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FOULING OF HVAC FIN AND TUBE HEAT EXCHANGERS

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ABSTRACT

Fin and tube heat exchangers are used widely in residential, commercial and industrial HVAC applications. Invariably, indoor and outdoor air contaminants foul these heat exchangers. This fouling can cause decreased capacity and efficiency of the HVAC equipment as well as indoor air quality problems related to microbiological growth. This paper describes laboratory studies to investigate the mechanisms that cause fouling. The laboratory experiments involve subjecting a 4.7 fins/cm (12 fins/inch) fin and tube heat exchanger to an air stream that contains monodisperse particles. Air velocities ranging from 1.5 - 5.2 m/s (295 ft/min - 1024 ft/min) and particle sizes from 1 – 8.6 µm are used. The measured fraction of particles that deposit as well as information about the location of the deposited material indicate that particles greater than about 1 µm contribute to fouling. These experimental results are used to validate a model that describes the relative importance of several deposition mechanisms including impaction, Brownian diffusion, turbophoresis and gravitational settling. The analysis is extended to apply to different fin spacings and particle sizes typical of those found in indoor air.

INTRODUCTION

Fouling of heat exchangers used in heating, ventilating, and air conditioning (HVAC) systems is important both because of their widespread use in commercial, residential and industrial buildings and the energy and indoor air quality impacts that can result from fouling. Fouling of indoor fin and tube heat exchangers, particularly air conditioner evaporators, is especially important as space cooling in buildings is an important contributor to overall energy use and peak electric demand. Furthermore, the location of heat exchangers in HVAC systems means that if bioaerosols containing bacteria, fungi, and viruses deposit on heat exchangers and remain viable, they can quickly spread through an indoor space if they are re-entrained in the airflow.

Before discussing the details of particle deposition on air conditioner evaporators, it is important to clearly describe the system being studied. The HVAC heat exchangers of interest are designed to exchange energy between a refrigerant and an air stream that is in turn used to condition an indoor space. Typical heat exchangers consist of horizontal refrigerant tubes with attached thin vertical fins to increase heat transfer. A typical residential heat exchanger has two staggered sets of 0.95 cm (3/8 inch) copper refrigerant tubes that run horizontally through vertical aluminum fins. Commercial and industrial systems can have much larger tubes. Fin spacings range from 2.4 to 7.9 fins/cm (6 - 20 Fins/inch or FPI), with typical systems having 4.7 fins/cm (12 FPI). The fins are approximately 100 µm thick and are often corrugated to increase surface area for heat transfer. Heat exchanger depth can vary, but typical residential and small industrial and commercial heat exchangers are about 5 cm (2 inch) thick and are often grouped together for larger capacities. Air velocities range from 1 to 5 m/s (200 – 1000 ft/min) in these systems.

There are two important non-dimensional parameters that characterize the airflow in the system. Re_{duct} and Re_{fin} are the Reynolds numbers associated with the characteristic dimension of the duct and the fin spacing, respectively. Mathematical formulae for, and typical values of, these parameters are shown in Table 1. These parameters occur have a wide range of values in typical residential and commercial systems.

Despite its importance, there has been relatively little research on air-side coil fouling. There have been several anecdotal reports of HVAC heat exchanger fouling (i.e. Anonymous, 1987; Neal, 1992). In the engineering literature, Krafthefter & Bonne (1986) report that a typical residential heat pump system will foul sufficiently to cause a 20 % reduction in performance over a 4 to 7 year period. They further report substantial reductions in fouling from large dust particles by the installation of electric air cleaners. Muyshondt et al. (1998) used a computational fluid dynamic (CFD) approach to predict aerosol deposition on fin and tube heat exchangers for a matrix of velocities, heat exchanger geometries, and horizontal and vertical fin orientations. Their work suggests, "the evaporator coil will collect significant amounts of aerosol in the particle size range of 5 – 100 µm." While both of these works contribute significantly to our understanding of coil fouling, there is still a gap in the literature of our understanding of the mechanisms that lead to coil fouling as well as a lack of particle size resolved experimental data of deposition on a typical fin and tube heat exchanger.

