Finding Fault: An investigation of the prevalence and energy impacts of residential HVAC installation faults

Eric Martin, Charles Withers, Jr., Karen Fenaughty, Pacific Northwest National Laboratory
Karthik Panchabikesan, University of Central Florida
Tom Downey and Abram Conant, Proctor Engineering Group

ABSTRACT

Millions of air conditioners and heat pumps are installed in new and existing homes each year, with many installed as part of utility or other programs having high efficiencies. A body of past work has documented that many of these systems are designed and installed improperly, likely compromising system efficiency, equipment longevity, and occupant comfort. These studies have shown that various design faults such as improper equipment and duct sizing, along with various installation faults including improper refrigerant charge and evaporator coil air flow, and presence of non-condensables and duct leakage, are commonplace. Data quantifying the prevalence of various faults, along with the effects on energy use are scarce, and additional data is useful to inform efforts to develop and implement quality installation programs and workforce training. This paper describes results from a field study evaluating prevalence and severity of design and installation faults in a sample of 83 homes in Arizona, California, and Florida with a newly installed air conditioner or heat pump. A portion of the systems were installed as part of a quality installation program. In addition, experimental results obtained by imposing common installation faults, individually and in combination, in a heat pump providing space conditioning for a simulated occupancy laboratory home are described. Measured increases in cooling energy use ranged from 4% for a single fault to 41% for a combination of faults.

Introduction

Properly designing and installing space conditioning systems is critical to ensure the expected efficiency and life expectancy is obtained, however several studies have documented that a significant fraction of newly installed residential comfort systems are likely installed improperly (US DOE 2018). These studies have documented that various *design* faults such as improper equipment and duct sizing and improper equipment selection, along with various *installation* faults including improper refrigerant charge, presence of non-condensables[1], improper air flow, and duct leakage, are commonplace. While a great deal of field work has already been done to identify the presence of comfort system installation faults, data quantifying the prevalence of these faults, along with their effect on energy use, are scarce. A study by the National Institute for Standards and Technology (NIST), based on simulations informed by laboratory data, concluded that installation faults can result in increased energy use of up to 30% (Domanski 2014). Since estimates of the prevalence and impacts of installation faults point to the potential for significant improvement in installation quality through the use and application of existing industry guidance such as the Air Conditioning Contractor's Association (ACCA)

[1] Non-condensable gas is typically ambient air that can be introduced into the refrigerant system of an air conditioner or heat pump if the system is not properly evacuated prior to charging with refrigerant.

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HVAC Quality Installation Specification (ACCA 2015) it is important to collect data to provide insight into how common installation faults are, and how they might be controlled.

One method to minimize the occurrence of installation faults is HVAC quality installation (QI) programs that provide combinations of contractor training, support during installation for more accurate measurements of critical aspects of equipment performance, and follow-up quality control checks. Such programs have been implemented by utilities, municipalities, code and above code programs, builders and contractors, and have proven successful at improving installation quality (Proctor 2001a).

This report describes efforts to add to the body of knowledge on the prevalence and energy impacts of residential HVAC installation faults. Proctor Engineering Group (PEG) led a field study that was conducted in 83 homes in AZ, CA, and FL that had an air conditioner or heat pump newly installed in the last 12 months. Field teams visited the homes after installation and collected data to determine if typical industry design and quality installation practices were followed. Specific design parameters of interest included the rated cooling capacity of the installed system as compared to the ACCA Manual J (ACCA 2016) design load of the home, and the duct system static pressure and blower fan watt draw as an indicator of the adequacy of the duct design. Specific installation quality parameters of interest included the refrigerant charge, evaporator coil air flow, and duct leakage.

In addition, the Florida Solar Energy Center (FSEC), a research institute of the University of Central Florida led a lab home study in which common installation faults were intentionally imposed in a heat pump, and cooling energy use was measured and compared to that from an identical lab home with an identical heat pump that was properly commissioned and operating alongside. Faults imposed included duct leakage, refrigerant under/overcharge, and non-condensables. While it was intended to conduct the field study first and use measured fault intensities to inform the lab study, the COVID-19 pandemic delayed field data collection until after the lab study. Fault intensities imposed were determined from literature and stakeholder input.

