2017

Ultraviolet Germicidal Coil Cleaning: Impact on Heat Transfer Effectiveness and Static Pressure Drop

Julia Cristina Luongo

Jason Brownstein

Shelly L Miller

Ultraviolet Germicidal Coil Cleaning: Impact on Heat Transfer Effectiveness and Static Pressure Drop

Julia C. Luongo[1], Jason Brownstein[1], Shelly L. Miller[1,]*

[1]Department of Mechanical Engineering, University of Colorado at Boulder, 427 UCB, Boulder, CO 80309

*Corresponding author, E-mail: Shelly.Miller@colorado.edu, Phone: 303-492-0587, Fax: 303-492-3498

Abstract

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

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

1.0 Introduction

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

In many climates where the outdoor air must be dehumidified prior to entering the building space, air is cooled below the dew point to condense moisture out of the air. This moisture can linger within the densely packed fins of a cooling coil and eventually form biofilms from deposited environmental bacteria and fungi present in the air. Heat exchanger surfaces are an ideal site for biofilms due to the presence of adequate nutrients (i.e., debris inherent on coil surfaces) and moisture [6]. High bacterial and fungal concentrations have been documented within HVAC systems, specifically on cooling coils and drain pans [7–9].

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

While health benefits of UVG-CC have been shown in the literature [9,10], little evidence exists of the potential energy efficiency benefits of this technology. Anecdotal evidence describes “visibly cleaner” cooling coils and energy savings after the installation of a UVG-CC system [11]. An increase in energy efficiency of 10-15% from coil cleaning has also been reported, but not specifically using UVG-CC [12]. A recent paper reported on a field study applying UVG-CC to an air handling unit in a building in Singapore. Results show that the coil overall thermal conductance increased by 10%, the pressure drop decreased by 13% and the fan energy used fell by 9% [13].

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

2.0 Materials and methods

2.1 Test Facility. A custom HVAC test apparatus was built in the Air Quality Laboratory at the University of Colorado Boulder, consisting of two parallel ducts, each with its own cooling coil, but supplied by the same temperature and relative humidity controlled airstream (Figure 1). Similar custom testing ducts have been used to investigate heat exchanger performance [14-16]. The coils were steam cleaned prior to starting the tests. The test apparatus was equipped with sensors to measure duct velocities using pitot tubes (BAPI ZPS-ACC12, ±1% on 0-1 psi range, www.bapihvac.com) connected to differential pressure sensors (OMEGA PX2650, ±1% best fit straight line (BFSL), www.omega.com), static pressure drops (OMEGA PX2650), entering and exiting water temperatures (OMEGA TH-44000-NPT, ±0.1°C, www.omega.com), and entering and exiting air temperatures (OMEGA ON-405, ±0.1°C, www.omega.com) and relative humidity (OMEGA HX71, ±4%, www.omega.com) for each branch. Voltage output from the sensors was fed into a data acquisition system (NI cDAQ-9171 with NI 9205 module, www.ni.com) and processed with LabView to export data for analysis in MATLAB. Both coils were TRANE light commercial tube and fin coils (Type P2) with aluminum fins (Prima-flo H) and copper tubes, one-ft^2 face area, 12 fins/inch, and were four rows deep. One UVC lamp (ALTRU-V V-Ray Model 23-1100, 25W) was installed ten inches away from the coil on the downstream side. The lamp was burned in for 100 hours prior to use. The lamp was shielded with mesh to achieve the desired level of surface irradiance.

