



**10 Abstract**

11 Cooling coil surfaces are ideal sites for biofilm formation due to the presence of adequate  
12 nutrients (deposited particles) and moisture (condensate), causing adverse impacts on heating,  
13 ventilation and air-conditioning (HVAC) energy usage and performance. In this study, an HVAC  
14 test apparatus was built in our laboratory to investigate the hypothesis that ultraviolet germicidal  
15 coil cleaning (UVG-CC) of heat exchanger surfaces improves heat transfer effectiveness and  
16 reduces the static pressure drop across the coil. The test apparatus consisted of two parallel ducts,  
17 each with its own cooling coil. One coil was treated with UVG-CC while the other was the  
18 control and left untreated. Thermodynamic properties of the air and water flowing through both  
19 heat exchangers were monitored over the course of two years with sensors and a data acquisition  
20 system. Differences in static pressure drop and coil effectiveness between the UV-treated and  
21 control coil were compared across multiple modes of coil operation (defined by presence of  
22 condensate). The effectiveness of UVG-CC was drastically affected by the presence of  
23 condensation on coil fins. We observed a statistically significant difference in the heat transfer  
24 effectiveness between the UV-treated and control coils in wetted conditions while no difference  
25 was observed in dry conditions. Sensor accuracy, however, contributed to large uncertainty in  
26 our result. The average heat transfer of the UV-treated coil was 3.0 to 6.4% higher compared to  
27 the control coil, with an uncertainty of +/- 2.7%. UVG-CC, however, did not significantly reduce  
28 static pressure drop.

29 **Keywords:** Heat exchanger, Efficiency, UVGI, UVC, Coil fouling

30

31 **1.0 Introduction**

32           Ultraviolet germicidal irradiation (UVGI) has a long history of being used for the  
33 disinfection of both water and air streams, primarily in environments with higher risk of airborne  
34 pathogen transmission such as water treatment plants, healthcare facilities, schools, and prisons  
35 [1]. UVGI systems use low-pressure mercury vapor lamps that emit shortwave ultraviolet-C,  
36 peaking at 253.7 nm. Using ultraviolet germicidal coil cleaning (UVG-CC) technology in  
37 heating, ventilation, and air-conditioning (HVAC) systems has recently gained popularity [2].  
38 While air disinfection may still occur as air passes by the UVG-CC system, the primary focus of  
39 UVG-CC is surface disinfection and, in turn, maintenance cost savings, and increased or  
40 prolonged system capacity due to cleaner heat exchanger surfaces, resulting in an overall system  
41 energy savings due to better heat transfer and reduced load on the chiller, pump, and/or fan. Life  
42 cycle cost simulations of UVGI in HVAC systems for air disinfection (requiring higher levels of  
43 irradiance than UVG-CC) found the annual energy cost of a UVGI system to be relatively small  
44 compared to a typical whole-building energy cost and, for comparison, found UVGI to be  
45 significantly more cost effective than the equivalent high efficiency filtration for removing  
46 microbial air contaminants [3]. The buildings sector accounted for 41% of primary energy  
47 consumption in the US in 2010 [4]. More than half of the energy used in buildings is for heating,  
48 ventilating and/or air-conditioning the indoor environment [5], so energy savings for HVAC  
49 systems could have large implications for total building energy consumption.

50           In many climates where the outdoor air must be dehumidified prior to entering the  
51 building space, air is cooled below the dew point to condense moisture out of the air. This  
52 moisture can linger within the densely packed fins of a cooling coil and eventually form biofilms

53 from deposited environmental bacteria and fungi present in the air. Heat exchanger surfaces are  
54 an ideal site for biofilms due to the presence of adequate nutrients (i.e., debris inherent on coil  
55 surfaces) and moisture [6]. High bacterial and fungal concentrations have been documented  
56 within HVAC systems, specifically on cooling coils and drain pans [7–9].

57 Biological fouling of heat exchangers can affect HVAC system energy efficiency and  
58 usage in a variety of ways. Two direct effects are a loss in heat transfer effectiveness due to  
59 lower thermal conductivity of heat exchange surfaces and an increase in pressure drop across the  
60 heat exchanger due to increased fin thickness, both caused by biofilm covering the fins.  
61 Increased energy usage occurs when subsequent actions are taken to maintain the same system  
62 performance with fouled equipment. One such action may be to lower the temperature of the  
63 cooling fluid to maintain the desired supply air temperature, causing the chiller to work harder to  
64 provide additional cooling, and thus use more energy. Alternatively if the cooling fluid  
65 temperature is not lowered, higher flow rates needed to meet the load would result in an increase  
66 in pump energy usage. Additionally, an increased pressure drop may lead to increased fan energy  
67 usage by a variable speed fan to maintain the desired air flow rate or meet the cooling load.  
68 Although a constant speed fan would result in reduced fan power due to increased flow  
69 resistance, the system would no longer be achieving design airflow.