Table 1: Heat Exchanger Reynolds Numbers

Parameter and Formula	Typical Ranges	
	Residential	Commercial
$Re_{duct} = \frac{U_{bulk} \cdot h_{duct}}{\nu_{air}}$	20 000 - 100 000	10 000 - 300 000
$Re_{fin} = \frac{U_{fin} \cdot h_{fin}}{\nu_{air}}$	100 - 500	100 - 2 000

Data in table calculated from information in ASHRAE (2001) and McQuiston et al. (2000).

This research described in this paper builds on previous research by trying to address the fundamental causes of coil fouling. The purpose of the research is to explore how likely a particle is to deposit on a fin and tube coil. The central question being addressed is given an air velocity, a coil geometry (i.e. fin spacing), and a particle size what is the probability that a particle will deposit on a heat exchanger? A secondary question is what mechanisms are responsible for particulate fouling in these systems. The answers to these questions are approached first analytically and then with experimental data.

MODELING METHODOLOGY AND RESULTS

In order to predict particulate fouling and to better understand which deposition mechanisms are most important for fouling of fin and tube heat exchangers, a mathematical model of the system was constructed. The model focuses on deposition associated with the particle inertia, such as impaction and interception on fin edges and refrigerant tubes, gravitational settling of large particles, and Brownian diffusion of small particles in the heat exchanger core. Additional deposition, caused by turbophoresis, the motion of large particles down a turbulence intensity gradient, near the leading edge of the fins is also included. If the heat exchanger is in heating or cooling mode, there can be additional deposition from thermal effects (thermophoresis) and humidity concentration gradients (diffusiophoresis). These effects are included in the model, but, in the interest of simplicity and available space, they are left out of this analysis.

Given the complex geometry and air flow in a typical fin and tube heat exchanger, the model takes a relatively simple approach to the system being studied. The analysis is broken down into two distinct parts: the modeling of deposition associated with impaction on the fins and tubes and the deposition associated with all of the other mechanisms.

There are three important non-dimensional parameters associated with impaction of particles in the system. The first, Re_{part}, is the Reynolds number associated with the particle size. Re_{part} determines the regime of particle drag, which affects the calculation of the Stokes number, stopping distance, and particle relaxation time. The Stokes numbers for particles approaching both the fins and the refrigerant tubes are also important. Stokes numbers much greater than unity would likely lead to particle impaction and Stokes numbers much less than one would likely not impact. The formulae for these parameters, as well as the range of values for air flows from 1- 5 m/s (200 – 1000 ft/min) and particle diameters from 0.01 – 100 µm, are listed in Table 2.

Table 2: Non-dimensional Parameters Associated With Impaction Deposition of Particles on the Coil

Parameter and Formula	Typical Range in HVAC Systems
$Re_{part} = \frac{U_{air} \cdot d_{part}}{\nu_{air}}$	0.001 - 30
$St_{fin} = \frac{\rho_{part} \cdot d_{part}^2 \cdot U_{bulk} \cdot C}{18 \cdot \mu_{air} \cdot t_{fin}}$	0.000 01 - 2 000
$St_{tube} = \frac{\rho_{part} \cdot d_{part}^2 \cdot U_{fin} \cdot C}{18 \cdot \mu_{air} \cdot d_{tube}}$	0.000 000 1 - 20

It should be noted that the calculation of the two Stokes numbers in Table 2 assume the particle Reynolds number is less than about unity. Although this is not strictly true for the largest particles at the highest velocities being considered, the amount of deviation of the actual values from the listed Stokes numbers is small.

The analysis of impaction is adapted from Hinds' (1982) analysis of cascade impactors. Although it is tempting to instead use a more rigorous single fiber deposition analysis for each fin edge (i.e. Yeh & Liu, 1974), the required Reynolds numbers (based on the fin thickness) for such analyses is less than 1, and thus much lower than is typical in HVAC systems. Although the cascade impactor analysis is relatively crude, it is adequate for this system because of the limits to deposition by impaction that are described below with the modeling results. The deposition associated with impaction on each fin edge and tube is calculated as $\pi/2$ times the relevant Stokes number. This deposition is multiplied by the number of tubes/fins in the coil to get the overall deposition as a result of impaction on tubes and fins.

