EFFECT OF INSTALLATION FAULTS ON THE PERFORMANCE OF A SPLIT AIR CONDITIONER

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ABSTRACT

The study analyzed the effect of common installation faults on the cooling performance of a residential, split air conditioner installed in a single-family, slab-on-grade house. We considered five climatic regions in the United States from hot/humid to cold. Through seasonal simulations of the house/air conditioner system, the study found that duct leakage, refrigerant undercharge, undersized refrigerant expansion valve, low indoor airflow, and air conditioner oversizing with non-oversized ductwork have the most potential for causing significant performance degradation and increased seasonal energy consumption. Seasonal energy use can significantly increase even more when the homeowner lowers the thermostat setting to remove excessive indoor humidity caused by certain installation faults.

1. INTRODUCTION

Building energy efficiency goals and environmental concerns require that installed space-conditioning equipment be highly efficient. Historically, state and municipal governments and utility partners have implemented various initiatives that promote sales of high-efficiency systems. More recently, stakeholders realize that merely increasing equipment's laboratory-measured efficiency without ensuring that the equipment is installed and operated correctly in the field is ineffective. Numerous field studies have documented degraded performance and increased energy usage for typical air conditioners (ACs) and heat pumps (HPs) installed in the United States. For example, Proctor (1997) measured the performance of 28 ACs that had been newly installed in 22 residential homes. Indoor airflow averaged 14 % below specifications, and only 18 % of the systems had a correct amount of refrigerant. The supply duct leakage averaged 9 % of the air handler airflow, and the return leakage amounted to 5 %. Rossi (2004) measured the performance of 1468 unitary ACs during routine maintenance visits and found that 67 % needed service. Of those units, 15 % required major repairs (e.g., compressor or expansion device replacement), and 85 % required a tune-up type service (e.g., coil cleaning or refrigerant charge adjustment). Approximately 50 % of all units operated with efficiencies of 80 % or less, and 20 % of all units had efficiencies of 70 % or less of their fault-free efficiency. Mowris et al. (2004) performed field measurements on 4168 new and existing ACs and HPs. They reported that 72 % of the tested units had improper refrigerant charge, and 44 % had improper airflow. Approximately a 20 % efficiency gain was estimated after refrigerant charge and airflow were corrected.

This paper discusses the effects of different commissioning faults on AC performance and seasonal energy use in a single-family, residential, slab-on-grade house. The results presented here are supplemental to a detailed report by Domanski et al. (2014), which presented effects of installation faults on annual (cooling and heating) energy consumption by a heat pump.

2. TECHNICAL APPROACH

2.1 Scope

We performed simulations of the house/AC system to determine the energy use during the entire cooling season for fault-free and faulty AC installations. We used a fault-free installation as a reference for normalizing energy use in faulty installations and for indicating the impact of specific installation parameters on energy consumption. The evaluated installation parameters include: AC sizing, indoor coil airflow, refrigerant charge, presence of noncondensable gases, electrical voltage, thermostatic expansion valve (TXV) undersizing, and duct leakage. The study considered five cities, which represent U.S. climate zones 2 through 6, from a hot-and-humid climate to a cold climate (Table 1), of the International Energy Conservation Code (IECC, 2012).

Table 1. IECC U.S. climates and locations considered with thermostat set points and electricity cost

Zone	Climate	Location	Air conditioner capacity (kW)	Thermostat set point (°C)	Electricity cost ($/kWh)
2	Hot-and-humid	Houston, TX	10.6	25.6	0.085
3	Hot-and-dry	Las Vegas, NV	12.3		0.126
4	Mixed	Washington, DC	8.8	24.4	0.141
5	Heating-dominated	Chicago, IL	8.8		0.128
6	Cold	Minneapolis, MN	8.8		0.108

Note: (1) Thermostat set points selected based on Rudd et al. (2013); (2) Electric costs from Form 826 data for local utility in 2010 for residential sector (EIA, 2012)

2.2 Building specifications

The simulated residential building corresponded to a code-compliant house with a Home Energy Rating System (HERS) score of approximately 100 (RESNET, 2006) and included the appropriate levels of insulation and other features required for the rating within each climate. The simulated house is a 185.8 m^2 three-bedroom structure with a separate, unconditioned attic zone (Figure 1). It had perimeter slab insulation in climate zones 4 and 5. A ‘fictitious layer’, represented in the figure by $R_{\text{fic-floor}}$, was added to create resistance between the conditioned zone and the ground. The R-value of this fictitious layer was set to provide the heat loss determined by the F-factor method ($R_{\text{effective}}$), as recommended by Winkelmann (1998).

