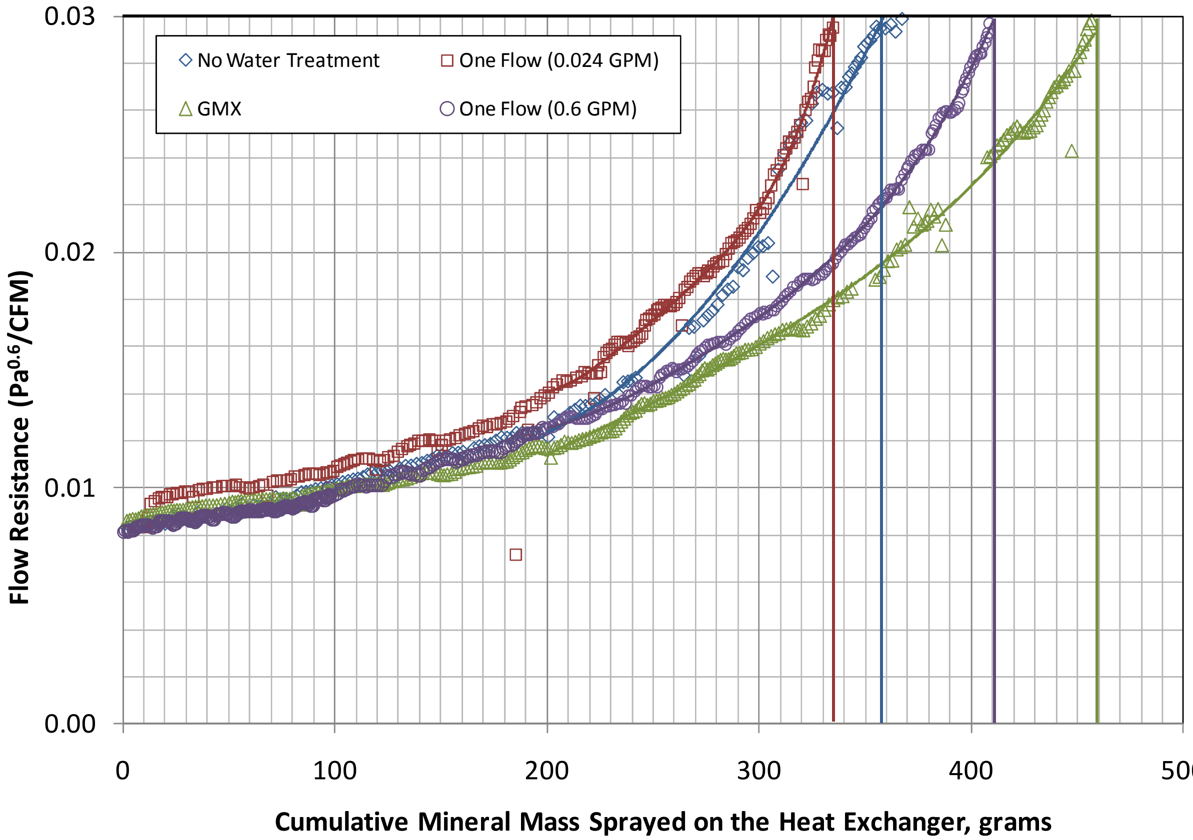

FIGURE 14 – RESULTING AIR FLOW RESISTANCE AS A FUNCTION OF CUMULATIVE MINERAL MASS SPRAYED ON THE HEAT EXCHANGER FOR EACH TEST

TABLE 3 - CUMULATIVE MINERAL MASS SPRAYED ON HEAT EXCHANGER AT FAILURE FOR EACH TEST

	Cumulative mineral mass sprayed on heat exchanger (g)	Uncertainty of Cumulative mineral mass sprayed on heat exchanger (g)	% Difference from Baseline
No Treatment	360	16.1	
OneFlow (0.024 GPM)	335	10.6	-7%
GMX	460	11.8	28%
OneFlow (0.6 GPM)	410	14.9	14%