The analysis for deposition from Brownian diffusion, gravitational settling, and turbophoresis are all calculated by first calculating the characteristic time for a particle to move through the heat exchanger core (heat exchanger depth divided by core velocity, U_{fin}) and then comparing this value to the characteristic times for particles to deposit by each deposition method being considered (one half fin spacing divided by deposition velocity component). The penetration based on each mechanism is defined as the ratio of the characteristic time for deposition divided by the characteristic time for the particle to move through the heat exchanger core. The velocities associated with each deposition mechanism and the appropriate characteristic lengths appear in Table 3. The settling velocity is described in Hinds (1982) and the diffusion velocity is adapted from the material from the same source. The turbophoretic velocity is described in Caporaloni et al. (1975). The calculation of turbophoretic velocity requires detailed knowledge of turbulence parameters, thus direct numerical simulation (DNS) data for flow in channels from Moser et al. (1995) was used. The turbulence from the duct was assumed to penetrate 5 mm (0.2 inch) into the heat exchanger core.

Table 3: Deposition Velocities and Characteristic Distances

Deposition Mechanism	**Deposition Velocity Component**	**Relevant Characteristic Distance**
Gravitational settling	Settling Velocity $v_s = \frac{d_p^2 \cdot g \cdot (\rho_{part} - \rho_{air}) \cdot C}{18 \cdot \mu_{air}}$	Half height of fin corrugation
Brownian diffusion	Diffusion Velocity $v_{diff} = \sqrt{\frac{2 \cdot k \cdot T_{air} \cdot C \cdot U_{fin}}{3 \cdot \mu_{air} \cdot d_p \cdot h_{fin}}}$	Fin spacing
Turbophoresis	Turbophoretic Velocity $v_{TF} = -\tau_{part} \cdot \frac{dv'^2}{dx}$	Turbulence intensity peak to wall

The overall penetration fraction is defined as the product of penetration (one minus the deposition for impaction mechanisms) for each individual mechanism. This assumes independence of each deposition mechanism. Many researchers, including Liu and Nazaroff (2001), use this multiplicative formulation and suggest that the resulting inaccuracies from assuming independence are small. Chen and Yu (1993) suggest an alternate superposition formulation for penetrations from different depositions, but they also report very little increased accuracy and present no physical basis for their superposition technique, thus we use the simpler multiplicative approach. The deposition fraction, the fraction of particles that enter the coil that eventually deposit, is defined as one minus the overall penetration fraction.

There are two major limitations to the modeling. The first is that the modeled deposition on the refrigerant tubes is likely an upper bound on the actual deposition. This is because the air streamlines around each tube are smoother and more rounded than those modeled. A more detailed analysis is being developed using experimental relationships from Israel and Rosener (1983) and Wang (1986). However, these results will likely predict a lower bound on the actual deposition because they assume fully developed laminar flow.

The second limitation is that there is an important deposition effect that is not included in the model. This mechanism is a result of inlet turbulence which leads to an aerosol lift force (called the Saffman lift force in laminar flow) as well as particle impaction on the walls from the initially turbulent flow just inside the coil. Even though Re_{fin} (see Table 1) suggests that the flow in the heat exchanger core would be laminar, surface roughness, geometric non-uniformities, and residual turbulence from the bulk flow can all lead to turbulence at low Reynolds numbers, particularly in the entry region of the heat exchanger core. These mechanisms are important in certain types of particle sampling, but are not well understood or described in the literature for a relevant geometry. Current experimental work is being done to estimate the magnitude of these mechanisms.

A heat exchanger coil was modeled that had the following geometric parameters. The parameters were chosen for comparison to the coil used in the experimental work discussed below. The fins were 114 μm thick and the refrigerant tubes were 0.95 cm (3/8 inch) in diameter. Each set of refrigerant tubes are spaced 2.5 cm (1 inch) apart with an associated staggered line 2.1 cm (0.86 inch) deeper into the coil. There are two sets of tubes in the simulated coil.