70 While health benefits of UVG-CC have been shown in the literature [9,10], little  
71 evidence exists of the potential energy efficiency benefits of this technology. Anecdotal evidence  
72 describes “visibly cleaner” cooling coils and energy savings after the installation of a UVG-CC  
73 system [11]. An increase in energy efficiency of 10-15% from coil cleaning has also been  
74 reported, but not specifically using UVG-CC [12]. A recent paper reported on a field study  
75 applying UVG-CC to an air handling unit in a building in Singapore. Results show that the coil

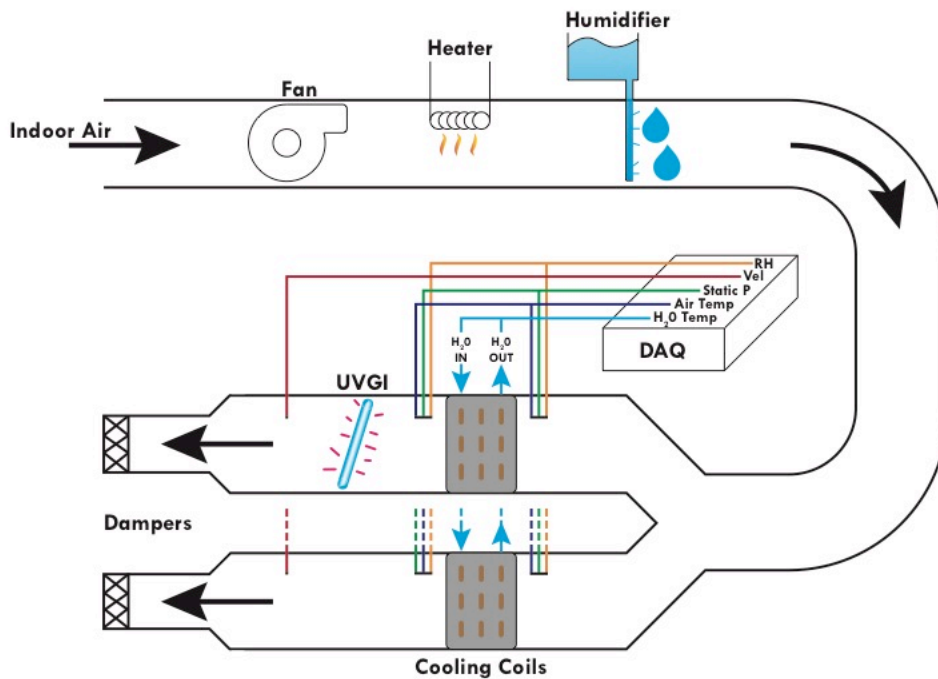
76 overall thermal conductance increased by 10%, the pressure drop decreased by 13% and the fan  
77 energy used fell by 9% [13].

78 The objective of this study was to investigate the hypothesis that UVG-CC increases heat  
79 transfer effectiveness and decreases static pressure drop across the coil. We built a lab-based test  
80 apparatus consisting of two identical heat exchangers, one being irradiated with UV and the  
81 other not; detailed thermodynamic measurements were collected. This study was conducted over  
82 the course of two years and was able to discern how slight changes in inlet air properties due to  
83 outdoor air variations affected heat exchanger performance. The results presented in this study  
84 allow us to provide recommendations for effective installation and operation of UVG-CC  
85 technology for reducing coil fouling and increasing or prolonging HVAC heat exchanger  
86 effectiveness.

## 87 **2.0 Materials and methods**

88 *2.1 Test Facility.* A custom HVAC test apparatus was built in the Air Quality Laboratory  
89 at the University of Colorado Boulder, consisting of two parallel ducts, each with its own cooling  
90 coil, but supplied by the same temperature and relative humidity controlled airstream (Figure 1).  
91 Similar custom testing ducts have been used to investigate heat exchanger performance [14-16].  
92 The coils were steam cleaned prior to starting the tests. The test apparatus was equipped with  
93 sensors to measure duct velocities using pitot tubes (BAPI ZPS-ACC12,  $\pm 1\%$  on 0-1 psi range,  
94 [www.bapihvac.com](http://www.bapihvac.com)) connected to differential pressure sensors (OMEGA PX2650,  $\pm 1\%$  best fit  
95 straight line (BFSL), [www.omega.com](http://www.omega.com)), static pressure drops (OMEGA PX2650), entering and  
96 exiting water temperatures (OMEGA TH-44000-NPT,  $\pm 0.1^\circ\text{C}$ , [www.omega.com](http://www.omega.com)), and entering  
97 and exiting air temperatures (OMEGA ON-405,  $\pm 0.1^\circ\text{C}$ , [www.omega.com](http://www.omega.com)) and relative

98 humidity (OMEGA HX71,  $\pm 4\%$ , www.omega.com) for each branch. Voltage output from the  
99 sensors was fed into a data acquisition system (NI cDAQ-9171 with NI 9205 module,  
100 www.ni.com) and processed with LabView to export data for analysis in MATLAB. Both coils  
101 were TRANE light commercial tube and fin coils (Type P2) with aluminum fins (Prima-flo H)  
102 and copper tubes, one-ft<sup>2</sup> face area, 12 fins/inch, and were four rows deep. One UVC lamp  
103 (ALTRU-V V-Ray Model 23-1100, 25W) was installed ten inches away from the coil on the  
104 downstream side. The lamp was burned in for 100 hours prior to use. The lamp was shielded  
105 with mesh to achieve the desired level of surface irradiance.