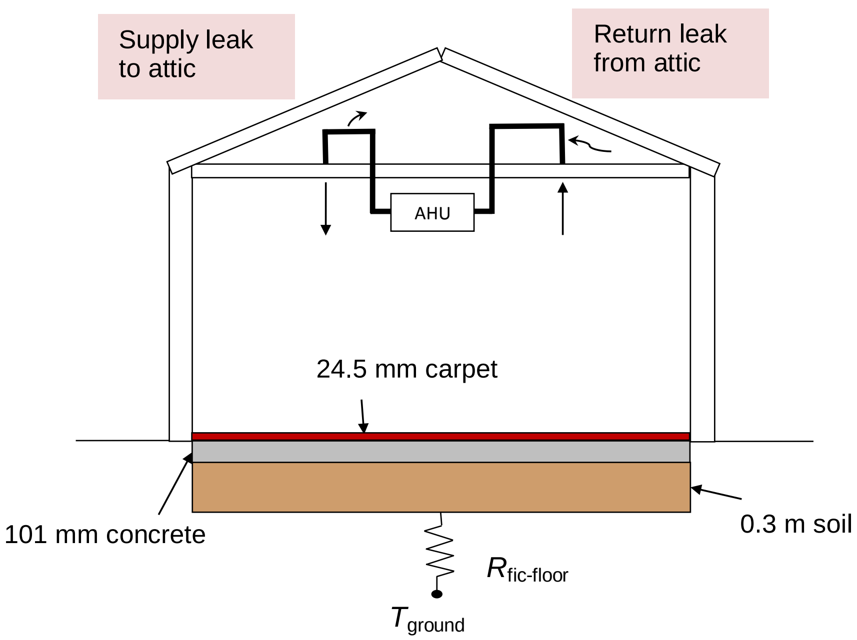

Figure 1. Schematic of a slab-on-grade house (ducts located in the unconditioned attic)

The above-ground portions of the houses had exterior walls with layers of drywall, framing and insulation [R(SI)-2.3 or R(SI)-3.3, depending on the climate zone], and stucco as the exterior finish. Windows comprised approximately 22 % of all of the exterior walls; 10.2 m^2 on the north and south facing walls, and 6.5 m^2 on the east and west facing walls. The ceiling (i.e. boundary between main zone and attic) was made up of a layer of drywall, framing and insulation [R(SI)-5.3 or R(SI)-6.7, depending on the climate zone]. The attic had gabled walls on the east and west ends and a pitched roof surface on the north and south sides. The roof was sheathed in plywood and then covered with asphalt shingles. The east and west surfaces (gables) were made up of plywood on the inside surface with stucco on the outside surface.

The ducts were located in the attic, and all the air leakage and thermal losses/gains were modeled to go into that zone. Duct leakage was assumed to be 10 % of flow, 6 % on the supply side and 4 % on the return side. Duct insulation was assumed to be R(SI)-1.1 with a supply duct area of 50.5 m^2 and a return duct area of 9.3 m^2 for a 10.6 kW-rated unit. The nominal duct areas were increased and decreased proportionally based on the size (nominal capacity) of the heat pump unit. The combined impact of duct leakage, ventilation, and infiltration was considered. An equivalent leakage rate of 0.06333 m^2 was chosen to provide the desired seven air changes per hour at 50 Pa pressure differential. All detailed house design specifications are given by Domanski et al. (2014).