Modeling coils with fin spacings from 3.1 – 6.2 fins/cm (8-16 FPI) and velocities from 1 – 5 m/s suggests the following broad general conclusions. Deposition of very small (sub micron) particles is typically very low (i.e. < 5%) and is predominantly caused by Brownian diffusion. Although aerosols of this size, typically caused by indoor combustion and gas to particle conversion processes (Hinds, 1982), are common in indoor environments they are unlikely to deposit, particularly 0.1 – 1 μm particles. Particles in the range of 1 – 10 μm, including household dust, common bioaerosols, and particles from cooking (Hinds, 1982), are likely to deposit on the leading edge of the evaporator by impaction, with minor contributions to deposition from the other mechanisms. Over the range of fin spacings and air velocities of interest, deposition fractions of 1 – 20 % are common in this particle size range. Very large particles, 10 – 100 µm such as those found in indoor dusts (Hinds, 1982), are very likely to deposit by turbophoresis near the leading edge of the fins, by gravitational settling in the corrugated channels of the fins, and by impaction on refrigerant tubes in the core of the heat exchanger. These particles, although less commonly suspended in indoor environments, contribute to the bulk of fouling because of their large size. Although such large particles are likely to be filtered, filter bypass, because of poor installation or duct leakage after the filter on the return (negative pressure) part of the HVAC system, is a common phenomenon.

Figure 1 shows the results of the modeling for 3 different fin spacings and an air velocity of 2 m/s (400 ft/min). The model predicts that particles are generally more likely to deposit for smaller fin spacings. Accumulation mode particles, those between 0.1 and 1 µm, are unlikely to deposit. For particle diameters in the range of 10 – 50 µm, deposition in the core of the heat exchanger is largely caused by impaction on the refrigerant tubes and is essentially fin spacing independent for a given velocity. There are two kinks in the deposition curves in Figure 1. The first occurs at 3 -5 µm and is caused by the fact that impaction on the leading edge of the fins becomes perfectly effective at removing particles from the air directly in front of each fin. Even though the Stokes number increases geometrically with increasing particle diameter, the maximum deposition that can result from impaction is reached when the air in front of each fin edge is completely swept of particles. The kink at 30 – 50 µm is caused by the same limit of impaction deposition on the horizontal refrigerant tubes that run through the heat exchanger.

Figure 2 shows three deposition curves at different velocities for a 4.7 fins/cm (12 FPI) coil. In order to compare with experimental results, the modeled coil has 1 cm tubes (0.4 inch) and 114 µm wide fins. The results are similar to those described above in Figure 1. The results for 5 m/s (984 ft/min) suggest that inertial effects, particularly impaction, are especially important for deposition for 1 – 20 µm particles. Impaction on refrigerant tubes is completely exhausted by 20 µm particles. Deposition for 30 µm and greater particles at high velocity is considerably lower than for these particles at lower velocities. This is because gravitational settling becomes important for very large particles and residence time in the evaporator coil decreases as the air velocity increases. Thus, gravitational settling is less likely to cause deposition at high air velocity.

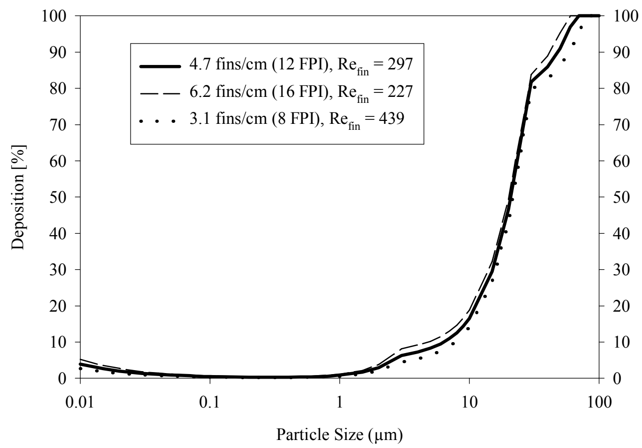

Figure 1: Modeled Deposition as a Function of Fin Spacing for a Bulk Air Velocity of 2 m/s (394 ft/min).

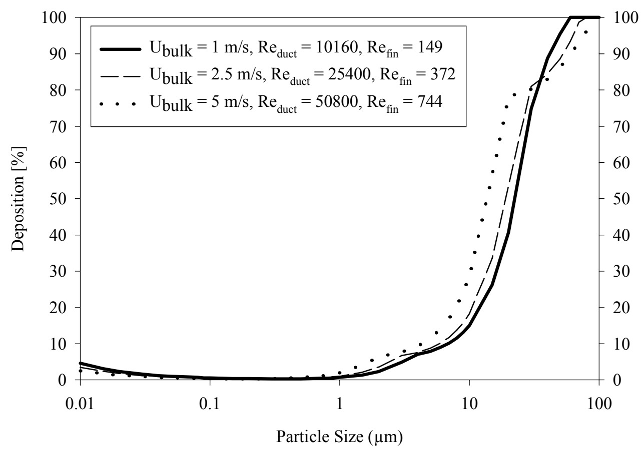

Figure 2: Modeled Deposition as a Function of Bulk Air Velocity on 4.7 fin/cm (12 FPI) Heat Exchanger