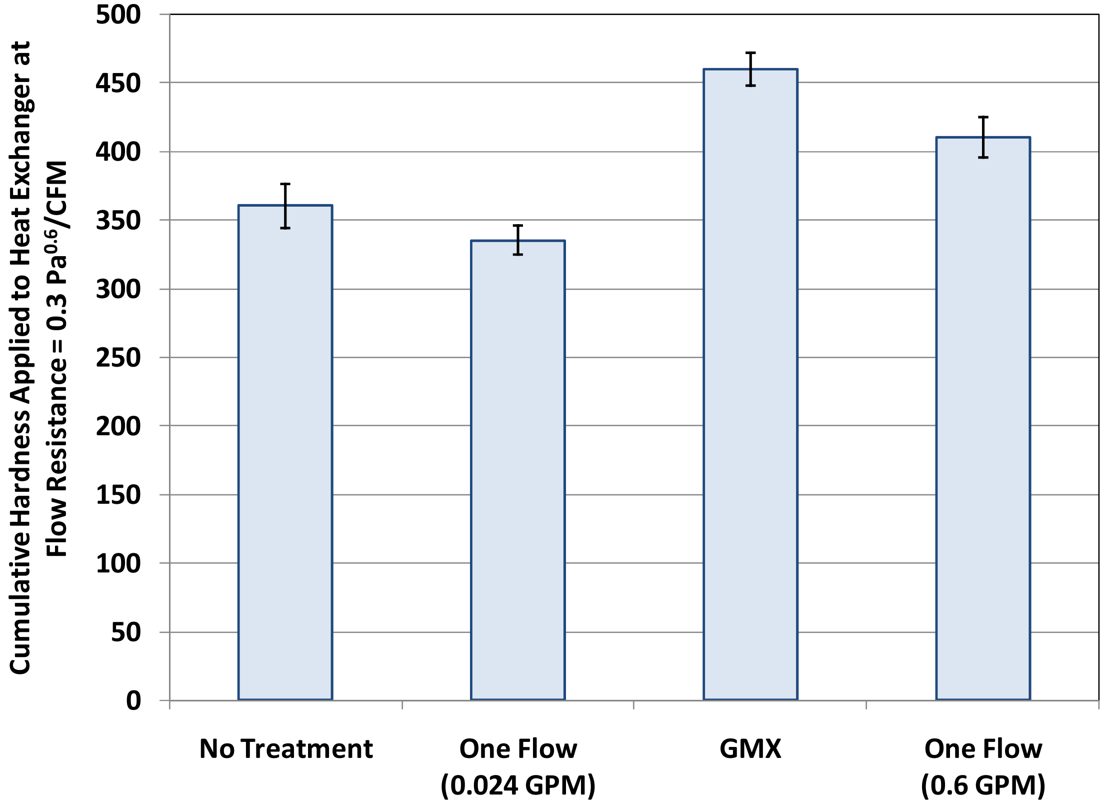

FIGURE 15 - HARDNESS APPLIED TO HEAT EXCHANGER AT A FLOW RESISTANCE OF 0.03 ($PA^{0.6}$/CFM) FOR EACH EXPERIMENT

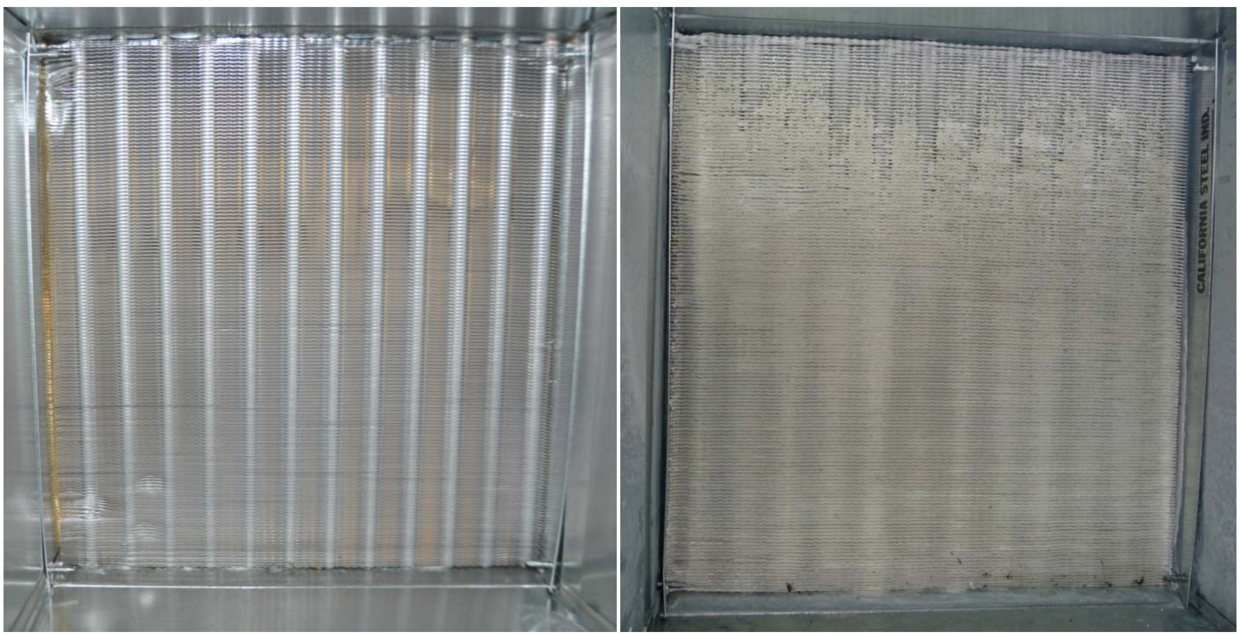

FIGURE 16 - NEW HEAT EXCHANGER (LEFT) COMPARED TO HEAT EXCHANGER COVERED WITH SCALE (RIGHT) AT FAILURE FLOW RESISTANCE OF 0.3 $PA^{0.6}$/CFM

TABLE 4 - AVERAGE VALUES OF INTEREST FOR EACH TEST

	Air flow (CFM)	Incoming Dry Bulb °F	Incoming Wet Bulb °F	Incoming WBD °F	Incoming RH	Exiting Dry Bulb °F	ΔP HE (Pa)	Total HD (mg/L)	Water Spray (Gal/Cycle)	Water Evap (Gal/Cycle)	Water Loss (Gal/Cycle)	Water on HE (Gal/Cycle)
No Treatment	915	78.0	59.1	18.9	0.31	73.2	73	578	1.44	0.53	0.28	0.62
One Flow (0.024 GPM)	898	77.7	57.7	20.0	0.28	72.7	79	437	1.44	0.55	0.28	0.61
GMX	921	76.1	57.8	18.4	0.32	71.2	77	394	1.44	0.57	0.25	0.63