Field Study Documenting the Prevalence and Severity of Installation Faults

Study Design and Participant Recruitment

The field study targeted primarily existing homes in AZ, CA, and FL to obtain a representative sample of homes in cooling dominated climates where most central air conditioners and heat pumps are installed. Homes with systems recently installed as part of a QI program were recruited, along with homes with systems installed according to regional standard practice (RS). Recruitment was primarily conducted in partnership with utility companies that the research team had relationships with. In the Southeast, the research team partnered with Orlando Utilities Commission (OUC) and the City of Tallahassee Electric Utility (TEU). In the Southwest, the research team partnered with the Los Angeles Department of Water and Power (LADWP) and Tucson Electric Power (TEP). FSEC and PEG worked with each utility to customize a study recruitment letter that was mailed to potential participants. Seven "other" homes that did not participate in a utility program were also recruited. Four were obtained from the City of Fresno permit data, and three contacted the research team after heating about the study. The TEP sample was exclusively QI homes, which used PEG's CheckMe!® quality assurance system, the only QI program identified in the Southwest. CheckMe is a portfolio of measures designed to ensure HVAC systems are operating at their designed capacity and efficiency and includes classroom and in-field training, in-field real-time feedback on installation quality, and post-installation quality assurance visits.

LADWP had a mix of QI, also using CheckMe, and RS. The remainder of the sample was RS. No QI program was identified in the Southeast. While none of the RS installations were subject to QI requirements, they may not be truly representative of regional standard practice. In some markets the majority of HVAC systems installed in existing homes do not have a permit pulled. For example, in the CA market, some estimate that 90% of residential systems do not involve a permit[2]. While un-permitted installations may represent true regional standard practice in these markets, there was no way to identify such installations for participant recruitment. Table 1 gives a geographic breakdown of the recruited sites along with whether systems were installed with regional standard practice or as part of a quality installation program.

Table 1. Breakdown of recruited study sites.

Field Sample Characteristic			**Number of Homes**
Hot Humid Climate (2A)			
FL	*OUC*	Existing homes, regional standard practice	15
		New Construction, regional standard practice	4
	TEU	Existing homes, regional standard practice	23
		Total Hot Humid QI	**0**
		Total Hot Humid Regional Standard	**42**
		Total Hot Humid Climate	**42**
Hot Dry Climate (2B, 3B)			
CA	*LADWP*	Existing homes, quality installation program	11
		Existing homes, regional standard practice	9
	Fresno	Existing homes, regional standard practice	4
	Other	Existing homes, regional standard practice	2
		Total CA homes	**26**
AZ	*TEP*	Existing homes, quality installation program	14
	Other	Existing homes, regional standard practice	1
		Total AZ homes	**15**
		Total Hot Dry QI	**25**
		Total Hot Dry Regional Standard	**16**
		Total Hot Dry Climate	**41**
Total Sample			
		Total QI	**25**
		Total Regional Standard	**58**
		Total Sample	**83**

Data Collection Methodology and General Home and HVAC System Characteristics

During 2021-2023 two-person field teams made a single visit to each home and collected data on home and HVAC system characteristics and performance over a period of six to eight

[2] https://www.nrdc.org/bio/kiki-velez/poor-quality-hvac-installs-are-costing-us-solution-within-reach

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hours. All field team staff were trained in the specific data collection methodology by PEG's field team manager. HVAC system data was collected exclusively by specially trained PEG employees. In AZ and CA, PEG employees also gathered home characteristic data. In FL, local home energy raters accompanied PEG staff and gathered home characteristic data.