The results presented in Figures 1 and 2 are roughly consistent with the published results from computational fluid dynamic simulations of Muyshondt et al. (1998). A rigorous comparison is not possible, because the details of the simulated heat exchanger geometry are not made explicit in Muyshondt et al. (1998). The current experimental results suggest slightly more deposition, particularly at the upper end of the particle size range. This discrepancy is likely caused by the fact that the corrugation of fins was not modeled in Muyshondt et al. (1998) and thus no gravitational settling on vertical fins was included. It is also possible that the effect of inlet turbulence discussed above is included in Muyshondt et al. (1998), but not in this work (see above) explains this aspect of the discrepancy. Muyshondt et al. (1998) also suggest more differentiation in deposition amounts for different fin spacings. This discrepancy might be caused by limitations of the CFD code that they used to accurately simulate turbulent boundary layers on the fin surfaces. The exact nature of these boundary layers is crucial for modeling deposition in such an analysis. Despite some minor differences, this modeling work agrees with the conclusions of earlier work that suggests that fouling of heat exchangers is caused by larger particles.

EXPERIMENTAL METHODS

Although the modeling described above is a useful predictor of important deposition mechanisms, many models tend to underpredict the deposition associated with real particles. Some researchers have suggested that this is due to the effects of surface roughness, or non-homogeneities associated with air turbulence. The bulk of work on this project is focussed on an experiment to directly measure particle deposition associated with fin and tube heat exchangers.

The apparatus used for this experiment is depicted in Figure 3. Monodisperse particles, tagged with fluorescein, are generated with a vibrating orifice aerosol generator and then charge neutralized. The particles are sized with an aerodynamic particle sizer (TSI model 3320). The particles are mixed with a HEPA (high efficiency particle arresting) filtered air stream designed to eliminate ambient particles,. Tests were done to confirm that all ambient particles were removed. The air was then sent into 24 m (80 ft) of straight 15 cm (6 inch) square duct. The duct air velocity can be varied continuously over the 1 - 5 m/s (197 - 984 ft/min) range of interest. These velocities correspond to Reynolds numbers of 10 160 to 50 800 in the duct (Re_{duct}) and Reynolds numbers of 149 to 744 in the core (Re_{fin}). From Table 1, this indicates that the tested heat exchanger is similar to those used in residential and small commercial applications. Several honeycomb flow straighteners are used to promote fully developed turbulent flow with a uniform concentration of test particles. The particle-laden air then passes through an experimental evaporator, which consists of a 4.7 fin/cm (12 FPI) coil that entirely fills the duct. The coil was not cooled or heated, although this will be done in future work.

Particle air concentrations are measured up and down stream of the duct by isokinetically sampling the air onto filter paper, which is later subjected to fluorometric techniques (Turner Designs model TD-700 fluorometer) to determine the particle concentration. The coil and the filters were washed repeatedly with sodium phosphate buffer until there was no measurable amount of fluorescein remaining. Because of non-uniformities associated with mixing downstream of the coil, three different samples are made along the vertical centerline of the duct. The deposition fraction is defined as:

$$D = 1 - \frac{C_{down}}{C_{up}} \qquad (1)$$

where: D = Deposition fraction []
C_{down} = Average downstream concentration [mg/m^3]
C_{up} = Upstream concentration [mg/m^3]

A confirmation of the results is made by removing the test coil from the duct and extracting the deposited particles with a buffer solution and using fluorometric techniques to determine the deposited mass. The penetration calculated by this technique is:

$$D = \frac{M_{coil}}{C_{up} \cdot U_{bulk} \cdot A_{duct} \cdot t} \qquad (2)$$

where: D = Deposition fraction []
M_{coil} = Mass deposited on coil [mg]
C_{up} = Upstream Concentration [mg/m^3]
U_{bulk} = Bulk Air Velocity [m/s]
A_{duct} = Cross sectional duct area [m^2]
t = Experimental Time [s]

Note that the experimental time, t, factors out of the equation because the upstream concentration, C_{up} is calculated as $M_{filter}/(Q_{filter}*t)$, where M_{filter} is the mass deposited on the sampling filter and in the nozzle upstream of the coil (also determined with fluorometric techniques) and Q_{filter} is the sampling pump flow rate.