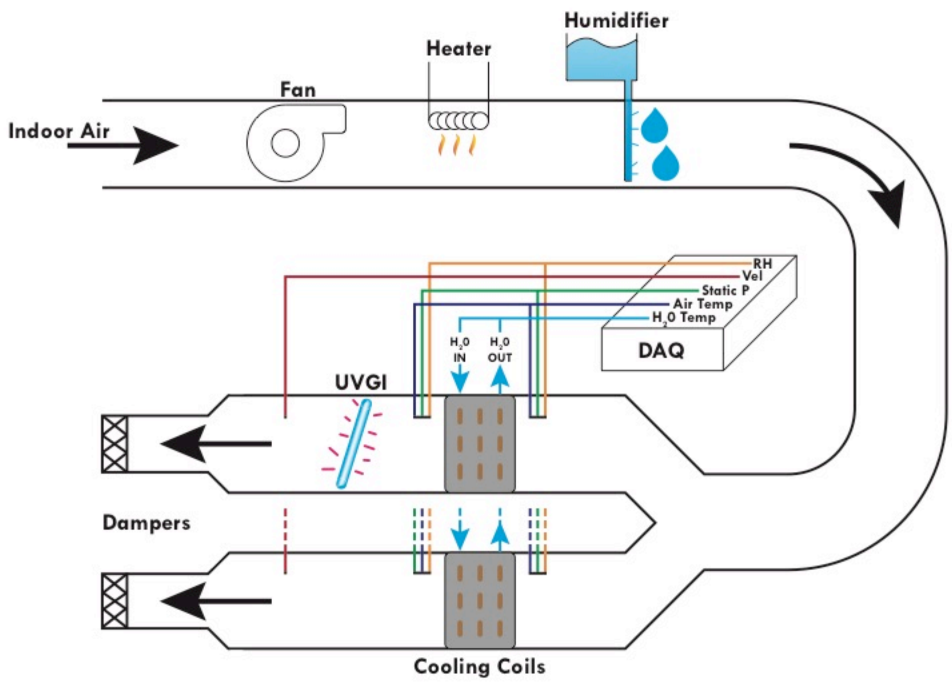

Figure 1. Schematic of HVAC test apparatus.

Table 1. Sampling subsets used for statistical analysis

Sampling Subsets	*Start Date*	*End Date*	*Mean Entering Air Temperature (Standard Deviation (SD)) and Rang, Mean Relative Humidity (SD) and Range, Mean Dew Point*[1]	*Description*
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

[1] 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.

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

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

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

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

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

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

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

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

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

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

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

$$\Delta P_s = aV^2 \qquad (3)$$

where ΔP_s is the static pressure drop across the coil, V is the volumetric flow rate of air through the coil, and a is a coefficient that describes the steepness of the curve. Equation 3 was fit to the data and is shown in Figure 2. Due to the quadratic relationship between flow and pressure drop, this coefficient, *a*, can be calculated for each coil for every sampling point. As the coil becomes more or less fouled, it is expected that its system curve will change, becoming steeper with increased fouling and shallower with decreased fouling. Figure 2 shows how both coils had very similar system curves during the baseline period (pre-UV treatment) and also demonstrates the quadratic relationship between pressure drop and flow rate.