2.3 Building/air conditioner simulation model

We used a building model developed in TRNSYS* to simulate the integrated performance of air conditioners in residential applications (CDH, 2010). The model is driven by typical meteorological year (TMY3) weather data sets (Wilcox et al., 2008). A time-step of 1.2 minutes was used. A detailed thermostat model turned the mechanical systems "on" and "off" at the end of each time step depending on the calculated space conditions. The "on" and "off" operation of the indoor unit was the same as that of the compressor. A conventional air conditioner unit with a 13 SEER rating was used in the simulations. The cyclic degradation coefficient, C_D, of the air conditioner was approximately 0.15. The required size of the unit was determined for each climate using ACCA Manual J (ACCA, 2011).

We modeled the performance of a fault-free AC using a detailed model derived from catalog data covering a series of single-speed products (Parker et al., 1997). To determine the performance of a faulty unit, we applied dimensionless multipliers to the fault-free performance parameters (capacity, COP, etc.), which were developed based on an extensive laboratory test effort (Kim et al., 2009; Cho et al., 2014). Refer to Domanski et al. (2014) for definitions of the studied faults and for more details on modeling of a fault-free and faulty heat pump. Table 2 lists the studied faults and their intensities with respect to fault-free state.

Table 2. Studied faults and their intensities

Fault Type	Fault Intensity (%)
AC Sizing (SIZ)	-20, 25, 50, 75, 100
Duct Leakage (DUCT)	0, 10, 20, 30, 40, 50
Indoor Coil Airflow (AF)	-36, -15, 7, 28
Refrigerant Undercharge (UC)	-10, -20, -30
Refrigerant Overcharge (OC)	10, 20, 30
Noncondensable Gases (NC)	10, 20
Electric Voltage (VOL)	-8, 8, 25
TXV Undersizing (TXV)	-60, -40, -20

3. EFFECTS OF INSTALLATION FAULTS ON PERFORMANCE

3.1 Effect of unit sizing

Changing the size of the air conditioner for a given house – either undersizing or oversizing – impacts unit performance in several ways:

- Cycling losses increase as the unit gets larger; the unit runs for shorter periods and the degraded performance at startup has more impact ($C_D \approx 0.15$).
- The shorter run periods impact the moisture removal capability, i.e., ability to control indoor humidity.

Unit sizing also affects the level of duct losses, and different oversizing scenarios are possible depending on the relative size of the unit and the ductwork. The scenario considered in this paper represents the case where the ductwork has been sized for an air conditioner of nominal capacity and remains unchanged for different size units. When the air conditioner is oversized, the indoor fan speed is increased but the airflow does not reach the target flow rate because the unit is not capable of overcoming the increased external static pressure. Since the indoor fan works against increased static pressure, the fan power changes per the fan curve, i.e., fan power increases with an increasing unit size. The increased pressure in the duct increases the duct leakage. The increased fan power (while working against increased static pressure) and the increased fan heat added to the load are the main factors contributing to the increase in energy used.

The effect of sizing on the total performance for a hot-and-humid climate (i.e., Houston) is compared in Table 3 with results for a cold climate (i.e., Minneapolis). To gain insight into the impact on moisture removal, a relative humidity threshold value of 55 % was selected as the level above which humidity might start to be a concern. The highest indoor temperature reached during the cooling season is recorded in the table's 'Space Temp Max' column. The 'AC Energy' entries contain the energy used by the compressor and

Table 3. Effect of AC sizing on seasonal energy use for a house with a fixed duct size