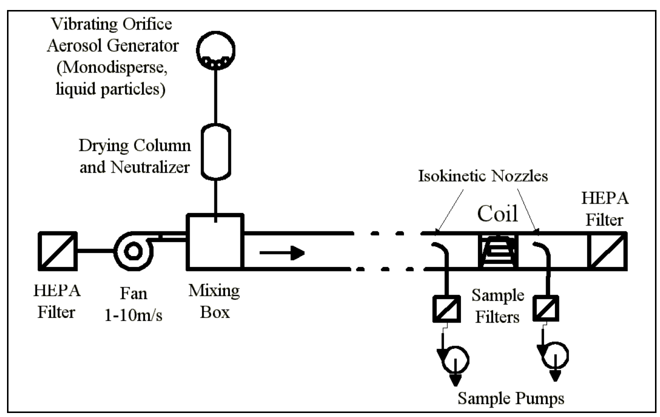

Figure 3: Evaporator Coil Deposition Apparatus

Additionally, in order to better understand the deposition mechanisms that lead to the accumulated material on the coil, the extraction of the coil was typically done to allow separate measurement of the material on the leading edge from material in the core of the coil.

The deposition described by Equation 1 is typically slightly higher than that calculated by Equation 2, because it also includes deposition on the duct between the particle samplers and the coil. For high velocities and large particles, this discrepancy can be quite large, particularly because of deposition downstream of the coil. For this reason, all of the experimental data presented here comes from Equation 2.

EXPERIMENTAL RESULTS

Deposition as a function of particle size is shown in Figures 4-6 for three air velocities and several particle sizes. The vibrating orifice did not generate perfectly monodisperse particles so one standard deviation in particle size is indicated on the plots with horizontal error bars The plots also include the results from the modeling analysis for each experimental air velocity.

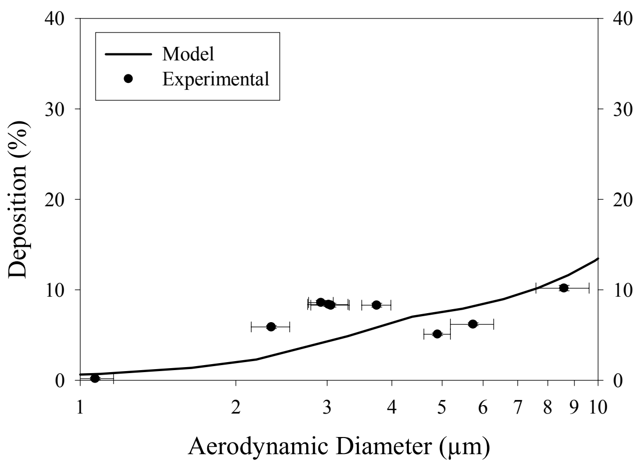

Figure 4: Experimental and Modeled Deposition for 1.5 m/s (295 ft/min), Re_{duct} = 15 240, Re_{fin} = 223. Horizontal error bars indicate one standard deviation in particle diameter.

An uncertainty analysis was also conducted out on the experimental results. All of the terms in Equation 2 area included in the analysis, with the dominant errors coming from uncertainty in velocity measurement, and, in some cases, the mass deposited on the coil and filters. This analysis yielded a 1-10% relative error range on the results, with typical values of about 2%. Error bars are indicated on the plots, but their small size often makes them difficult to see. Three repetitions of the experiment were completed at each of two data points, 3 µm particles at 1.5 m/s (295 ft/fin) and 5.5 µm particles at 5.2 m/s (1 024 ft/min), to confirm the validity of the uncertainty analysis.

