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IMPACT OF EXTERNAL STATIC PRESSURE ON RESIDENTIAL HEATING AND COOLING ENERGY USE IN HOT CLIMATES

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ABSTRACT

Extensive field measurements indicate that the external static pressures (ESPs) in residential air distribution systems are well above the requirements in equipment rating and testing standards. An excess ESP directly affects blower airflow rates and power draws. In addition, it has indirect impacts on system efficiencies, capacities, and operating hours because of the ESP effects on the system airflow rates. Hence, the objective of this study is to investigate how ESPs affect the residential heating and cooling energy use in hot climates. Experimental results on the aerodynamic and cooling performance of unitary equipment were integrated with a public-domain building simulation model to estimate the annual heating and cooling energy use at different ESP settings.

INTRODUCTION

Blowers in air conditioners and furnaces are used to circulate cooled or heated air through duct systems. Their performance is heavily affected by the external static pressure (ESP), which is the total flow resistance from ductwork components, including ducts, fittings, filters, supply registers, and return grilles. Figure 1 shows typical static pressure changes and gradients referenced to the atmospheric pressure in an air handling unit (AHU) and its adjacent ducts.

National standards specify ESP settings based on unit capacities when testing and rating residential forced-air heating and cooling equipment. For example, unitary air-conditioning and air-source heat pump equipment up to 65,000 Btu/h (19,050 W) are tested using an ESP in the range of 0.1 to 0.2 in. w.g. (25 to 50 Pa) in accordance with the Air-Conditioning, Heating and Refrigeration Institute (AHRI) Standard 210/240 (AHRI 2008). Also, gas furnaces up to 300,000 Btu/hr (87,921 W) are rated with an ESP in the range of 0.18 to 0.33 in. w.g. (45 to 82 Pa) following the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) Standard 103 (ASHRAE 2007a). In contrast to what is specified in standards, extensive field measurements have shown that many residential systems operate at much higher ESPs. Results from field surveys, including over 800 field measurements from 22 studies, indicate that ESPs varied from 0.31 to 1.12 in. w.g. (77 to 279 Pa) in installed conditions with a weighted average of 0.5 in. w.g. (125 Pa) (DOE 2012). The field measured ESPs are well above the pressure ranges used by AHRI Standard 210/240 (AHRI 2008) and ASHRAE Standard 103 (ASHRAE 2007a) for equipment testing and rating.

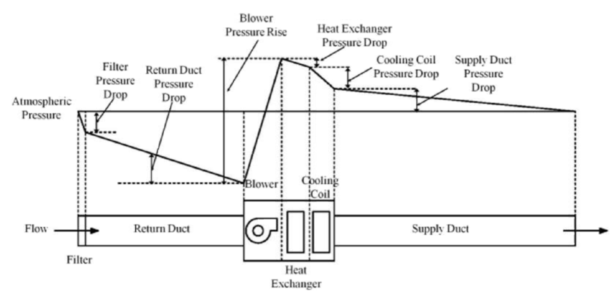

Figure 1 Theoretical static pressure changes for a typical residential system

Excess ESPs have significant energy impacts on residential heating and cooling systems. It directly affects blower power draws and airflow rates, especially for systems with electrically commutated motors (ECMs). Walker (2008) reported that the power draw of an ECM blower increased with increasing ESPs. For example, it was shown that the power draw doubled as the ESP was increased from 0.2 to 0.8 in. w.g. (50 to 200 Pa). In another study, Franco et al. (2008) estimated the energy consumption of an ECM blower in both heating and cooling seasons by integrating the experimental data with building energy simulations. The results indicated that the annual electricity consumption for the ECM blower increased from 361 to 523 kWh, which was a 45% increase, as a result of increasing ESPs from 0.22 to 0.8 in. w.g. (55 to 200 Pa).

Of special importance, the increased blower power draw from an excess ESP imposes an additional cooling load on a system (Kendall 2004), because the generated heat from an ECM motor is proportional to its power draw. This heat is directly released into the cooled air in

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an air conditioner so that it offsets the sensible capacity by reducing the temperature difference between the supply and return air. Capturing this effect is of special importance to the residential air conditioning systems that are controlled by thermostats due to the fact that thermostats can only sense the sensible load. Furthermore, for a given sensible load, system runtime will increase with reduced sensible capacity. Because the power draw of a condensing unit is much higher than the power draw of a blower, even a small increase in system runtime may offset any energy savings from ECM blowers.