Houston	Hours Above 55 % RH	Space Temp Max (C)	AC Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Cooling Fan Energy (MJ)	TOTAL Energy (MJ)	Total Costs ($)	Relative Energy (%)
Undersized 20 %	1,692	27.6	2,317	4.5	0.790	15,621	3,085	18,706	442	90
Normal	1,512	26.6	1,981	4.3	0.785	16,660	4,012	20,672	488	100
Oversized 25 %	1,443	25.8	1,687	4.3	0.780	17,709	5,394	23,103	545	112
Oversized 50 %	1,320	25.3	1,492	4.1	0.774	18,713	6,087	24,800	586	120
Oversized 75 %	1,244	25.2	1,343	4.0	0.769	19,587	6,677	26,265	620	127
Oversized 100 %	1,205	25.1	1,224	3.9	0.766	20,351	7,262	27,613	652	134
Minneapolis	**Hours Above 55 % RH**	**Space Temp Max (C)**	**AC Runtime (h)**	**AC COP (-)**	**AC SHR (-)**	**AC Energy (MJ)**	**Cooling Fan Energy (MJ)**	**TOTAL Energy (MJ)**	**Total Costs ($)**	**Relative Energy (%)**
Undersized 20 %	16	26.0	1,059	4.6	0.857	5,604	1,439	7,044	211	95
Normal	13	25.2	897	4.5	0.846	5,912	1,514	7,425	223	100
Oversized 25 %	13	24.5	761	4.4	0.840	6,255	2,161	8,417	252	113
Oversized 50 %	11	24.1	677	4.3	0.828	6,659	2,399	9,058	272	122
Oversized 75 %	8	24.0	612	4.1	0.820	7,000	2,564	9,564	287	129
Oversized 100 %	5	24.0	560	4.0	0.814	7,295	2,708	10,003	300	135

outdoor fan to provide cooling; 'Cooling Fan Energy' contains the energy used by the indoor fan over the cooling season. The column 'TOTAL Energy' is the sum of the two previous columns.

At first glance, the reader may be surprised by a decrease in the energy when the AC is undersized. The lower energy use is a result of the air conditioner being unable to handle all the cooling load (the indoor temperature increases on hot days). The coefficient of performance (COP) in Minneapolis is only somewhat higher than the value for Houston because the climate in Minneapolis is less humid and the AC handles a lower latent load than that in Houston. Otherwise, the relative increase in energy use due to oversizing is similar in all five studied climates and can be on the order of 30 % for an AC oversized by 100 % (Figure 2).

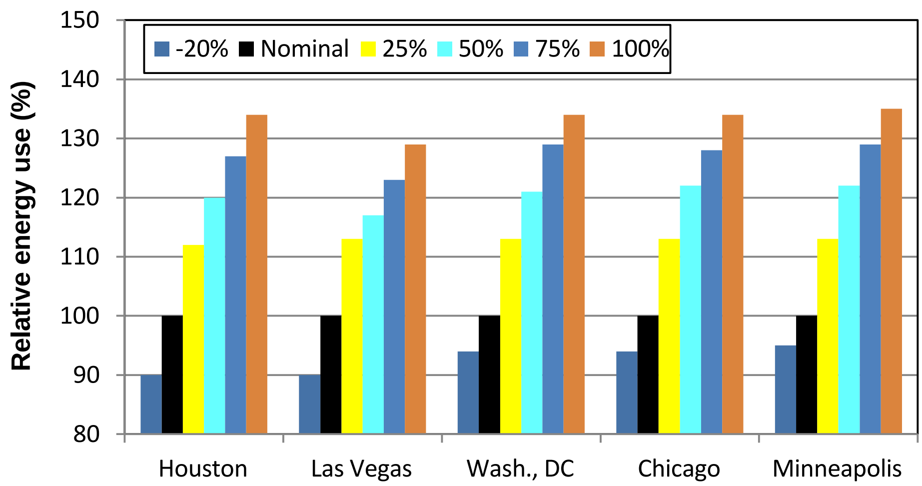

Figure 2. Air conditioner energy use for houses with different unit sizings

3.2 Effect of duct leakage

The baseline houses include ducts in the attic with a leakage rate of 10 % (leakage distributed 60 % on the supply side and 40 % on the return side) and thermal losses through the duct wall. Table 4 compares this base case to other levels of duct leakage with the thermostat set at the default set point temperature (Table 1) for the house in Houston. The entry '0 % & No thermal' in the left most column of Table 4 denotes an ideal installation with zero air leakage and no thermal loss.

As expected, the 10 % baseline duct losses increase energy use in the baseline houses for all five cities. Our simulations showed a 22 % increase in energy use as compared to the ideal '0 % & No thermal' case. As the duct leakage increases energy use increases by approximately 8 % for each 10 % increment in duct leakage.