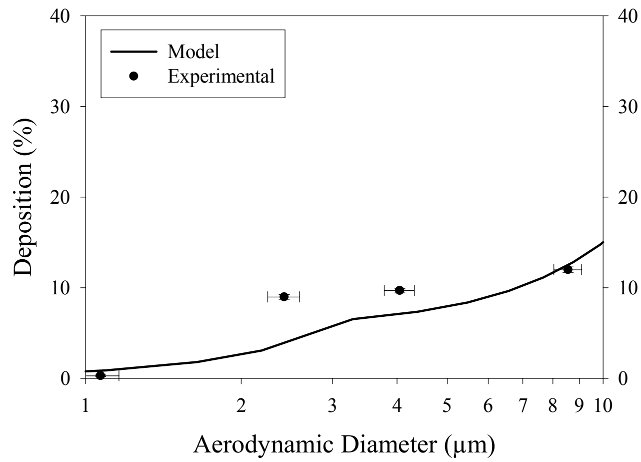

Figure 5: Experimental and Modeled Deposition for 2.1 m/s (413 ft/min) , Re_{duct} = 21340, Re_{fin} = 312. Horizontal error bars indicate one standard deviation in particle diameter.

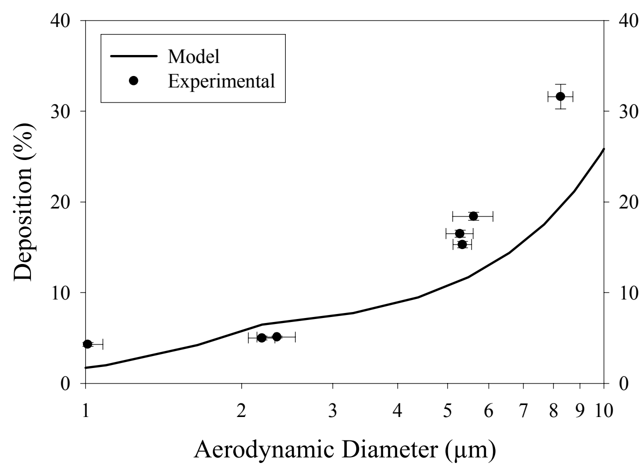

Figure 6: Experimental and Modeled Deposition for 5.2 m/s (1 024 ft/min) , Re_{duct} = 52 830, Re_{fin} = 773. Horizontal error bars indicate one standard deviation in particle diameter.

DISCUSSION

The model predictions, although usually outside of the range of the uncertainty for the experimental data, are typically quite close to the experimental results. The disagreement of the results seems to get worse with increasing velocity. Given the importance of impaction on fin edges for the particle size range of interest, this suggests a problem with the modeling of impaction. The impaction model is currently quite crude and work is underway to improve this part of the model. Another possible reason for the discrepancy between modeled and measured results is related to the fact that increasing velocity leads to increasing Re_{duct} and thus increasing duct turbulence. This turbulence might cause resuspension of particles from duct surfaces and fan blades. Comparing the repetitions done for 3µm particles at 1.5 m/s (295 ft/min) with those at 5.5 µm particles at 5.2 m/s (1 024 ft/min) suggest both that the uncertainty analysis is valid for the lower velocity and also that the actual uncertainty in the measured results is larger than predicted for higher velocities. Blank runs (experiments with no injected particles) at all three velocities with no particles was done to test this hypothesis. Only the 5.2 m/s (1 024 ft/min) velocity showed any resuspended particles. The amount of deposited material on the coil and the filters from resuspension is currently being integrated into an improved uncertainty analysis for high velocity runs. This preliminary analysis suggests a slightly larger relative uncertainty (5-15%) is appropriate for the higher velocity experiments. The blank trials also confirmed the completeness of the extraction of the coil as no fluorescein could be extracted from the coil after the low velocity blank trials.

There is also some indication in the plots that the model might underpredict deposition, particularly for high air velocities and larger particle sizes. Data collection is currently underway for 20 µm particles to test this hypothesis. The inclusion of inlet turbulence deposition, discussed above in the modeling section, will also likely improve the agreement between modeled and measured results.

Another suggestion of why this underprediction might be occurring is that typical evaporators, including the test coil often have discontinuities in the fins in the core of the heat exchanger. These discontinuities are part of the manufacturing process and would lead to increased deposition from impaction of particles at the resulting additional edges. Preliminary analysis from experiments where the leading edge of the fins, defined as the first 5mm, was extracted separately from the core suggest that 15-35 % of the material deposits on the leading edge. This is a smaller amount than would be predicted with the model which suggests increased deposition in the core and supports the fin discontinuity impaction theory.