One Flow (0.6 GPM)	927	81.7	62.1	19.7	0.32	76.7	69	352	1.25	0.59	0.27	0.40

TABLE 5 - UNCERTAINTY ANALYSIS FOR EXPERIMENTAL RESULTS USING AVERAGE DATA FROM NO TREATMENT EXPERIMENT

Total Fan Flow = GF1*sqrt(ΔP1) + GF2*sqrt(ΔP2)			
Independent variable	Average value	Sensitivity Index, Θ	Uncertainty, u (±)
x1 = Pressure FanA, P1	40	5.7	0.4
x2 = Pressure FanB, P2	40	5.7	0.4
x3 = Gate Factor Fan A, GF1	72	8.5	1.4
x4 = Gate Factor Fan B, GF2	72	17.0	1.4
Flow (cfm)	915		27.5
Uncertainty (%)			3%

Air Flow Resistance = ΔP_{HE} ^ 0.6 / Air Flow, n=0.6038			
Independent variable	Average value	Sensitivity Index, Θ	Uncertainty, u (±)
x1 = ΔP heat exchanger (Pa)	73	1.2E-04	0.7
x2 = Air Flow (CFM)	915	-1.6E-05	27.5
Air Flow Resistance (Pa^0.6/CFM)	0.014		4.39E-04
Uncertainty (%)			3.1%