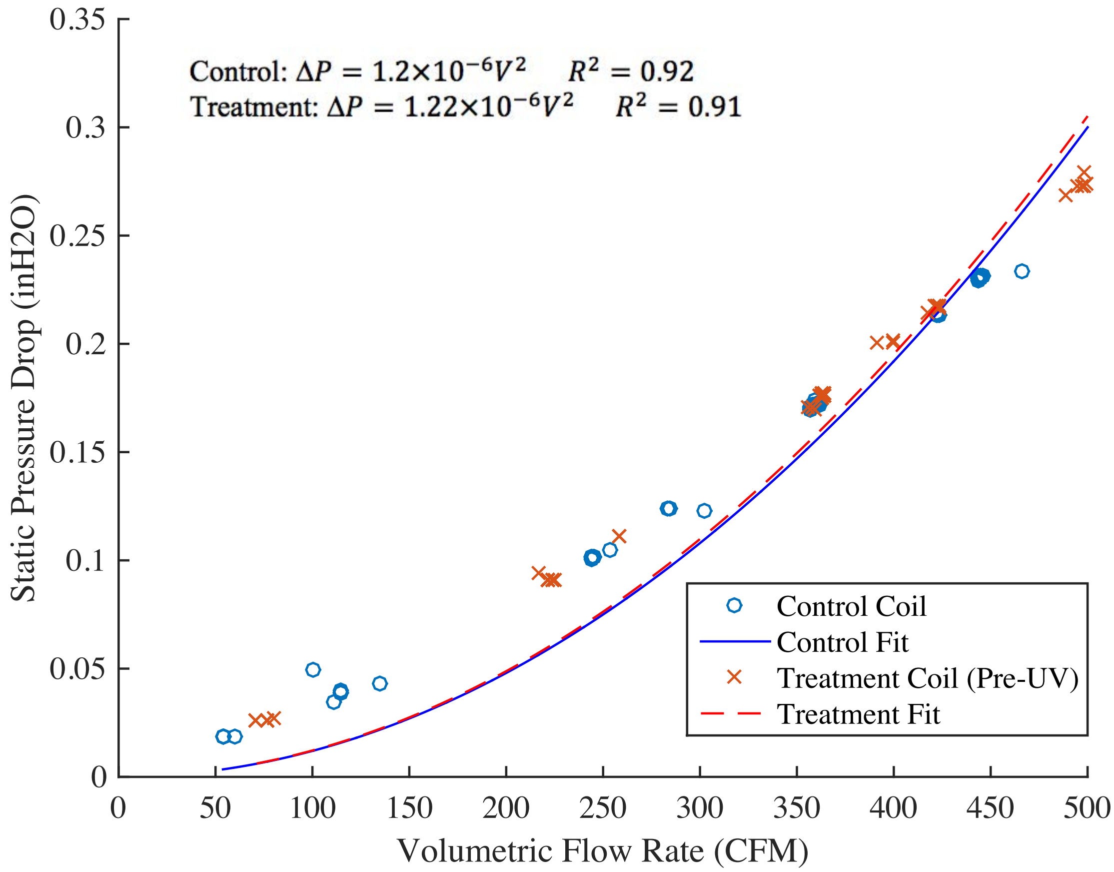

Figure 2. Static pressure drop versus volumetric flow rate for both cooling coils during the baseline period. The two coils displayed very similar system curves at the start of the experiment. Pressure drop and velocity uncertainty was ±1%.

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

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

2.5 Statistical Analyses. All data were smoothed using a moving average approach with a boxcar window that averaged each sampling point with the 5000 points (approximately 28 hours) to its right and left to mute short-term fluctuations due to the building's HVAC system operation and focus on long-term UVG-CC effects. An example of raw versus smoothed data can be found in Figure A4.

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

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

2.6 Sensor Uncertainty Propagation. Each measured parameter had an associated accuracy (obtained from the sensor manufacturer, see 2.1) that was propagated through all of our analyses to determine the uncertainty of calculated values of heat transfer effectiveness and coil system curve coefficients due to sensor accuracy. The standard variance formula was used, which assumes variables are not correlated and are independent [22].