The slight improvement in COP with the increasing leakage is caused by a somewhat higher refrigerant saturation temperature (and pressure) in the evaporator, which results from the return-side duct leak raising the evaporator entering air temperature. This COP improvement, however, can't compensate for the significant increase in the cooling load, which is the cause of the increased energy use. Simulations for other cities showed similar trends and values.

Table 4. Effect of duct leakage on seasonal energy use at the default cooling set point

Houston	Hours Above 55 % RH	AC Runtime (h)	AC COP (-)	AC SHR (-)	AC Energy (MJ)	Cooling Fan Energy (MJ)	TOTAL Energy (MJ)	Total Costs ($)	Relative Energy (%)
0 % & No thermal	1,715	1,555	4.3	0.789	13,007	3,148	16,155	381	78
0 % Leak	1,537	1,794	4.3	0.812	15,046	3,633	18,679	441	90
10 % Leak	1,512	1,981	4.3	0.785	16,660	4,012	20,672	488	100
20 % Leak	1,632	2,160	4.4	0.767	18,179	4,375	22,553	533	109
30 % Leak	1,922	2,327	4.5	0.753	19,574	4,711	24,285	573	117
40 % Leak	2,738	2,489	4.5	0.743	20,922	5,040	25,962	613	126
50 % Leak	3,364	2,649	4.6	0.734	22,231	5,364	27,595	652	133

Note: All simulation cases account for thermal losses along with leakage losses except the case denoted "0 % & No thermal".

Simulations for Houston showed a significant increase in hours above 55 % relative humidity with increased duct leakage, which may motivate the homeowner to lower the thermostat set point temperature to improve indoor comfort. However, lowering the thermostat setting results in a significant increase in energy use (Figure 3). For the house with a 30 % duct leakage, the energy use is predicted to be 42 % and 88 % higher than for the reference house when the set point is lowered by 1.1 °C and 2.2 °C (to 24.5 °C and 23.4 °C), respectively.

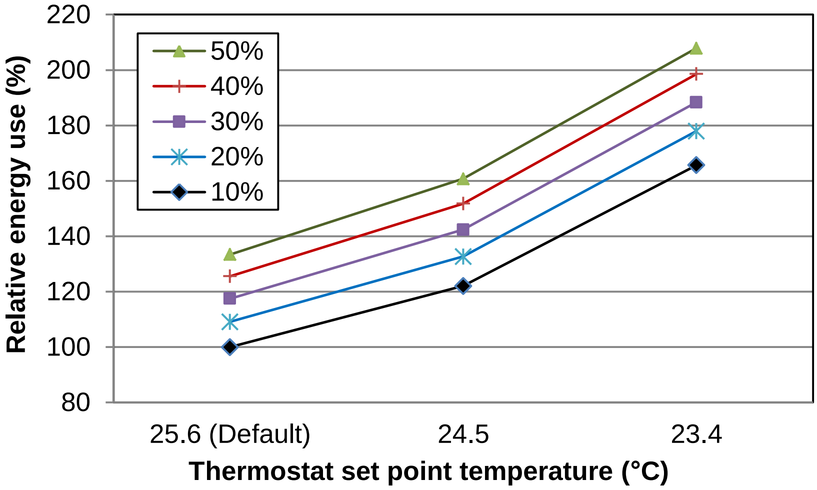

Figure 3. Energy use for a house in Houston with duct leak rates from 10 % to 50 % at three thermostat set points relative to energy use for a house operated at the default thermostat set point and 10 % leakage rate

3.3 Effect of indoor coil airflow

This fault covers the case where the AC is properly sized for the building load but a restriction in the ductwork reduces airflow (in a way that does not affect duct leakage or losses). The simulated indoor airflows, ranging from -36 % to +28 % of the design flow, corresponded to external static pressures of (177, 171, 168, 165, and 149) Pa, respectively. Fan power increases as the fan rides its curve, as discussed above. Reduced airflow results in increased energy use (Figure 4). For the lowest airflow, 36 % below the design value, the energy use increased from 8 % to 12 % for the five cities considered. Reducing the airflow below the design value causes a decrease in the indoor coil temperature and provides better humidity control, but results in higher energy use because the sensible capacity is reduced and running time increased. Conversely, providing more airflow degrades humidity control in the house but decreases energy use.

