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Li Yang
Purdue University

James E. Braun
Purdue University

Eckhard A. Groll
Purdue University

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Li Yang*, James E. Braun, Eckhard A. Groll

Ray W. Herrick Laboratories
School of Mechanical Engineering
Purdue University
West Lafayette, IN 47906, USA

*Author for correspondence:
Phone: (765)-496-7515; Fax: (765)-494-0787; Email: lyang@purdue.edu

ABSTRACT

The goal of the study presented in this paper was to evaluate the impact of different filter types on the performance of packaged air conditioners under both clean and fouled conditions. In a companion paper, combinations of 6 different levels of filtration and 4 different coils were tested at fouled conditions. From the tests, it was found that fouling has a relatively small impact on air-side effective heat transfer coefficient, but can have a large impact on pressure drop. The equipment cooling capacity is reduced with fouling primarily because of a decrease in airflow due to the increased pressure drop. In most cases, EER (Energy Efficiency Ratio) was reduced with fouling primarily due to increased fan power. However, the changes in EER were relatively small, in the range of 1%-10%. Equipment having low efficiency filters had higher EER after fouling than equipment with high efficiency filters, because high efficiency filters result in significantly higher pressure drops than low efficiency filters.

1. INTRODUCTION

The buildup of dust on an evaporator leads to an increase in air-side pressure drop and eventual reduction of heat transfer. The impact of fouling on evaporator coils with different upstream filters was presented in a companion paper (Yang et al. (2004)) and correlations of coil pressure drops and air-side effective heat transfer coefficients were obtained. This paper focused on equipment modeling and investigated the impact of the fouling of filter-coil combinations on packaged air conditions.

There are several references that address the effects of filter and coil fouling on overall air conditioning system performance. Krafthefer et al. (1986) studied the buildup rate on coil surfaces, the need for scheduled cleaning and fouling effects on air pressure drop and system energy consumption for heat pumps. The paper showed that particulate accumulation influences peak electricity demand by reducing indoor fan power and compressor power in cooling mode while reducing indoor fan power and increasing compressor power in heating mode. The use of an air cleaner sharply reduced these effects. They estimated a 10-13% decrease in COP for typical evaporator filter fouling of a heat pump. Furthermore, they estimated operating cost savings of 10-25% through use of a high efficiency air filter upstream of the evaporator. Breuker and Braun (1998) conducted an experiment with a three-ton rooftop unit under 96 conditions (4 load levels$\times$24 fault levels, including fouling fault). For the case of fouling, uniform condenser fouling was simulated in the test by blocking the condenser coil with strips of paper. The level of condenser fouling was expressed as a total percent reduction in the surface area. Evaporator fouling was simulated by reducing the airflow rate and it was expressed as a percent reduction from the nominal air flow rate. A 12% reduction in both cooling capacity and COP occurred for a 25% loss of evaporator airflow caused by fouling, while only around a 5% loss in capacity and an 8% loss in COP occurred when about 25% of the condenser coil was block due to fouling.

For this study, three prototypical packaged air conditioners were modeled: a 35-ton unit (medium to large commercial), a 5-ton unit (small commercial) and a 3-ton unit (small commercial or residential). Experimental results for pressure drops and heat transfer coefficients were correlated and the correlations were implemented within computer models of the packaged units and used to evaluate the impact of fouling on cooling capacity and EER. In addition, it was found that the indoor filter and fan efficiency curve had significant impacts on the equipment performance.

2. IAQ EFFECT

Filtration impacts IAQ (Indoor Air Quality) and overall equipment performance. IAQ effects were quantified from the tests by obtaining the dust quantities passing the upstream filter and coil, which would be supplied to the indoor area.

Figure 1 shows the percentage of the dust quantity passing the filter and coil relative to the total 600 grams of injected dust for all cases. Approximately 0.5%-1.8% of dust passed through the coil with an upstream filter of MERV14 or MERV11; approximately 4.8%-7.3% of dust passed through the coil with a MERV8 or MERV6 filter; approximately 15.5%-18.8% of dust passed coil for MERV4 cases and 30% of dust passed coil without any upstream filter. The mass of dust that would enter the indoor space for no-filter cases was nearly 60 times of the mass of dust for MERV14 cases. The difference is extremely large compared to the equipment system impact, which will be discussed in the next section.