The majority of homes recruited were single-family, detached homes, although a small number of single family attached homes were recruited in each region (three in AZ, three in CA, and six in FL). The general characteristics of homes recruited were representative of the housing stock in each region, however CA homes had significantly more variation in age (median 58 years old vs 30), foundation type (more crawlspaces), and insulation values (lower wall and attic insulation) than the homes in either AZ or FL. The homes in AZ and FL tended to be more airtight (median 9.5 and 7.0 ACH50 respectively) than the homes in CA (median 11.1 ACH50), and in general, newer homes in the sample tended to be 5-7 ACH50 more airtight than older homes.

All homes had a single, newly installed, centrally ducted air conditioner or heat pump. Heat pumps were more common in the Southeast (79% of sample vs. 41%). All air conditioner systems were comprised of gas furnaces with external evaporator coils. The sample contained a mix of different system manufacturers and installation contractors. The majority of the systems had single-stage compressors, although some were two-stage (25%) or inverter driven compressors (14%). The rated average seasonal energy efficiency ratio (SEER) for the systems was 16 and the average rated cooling capacity was 3.5 tons. Except for the small Fresno sample group, 50% or more in each group are in the 4-5 ton range, and 82% of the LADWP AC Opt sample are in the 4-5 ton range. The characteristics of the duct systems were consistent across all regions, with R-6 insulated flex duct located in vented attics or crawlspaces, and air handlers and furnaces located in attics, garages, or interior closets. Seventy-one percent of the systems utilized a single return and twenty-three percent utilized two returns. Only five systems had more than two returns. Overall, the duct systems in FL had a design with more branch ducts. The average system in FL had 13 supply registers or four supply registers per ton of rated cooling capacity. AZ and CA systems both had an average of 10 supply registers per system, with the AZ systems having an average of three supply registers per ton of rated cooling capacity and CA systems averaged two supply registers per ton of rated cooling capacity.

Field Study Results and Discussion

System sizing. The main reason to correctly size retrofit replacement air conditioners and heat pumps is to enable an existing duct system to have a better chance of delivering adequate air flow without placing additional load on blower fans due to increased duct system static pressure. Another reason is to reduce peak demand, as oversized single stage HVAC systems have higher watt draws while operating during peak periods. ACCA Manual J calculations were performed for each home to compare the calculated cooling loads to the installed capacity of the cooling systems, and results show that most systems continue to be oversized. ANSI/ACCA Manual S – Residential Equipment Selection (ACCA 2023) dictates that single speed systems should be selected with capacities between 90-115% of the calculated load or be the next nominal size available. Two-stage and inverter systems can be up to 120% and 130% of the calculated load respectively. The average system in this study was found to be oversized by 9,483 Btu/h, providing an average of 135% of the calculated cooling load. While the average oversizing for the QI sample (141%) is slightly higher than the RS sample (133%), the QI sample has a larger fraction of two-stage systems. Fifty three (76%) of the 70 oversized systems were oversized by more than the half ton a contractor would need to round up if the cooling load exceeded the cooling capacity of the next smaller system. Results show that oversizing of air conditioners and heat pumps continues to be pervasive. A 1999 review of previous research from across the country found average oversizing in 11 research projects of 147%, ranging from 116% to 170% (Neme 1999). Average system oversizing in eight of these research projects was 0.91 tons over the design load of the house. The rated cooling capacity of the average system in this study is 0.79 tons greater than the design load of the house.

Evaporator coil airflow and blower fan watt draw. For a system to deliver its intended capacity, the designed air flow must be delivered across the evaporator coil. There are many factors that influence the air flow through an air handler or furnace including: filter condition, filter efficiency, supply register closure, return grille obstruction, blower motor speed, blower cleanliness, and undersized or otherwise restricted duct systems. If system static pressure is elevated as a result, it can cause the blower fan motor to have an elevated watt draw.