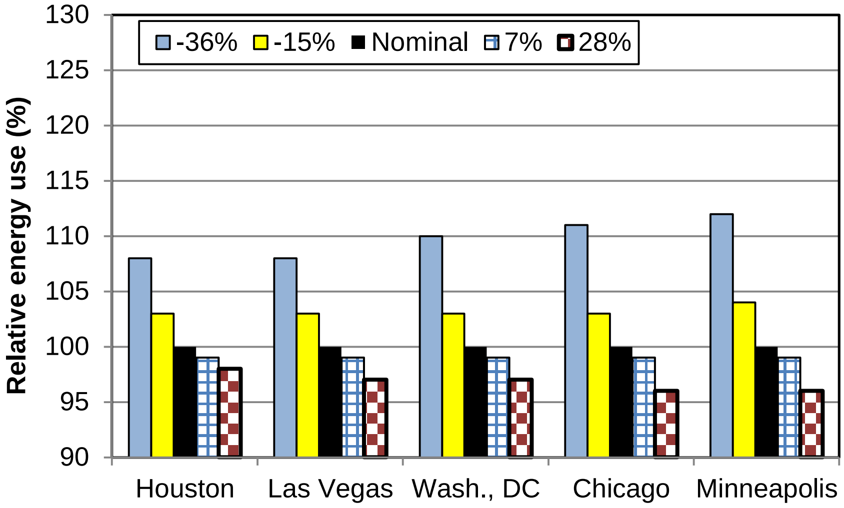

Figure 4. Air conditioner energy use for houses with different indoor coil airflows relative to energy use for the house in the same location with design airflow rate

3.4 Effect of refrigerant undercharge and overcharge

When the amount of refrigerant charge is below the design value, the performance of the unit is degraded. Simulation yielded the same relative energy use increases for all five locations studied. The energy use increased by 3 %, 8 %, and 16 % for refrigerant undercharges of 10 %, 20 %, and 30 %, respectively. The moisture removal capacity of the unit is also degraded when the unit is undercharged. This relationship was demonstrated in Houston, for example, where the hours with relative humidity above 55 % increased from 1512 h for a properly charged system to 1811 h for a system undercharged by 30 %.

When the amount of refrigerant charge in the system is above the correct value, the performance of the unit is also degraded; however, by a smaller degree than for undercharging. For 30 % refrigerant overcharge, the simulations showed 4 % energy use increase for all studied locations. The moisture removal capability of the unit is not affected by the overcharge fault.

3.5 Effect of noncondensable gases, voltage, and TXV sizing

If the AC contains noncondensable gases (e.g., air), the performance of the unit is degraded. The simulation results showed a (3 to 4) % and (6 to 7) % energy use increase for 10 % and 20 % fault levels, respectively. The moisture removal capability of the unit is minimally affected by the noncondensable gases in the system.

When input voltage to the unit is changed from the nominal value, the performance of the unit is degraded. A 25 % overvoltage condition results in a (8 to 11) % increase in seasonal energy consumption. This effect on the energy use does not include an adjustment for indoor fan power change with voltage. An undervoltage of 8 % resulted in an insignificant (within 1 %) change in the energy use. Higher levels of undervoltage were not studied because of a possible heat pump catastrophic failure.

Undersizing a TXV by 20 % has a modest effect on the energy use (within 4 %) because for this degree of undersizing the TXV can open sufficiently at most AC operating conditions. However, the impact becomes significant at a 40 % undersizing (21 % to 28 % increase in energy consumption) because, in this case, the maximum opening of TXV is too small for a significant part of the cooling season. Moisture removal is not much affected.

3.6 Discusson of effect of single faults

Figure 5 shows examples of seasonal energy used by an air conditioner installed with different installation faults. The levels of individual faults were selected to reflect the installation condition which might not be noticed by a poorly trained technician. (The authors recognize the speculative aspect of this selection.)