106

107 **Figure 1.** Schematic of HVAC test apparatus.

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109 **Table 1.** Sampling subsets used for statistical analysis

<i>Sampling Subsets</i>	<i>Start Date</i>	<i>End Date</i>	<i>Mean Entering Air Temperature (Standard Deviation (SD)) and Rang, Mean Relative Humidity (SD) and Range, Mean Dew Point<sup>1</sup></i>	<i>Description</i>
Baseline Region	6/6/2014	7/6/2014	78.8 F (1.42 F) 73.7-83.8 F  46% (5.1%) 36.4-56.3%  56 F dew point	After allowing both coils to foul for four months, a one month period prior to turning the UV lamp on is the baseline period
Humid Region 1	8/27/14	9/27/14	74 F (1.1 F) 68-76.5 F  55% (3.6%) 41.3-56.5%  57 F dew point	A one month period after UV was turned on for the treatment coil while condensation was occurring
Dry Region	12/15/14	1/15/15	73 F (1.1 F) 66.7-76.8F  34% (5%) 24.5-43.7%  43 F dew point	One month during the winter while no condensation was occurring due to dry ambient air conditions
Humid Region 2	7/19/15	8/19/15	80 F (0.56 F) 78.6-82.9 F  45% (4.5%) 33.7-51.9%  57 F dew point	The following summer after returning to condensing conditions

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<sup>1</sup> Standard deviation and range are only presented for temperature and relative humidity as those parameters were continuously monitored. The average dew point for the period is presented.

111 The test apparatus used indoor air from the room as the inlet air. The room HVAC system  
112 supplied 100% outdoor air filtered with MERV 14 filters. Air entered each cooling coil, on  
113 average, at 24 °C (75°F) and 44% relative humidity and chilled water entered at 10 °C (50°F),  
114 satisfying conditions for condensation onto the coils. The system mimicked a constant volume  
115 HVAC system, meaning the volumetric flow rate is held constant at 350 CFM. The flow rates  
116 through each coil were held equal to one another using dampers since the static pressure drop  
117 across the coils may not be equal given equivalent flow rates. The dampers were positioned fully  
118 open at the start of the experiment and were only closed by a few degrees to equilibrate flow  
119 rates throughout the experiment. Pitot tubes were placed four feet downstream of the coils in a  
120 six foot section of straight ductwork to allow for uniform flow at the location of measurement.  
121 Air and water inlet temperatures, inlet relative humidity, and water flow rate were held as  
122 constant as possible (see Table 1 for mean values). Fluctuations in outdoor air conditions slightly  
123 affected conditions within the apparatus. During summer months, both coils had water actively  
124 condensing onto fin surfaces at nearly all times and drain pans were wet. In the winter months  
125 when outdoor air became very dry (~10% RH), the apparatus was unable to humidify the air  
126 sufficiently to continue condensing water onto the cooling coils. These test periods of desiccation  
127 revealed interesting results, described in the Results section.

128 The system ran undisturbed for four months without UVG-CC on either coil to ensure  
129 that both coils fouled at an equivalent rate and to establish a robust baseline dataset. After four  
130 months of operation, the UV lamp was turned on, irradiating the downstream side of one of the  
131 cooling coils (called the *treatment* coil). The *control* coil was never irradiated. The irradiance at  
132 the surface of the treatment coil was on average 200  $\mu\text{W}/\text{cm}^2$ , being roughly 280  $\mu\text{W}/\text{cm}^2$  at the  
133 center but 180  $\mu\text{W}/\text{cm}^2$  at the corners, just above levels referenced as “typical” in the ASHRAE



134 HVAC Applications Handbook [17] at 50 to 100  $\mu\text{W}/\text{cm}^2$ . A factory calibrated radiometer  
135 (model 1400 International Light Inc., Newburyport, MA) was used to measure UV irradiance.  
136 Measurements were made in a grid vertically across the ducting at 0.3 meters way from the UV  
137 lamp, on the surface of the coil.

138 *2.2 Coil Effectiveness.* One of the main challenges in assessing changes in flow  
139 characteristics and effectiveness in two heat exchangers over time is that all variables affecting  
140 these qualities are never exactly the same and cannot be held completely constant. For this  
141 reason, small fluctuations in temperature, relative humidity, or flow rate affected static pressure  
142 drop and the calculated value of heat transfer, making it difficult to compare. To remedy this,  
143 comparisons between the control and UV-treated coils were only made with dimensionless  
144 quantities, including heat exchanger effectiveness and the coefficient of an assumed quadratic  
145 relationship between static pressure drop and velocity.