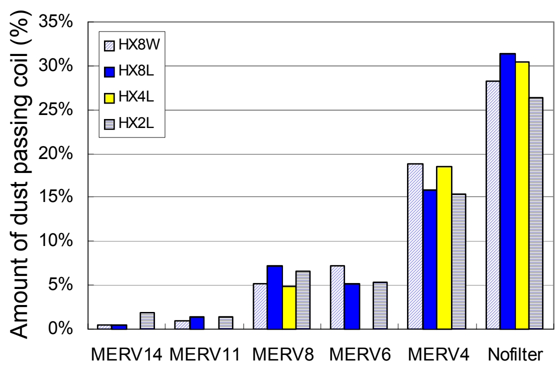

Figure 1: Dust quantities passing coil and filter for all test cases

3. EQUIPMENT ENERGY ANALYSIS

3.1 Equipment Descriptions

The coils tested in this project, eight-row (HX8W, HX8L), four-row (HX4L) and two-row (HX2L) would be employed in medium commercial, small commercial and residential packaged units. Table 1 gives information on the units considered in the simulations.

The 35-ton unit that was employed in this study was a typical commercial rooftop unit. All physical data including the compressor equations, condenser and evaporator configurations and fan performances were provided by the manufacturer. The 3-ton and 5-ton units were from a different manufacturer and they have been used in previous studies for validating system model predictions. The physical data were collected and integrated into a system simulaton model, ACMODEL, developed at Purdue. The fan curves for actual fans employed in the 3-ton and 5-ton units were not available. Thus, performance curves for similar fans were employed.

3.2 Modeling Process

Cooling capacity (Q_c), compressor power (W_c), evaporator-side fan power (W_{fe}) and condenser-side fan power (W_{fc}) were four critical performance factors obtained in the modeling process. Q_c and W_c were determined by ACMODEL and W_{fe} and W_{fc} were determined using fan characteristics and system pressure drop. EER was obtained from the capacity and total power consumption for clean and fouled cases.

ACMODEL is a public-domain computer model developed at Purdue, which can predict the system performance of unitary air conditioners and heat pumps. It has been validated by Rossi (1995) and LeRoy (1997). It was used to obtain cooling capacity and compressor power for the three units described in Table 1. ACMODEL is extremely modular and uses separate subroutines to model each of the components of a packaged air conditioner or heat pump. For this project, the experimental correlations and fouling factors of the different filter-coil combinations were incorporated into the program to replace the original built-in correlations.

Table 1: Descriptions of the equipment units

	3-ton rooftop	5-ton rooftop	35-ton rooftop
Refrigerant		R22	
Compressor	Reciprocating*1	Scroll*1	Scroll*2
Exp. Device		Thermal Exp. Valve	
Condenser rows	1	2	4
Evaporator rows	2	4	8
Evap: tube diameter (mm)	9.53 (0.38")	12.7 (0.50")	12.7 (0.50")
Evap: tube thickness (mm)	0.65 (0.0256")	0.65 (0.0256")	0.55 (0.0217")
Evap: fin density (fin/cm)	5.51(14 fin/inch)	4.72 (12 fin/inch)	3.15 (8 fin/inch)
Evap: fin thickness (mm)	0.19 (0.0075")	0.114(0.0045")	0.15(0.0059")
Evap: face area (m^2)	0.372 (4 ft^2)	0.372 (4 ft^2)	1.858 (20 ft^2)
Evaptor fan norminal (m^3/s)	0.57(1200 Cfm)	0.94(2000 Cfm)	4.72(10000 Cfm)

For each coil, the clean case was simulated within an ambient temperature range (27 °C -45 °C (81 °F –113 °F)) to obtain cooling capacity and compressor power at the nominal air flow rate. Then, the six fouling cases (five filter-cases and one no-filter-case) were simulated successively and the capacity and compressor power at fouled conditions were predicted. In these cases, the air flow rate decreased somewhat from the nominal flow rate and was determined from the fan curve and system pressure drop. In the next step, the evaporator-side fan power and condenser-side fan power was calculated. The condenser was assumed to be clean and the fan operated with a fixed air flow rate and thus, the condenser fan power was a constant. The condition was more complicated for the evaporator-side fan because the evaporator air-side pressure drop changed after fouling. The following describes the process to determine the evaporator-side air flow and fan power.