The nominal rated cooling capacity for most systems is determined at an evaporator coil air flow of 400 cfm/ton, unless specified otherwise by the manufacturer (AHRI 2020). In humid climates, a lower air flow of 350 cfm/ton is often targeted for improved moisture removal. While the research team does not know what the installation contractors intended as design air flow, results are referenced to 400 cfm/ton nominal. Overall, evaporator coil air flow was found to be lower than this nominal value, similar to what has been found in various past studies. Fifty three percent of the systems tested had less than 350 cfm/ton, a common point at which air flow improvement is recommended. This is consistent with a 2011 study of 80 CA homes that found average air flow of only 322 cfm/ton (Proctor 2011).

Figure 1 shows a binned distribution of measured air flow rates as a percentage of nominal. The systems with air flow closest to nominal rated air flow were the systems in the TEP QI sample, with 12 of the 14 systems having at least 350 cfm/ton. On the other hand, the LADWP AC Opt QI sample had among the lowest percentage of systems with correct air flow with 9 of the 11 sites having less than 350 CFM/ton. Part of the reason for the poor air flow in the AC Opt sample had to do with system sizing. The AC Opt sample has the largest average installed capacity (49,364 Btu/h). The homes in CA are also among the oldest in the sample, with ducts likely designed for heating only, and likely undersized for cooling. Ducts in the TEP sample were likely originally designed for evaporative cooling systems, and hence able to handle higher flow. The field study evaporator coil air flow data was collected using the Energy Conservatory TrueFlow grid, which tends to provide more accurate results than the temperature split method commonly used by contractors including those participating in the CheckMe program (Downey 2002). The temperature split method is a qualitative indicator to see if air flow is likely to be outside a reasonable lower limit. While utilizing the True Flow device or duct blaster to measure air flow is more accurate, contractors have resisted their use due to the additional time, complexity, and expense involved.

Another measure of air flow efficiency is the amount of blower fan motor power consumed per unit air flow, or fan efficacy. Only three of the systems have a constant volume blower designed to respond to static pressure in the duct system and deliver target flow (at the expense of higher power consumption). The remaining blower fans will deliver less than target flow when placed under higher static pressures, which can often result in lower power consumption. Title 24 requires blower motors to perform at or below a 0.45 W/cfm for gas furnaces and at or below 0.58 W/cfm for air handling units that are not gas furnaces. While the 0.36 W/cfm average fan efficacy for homes in this study is lower than that found in past studies,

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the average total external static pressure of 0.61 inches of water column is higher (Proctor 2001b). Only 4 of the systems tested had fan efficacy >0.58 W/cfm, one in the LADWP AC Opt program, one in the LADWP CRP, and two in TEU.