Evaporated water = CFM*ΔW*(0.0752 lb/ft^3)*(gal-w/8.33lb-w)*(60min)			
Independent variable	Average value	Sensitivity Index, Θ	Uncertainty, u (±)
x_1 =Air Flow (CFM)	915	5.8E-04	27.5
x_2 =$W_{air\ out}$ (lb-water/lb-air)	7.48E-03	495.6	3.8E-04
x_3 = $W_{air\ in}$ (lb-water/lb-air)	6.40E-03	-495.6	2.3E-04
Evaporated water (gal/cycle)	0.53		0.2
Uncertainty (%)			41%

Water Applied to Heat Exchanger (V_{HE}) = $V_{sprayed}$-$V_{drained}$-V_{evap}			
Independent variable	Average value	Sensitivity Index, Θ	Uncertainty, u (±)
x1 = Sprayed Water (gal/cycle)	1.4	1	0.03
x2 = Drained Water (gal/cycle)	0.3	1	0.01
x3 = Evaporated Water (gal/cycle)	0.5	1	0.2
Water Applied to HE (gal/cycle)	0.6		0.2
Uncertainty (%)			36%

Mineral Mass Sprayed on Heat Exchanger (One Cycle) = ($HD_{sprayed}$×$V_{sprayed}$)/($V_{sprayed}$ - V_{evap})*(V_{HE})*3.785			
Independent variable	Average value	Sensitivity Index, Θ	Uncertainty, u (±)
x1 = HDspray (mg/l)	578	3.8	57.8
x2 = Water Sprayed (gal/cycle)	1.4	-890.2	0.03
x3 = Water Evaporated (gal/cycle)	0.5	-2396.6	0.2
x4 = Water Applied to HE (gal/cycle)	0.6	-3481.3	0.2
HD on HE (g as $CaCO_3$)	2169		965.6
Uncertainty (%)			45%

Uncertainty of Incoming Humidity Ratio lb_W/lb_A							
Variable, xi	x_i^0	W_i^0	W_i^+	W_i^-	δW_i^+	δW_i^-	δW_i
x_1=T_{db} (F)	77.95	6.4E-03	6.5E-03	6.3E-03	1.1E-04	-1.1E-04	1.1E-04
x_2=RH	0.31	6.4E-03	6.6E-03	6.2E-03	2.1E-04	-2.1E-04	2.1E-04
Uncertainty Humidity Ratio lb_W/lb_A							2.3E-04
Uncertainty %							4%

Uncertainty of Incoming Wet Bulb (°F)							
Variable, xi	x_i^0	W_i^0	W_i^+	W_i^-	δW_i^+	δW_i^-	δW_i
x_1=T_{db} (F)	77.95	5.9E+01	5.9E+01	5.9E+01	3.6E-01	-3.6E-01	3.6E-01
x_2=RH	0.31	5.9E+01	5.9E+01	5.9E+01	3.3E-01	-3.4E-01	3.4E-01
Uncertainty Wet Bulb °F							4.9E-01
Uncertainty %							1%

Uncertainty of Exiting Humidity Ratio lb_W/lb_A							
Variable, xi	x_i^0	W_i^0	W_i^+	W_i^-	δW_i^+	δW_i^-	δW_i
x_1=T_{db} (F)	73.2	7.5E-03	7.3E-03	7.7E-03	-2.3E-04	2.3E-04	2.3E-04
x_2=T_{wb}	59.1	7.5E-03	7.8E-03	7.2E-03	3.0E-04	-3.0E-04	3.0E-04
Uncertainty Humidity Ratio lb_W/lb_A							3.8E-04
Uncertainty %							5%

Outcome 2: Conductivity of treated water

The electrical conductivity of the water entering the treatment system was compared to the water exiting the treatment system. The results for 5 samples collected for both the OneFlow test at 0.60 gpm and the GMX test are presented in Figure 17 for OneFlow and in Figure 18 for GMX. In both cases, no significant change in conductivity from pre-treatment to post-treatment was observed. In addition to the pre- and post-treatment conductivity measurements, a separate experiment that re-circulated the water continuously through the treatment device was conducted. No significant change in conductivity was observed for either the OneFlow or the GMX.