(1) System pressure drop and fan power at clean conditions

The system static pressure drop included three parts: filter pressure drop, coil air-side pressure drop and additional pressure drop in the air distribution system as:

$$\Delta P_{sys,c} = \Delta P_{f,c} + \Delta P_{c,c} + \Delta P_{dist} \quad (1)$$

where:

$$\Delta P_{f,c} = e_c V_c^{g_c} \quad (2)$$

$$\Delta P_{c,c} = a_c V_c^{b_c} \quad (3)$$

V_c is the air velocity at clean conditions and the coefficients a_c, b_c, e_c, g_c were determined from experiments. The additional system pressure drop includes friction losses that occur along the entire air duct length and within fittings. This distribution pressure loss would depend on the installation, but would also depend on the square of the velocity according to:

$$\Delta P_{dist} = K V_c^2 \quad (4)$$

ΔP_{dist} was assumed not to be impacted by fouling. For the modeling presented here, the value of ΔP_{dist} was set at the design air velocity so that $\Delta P_{sys,c}$ was determined and the fan ran at the its peak efficiency η_c for the highest filter tested (MERV 14 for the 2 and 8-row coils and MERV8 for the 4-row coil). ΔP_{dist} was then fixed for all other filter cases, so for those filter cases, the fan efficiency was not at the peak value. The factor K was obtained from ΔP_{dist} and V_c.

Corresponding to the design air velocity and system pressure drop at clean conditions, the required fan speed was then determined according to the performance data provided by the manufacturer. Fan power $W_{fe,c}$ was also determined with performance data. Figure 2 shows a diagram of the system pressure drop and fan curve for both clean and fouled conditions. The fan operates at the intersection of the fan and system characteristic. As the coil and filter foul, the pressure drop increases and the air flow is reduced.

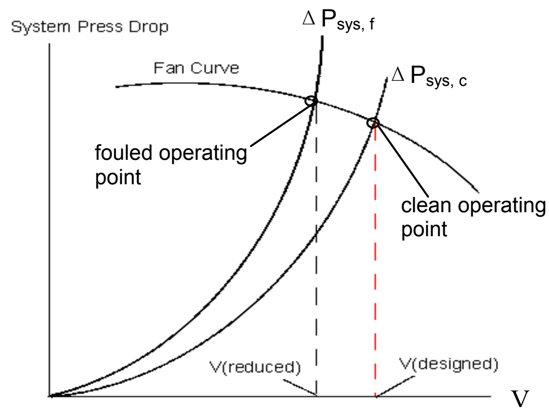

Figure 2: System pressure drop and fan curve

(2) System pressure drop and fan power at fouled conditions
After fouling, the system pressure drop increases while the fan speed does not change. Therefore, the new system pressure drop line intersects the original fan curve at a lower air velocity V_f (reduced) and higher system pressure drop $\Delta P_{sys,f}$.

The system static pressure drop at fouled conditions was determined with:

$$\Delta P_{sys,f} = \Delta P_{f,f} + \Delta P_{c,f} + \Delta P_{dist} \tag{5}$$

where:

$$\Delta P_{f,f} = e_f V_f^{\,g_f} \tag{6}$$

$$\Delta P_{c,f} = a_f V_f^{\,b_f} \tag{7}$$

$$\Delta P_{dist} = K V_f^{\,2} \tag{8}$$

Here, V_f is the reduced air velocity and factors e_f, g_f, a_f, b_f were determined from experiments.