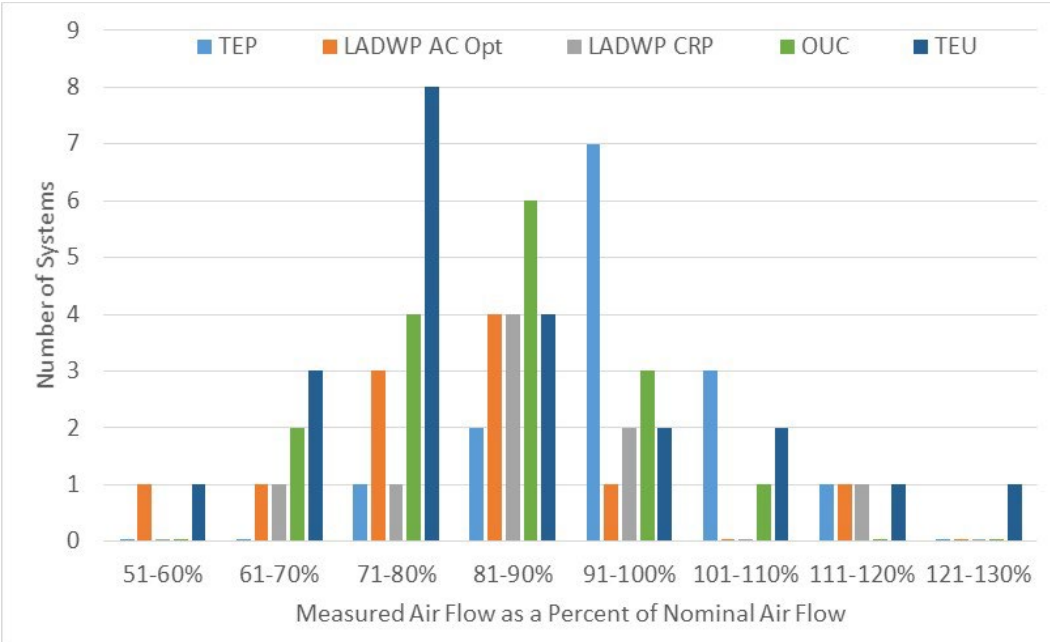

Figure 1. Distribution of measured air flow as a percent of nominal air flow.

Duct leakage. When ductwork is located outside of the conditioned space, duct leakage results in increased heating and cooling energy use and pressure imbalances resulting in uncontrolled infiltration of outdoor air. Title 24 code requires that when an existing HVAC system is replaced, the total duct leakage expressed as cfm25 has to be reduced to 15% or less of nominal system air flow, or leakage to outside reduced to 10% or less of system air flow. Duct leakage measured with the Energy Conservatory Duct Blaster was found to be similar across all sample groups and averaged 21% total and 14% to outside. Only the TEP and the small Fresno sample were found to meet Title 24 requirements on average. When leakage is expressed in terms of cfm25 per 100 square feet of conditioned floor area, ductwork in the Southeast tended to be slightly tighter than in the Southwest, both in terms of total duct leakage (15 vs. 16.5 cfm25), and duct leakage to outside (9.5 vs. 11 cfm25). Overall, these numbers are similar to what has been found in past studies (Proctor 1995, Cummings 2006, Mcilvaine 2013). Figure 2 shows a binned distribution of total duct leakage and duct leakage to outside as a percent of system air flow for the sample.

Refrigerant charge. New air conditioning systems come pre-charged with enough refrigerant for a typical line set length. During installation, contractors need to check system subcooling (for TXV systems) or superheat (for fixed orifice systems) to determine if refrigerant needs to be added or removed for the actual line set length and installed evaporator coil. While 92% of the systems in the QI sample were found to be correctly charged based on subcooling/superheat (+/- 3 degrees from manufacturers target), slightly over half (55%) of the systems in the RS sample were correctly charged. This is consistent with past studies (Pigg 2008, Downey 2002). Most of the incorrectly charged systems were undercharged (77%). It is possible that installation contractors are not adjusting charge according to the actual refrigerant line length. However, while the average refrigerant line length is longer than the 15-25 feet that manufacturer's pre-charge systems for, long refrigerant lines do not appear to be a definitive reason for the high percentage of systems with incorrect charge. Average line set length for incorrectly charged systems and correctly charged systems in the RS sample is 38 ft and 32 feet respectively. Results were also tabulated to determine if the type of compressor had any influence on the charge results. Slightly different procedures must be used to check the refrigerant charge with multi-stage and variable speed systems compared to single-stage systems. While the percentage of systems in the RS sample with correct charge decreases with increasing compressor complexity, the vast majority of systems in the RS sample are single stage.

