Building and Environment 73 (2014) 151–161

Contents lists available at ScienceDirect

Building and Environment

journal homepage: www.elsevier.com/locate/buildenv

The relationship between filter pressure drop, indoor air quality, and energy consumption in rooftop HVAC units

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ARTICLE INFO

Article history:
Received 3 October 2013
Received in revised form
16 December 2013
Accepted 18 December 2013

Keywords:
Filtration
Packaged terminal air conditioner (PTAC)
HVAC energy use
Clean air delivery rate
Commercial buildings

ABSTRACT

HVAC filters are commonly used to decrease exposure to particulate matter, yet little is known about the energy impacts and air quality consequences of high efficiency filters installed in commercial buildings. To explore these effects, system airflow, filter and coil pressure drop, fan pressure rise, and power draw were measured, and cooling capacity and compressor power were modeled for at least four filter pressure drops in 15 rooftop units equipped with and without fan speed control. Energy implications and clean-air-delivery-rate were estimated for a large dataset of filters divided into four efficiency (MERV) categories. Field measurements conducted on units without fan speed control showed that increased filter pressure drop decreased flow, cooling capacity, and power. For a unit with fan speed control, the same increase in pressure drop resulted in the same magnitude change of fan power but in the opposite direction, and other parameters were unchanged. Replacing MERV 8 with MERV 13/14 resulted in higher energy consumption (2–4%) during cooling mode for both unit types, energy savings during fan-only mode (8–13%) in units without fan speed control, and increased energy consumption in fan-only mode (11–18%) in the unit with fan speed control. Energy consumption increases were offset by improvement in clean-air-delivery-rate, especially for $PM_{2.5}$ (2.9–3.8 times increase going from MERV 8 to MERV 13/14), with larger benefits achieved for the unit with fan speed control. A comprehensive understanding of the impact of filtration is essential to selecting the appropriate efficiency of filters that ensures low-energy use and a healthy indoor environment.

1. Introduction

Particles are an important pollutant of concern in commercial buildings due to their significant health effects. To decrease particles of outdoor and indoor origin, the use of high-efficiency heating ventilation and air-conditioning (HVAC) filters is often recommended as an alternative to supplying additional ventilation because such filters (1) can lower particle matter (PM) concentrations in a less energy intensive way and (2) are effective even when outdoor concentrations of PM are high. However, high-efficiency HVAC filters generally have a higher-pressure drop and are presumed to have large energy penalties. This is particularly important because fan energy necessary to move air throughout commercial buildings equipped with rooftop units accounts for at least 7% of total site energy consumption (261 trillion BTUs annually; Refs. [1,2]). As energy conservation becomes essential, building designers and operators need to understand the operational power requirements and the underlying benefits in improving indoor air quality through the use of high efficiency filters.

Quantifying the relationship between filter pressure drop, indoor air quality, and energy cost of air filters has been the focus of a number of studies. Field, lab, and computer simulation work have mostly focused on small systems (<30 kW) with no fan speed control [3–11]. Only one recent field study tested small systems equipped with and without fan speed control in residences [12]. Three small-scale studies of large commercial systems investigated the effect of reducing the pressure drop across filters on energy consumption [11,13,14]. Of those studies, however, only Lam et al. [14] performed field measurements. They conducted a field study on a 40-floor building with a centralized HVAC system; however, the results from this study are difficult to apply to most of buildings in U.S. that are smaller, because the filter pressure drop on the studied 40-floor building is negligible when compared to the high static pressure drop of the entire supply ducting system of this building.

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http://dx.doi.org/10.1016/j.buildenv.2013.12.010

Other studies have focused on determining an approximate cost of filter operation (life cost analysis) by applying a filter model [15–23]. In these studies, the filter models used assumed the airflow rate through the filter to be constant over the life of the filter, and therefore the impact on power for systems without fan speed control was not captured. In addition, the actual system operating point of the fan and duct curves and the resulting system efficiency can significantly impact the fan power consumption. Also, the fan efficiency reported in the literature varies widely, with values that range from 22 to 49% for residential and light-commercial buildings (e.g., Ref. [8]), and 30–80% for modeled commercial systems (e.g., Refs. [6,13,15,17]).

There is a clear lack of measured data and analysis of energy and indoor air quality consequences of filters used in big commercial systems (rated cooling capacity >30 kW) equipped with or without fan speed control. The purpose of this paper is to (1) quantify the relationship between filter pressure drop and fan pressure, airflow rate, power, and efficiency through fieldwork conducted on rooftop units equipped with and without speed control; (2) estimate the impact of filter pressure drop on cooling capacity and compressor power through modeling the vapor compression cycle; and (3) compare the energy and air quality performance of different filtration efficiencies in these systems.