Fan power ($W_{fe,f}$) was obtained from the performance data with a known $\Delta P_{sys,f}$ and reduced air velocity. Fan efficiency changed from η_c to η_f. $W_{fe,f}$ could increase or decrease compared to $W_{fe,c}$ according to the following equation:

$$W_{fe,c(f)} = \frac{\dot{V}_{c(f)} \Delta P_{sys,c(f)}}{\eta_{c(f)}} \tag{9}$$

where $\dot{V}_{c(f)}$ is air volumetric flow rate (m^3/s) at clean or fouled conditions.

$\dot{V}$ dropped after fouling while ΔP_{sys} increased. η decreased if η_c was set at the peak value but it was possible to increase if η_c was not at the peak to begin with. Therefore, as a result of the tradeoffs between these factors, W_{fe} could increase or decrease after fouling, but it generally increased in this study.

The final step was to compute EER for the equipment. EER is the difference of cooling capacity and evaporator-side fan power in Btu per hour divided by the power input in Watts at rating conditions, expressed in Btu/hr per Watt:

$$EER = \frac{Q_c - w_{fe}}{w_c + w_{fe} + w_{fc}} \times 3.412 \tag{10}$$

It was found that W_{fe} had a very critical effect on EER. At a typical fan efficiency of 30% to 40%, W_{fe} was around 13% to 37% of the total power.

4. FOULING IMPACT ON EQUIPMENT CAPACITY AND EER

4.1 Capacity Impact

Fouling affects equipment cooling capacity in two ways: through reduction in heat transfer coefficient and through reduction in air flow. It was found from the experimental tests that the heat transfer coefficient could actually increase with limited fouling. Even with significant fouling, the degradation in heat transfer coefficient was small. Furthermore, this impact was small compared to the impact on coil air-side pressure drop. As a result of the increase of coil and filter pressure drop, the air flow rate decreases so that the total cooling capacity decreases as well.

For one year's dust loading (600 grams of dust), the degradation in cooling capacity after fouling was not very significant. The decrease in cooling capacity was approximately 2% to 4%, 2% to 3%, 5% to 7%, and 4% to 5% for the 35-ton (HX8L), 35-ton (HX8W), 5-ton (HX4L) and 3-ton (HX2L) units, respectively. The effect is greater for smaller systems that employ shallower coils with higher fin densities. Figure 3 shows the degradation for all filter cases for the 3-ton (HX2L) unit as an example. The capacity ratio (capacity divided by the capacity of the no-filter case at clean conditions) was used as the vertical axis. Since for all cases at clean conditions the air flow rate was set at the design value (1.52 m/s for 3-ton unit) and the coil had the same performance, all clean cases had the same cooling capacity ratios.

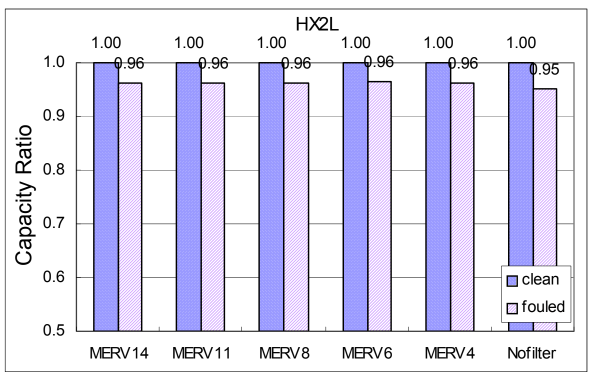

Figure 3: Capacity ratio of all coil-filter cases for HX2L (T_{ci}=35^o C)

4.2 EER Impact

EER values for all cases (for different coil-filter combinations and at clean and fouled conditions) were obtained. For each coil, all EER values were compared with EERs for no-filter clean cases so that EER ratios were obtained and are presented in Figures 4 through 7.