3.0 Results

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

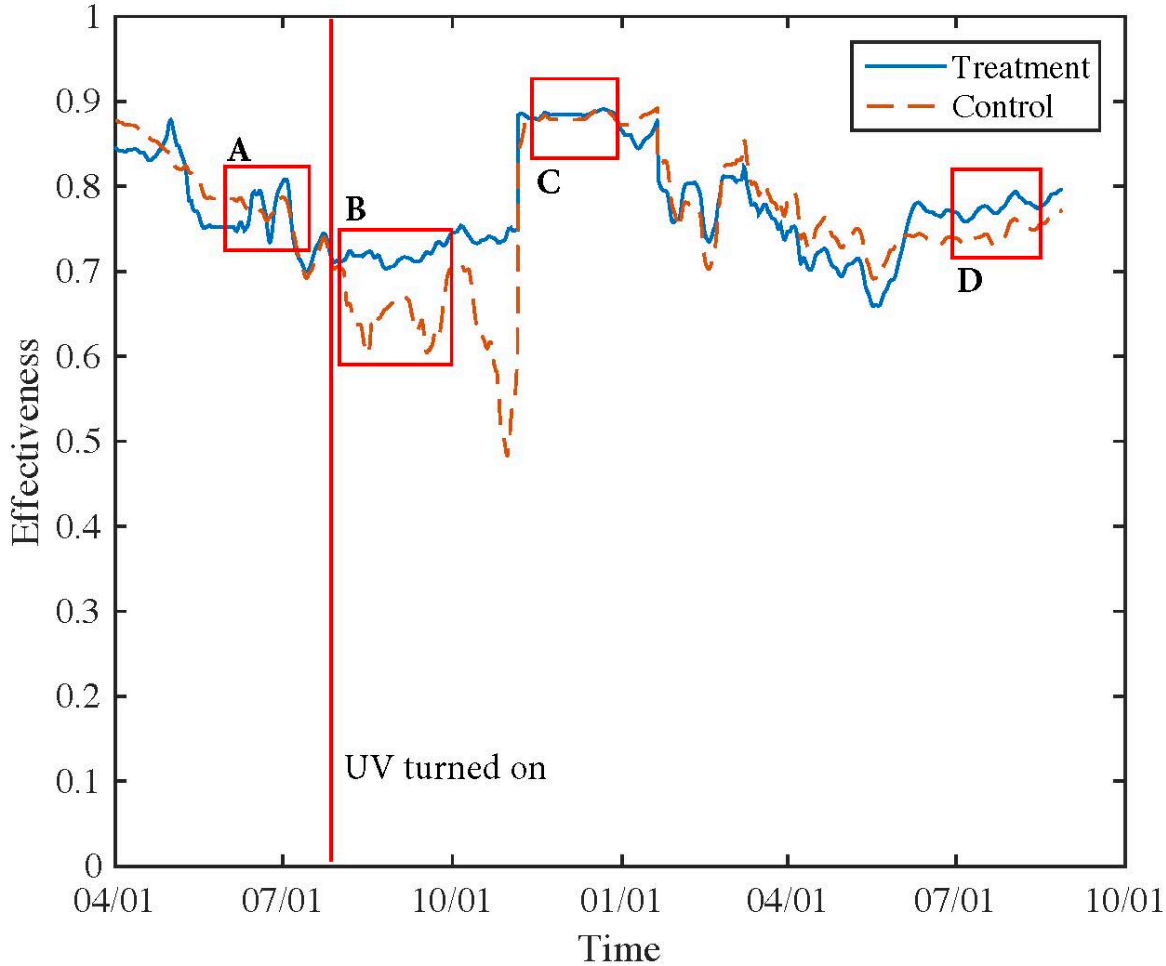

Figure 3. Time series of calculated effectiveness for UV-treated and control coils over the course of 18 months.