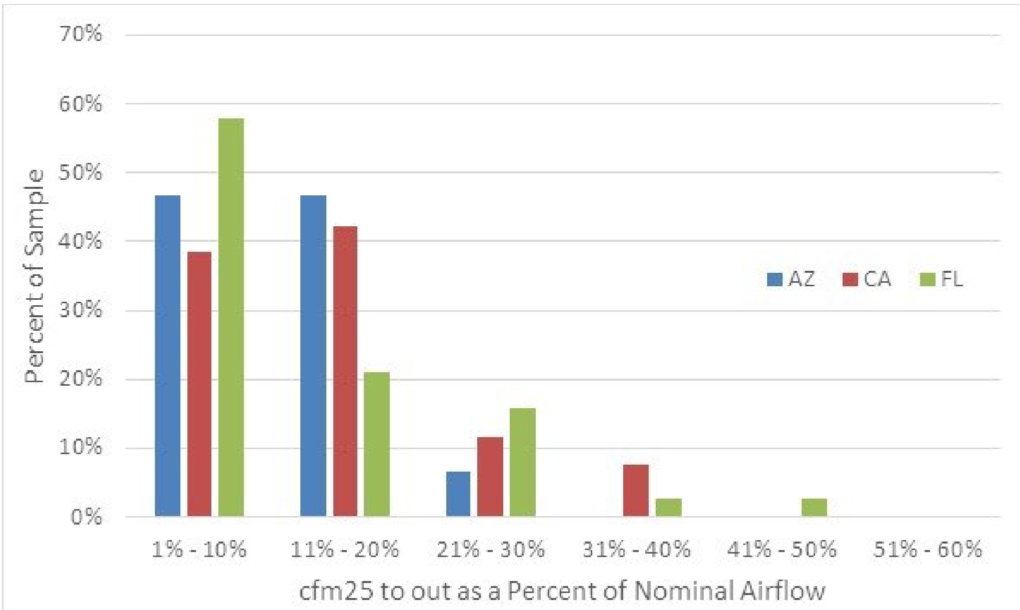

Figure 2. Distribution of cfm25 to outside as a percent of nominal air flow by state.

Field teams intended to work directly with installation contractors to correct the refrigerant charge in all systems that were determined to be incorrectly charged and quantify the amount of refrigerant added/removed by weight. The contractors that were part of QI programs were already incentivized by the program to ensure the charge was correct and participated in this process by default. For the RS sample, contractors were offered $400 for this service visit. Slightly more than half (54%) of contractors for the RS sample did not agree to participate. Refrigerant charge was able to be corrected in 10 systems, all of which were undercharged. Refrigerant charge added to correct the 10 undercharged systems ranged from 14-68% of manufacturer's nameplate charge (based on weight, unadjusted for line set length) and averaged 28%.

Lab Evaluation of the Cooling Energy Impacts of Installation Faults

Study Design and Test Facility Description

In the summers of 2021 and 2022, different combinations of common installation faults were intentionally imposed to the heat pumps serving as the space conditioning systems for FSEC's Flexible Residential Test Facility (FRTF). The two FRTF buildings (referred to as East and West) are identical, 1,536 ft^2 single-story, site built residential structures. The slab-on-grade buildings were built in 2010 and designed to mimic a typical FL existing home. They have painted, uninsulated concrete block walls with no exterior cladding, single pane windows, and vented attics with shingle roof coverings and R-19 kraft-faced fiberglass batt ceiling insulation. Each lab home was cooled with a two-stage SEER 16 central heat pump with heating efficiency of 9.0 HSPF. The air handlers have fans driven with electronically commutated motors (ECM) and are located in the garage on top of a central return support platform. The R-6 flexible duct systems are located in the attic and were designed to best practice with minimal sag and compression and proper turning radii. Internal sensible and latent gains are imposed to ensure realistic operation of the heat pumps. Commissioning and null testing was conducted to ensure the buildings and heat pumps were performing close to each other prior to the start of experimentation.

Various individual and combination faults were imposed for differing periods of time. Refrigerant faults included 30% undercharge and 20% overcharge (by refrigerant weight). Duct leakage was imposed at 10% of evaporator coil air flow on both the return and supply side (20% total), or 15 cfm25 per 100 square feet of conditioned floor area. Non-condensables were imposed using nitrogen at 50% of atmospheric pressure. Space cooling energy use, refrigerant conditions, and indoor/outdoor environmental conditions were monitored to investigate energy and comfort impacts related to the imposed faults. The bulk of the data collection features a side-by-side arrangement where the East system was faulted and the West system remained as a baseline. The number of side-by-side test days, along with the minimum and maximum daily average temperatures, are shown in Table 2. As the outdoor temperature range during data collection was insufficient to model cooling energy use over an entire year for some of the faults, the West system was also faulted for a period of time during summer 2022 and was operated with a different fault than the East system. While no baseline data was collected during this time, baseline data over a sufficient range of outdoor temperatures had already been obtained.