146 Heat exchanger effectiveness compares the actual airside heat transfer to the maximum  
147 heat transfer theoretically possible. The equation for calculating heat transfer effectiveness is  
148 different for a wet coil versus a dry coil. Equation 1 was used to calculate the effectiveness of the  
149 heat exchangers while condensate was present in the drain pans. Effectiveness is defined as the  
150 ratio between the actual airside heat transfer,  $q$ , to the maximum possible heat transfer given the  
151 temperature of the cooling fluid entering the heat exchanger,  $q_{max}$ .

$$152 \quad \varepsilon = \frac{q}{q_{max}} = \frac{\dot{m}_a(h_{a,i} - h_{a,o})}{\dot{m}_a(h_{a,i} - h_{sat@T_{w,i}})} \quad (1)$$

153 where  $\dot{m}_a$  is the mass flow rate of air through the heat exchanger ( $\text{kg}/\text{m}^3$ ),  $h_{a,i}$  and  $h_{a,o}$   
154 are the enthalpies of the air entering the heat exchanger at the inlet and exiting at the outlet

155 (J/kg), and  $h_{sat@T_{w,i}}$  is the saturation enthalpy (J/kg) at the temperature that the water enters the  
156 heat exchanger. Alternatively, Equation 2 was used to calculate the heat transfer effectiveness  
157 when the cooling coil surfaces were dry.

$$158 \quad \varepsilon = \frac{(\dot{m}c_p)_{air}(T_{a,i} - T_{a,o})}{(\dot{m}c_p)_{min}(T_{a,i} - T_{w,i})} \quad (2)$$

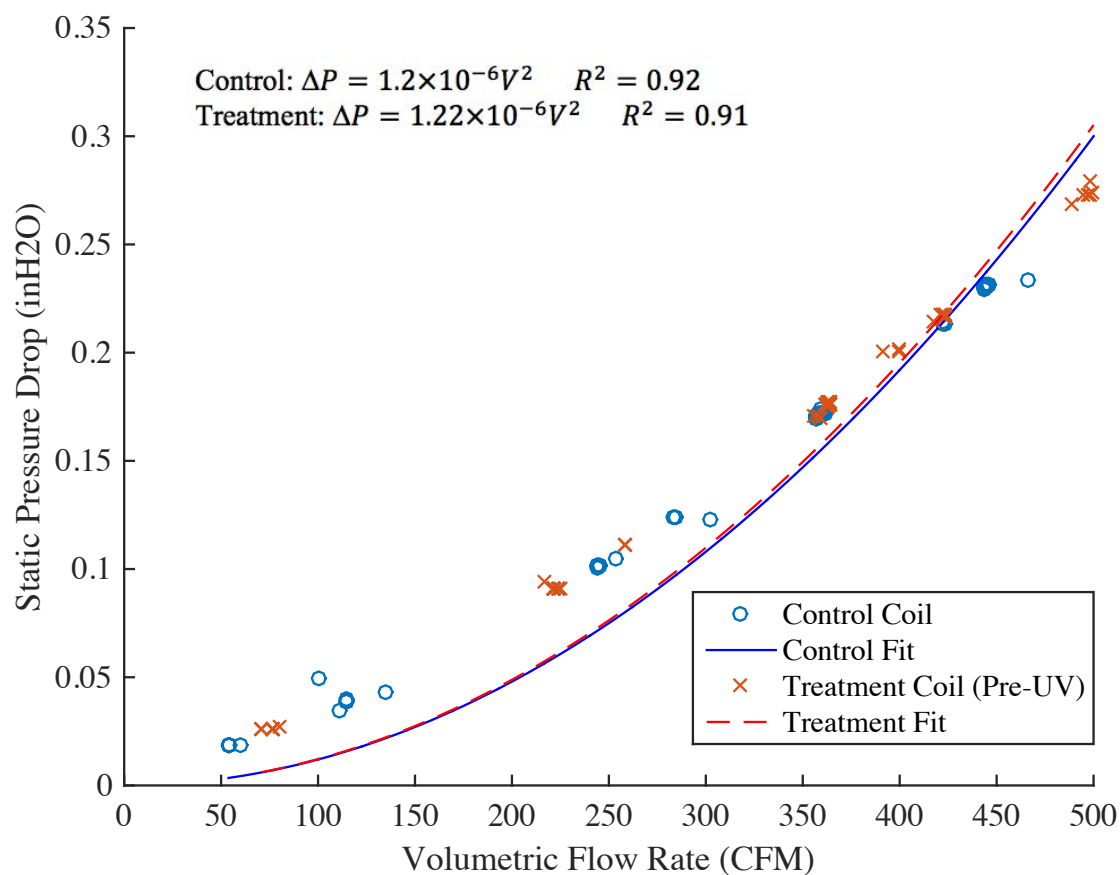
159 where  $(\dot{m}c_p)_{air}$  is the mass flow rate (kg/s) multiplied by the specific heat of air through  
160 the heat exchanger (J/kg-°C),  $(\dot{m}c_p)_{min}$  is the minimum mass flow rate multiplied by the  
161 specific heat between both fluids (air or water),  $T_{a,i}$  is the temperature of the air entering the heat  
162 exchanger,  $T_{a,o}$  is the temperature of the air leaving the heat exchanger, and  $T_{w,i}$  is the  
163 temperature of the water entering the heat exchanger (°C). Heat exchanger effectiveness was  
164 monitored throughout the entire experiment for both coils.