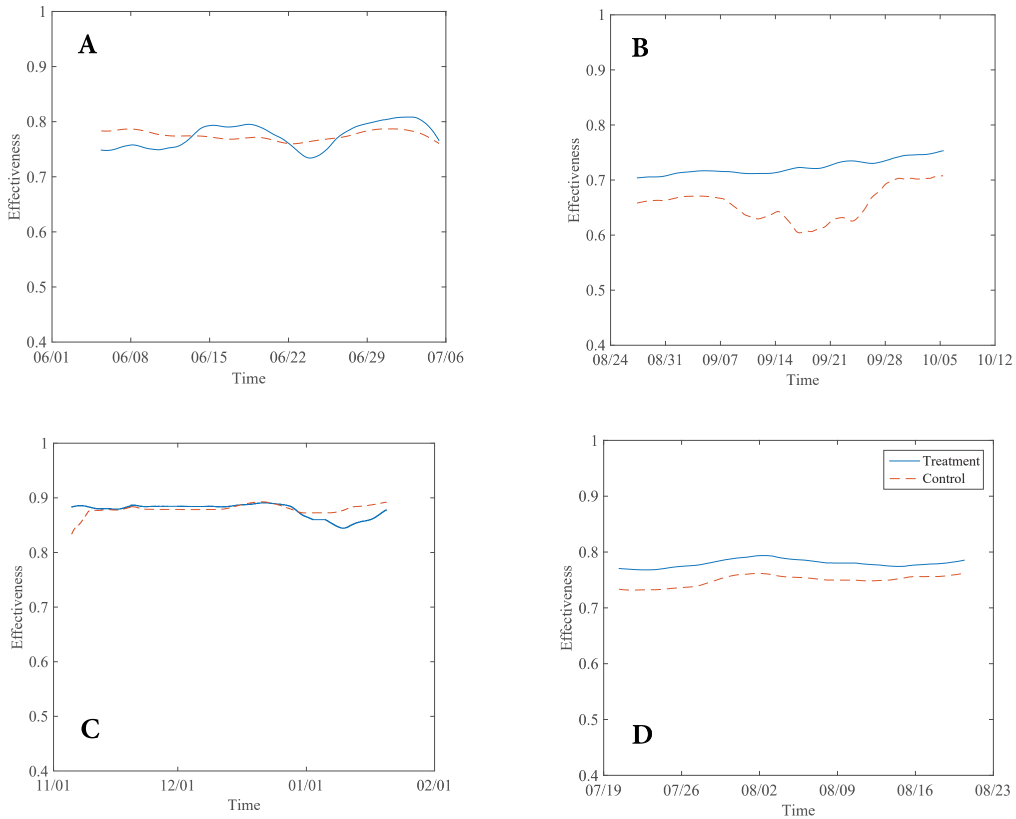

Figure 4. Sampling subsets used for statistical analyses. (A) Baseline Region – both coils had been running for four undisturbed months to build up fouling, UV lamp had not yet been turned on. (B) Humid Region 1 – immediately after UV lamp was turned on for the treatment coil, still in condensing conditions.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

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

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

Mode of Operation			Effectiveness[2]		System Curve Coefficient	
	Mean of Differences	Standard Deviation of Differences	T-test p value	ANOVA p value	T-test p value	ANOVA p value
Baseline Region	8.2e-05	0.021	*0.49*		7.5e-05	
Humid Region 1	0.067	0.026	1.3e-43		2.1e-25	
Dry Region	-0.0019	0.015	*0.19*		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

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

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 ($\Delta_{a_i} = a_{\mathrm{Control}_i} - a_{\mathrm{Treatment}_i}$). The larger the positive difference, the lower the UV-treated coil pressure drop was in comparison to the control coil.

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

[2] 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.

Baseline, Humid Region 1, and Humid Region 2 subsets were not significantly different from each other (Figure A3). The Dry Region was statistically significantly less than the other three regions, but not outside the range of sensor measurement error (+/- 0.005 in-H_2O for static pressure drop reading). In summary, we were unable to detect any effect on static pressure drop resulting from UV treatment.

4.0 Discussion

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

The temperature and relative humidity of the air entering our test duct were mild compared to the condensing conditions of cooling coils in hot, humid climates. We think UVG-CC treatment is likely more effective in a location located in the International Energy Conservation Code (IECC) Climate Zone 2 such as Southern Florida, with high cooling latent loads, and possibly more persistent fin biofilms, compared to the IECC Climate zone 7 & 8 such as Alaska with little to no cooling days annually [24]. Our laboratory setup was located between these two extremes, in Boulder, Colorado, with average entering conditions of 24 ^{o}C (75^{o}F) and 44% RH compared to average August conditions of 29 ^{o}C (85^{o}F) and 72% RH in Miami, FL [25]. It is possible that increases in heat transfer effectiveness may be greater than 3.0-6.4% in climates with higher cooling latent loads, a question we urge researchers to investigate in future UVG-CC studies.