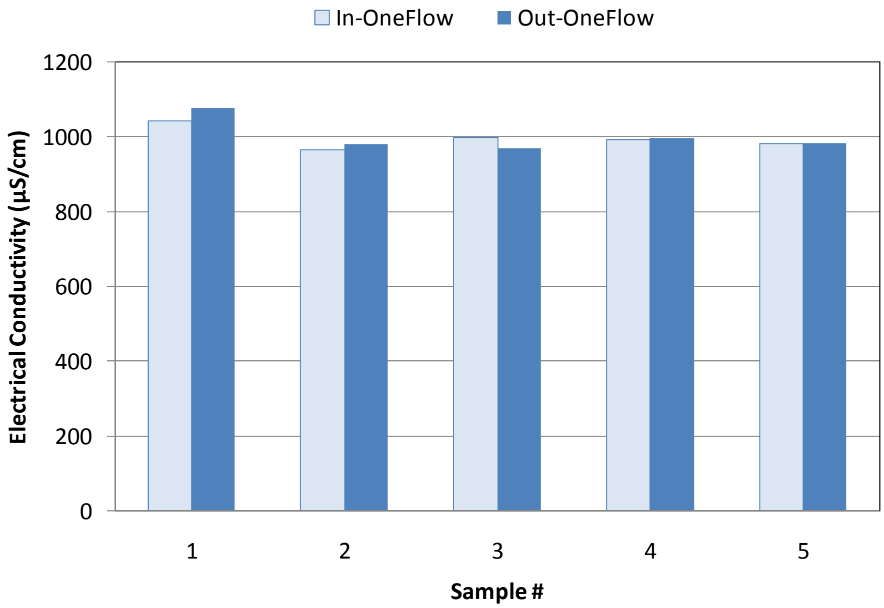

FIGURE 17 - COMPARISON OF ELECTRICAL CONDUCTIVITY OF WATER FOR PRE AND POST ONEFLOW TREATMENT AT 0.60 GPM IN 5 SAMPLES

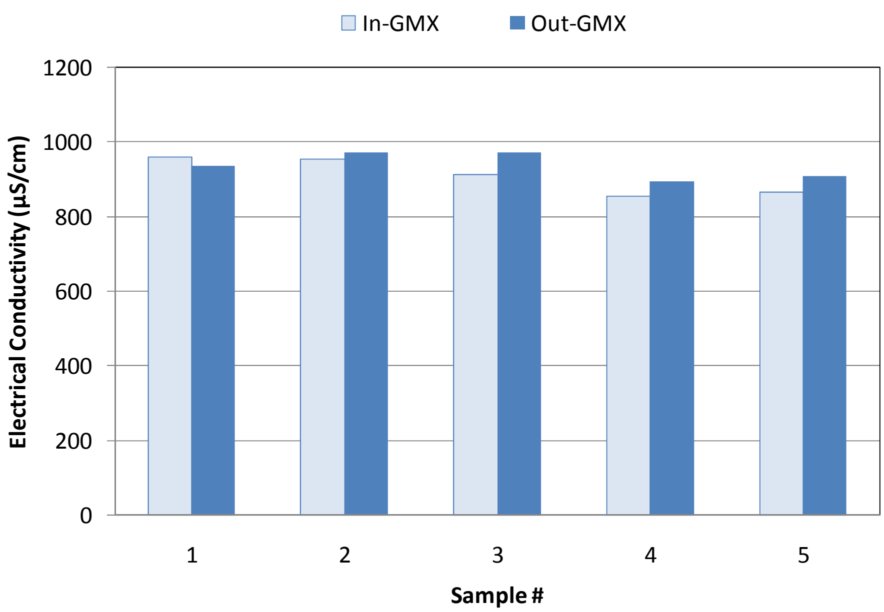

FIGURE 18 – COMPARISON OF ELECTRICAL CONDUCTIVITY OF WATER FOR PRE AND POST GMX TREATMENT IN 5 SAMPLES

4 Conclusions

The main conclusions from this project are:

1. The OneFlow test result at 0.024 gpm shows a reduced a reduced lifetime compared to the baseline, but the result is not statistically significant.
2. The second OneFlow test result at 0.060 gpm shows an improved lifetime 14%greater than the baseline. The result appears to be statistically significant.
3. The GMX test result shows an improved lifetime 28%greater than the baseline. The result appears to be statistically significant.
4. No change in conductivity of the water was observed for either treatment method in either the once-through or recirculation tests.

However, in analyzing the data, it became clear that many assumptions and corrections were needed in order to derive the result. While these assumptions and corrections were consistent in all the analyses, the results should be reviewed within this context. The following items should be considered:

1. Incoming water quality was not controlled and was highly variable. The results were normalized by total hardness, but this may not be the only factor affecting scale formation. A review of the City of Davis water quality report[3] shows the presence of metallic cations such as iron, copper, and zinc, which could affect in scale formation. The variance of these metallic ions in the water supply over the course of the tests is unknown.
2. Each experiment used a new spray nozzle that was positioned manually using a marking on the wall of the duct. Confidence in this assumption was gained by reviewing the pressure at the nozzle and the evaporated water metric, which was consistent between experiments.
3. Each experiment used a new heat exchanger, all of which were purchased from the same manufacturer.
4. Measurements from ten instruments were required to obtain the final results. Confidence in the results was gained by performing an uncertainty analysis and assuring that the same instruments were used throughout all experiments.
5. It was assumed that evaporated water did not remove any minerals from the system and that the minerals concentrated equally in the remaining water that was partially sprayed on the heat exchanger and partially drained from the duct. A one-time experiment in which the conductivity of the drained water was measured showed this assumption to be reasonable.
6. An additional correction was needed to obtain spray flow rate for the second OneFlow test. Although post-test experiments showed that this correction is valid, this is an inconsistency in the test procedure that adds to the uncertainty of the second OneFlow test result.

It is notable that a physical result (longer heat exchanger life) was obtained for the GMX magnetic treatment while no change in conductivity was observed, suggesting that the

[3] City of Davis Water Quality Report 2009
http://cityofdavis.org/pw/water/pdfs/2009_waterqualityreport.pdf

treatment is not resulting in the precipitation of minerals within the bulk water supply at the time the measurement is made. One possible explanation is that the magnetic treatment may affect the precipitation behavior of the ions once water begins to evaporate and the solution is driven toward saturation. Therefore, measuring the properties of the water as it passes through the treatment device does not necessarily indicate what will happen during the critical time when evaporation takes place. One hypothesis is that the magnetic treatment changes the behavior of the ions so that, when precipitation takes place, the ions are more likely to conglomerate onto particles in solution and less likely to precipitate on to heat exchange surfaces and cause scale.

5 Recommendations

Further tests evaluating this technology should consider the following recommendations to reduce uncertainty in the results:

1. Incoming water quality should be consistent between tests. This could be done either by running tests in parallel (baseline versus treatment under test) with the same incoming city water or by mixing hard water in the laboratory.
2. The spray mechanism is quite complicated. A rig that re-circulates water over a hot coil may be more practical. This assures that all the water hits the coil and that the complication of lost water is removed.
3. For minerals to reach the heat exchanger and not contribute to flow resistance, the assumption is that the minerals are not "sticking". Mineral dust was clearly seen in the laboratory but was not caught or measured. There may be a way to filter this dust from the exiting air stream and collect it for measurement.

While the results of these experiments have provided evidence that physical water treatment systems have applications to evaporative cooling, further research is needed to understand the mechanism by which scale buildup is reduced. Subsequent research, which will focus on understanding the mechanisms by which physical water treatment systems reduce scale, will allow improved experimental designs to quantify the performance and application of various technologies to reduce water consumption and/or improve the performance of evaporative coolers.