2. Methodology

2.1. Data collection/simulation: impact of pressure drop on unit performance

The sample of buildings for which field data were collected includes 14 rooftop units (RTUs) equipped with no fan speed control and one unit equipped with fan speed control. These units were installed in big box retail stores in Austin, Texas that were tested as part of ASHRAE RP-1596 [24]. Cooling capacity ratings on these units ranged from 30 to 84 kW. For each sampled system, system airflow, filter and coil pressure drop, fan pressure rise, and power draw were measured for at least four different filter pressure drops (induced by blocking airflow at the filter). Duct leakage impacts were excluded because HVAC distribution systems in retail stores typically have almost no ducts and instead deliver conditioned air directly to the space.

System airflow rates were measured with an Energy Conservatory TrueFlow metering plate and DG-700 digital manometer (with uncertainty of ±7% of measured value) for the baseline pressure drop. Changes in supply static pressure for different filter pressure drops were measured and, following the calculation procedure in the instrument manual, corrections were made based on changes in the supply plenum static pressure. Also, for baseline and each elevated pressured drop, filter and coil pressure drop and fan pressure rise were measured using a DG-700 manometer (±1% measurement uncertainty). An Onset HOBO Energy Logger was used to record the power draw of the air handler fan measured by a Continental Control Systems (CCS) Wattnode AC true power meter for approximately 30 min at 10-s intervals (±3% measurement uncertainty). The Energy Logger box was connected to pressure taps, voltage taps, and 0–20 amp CCS current transducers. For the unit equipped with fan speed control, fan speed was adjusted manually using the variable frequency drive installed on the unit to maintain approximately the same airflow rate delivered by the fan for each measured filter pressure drop. Using the pressure rise across the fan, the airflow rate, and fan power, the fan efficiency was calculated for different pressure drops.

The impact of filter pressure drop on cooling capacity, sensible heat ratio, and compressor power were modeled using two vapor compression models: the ACHP vapor compression system model [25] and Secondary HVAC Toolkit software [26]. Two additional parameters, outdoor temperature and relative humidity, were also measured during fieldwork and were used as inputs for the model. Other inputs include geometry of coils, refrigerant type (obtained from manufacturer specifications), and fan power and airflow rate (obtained from the collected dataset). Comparison of the modeling results of the two models showed very similar results, and in this paper the results from the ACHP vapor compression system model are reported.

To capture and analyze the effect of different filter pressure drops on system runtime and the total power draw, the collected and simulated data were used to calculate the change in length of the cooling cycle (i.e., duty cycle) and the energy efficiency ratio (EER). The change in duty cycle is used to see how much the system will run longer to achieve the same cooling capacity when the system airflow is decreased relative to the baseline case; the total power draw is then multiplied by the additional length cycle. Also, the energy efficiency ratio (EER) was calculated to assess the change in the net cooling capacity (difference between modeled cooling capacity and measured evaporator-side fan power) divided by the total power input (measured fan power, modeled compressor and condenser power) relative to the base case.

2.2. Energy and indoor air quality comparison of different efficiency filters

The established relationships between (1) filter pressure drop and (2) system parameters (including: airflow, fan speed, fan power, cooling capacity, compressor power, and duty cycle) study results were presented in the context of comparing the energy and particle reduction performance of different filters. Data from sixty filters obtained from 15 different manufacturers and an additional 15 filters provided by Rivers and Murphy [27] was investigated. For each filter, filtration efficiency data was obtained. Then, these filters were classified with an ASHRAE Standard 52.2 [28] Minimum Efficiency Reporting Value (MERV) and the sample consisted of MERV 8 ($n = 16$), MERV 11 ($n = 21$), MERV 13 ($n = 24$), MERV 14 ($n = 14$) filters. It should be noted that Rivers and Murphy conducted filter traverse tests in 1996 and thus did not necessarily do full test according to Standard 52.2 [28]. Accordingly, the MERV values that would results from a Standard 52.2 test may be slightly different. Besides conducting tests at clean conditions, Rivers and Murphy [27] provided an extra set of ASHRAE tests for laboratory-fouled filters that were loaded with a standard test dust. In all cases, fractional efficiency (η) for three particle size bins (E_1: 0.3 μm–1 μm; E_2: 1 μm–3 μm; E_3: 3 μm–10 μm) was reported by filter manufacturers or in the case of Rivers and Murphy it was calculated by averaging efficiencies of particle sizes within each corresponding size range group. Filter efficiencies were typically measured at a face velocity of 2.5 m s^{-1}, which is generally higher than the velocities considered in this work and typically found in rooftop HVAC units. Based on findings from Rivers and Murphy [27] and Hanley et al. [29] on the relationship between velocities and filter efficiency for particle sizes larger than 0.3 μm, the impact of velocity on filter efficiency was determined to be minimal and the filtration efficiency at 2.5 m s^{-1} was used with no further adjustment.

Filter efficiency for integrated particle sizes $PM_{2.5}$ and PM_{10} was calculated using Equation (1); where the efficiency for each particle size (η_i) was combined with indoor ($N_{\mathrm{in},i}$) and outdoor ($N_{\mathrm{out},i}$) particle size distribution, particle geometric mean (GM_i), and outdoor air fraction (OA); these parameters were all measured in ASHRAE RP-1596 [24]. In the present work, particle density was assumed to be 1 g cm^{-3} for all particle sizes.