Several studies have tried to quantify the energy impact of excess ESP for the purpose of ductwork design and optimization (Walker 2008, Franco et al. 2008, Lutz et al. 2006). However, these studies have been largely limited to blower energy use only. The energy consumed by non-blower components, such as electricity used by condensing units in cooling seasons or the fuel/electricity consumed by furnaces in heating seasons, takes up to 80-95% of the energy use in residential heating/cooling systems (Stephens et al. 2010, Parker et al. 2005) and has not been thoroughly investigated.

The objective of the study reported herein is to quantify the impact of ESPs on residential heating and cooling energy consumptions in hot climates. Both experimental and simulation approaches were used to achieve this research goal. Specifically, the aerodynamic and cooling performance of a unitary unit with gas heating and electric cooling was experimentally evaluated at various ESPs and airflow rates. These laboratory test results were then integrated with a public-domain building energy simulation model to estimate the annual heating and cooling energy use associated with four different duct types: low, medium low, medium high, and high pressure drop ducts.

EXPERIMENTAL STUDY RESULTS

The purpose of the experimental study was to characterize the airflow and power performance of the blower. It also provided performance curves to describe the relationships between evaporator airflow rates and cooling capacities. The unit tested in this study was a packaged air-conditioner and gas-furnace system. The nominal heating and cooling capacities were 73,000 Btu/hr (21,400 W) and 57,000 Btu/hr (16,700 W), respectively. The indoor blower was directly driven by a 1-hp (745 W) ECM motor with a nominal airflow rate of 1750 ft^3/min (0.83 m^3/s).

Well-instrumented laboratory measurements were taken on the aerodynamic and cooling performance of the unit. In the aerodynamic tests, the airflow and power draw of the blower were measured over an ESP range from 0.1 to 1.1 in. w.g. (25 to 275 Pa) in accordance with ASHRAE Standard 51 (ASHRAE 2007b). In the cooling performance tests, cooling capacities and power consumptions were determined as the airflow rate was varied from 2250 to 1000 ft^3/min (1.06 to 0.47 m^3/s) by following the A-test conditions specified in AHRI Standard 210/240 (AHRI 2008). Detailed descriptions of the experimental setups and test procedures can be found in Yin et al. (2014).

Figures 2 and 3 show the airflow rate and power draw of the ECM blower over an ESP range from 0.1 to 1.1 in. w.g. (25 to 275 Pa). The airflow rate continuously decreased in response to increasing ESPs. For the cooling speed mode, a 15% airflow reduction occurred when the ESP was increased from 0.2 to 0.7 in. w.g. (50 to 175 Pa). Similar trends were observed in the heating speed mode but to a lesser degree. As a result of increasing ESPs from 0.1 to 1.1 in. w.g. (25 to 275 Pa), the power draw of the blower in the heating mode continuously increased from 459 to 549 W. In the cooling mode, the power draw first increased with increasing ESPs. After reaching its peak power of 745 W at 0.6 in. w.g. (150 Pa), the power draw gradually decreased to 593 W at 1.1 in. w.g. (275 Pa).

The gross cooling capacity and condensing unit power draw obtained at various airflow rates were normalized against the results at the nominal airflow rate. These normalized results were plotted against the flow fraction (FF), which was the ratio of the actual airflow rate to the rated airflow rate. Figures 4 and 5 show the normalized gross cooling capacity and condensing unit power draw as functions of FF. Two performance curves were generated by fitting the normalized data into a second order polynomial (Equation 1) with FF as the only input variable. The estimated coefficients and R^2 values for the developed models are shown in Table 1.

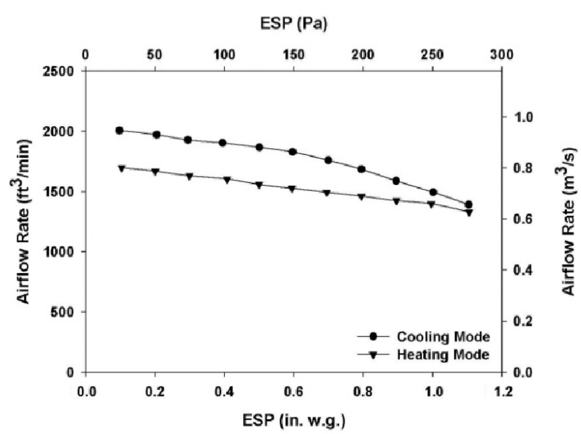

Figure 2 Airflow performance of the ECM blower

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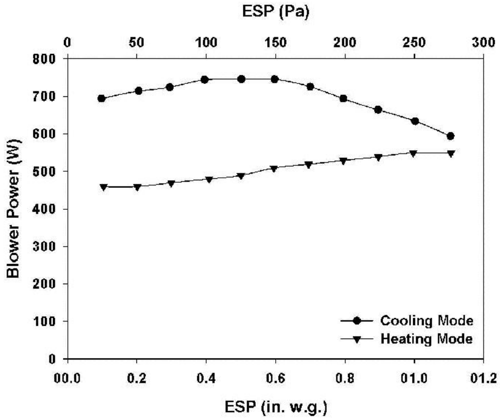