From the results in the figures, it was concluded that:

(1) The clean equipment without an upstream filter worked at the highest EERs for all coil cases.
(2) The equipment with the highest MERV filters got the lowest EERs for all clean coil cases.
(3) The EER decreased significantly after fouling. For the 35-ton units (HX8L and HX8W), the degradations ranged from 2% to 10% (2% for no-filter case); for the 5-ton unit, the degradation ranged from 3% to 8% (8% for no-filter case); for the 3-ton unit, the degradation ranged from 6% to 10% (10% for no-filter case). It was found that the 3-ton unit had the greatest degradation and the 35-ton units had the smallest degradation with fouling. In addition, for the larger equipment, the impact of fouling was greater for the high efficiency filter, whereas the opposite was true for the smaller units. This is due to that fact that fouling has a greater impact on shallower designs.
(4) The greatest impact of filter choice on EER occurred for the larger capacity equipment. This was caused by different fan performance: the 35-ton unit had a steep fan efficiency curve so that EER varied greatly with system pressure drop, which was influenced by both filter choice and fouling. The 3-ton unit had a more flat

fan efficiency curve. The differences would be eliminated if these units had similar fan performance characteristics. The influence of fan curves will be discussed in the following section.

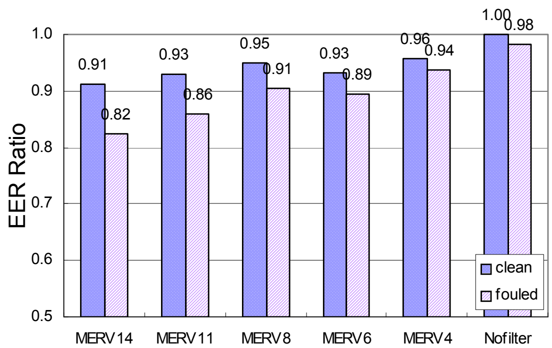

Figure 4: EER ratio of all coil-filter cases for HX8L (T_{ci}=35^{o} C)

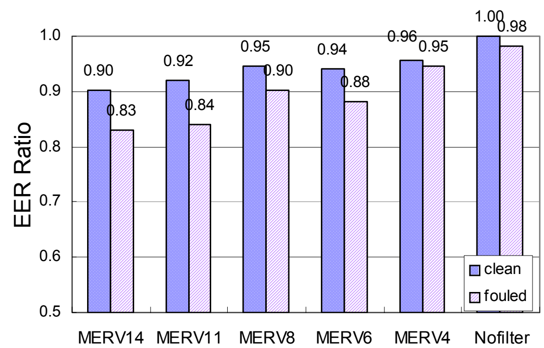

Figure 5: EER ratio of all coil-filter cases for HX8W (T_{ci}=35^{o} C)

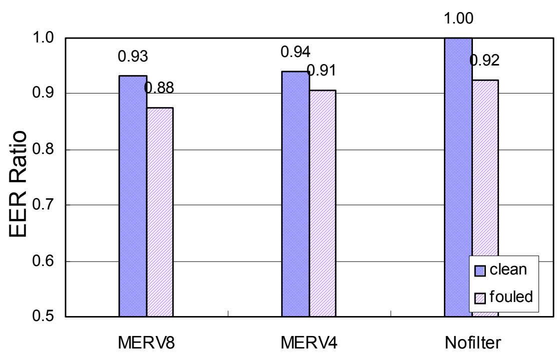

Figure 6: EER ratio of all coil-filter cases for HX4L (T_{ci}=35^{o} C)

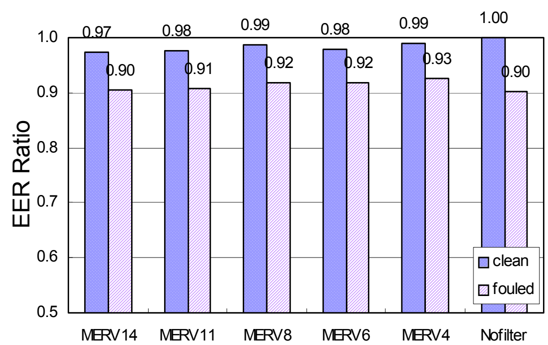

Figure 7: EER ratio of all coil-filter cases for HX2L (T_{ci}=35^{o} C)

4.3 Influence of Fan Efficiency Curve on EER

For the previous results, the evaporator side fan power W_{fe} was approximately 13% to 37% of the total power with fan efficiencies ranging from about 30% to 40%. Therefore, the results are sensitive to the fan curve and fan efficiency. To study the influence of fan efficiency, two cases were considered:

(1) The fan efficiency was assumed to not change with air flow rate and pressure drop: 38% for 35-ton (HX8L and HX8W) units, 28% for 5-ton (HX4L) unit and 29% for 3-ton (HX2L) unit. These efficiencies were the peak values for the actual fan curves.
(2) The fan efficiency was assumed to be 100% and did not change with air flow rate for all cases.