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

This study investigated the effect of UV-treatment on airside heat transfer effectiveness but additional energy savings are likely due to a reduced load on the chiller supplying the cooling fluid to the cooling coil. The effect of UV-treatment on pressure drop may be more pronounced when biofilms are more robust than with our laboratory setup and when scaled up to commercial-sized cooling coils that have much more surface area than the coils in our setup, resulting in potential fan energy savings as well. A recent simulation of UVG-CC in a representative office building in Philadelphia found that eliminating biofouling led to a decrease in pump energy use between 15% and 21% as well as a decrease in fan energy use ranging between 15% and 23% [27]. Wang and colleagues [13] found that the fan energy use fell by 9% during a 10-month period in an air handling unit with UVG-CC in Singapore.

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

Acknowledgements

We gratefully acknowledge financial support for this project from the University of Colorado Boulder Innovative Seed Grant Program, the Department of Mechanical Engineering, University of Colorado Boulder, and an Industry Consortium consisting of four UV companies. A sincere thank you to Trane for donating the cooling coils.

References

[1] N.G. Reed, The history of ultraviolet germicidal irradiation for air disinfection., Public Health Rep. 125 (2010) 15–27.

[2] S.L. Miller, J. Linnes, J. Luongo, Ultraviolet germicidal irradiation: future directions for air disinfection and building applications., Photochem. Photobiol. 89 (2013) 777–81. doi:10.1111/php.12080.

[3] B. Lee, W. Bahnfleth, K. Auer, Life-cycle cost simulation of in-duct ultraviolet germicidal irradiation systems, in: Build. Simul., 2009: pp. 1159–1166.

[4] DOE, Buildings Energy Data Book, 2011.

[5] EIA, Commercial Buildings Energy Consumption Survey (CBECS) Data, 2003.

[6] P.R. Morey, Microorganisms in buildings and HVAC systems: A summary of 21 environmental studies., in: Proc. ASHRAE Conf. Indoor Air Qual. Am. Soc. Heat. Refrig. Air-Cond. Eng., Atlanta, GA, 1988: pp. 10–24.

[7] P. Hugenholtz, J. a Fuerst, Heterotrophic bacteria in an air-handling system., Appl. Environ. Microbiol. 58 (1992) 3914–20.

[8] E. Levetin, R. Shaughnessy, C.A. Rogers, R. Scheir, Effectiveness of Germicidal UV Radiation for Reducing Fungal Contamination within Air-Handling Units, Appl. Environ. Microbiol. 67 (2001) 3712–3715. doi:10.1128/AEM.67.8.3712.

[9] D. Menzies, J. Popa, J.A. Hanley, T. Rand, D.K. Milton, Effect of ultraviolet germicidal lights installed in office ventilation systems on workers' health and wellbeing: double-blind multiple crossover trial., Lancet. 362 (2003) 1785–91. doi:10.1016/S0140-6736(03)14897-0.

[10] R.M. Ryan, G.E. Wilding, R.J. Wynn, R.C. Welliver, B. a Holm, C.L. Leach, Effect of enhanced ultraviolet germicidal irradiation in the heating ventilation and air conditioning system on ventilator-associated pneumonia in a neonatal intensive care unit, J. Perinatol. Off. J. Calif. Perinat. Assoc. 31 (2011) 607–14. doi:10.1038/jp.2011.16.

[11] F. Keikavousi, UVC : Florida Hospital Puts HVAC Maintenance Under a New Light, Eng. Syst. (2004) 60–66.

[12] R.D. Montgomery, R. Baker, Study Verifies Coil Cleaning Saves Energy, ASHRAE J. (2006) 34–37.