165 *2.3 Coil System Curves.* A typical system resistance curve is a representation of how an  
166 air handling system responds to a given airflow, and is the sum of all pressure losses through the  
167 duct, elbows, filters, dampers, coils and any other device that resists flow [18]. To rule out  
168 changes in pressure drop from other system components, focus was placed solely on the system  
169 curve for the cooling coil. The following equation shows the relationship between static pressure  
170 drop and volumetric flow rate for the system curve of each coil:

$$171 \quad \Delta P_s = aV^2 \quad (3)$$

172 where  $\Delta P_s$  is the static pressure drop across the coil,  $V$  is the volumetric flow rate of air through  
173 the coil, and  $a$  is a coefficient that describes the steepness of the curve. Equation 3 was fit to the  
174 data and is shown in Figure 2. Due to the quadratic relationship between flow and pressure drop,

175 this coefficient,  $a$ , can be calculated for each coil for every sampling point. As the coil becomes  
176 more or less fouled, it is expected that its system curve will change, becoming steeper with  
177 increased fouling and shallower with decreased fouling. Figure 2 shows how both coils had very  
178 similar system curves during the baseline period (pre-UV treatment) and also demonstrates the  
179 quadratic relationship between pressure drop and flow rate.



180

181 **Figure 2.** Static pressure drop versus volumetric flow rate for both cooling coils during the  
182 baseline period. The two coils displayed very similar system curves at the start of the  
183 experiment. Pressure drop and velocity uncertainty was  $\pm 1\%$ .

184 Pressure and flow have a strictly quadratic relationship in the case of fully developed  
185 turbulent flow. If turbulent flow is not fully developed (at lower Reynolds numbers) the  
186 exponent in Equation 3 may fall between 1 and 2. This results from flow that falls between the  
187 linear regime for the laminar flow and the quadratic regime for the fully developed turbulent  
188 flow. For this analysis, a quadratic relationship was assumed since flow was held constant and  
189 changes in Reynolds numbers between the two coils relative to one another were assumed to be  
190 negligible. Although this assumption is a limitation of our study and results in slightly lower  
191 values in  $R^2$  from the quadratic fit, by focusing on differences between the two coils with respect  
192 to a baseline period instead of absolute values, this quadratic model still serves as a useful  
193 measure of the effect of UV on the pressure drop.

194 *2.4 Modes of Operation.* We explored the effectiveness of UV cleaning over a range of  
195 supply air conditions (temperature and RH) to reflect the operation of an air handling unit during  
196 different seasons. For our analysis, we split our time series data on a month-to-month basis and  
197 classified each month as either a *humid* month, where water was condensing out of the air for the  
198 majority of that month, or a *dry* month, where water was not condensing out of the air for the  
199 majority of the month. Whether the system was operating in the humid or dry regime was  
200 determined by visually observing the drain pans: if they were dry, then the system was running  
201 dry and if they were wet, then the system was running humid. To simplify our analysis, we  
202 excluded months during the transition period from humid to dry months or vice versa.

203 *2.5 Statistical Analyses.* All data were smoothed using a moving average approach with a  
204 boxcar window that averaged each sampling point with the 5000 points (approximately 28 hours)  
205 to its right and left to mute short-term fluctuations due to the building's HVAC system operation

206 and focus on long-term UVG-CC effects. An example of raw versus smoothed data can be found  
207 in Figure A4.

208 While both treatment and control cooling coils were manufactured to the same  
209 specifications, they were not identical in the measured properties of heat transfer effectiveness  
210 and pressure drop. To account for this, we analyzed the difference between measured properties  
211 of each coil across discrete one month periods that fell into *humid* or *dry* month categories. One  
212 year of sampling was split into four one-month subsets described in Table 1.