Figure 3 Power performance of the ECM blower

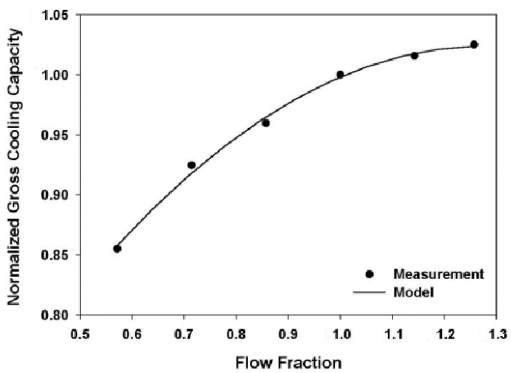

Figure 4 Normalized gross cooling capacity as a function of FF

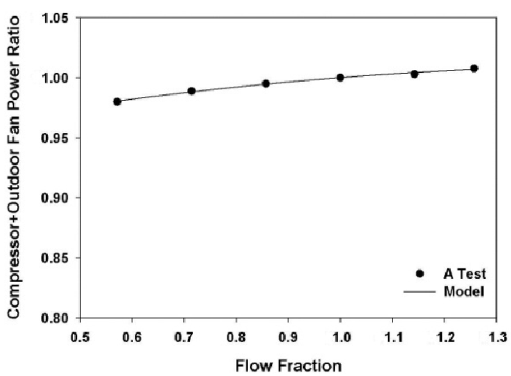

Figure 5. Normalized outdoor fan/compressor power draw as a function of FF

$$\text{Function} = C1+C2\times FF+C3\times FF^2 \tag{1}$$

BUILDING ENERGY SIMULATION RESULTS

A model of a single story residential house was constructed, as shown in Figure 6. It has one conditioned zone with dimensions of 35.5 ft ×84.5 ft ×8 ft (10.8 m ×25.8 m ×2.4 m) and two unconditioned zones, namely the attic and garage. The building envelop was designed compliant with the requirements in ASHRAE Standard 90.2 (2007c). The window to wall area ratio is 25%. The overall daily internal heat gains from lights, people, and equipment were determined from ASHRAE Standard 90.2 (2007c) by using Equations (2) and (3). Only one living unit was assumed in the designed model. The overall internal heat gains were calculated to be 78,000 Btu/day (22.8 kWh/day), corresponding to 3250 Btu/hr (952.5 W). An effective leakage area (ELA) of 1.55 ft^2 (0.14 m^2) was estimated from the regression analysis reported by Chan et al. (2012). In addition, a constant infiltration of 0.15 ach was added to account for the occupancy-caused infiltration through door openings, exhaust fans, etc.

$$\begin{aligned}\text{Sensible Heat Gains} &= (\text{Conditioned Zone Floor Area} \times 15\ \text{Btu/day}\cdot\text{ft}^2) \\ &+ (\text{Number of Living Units} \times 20000\ \text{Btu/day})\end{aligned} \tag{2}$$

$$\text{Latent Heat Gains} = 0.2\times\text{Sensible Heat Gains} \tag{3}$$

Load calculations were conducted to verify the system selection. The 99% heating dry-bulb temperature and the 1% cooling dry-bulb with mean coincident wet-bulb temperature in College Station, Texas were used for heating and cooling load calculations. The thermostat was set at 78°F (25.6°C) for cooling and 68°F (20°C) for heating. A heating oversizing factor of 1.7 and a cooling oversizing factor of 1.1 (Franco et al. 2008) were applied to the calculated loads, which resulted in a heating load of 73,700 Btu/hr (21,599 W) and a cooling load of 53,300 Btu/hr (15,621 W). The estimated annual heating load was 17.5 MMBtu (5125 kWh), and the estimated annual total cooling load was 72.6 MMBtu (21,292 kWh).