[13] Y. Wang, C. Sekhar, W. P. Bahnfleth, K. W. Cheong, J. Firrantello, Effectiveness of an ultraviolet germicidal irradiation system in enhancing cooling coil energy performance in a hot and humid climate, Energy Build. 130 (2016) 321–329.

[14] I.H. Bell, E. a. Groll, Air-side particulate fouling of microchannel heat exchangers: Experimental comparison of air-side pressure drop and heat transfer with plate-fin heat exchanger, Appl. Therm. Eng. 31 (2011) 742–749. doi:10.1016/j.applthermaleng.2010.10.019.

[15] B.C. Pak, E. A. Groll, J. E. Braun, Impact of Fouling and Cleaning on Plate Fin and Spine Fin Heat Exchanger Performance, ASHRAE Trans. 111 (2005).

[16] H. Pu, G. Ding, X. Ma, H. Hu, Y. Gao, Effects of biofouling on air-side heat transfer and pressure drop for finned tube heat exchangers, Int. J. Refrig. 32 (2009) 1032–1040. doi:10.1016/j.ijrefrig.2008.10.007.

[17] ANSI (American National Standards Institute)/ASHRAE, Chapter 60: Ultraviolet air and surface treatment, in: HVAC Appl. Handb., American Society of Heating, Refrigerating and Air-Conditioning Engineers, Atlanta, GA, 2015.

[18] W. Cory, Fans and Ventilation: A Practical Guide, Elsevier. (2010) 109.

[19] N. Clements, J.B. Milford, S.L. Miller, W. Navidi, J.L. Peel, M.P. Hannigan, Errors in coarse particulate matter mass concentrations and spatiotemporal characteristics when using subtraction estimation methods, J. Air Waste Manag. Assoc. 63 (2013) 1386–1398. doi:10.1080/10962247.2013.816643.

[20] D. Alcaraz-Segura, E. Chuvieco, H.E. Epstein, E.S. Kasischke, A. Trishchenko, Debating the greening vs. browning of the North American boreal forest: differences between satellite datasets, Glob. Change Biol. 16 (2010) 760–770. doi:10.1111/j.1365-2486.2009.01956.x.

[21] C.J.F. ter Braak, Permutation Versus Bootstrap Significance Tests in Multiple Regression and Anova, in: P.D.K.-H. Jöckel, P.D.G. Rothe, P.D.W. Sendler (Eds.), Bootstrapping Relat. Tech., Springer Berlin Heidelberg, 1992: pp. 79–85. http://link.springer.com/chapter/10.1007/978-3-642-48850-4_10 (accessed September 20, 2016).

[22] H.H. Ku, Notes on the use of propagation of error formulas, J. Res. Natl. Bur. Stand. 70C (1966) 263–273. doi:10.6028/jres.070c.025.

[23] J.C. Luongo, S.L. Miller, Ultraviolet germicidal coil cleaning: Decreased surface microbial loading and resuspension of cell clusters, Building and Environment 105 (2016): 50-55.

[24] IECC, International Energy Conservation Code, (2009). https://energycode.pnl.gov/EnergyCodeReqs/, accessed 7 September 2016.

[25] NCEI, Climate Data Online, National Centers for Environmental Information, NOAA, n.d. http://www.ncdc.noaa.gov/cdo-web/ (accessed November 12, 2015).

[26] Julia C. Luongo, Towards Defining Healthy Buildings: Investigating the Effect of Building Characteristics and Interventions on Indoor Air Microbial Exposures and Energy Efficiency, Doctoral dissertation, University of Colorado, 2016.

[27] J. Firrantello, J. DeGraw, W. Bahnfleth, Modeled Air Quality and Energy Use Impacts of UVGI for Cooling Coil Biofouling Mitigation, in: ASHRAE IAQ 2013 Proc. Environ. Health Low Energy Build., 2013.