Figures 8-11 show the EER ratios for HX8L and HX2L for these two cases. Compared to the actual fan curves, the penalty associated with using high efficiency filters was reduced. This was particularly true for the higher capacity unit. Given a perfect fan curve (η_{fe}=100%), the differences among all filter-cases were much smaller than for the original analysis with actual fan curves.

The absolute value of EER increased for these two cases compared to the EER with the actual fan efficiency. For the first case, EER increased slightly, but for the second case, EER increased approximately 11% and 27% for 3-ton and 35-ton units respectively. It indicates that EER values increase significantly with fan efficiencies. With higher efficiency fans, the energy penalty associated with high efficiency filters was reduced more considerably than that for low efficiency filters.

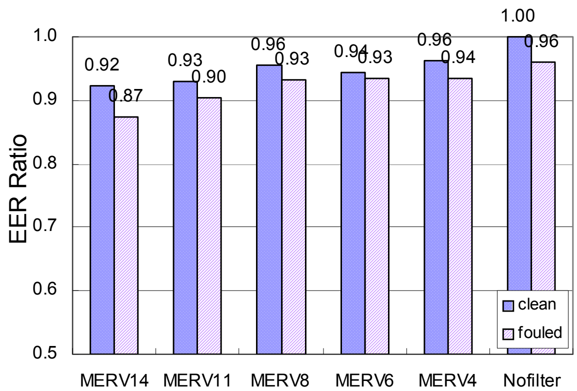

Figure 8: EER ratio of all coil-filter cases for HX8L with constant fan efficiencies of 38% (T_{ci}=35° C)

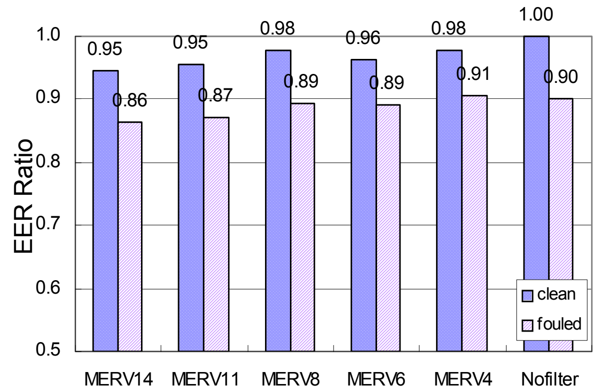

Figure 9: EER ratio of all coil-filter cases for HX2L with constant fan efficiencies of 29% (T_{ci}=35° C)

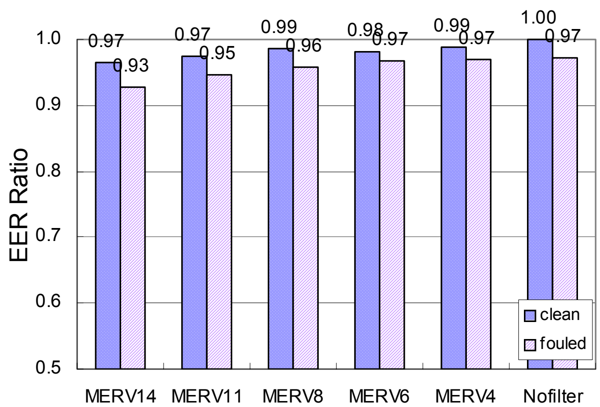

Figure 10: EER ratio of all coil-filter cases for HX8L with constant fan efficiencies of 100% (T_{ci}=35° C)