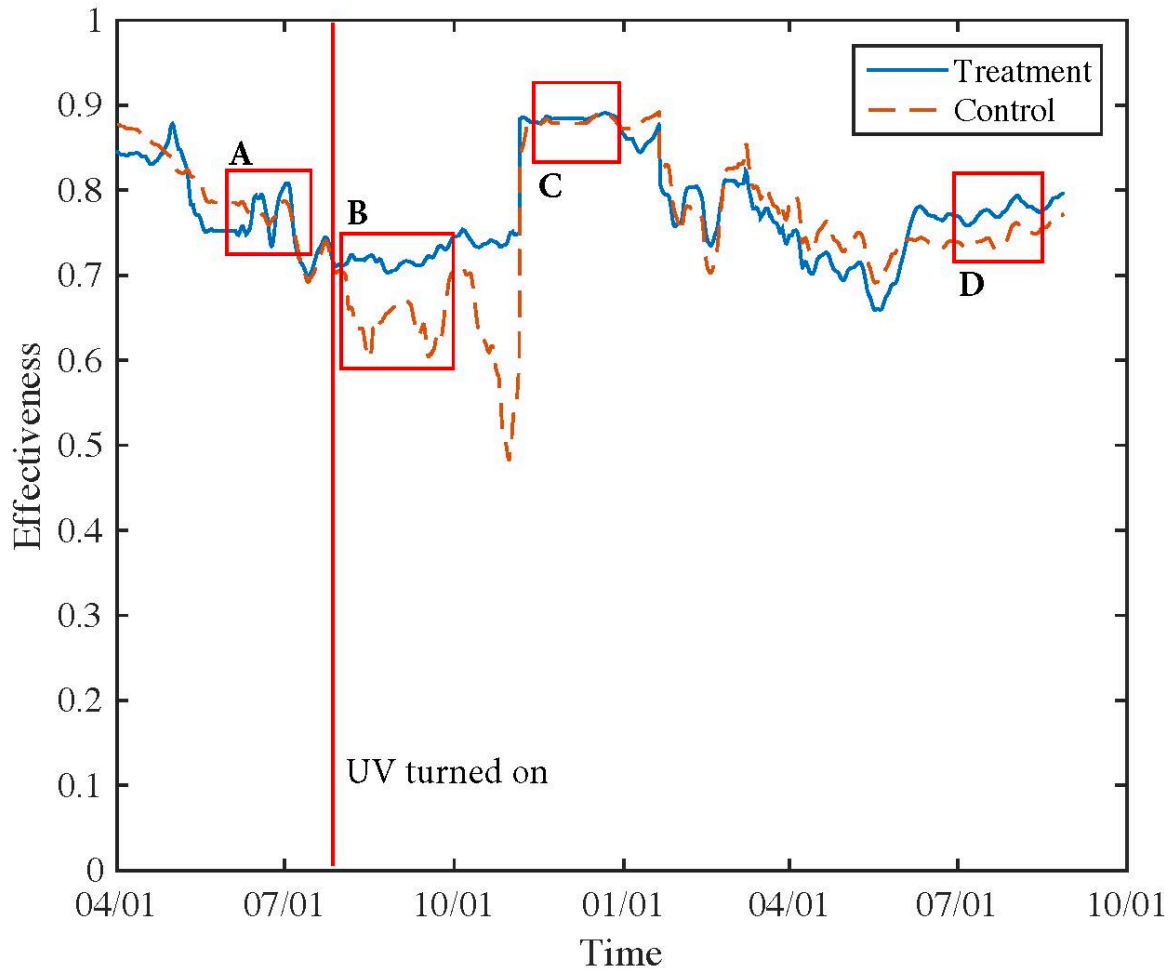
213 We used a bootstrapping approach in our statistical analyses due to very large sample  
214 sizes (roughly 134,000 points per sampling subset) [19–21]. A one-way analysis of variance  
215 (ANOVA) statistical test was used to compare the difference in heat transfer effectiveness and  
216 system curve coefficient between treatment and control coils across sampling subsets. Student t-  
217 tests were used to test if sampling subset means were significantly different than zero. Rather  
218 than perform an ANOVA across all sampling subsets, we randomly sampled 1000 points without  
219 replacement from each subset and performed an ANOVA on those samples. This subsampling  
220 process was repeated 1000 times, generating a distribution of 1000 p-values from all ANOVAs.  
221 When the mean ANOVA p-value rejected the null hypothesis that all sampling subset means  
222 were equal, we conducted multiple comparisons across sampling subsets using Tukey’s Honestly  
223 Significant Difference (HSD) procedure. Similarly, t-tests were bootstrapped by randomly  
224 sampling 1000 points from each sampling subsets and performing a t-test 1000 times to produce  
225 a distribution of t-test p values. All analyses were performed in MATLAB.

226 *2.6 Sensor Uncertainty Propagation.* Each measured parameter had an associated  
227 accuracy (obtained from the sensor manufacturer, see 2.1) that was propagated through all of our

228 analyses to determine the uncertainty of calculated values of heat transfer effectiveness and coil  
229 system curve coefficients due to sensor accuracy. The standard variance formula was used,  
230 which assumes variables are not correlated and are independent [22].

### 231 **3.0 Results**

232 *3.1 Coil Effectiveness.* Under the hypothesis that the UV-treated coil would have a higher  
233 effectiveness than the control coil, the difference between the coils was calculated as treatment  
234 coil effectiveness minus control coil effectiveness at every sampling point during the sampling  
235 subset ( $\Delta_{\varepsilon_i} = \varepsilon_{\text{Treatment}_i} - \varepsilon_{\text{Control}_i}$ ). The larger the positive difference, the higher the UV-treated  
236 coil effectiveness was compared to the control coil effectiveness. Figure 3 shows the entire time  
237 series of calculated effectiveness for both coils and Figure 4 shows each sampling subset  
238 described in Table 1.