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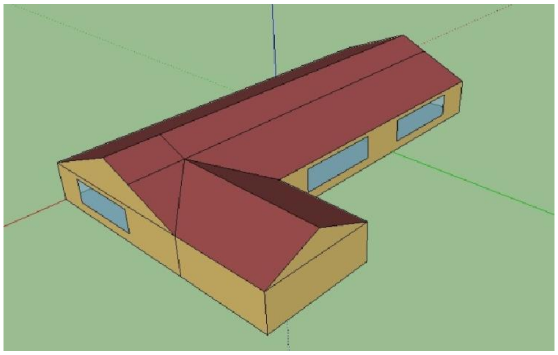

Figure 6 The developed model of a single story residential house

Four duct pressure drop curves were generated for the representation of ducts with low, medium low, medium high, and high flow resistance. The operating points under each scenario were determined from the intersections of pressure drop and blower airflow curves, as displayed in Figure 7. At each point, the airflow rate and blower power draw were interpolated from the experimental data. The corresponding heating capacity was determined by using the actual airflow rate and the interpolated heating temperature rise from the tabulated catalog data. The corresponding cooling performance was estimated by using the rated performance data and five performance curves. Two of the curves, namely the total cooling capacity and energy input ratio as functions of FF, were determined from the experimental data. The total cooling capacity and energy input ratio as functions of temperatures were developed by using manufacturer's catalog data. The default setting was used for the curve of part load ratio. Table 2 lists the ESP, airflow rate, blower power draw, and heating capacity at each operating point. Blower standby power and furnace inducer fan power were assumed to be 9 W and 75 W (Franco et al. 2008).

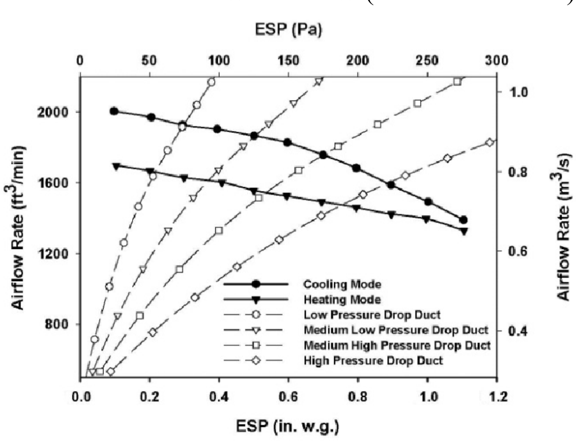

Figure 7 System operating points

RESULTS SUMMARY

The performance data at each operating point was integrated with the established building model to simulate a residential house with a duct system of low, medium low, medium high, and high pressure drops. A total of four simulations were conducted by using public-domain building energy simulation software (DOE 2013) with the TMY3 weather file of College Station, Texas. In each simulation, the energy consumption of the heating and cooling equipment over the whole year was predicted. The energy impact of ESPs was characterized in terms of the system electricity and natural gas consumptions. Key parameters reported from the simulations included the electricity consumptions of the blower and condensing unit, furnace gas consumption, and system operating hours.

Table 3 summarizes the annual blower electricity consumption and operating hours with different ducts of low, medium low, medium high, and high flow resistance. For the College Station house, the blower electricity used in the cooling mode used about 81% of the total blower electricity consumption. The heating mode and standby consumptions were about 13% and 6%, respectively. As a result of increasing ESPs, the blower electricity consumption in the heating mode increased from 145.5 to 160.3 kWh, accounting for a 10.2% increase. The consumption in the cooling mode first increased from 903.3 to 922.1 kWh, and then decreased to 884.9 kWh. This pattern in the cooling mode electricity consumption was consistent with the trend in the blower power draw revealed from the experimental data in Figure 3. The total operating hours increased by 5% with the increase in ESPs, leading to a 1% decrease in the standby electricity consumption.

Table 4 lists the annual heating and cooling equipment operating hours under four different duct types: low, medium low, medium high, and high duct pressure drops. The increase in ESPs resulted in airflow reductions of 11.2% in the heating mode and 17.7% in the cooling mode, as indicated in Table 2. Because of this airflow reduction, the heating capacity of the furnace increased by 2.1%, but the cooling capacity of the air conditioner was adversely impacted, as shown in Figure 4. For a constant heating or cooling load, the increase in the capacity would decrease the system runtime. Conversely, a reduced capacity would result in extra operating hours. As a result, the heating hours decreased from 311 to 304, corresponding to a 2.4% decrease. The cooling hours increased from 1243 to 1328, corresponding to a 6.8% increase.