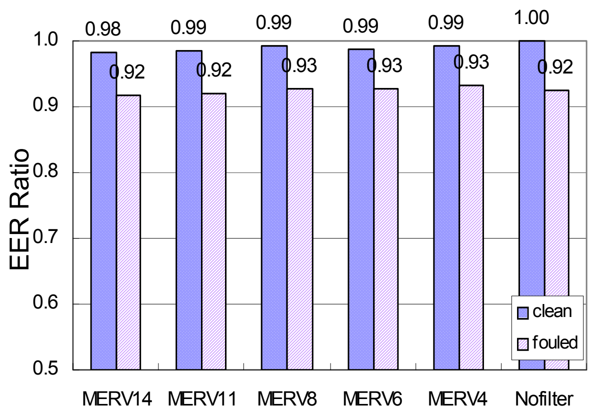

Figure 11: EER ratio of all coil-filter cases for HX2L with constant fan efficiencies of 100% (T_{ci}=35° C)

5. CONCLUSIONS AND RECOMMENDATIONS

The following significant conclusions were obtained from the equipment modeling and test results.

(1) Large equipment seems to be affected less by fouling than small equipment.
(2) Fouling decreases equipment cooling capacity because of reduced air flow rate. For a given air velocity, the heat transfer coefficient could decrease with fouling. However the decrease in air flow is a more significant effect. An average decrease of 8% of the air flow rate was determined from the simulations and test results.
(3) The impact of filter choice on cooling capacity is relatively small. However, using high efficiency filters results in significantly higher EER penalties for fouling especially for large equipment. For a 35-ton unit, the EER values decreased by 8%-10% after fouling for MERV14 cases, but by 1%-2% for MERV4 and no-filter cases. This was due to an increase in pressure drop and the influence of fan efficiency.
(4) Fouling affects evaporator-side fan power which in turn affects the equipment EER significantly. Given actual fan curves with an efficiency of approximately 30%, the fan power was 13%-37% of the total power. Comparing the fan power for fouled conditions to the fan power for clean conditions, the variation ranged from approximately –7% to a value as high as 40% in one case. The actual fan efficiency was low (approximately 30%) and varied with air flow rate. The effect of fan efficiency was considered through simulation. The energy penalty associated with high efficiency filters was reduced considerably with higher efficiency fans.
(5) Equipment with high efficiency upstream filters has lower EERs than equipment with low efficiency filters. This is because of increased pressure drop.

(6) Equipment with high MERV upstream filters will provide significantly better air quality. For HX8L, the quantity of dust passing through the coil with a MERV4 filter was approximately 30 times the dust passing the coil with a MERV14 filter. Without an upstream filter, the quantity of dust passing through the coil was approximately 60 times the value for a MERV14 filter.

Only one year's dust loading (600 grams) was considered in this project. At this level, the fouling impacts on coil performance are relatively low. For 35-ton units (HX8L and HX8W), the EER was highest without any upstream filter at both clean and fouled conditions compared to any filter case. Therefore, further study with more dust loading is recommended. Furthermore, in this study, ASHRAE standard dust was used which was made of coarse particles (with sizes of 6 μm and up). Smaller particles can be employed in the future work so that the whole size range of ambient particles is considered.

NOMENCLATURE

a	coil pressure drop factor, $Pa \cdot s^{b_{c(f)}} / m^{b_{c(f)}}$
b	coil pressure drop exponent
e	filter pressure drop factor, $Pa \cdot s^{g_{c(f)}} / m^{g_{c(f)}}$
g	filter pressure drop exponent
K	air duct pressure drop factor, $Pa{\cdot}s^2/m^2$
Q_c	equipment cooling capacity, W
T_{ci}	condenser side air inlet temperature, ºC
V	air velocity, m/s
$\dot{V}$	air volumetric flow rate, m^3/s
W	power, W
W_c	compressor power, W
ΔP	pressure drop, Pa
η	fan efficiency
Subscripts	
c	clean
dist	air distribution system
f	fouling
fe	evaporator side fan
fc	condenser side fan
sys	system

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