Although very large particles (>10 µm) likely contribute to most of the air flow reduction and thus capacity and efficiency degradation of these systems, smaller particles also deposit with as much as a 30 % probability. Thus, biologically active aerosols, often in the size range of 1-10 µm, are likely to deposit on heat exchangers. Given the presence of dust and other deposited nutrient containing material, as well as water being condensed out of the air stream, biological growth is a distinctly possible outcome.

In addition to the biological indoor air impacts, there are also heat exchanger performance impacts from coil fouling. There are two potential effects that would lead to reduced HVAC system performance from coil fouling. The first effect is a reduction of heat transfer from the air to the refrigerant because of the buildup of an insulating layer of deposited material and the second is the reduced heat transfer due to decreased air velocity through the coil. The build up of an insulating layer has a very limited effect because the total surface area of an evaporator is very large and thus any deposited material only affects a small amount of the heat transfer surface. In particular, the experimental results and analysis have indicated that particles do not deposit uniformly over the fin surface. In particular, deposition at the leading edge and at the fin discontinuities are in locations that are not particularly important for heat transfer. The second effect, reduced airflow is caused by deposited particles increasing the pressure drop across an evaporator coil and has been shown to be about an order of magnitude greater effect than the insulating layer (Krafthefter & Bonne, 1986).

The experimental work described in this paper will be combined with indoor size-resolved particle concentration data, additional experimental data linking particle deposition and pressure drop, and existing research on air conditioner performance degradation as a result of air flow reduction.

CONCLUSIONS

The experimental and theoretical work presented suggest that fouling of HVAC heat exchangers is predominantly caused by super micron particles. A simple model does reasonably well at predicting particle deposition for the tested 1 – 8.6 µm particle size range and air velocities from 1.5 – 5.2 m/s (295 - 1 024 ft/min), corresponding to Re_{fin} of 223 – 773 and Re_{duct} of 15 240 – 52 830. Particle deposition ranges from less than 1% for 1.1 µm particles at low velocities (1.5 and 2.1 m/s) to over 30% for 8.5 µm particles at high air velocity (5.2 m/s). These deposition rates suggest that particles, particularly larger particles, contribute to fouling. The fouling leads to increased pressure drop and decreased air conditioner performance. Furthermore, deposition of 1 – 10 µm particles that may be biologically active is a likely scenario. Given these significant potential energy and indoor air quality impacts, more research on HVAC coil fouling is required.

NOMENCLATURE

A_{duct}	cross sectional duct area, m^2
C	Cunningham slip correction factor ~ 1 for d_{part} > 10µm, dimensionless
C_{down}	average downstream concentration, mg/m^3

C_{up} upstream concentration, mg/m^3
d_{part} particle diameter, µm
d_{tube} refrigerant tube diameter, cm
D deposition fraction, dimensionless
g Acceleration due to gravity = 9.8, m/s^2
h_{fin} fin spacing, mm
h_{duct} duct characteristic dimension, cm
k Boltzmann's constant = 1.38 x 10^{-23}, J/K
M_{coil} mass deposited on coil, mg
M_{filter} mass deposited on coil, mg
Q_{filter} sample pump flow rate, L/min
Re_{duct} Reynolds number based on h_{duct} and U_{bulk}, dimensionless
Re_{fin} Reynolds number based on h_{fin} and U_{fin}, dimensionless
Re_{part} Reynolds number based on d_{part}, dimensionless
St_{fin} Stokes number based on t_{fin} and U_{bulk}, dimensionless
St_{tube} Stokes number based on d_{tube} and U_{fin}, dimensionless
t experimental time, s
t_{fin} fin thickness, µm
T_{air} air temperature, K
U_{air} air velocity, m/s
U_{bulk} bulk air velocity, m/s
U_{fin} heat exchanger core air velocity, m/s
v' fluctuating air velocity component in wall normal direction, m/s

μ_{air} air dynamic viscosity = 1.8 x 10^{-4} @ STP, kg/m s
ν_{air} air kinematic viscosity = 1.5 x 10^{-5} @ STP, m^2/s
ρ_{air} air density = 1.2@ STP, kg/m^3
ρ_{part} particle density, kg/m^3
τ_{part} particle relaxation time, s

FPI Fins per inch
HVAC Heating, Ventilating and Air Conditioning

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