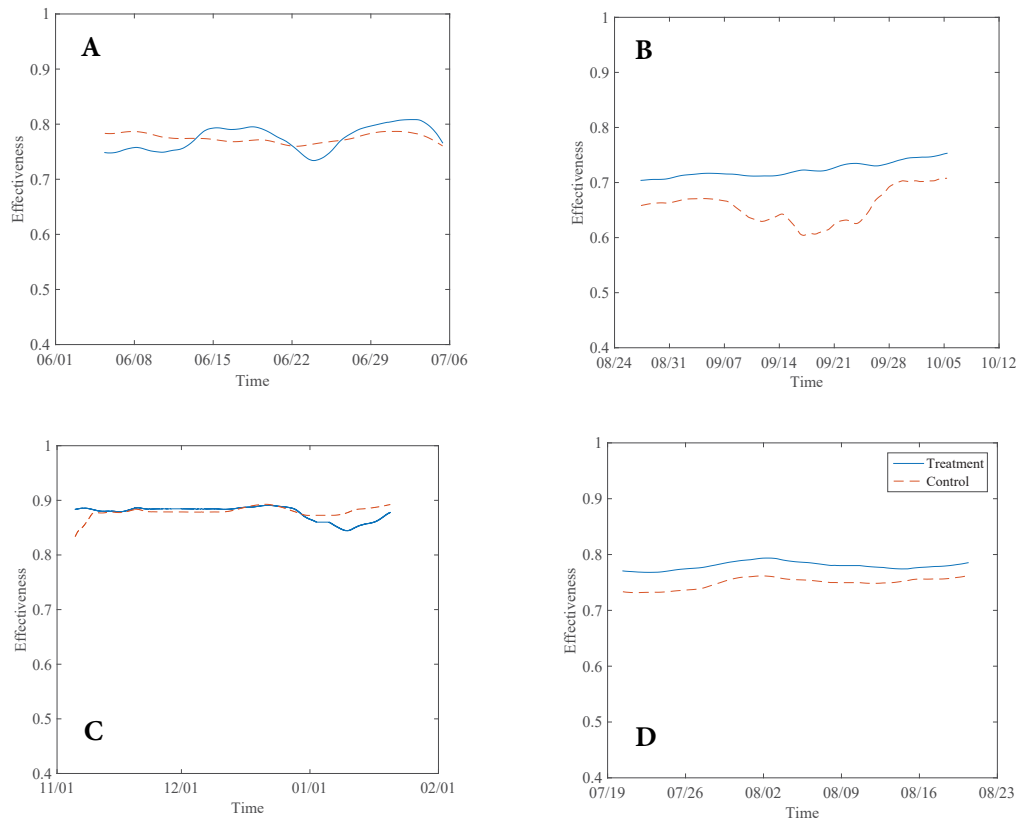


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241 **Figure 3.** Time series of calculated effectiveness for UV-treated and control coils over the  
 242 course of 18 months.

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247 **Figure 4.** Sampling subsets used for statistical analyses. (A) Baseline Region – both coils had  
 248 been running for four undisturbed months to build up fouling, UV lamp had not yet been turned  
 249 on. (B) Humid Region 1 – immediately after UV lamp was turned on for the treatment coil, still  
 250 in condensing conditions. (C) Dry Region – period of time in the winter when coil surfaces were  
 251 dry due to very low humidity level of inlet air. (D) Humid Region 2 – the second period of time  
 252 when condensing conditions were achieved the following summer.



253           The bootstrapped ANOVA showed that all sampling region means were not equal to each  
254 other (Table 2). A sampling region mean refers to the mean of differences between the control  
255 and treatment coils on a point by point basis within a specific sampling period of time and mode  
256 of operation. Tukey’s HSD procedure found that the Dry Region mean was not significantly  
257 different than the Baseline Region mean, indicating that UV treatment had no effect on heat  
258 transfer effectiveness during dry conditions (Figure A1). The two humid region means were  
259 statistically significantly greater than the Baseline Region mean. This result suggests that  
260 treatment with UV was most effective during humid operating conditions, when water was  
261 actively condensing out of the air. Based on the data from the two humid regions of operation,  
262 the average heat transfer of the UV-treated coil was 3.0% to 6.4% more effective compared to  
263 the control coil (Figure A1). Uncertainty due to sensor error was estimated to be +/- 2.7%.  
264 Bootstrapped t-tests found that the Baseline and Dry Regions were not significantly different  
265 than zero while both humid regions were significantly greater than zero (Table 2). The  
266 distributions of the t-test p values are shown as histograms in Figure A2.

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Table 2. Mean p-values from bootstrapped T-tests and bootstrapped one-way analysis of variance

Mode of Operation	Mean of Differences	Standard Deviation of Differences	Effectiveness <sup>2</sup>		System Curve Coefficient	
			T-test p value	ANOVA p value	T-test p value	ANOVA p value
Baseline Region	8.2e-05	0.021	<i>0.49</i>		7.5e-05	
Humid Region 1	0.067	0.026	1.3e-43		2.1e-25	
Dry Region	-0.0019	0.015	<i>0.19</i>		1.3e-23	
Humid Region 2	0.031	0.0055	1.5e-72		1.3e-08	
All Regions ANOVA				4.1e-78		6.9e-75

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Table 2. Mean p-values from bootstrapped T-tests and bootstrapped one-way analysis of variance

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*3.2 Coil System Curve Coefficient.* Since the hypothesis was that UV treatment would reduce pressure drop, the difference between coils was calculated as control coil system curve coefficient minus treatment coil system curve coefficient ( $\Delta a_i = a_{\text{Control}_i} - a_{\text{Treatment}_i}$ ). The larger the positive difference, the lower the UV-treated coil pressure drop was in comparison to the control coil.