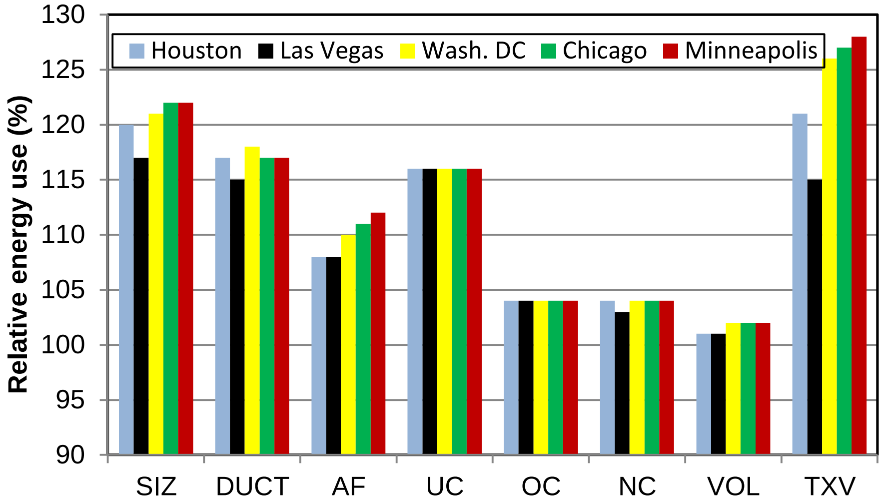

Figure 5. Air conditioner energy use increase resulting from a single-fault installation.
(Fault level used (%): SIZ (+50), DUCT (30), AF (-30), UC(30), OC(30), NC(10), VOL(+8), TXV(-40))

The undersized TXV fault, air conditioner oversizing, duct leakage, and refrigerant undercharge have the potential to increase energy use by 15 %, with the last three of these faults probably being most common. Somewhat less detrimental is the indoor airflow fault, although it can still degrade the COP by 10 %. It should be noted that the impact of installation faults can be compounded when they occur simultaneously. Our selected cooling mode simulations for a slab-on-grade house showed dual fault having an additive effect.

4. CONCLUDING REMARKS

Extensive simulations of the house/air conditioner system lead to the following conclusions:

- The undersized TXV fault, air conditioner oversizing with non-oversized ductwork, duct leakage, refrigerant undercharge, and reduced indoor airflow have the most potential for causing significant performance degradation and increased seasonal energy consumption.
- The detrimental effect of installation faults can be compounded when multiple faults occur simultaneously.
- The relative increase of seasonal energy use due to installation faults is similar for the five climates studied.
- A significant increase in seasonal energy use can be caused by lowering the thermostat to improve indoor comfort in cases of excessive indoor humidity levels cause by installation faults.

Future work should include laboratory characterization of performance degradation of an air conditioner operating under multiple faults to show their effect on seasonal energy use.

The goal of this study was to assess the impacts that HVAC system installation faults have on equipment electricity consumption. The effect of the installation faults on occupant comfort was not the main focus of the study, and this research did not seek to quantify any impacts on indoor air quality, or noise generation (e.g., airflow noise from air moving through restricted ducts). Additionally, the study does not address the effects that installation faults have on equipment reliability/robustness (number of starts/stops, etc.), or costs of initial installation and ongoing maintenance.

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6. NOMENCLATURE

AC = air conditioner
AHU = air handling unit
AF = improper indoor airflow rate fault (%)
C_D = AC cyclic degradation
COP = coefficient of performance
DUCT = duct leakage fault
ELA = equivalent leakage area (m^2)
HP = heat pump
NC = presence of noncondensable gases fault
OC = refrigerant overcharge fault (%) departure from the correct value
R(SI) = thermal resistance in SI system of units ($K \cdot m^2 \cdot W^{-1}$)
RH = relative humidity (%)
SEER = seasonal energy efficiency ratio ($Btu \cdot W^{-1} \cdot h^{-1}$)
SHR = sensible heat ratio (sensible capacity divided by total capacity)
SIZ = heat pump sizing fault (%) above or below the correct capacity
T = temperature (°C)
TMY3 = data set 3 with typical meteorological year weather data
TXV = thermostatic expansion valve or TXV undersizing fault in cooling
UC = refrigerant undercharge fault (%) departure from the correct value
VOL = electric line voltage fault (%)

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