Table 5 shows the system annual electricity and gas consumptions associated with ducts of different

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pressure drops. The electricity consumed by the condensing unit accounted for more than 81% of the annual system electricity consumption. The blower consumption was about 18%. The electricity utilized by the furnace inducer fan was less than 1%. Because the increase in the blower electricity consumption in the heating mode was offset by the decrease in the cooling mode, the overall impact of ESPs on the annual blower electricity consumption was relatively small. Due to the 6.8% increase in the cooling equipment runtime, the condensing unit electricity consumption increased by 7.5%. The total system electricity consumption was penalized by 6% as a result of increasing ESPs. Because of the increased blower electricity consumption in the heating mode, the responding increased heat gain from the blower tended to offset the natural gas consumption, which only had a minor decrease (0.3%) for increases in ESPs.

CONCLUSIONS

This study used both experimental and simulation approaches to investigate the impact of ESPs on the residential heating and cooling energy use. In the experimental study, the aerodynamic and cooling performance of a unitary unit with gas heating and electric cooling was experimentally evaluated at various ESPs and airflow rates. These laboratory test results were then integrated with a public-domain building simulation model to estimate the annual heating and cooling energy use at different ESP settings.

Simulations were conducted with four different duct types: low, medium low, medium high, and high duct pressure drops. The results indicated that ESPs have great impact on residential heating and cooling energy consumptions. The increased ESPs resulted in an airflow rate decrease of 11.2% in the heating mode and 17.7% in the cooling mode. Due to the degradation in the cooling capacity and efficiency, the system runtime increases by 6.8%, leading to a 7.5% rise in the condensing unit electricity consumption. The overall annual electricity utilization of the system (including the blower and condensing unit) was penalized by 6% as a result of increasing ESPs. In contrast, the natural gas consumption was reduced by 0.3% because of the increased heat gain from the blower as ESPs were increased.

In summary, this study quantified the impact of ESPs on residential heating and cooling energy consumptions in hot climates. The results generated in this study could be used to investigate how ESPs affect the residential energy consumptions and annual utility cost. In addition, the results can also be used for the development of a cost-effective design for residential air distribution systems.

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Table 1 Coefficient and R^2 Values for the DX Cooling System Performance Curves

PERFORMANCE CURVE	COEFFICIENTS			R^2
	C1	C2	C3	
GROSS COOLING CAPACITY	0.4898	0.8367	-0.3266	0.99
OUTDOOR FAN/COMPRESSOR POWER	1.4544	-0.6969	0.2424	0.98

Table 2 Blower Performance and Heating Capacities at each Operating Point

OPERATING POINT	COOLING MODE			HEATING MODE			
	ESP, in. w.g. (Pa)	Airflow Rate, ft^3/min (m^3/s)	Blower Power, W	ESP, in. w.g. (Pa)	Airflow Rate, ft^3/min (m^3/s)	Blower Power, W	Actual Heating Capacity, Btu/hr (W)
LOW	0.30 (75)	1941 (0.92)	734	0.22 (55)	1659 (0.78)	462	66548
MEDIUM LOW	0.50 (125)	1863 (0.88)	743	0.37 (93)	1607 (0.76)	475	66925
MEDIUM HIGH	0.70 (175)	1752 (0.83)	720	0.54 (135)	1549 (0.73)	497	67370
HIGH	0.90 (225)	1598 (0.75)	670	0.77 (193)	1473 (0.70)	527	67952

Table 3 Annual Blower Electricity Consumption and Operating Hours

DUCT PRESSURE DROP	ANNUAL BLOW ELECTRICITY CONSUMPTION, kWh			OPERATING HOURS
	HEATING MODE	COOLING MODE	STANDBY MODE	
LOW	145.5	903.3	64.9	1554
MEDIUM LOW	146.9	922.1	64.7	1569
MEDIUM HIGH	152.7	916.9	64.5	1593
HIGH	160.3	884.9	64.2	1632

Table 4 Annual Heating and Cooling Equipment Operating Hours

DUCT PRESSURE DROP	EQUIPMENT OPERATING HOUR	
	HEATING HOUR	COOLING HOUR
LOW	311	1243
MEDIUM LOW	309	1260
MEDIUM HIGH	307	1286
HIGH	304	1328

Table 5 Annual System Electricity and Gas Consumption

DUCT PRESSURE DROP	ANNUAL SYSTEM ELECTRICITY CONSUMPTION, kWh				ANNUAL SYSTEM GAS CONSUMPTION, mcf
	BLOWER	CONDENSING UNIT	FURNACE INDUCER	TOTAL	
LOW	1113.7	4937.7	23.3	6074.7	25.8
MEDIUM LOW	1133.7	5016.1	23.2	6173.0	25.8
MEDIUM HIGH	1134.1	5131.8	23.0	6288.9	25.7
HIGH	1109.4	5307.6	22.8	6439.8	25.7

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