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Bootstrapped t-tests revealed that all region means were significantly different than zero (Table A1), meaning that the two coils did not have the same pressure drop at the beginning of the experiment during the Baseline period. The bootstrapped ANOVA showed that all four region means were not equal to each other but the Tukey's HSD procedure found that the

<sup>2</sup> Italicized values signify that the region mean was not significantly different than zero, meaning the treatment and control coils were not significantly different for those sampling regions.

282 Baseline, Humid Region 1, and Humid Region 2 subsets were not significantly different from  
283 each other (Figure A3). The Dry Region was statistically significantly less than the other three  
284 regions, but not outside the range of sensor measurement error ( $\pm 0.005$  in-H<sub>2</sub>O for static  
285 pressure drop reading). In summary, we were unable to detect any effect on static pressure drop  
286 resulting from UV treatment.

#### 287 **4.0 Discussion**

288 While our analyses show statistical significance between baseline and wet region groups for the  
289 calculated heat transfer effectiveness, sensor accuracy and precision obscures somewhat the  
290 difference detected between control and treatment coils. Additionally, any statistical significance  
291 between the coils for static pressure drop was not outside the range of error associated with our  
292 pressure differential sensors. We observed that coil fin biofouling was reduced with UVG-CC  
293 [23], thus we think that there is a small effect on pressure drop that went undetected due to the  
294 sensitivity and detection limits of our instrumentation. In our companion paper we report on the  
295 microbial loading observed during this study. We saw higher microbial loading downstream during  
296 condensing conditions, and higher loading on upstream surfaces in dry conditions. Microbial surface  
297 concentrations were not statistically different on the UV-treated coil versus the control except for on  
298 just one sampling day.

299         The temperature and relative humidity of the air entering our test duct were mild  
300 compared to the condensing conditions of cooling coils in hot, humid climates. We think UVG-  
301 CC treatment is likely more effective in a location located in the International Energy  
302 Conservation Code (IECC) Climate Zone 2 such as Southern Florida, with high cooling latent  
303 loads, and possibly more persistent fin biofilms, compared to the IECC Climate zone 7 & 8 such

304 as Alaska with little to no cooling days annually [24]. Our laboratory setup was located between  
305 these two extremes, in Boulder, Colorado, with average entering conditions of 24 °C (75°F) and  
306 44% RH compared to average August conditions of 29 °C (85°F) and 72% RH in Miami, FL  
307 [25]. It is possible that increases in heat transfer effectiveness may be greater than 3.0-6.4% in  
308 climates with higher cooling latent loads, a question we urge researchers to investigate in future  
309 UVG-CC studies.

310 Another possible source of error may be using indoor air that had been prefiltered (with a  
311 MERV 14) as inlet air. We were unable to use outside air in our experiment due to the location  
312 and design of the lab. The lab was frequently occupied by personnel and was used to build and  
313 calibrate instruments, so it was not a sterile environment. In a parallel experiment in a real  
314 building air handling unit in West Virginia, consisting of a dual ducted VAV air handling unit  
315 with both a control and UV-treated coil installed. This system ran for over two years, however  
316 we were unable to detect any difference in performance between the UV-treated and control  
317 cooling coils [26]. Wang et al [13] reported on a 10-month investigation in a real building air  
318 handling unit in Singapore that showed thermal performance of the coil increased by 10%. These  
319 results are comparable to our highest possible estimate of heat transfer effectiveness  
320 improvement with UV ( $6.4\% + 2.7\% = 9.1\%$ ). This study, design, however, was different in that it  
321 compared before UV to after UV.

322 This study investigated the effect of UV-treatment on airside heat transfer effectiveness  
323 but additional energy savings are likely due to a reduced load on the chiller supplying the cooling  
324 fluid to the cooling coil. The effect of UV-treatment on pressure drop may be more pronounced  
325 when biofilms are more robust than with our laboratory setup and when scaled up to  
326 commercial-sized cooling coils that have much more surface area than the coils in our setup,

327 resulting in potential fan energy savings as well. A recent simulation of UVG-CC in a  
328 representative office building in Philadelphia found that eliminating biofouling led to a decrease  
329 in pump energy use between 15% and 21% as well as a decrease in fan energy use ranging  
330 between 15% and 23% [27]. Wang and colleagues [13] found that the fan energy use fell by 9%  
331 during a 10-month period in an air handling unit with UVG-CC in Singapore.

332 In summary, UVG-CC is effective at reducing biofouling [23] and increases heat transfer  
333 effectiveness in wetted conditions. We found an effectiveness increase between 3.0-6.4% during  
334 condensing conditions in our laboratory setup under mild climate conditions, with an uncertainty  
335 of +/- 2.7% resulting from the accuracy and precision of our instrumentation. We did not observe  
336 any differences in heat transfer effectiveness between the UV-treated and control coils during  
337 dry conditions, suggesting that installation of this technology should be carefully considered  
338 depending on the climatic region, and may not need to be operated during non-condensing states.  
339 UVG-CC also had no effect on static pressure drop in our laboratory setup with mild climatic  
340 conditions. Future studies of UVG-CC should pay careful attention to the sensitivity and  
341 detection limits of their instrumentation, and would benefit from studying environments prone to  
342 excessive biological fouling so that differences between UV and non-UV coils are more  
343 pronounced.

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