Lab Study Results and Discussion

Two different types of regression analysis were used to examine the impact of individual and combination faults. One method utilized the daily data collected side-by-side, in the same time domain, with one building acting as a control (baseline) and one building operating with faults. The model expressed daily cooling energy as a function of daily average outdoor temperature, and results were evaluated for a typical summer day in central FL, with average daily temperature of 80ºF. This is roughly representative of a typical day during the central FL cooling season and also approximates cooling season impacts in a variety of other climates. A second method was conducted using hourly data to evaluate seasonal impacts and utilized all baseline and fault data collected. This model included considerations for stage 1 and stage 2 heat pump operation, and included additional variables of solar insolation, as well as averages of short term, historical outdoor temperature and solar insolation to account for building thermal mass.

Side-by-Side, 80ºF Evaluation. Regression analyses were conducted using baseline and fault data from the same time domain, to weather-normalize the monitored cooling energy use so changes in energy use attributed to the fault could be quantified. Linear and quadratic models that applied daily average outdoor temperature to predict the daily total cooling energy use (condenser + air handler) were built for each fault, and for each set of baseline data that was collected alongside that fault. For cases with data over a more limited temperature range that did not include data below 75ºF, the linear model outperformed the quadratic model. The best model determined for each fault was used to evaluate impacts for an 80°F day, and results are provided in Table 2. Root mean square error (RMSE) of predictions ranges from 1-2 kWh/d for the linear models (Faults 20, 23, 24, 25, and 28) and 2-3 kWh/d for the quadratic models (Faults 21, 22, 26, and 27).

Table 2. Energy penalty of faults predicted by side-by-side evaluation for 80ºF average day.

Fault Code	Fault Name	Side-by-Side Daily Models @ 80°F						
		Sample	Baseline	Fault	Baseline	Fault	Penalty	
		N (days)	Adj. R^2	Adj. R^2	Daily Cooling Energy (kWh)	Daily Cooling Energy (kWh)	Daily Cooling Energy (kWh)	Daily Cooling Energy (%)
20	Duct leak	10	0.91	0.92	23.8	27.8	4.0	16.8%
21	Undercharge[1]	39	0.95	0.94	22.9	23.6	0.8	3.3%
22	Undercharge & Duct leak[1]	48	0.95	0.95	22.9	28.0	5.1	22.4%
23	Overcharge	15	0.89	0.89	24.4	26.6	2.1	8.8%
24	Overcharge & Duct leak	15	0.85	0.88	23.7	30.8	7.2	30.4%
25	Undercharge & Non-cond.	14	0.94	0.95	25.0	27.4	2.4	9.7%
26	Undercharge, Non-cond., & Duct leak	42	0.95	0.95	23.2	32.1	8.9	38.4%
27	Non-cond. & Duct leak	45	0.94	0.94	21.1	28.2	7.1	33.8%
28	Non- cond.	25	0.97	0.97	22.7	25.5	2.8	12.4%

[1]Small dataset with limited temperature range provided unreliable results. Modeled with all data collected, not just side-by-side data. Baseline includes all baseline days for all side-by-side evaluations, and has a sample size of 184 with a temperature range of 56.5-84.5°F.

Annual evaluation using the full data set. A full cooling season evaluation was conducted using the full dataset that included data over a wider range of outdoor temperatures. As the bulk of data was collected during summer, for it to be representative of an entire cooling season hourly models were developed for each fault and compared to models developed for a singular baseline. Data used to develop the singular baseline model is a composite dataset with all east and west data during 2021 and 2022 when they were in baseline mode, along with some additional data from 2019 that was collected during peak cooling weather.

Quadratic regressions were generated considering the influences of temperature, solar insolation, wind, and rainfall. Temperature and solar insolation were consistently significant in this modeling, wind was rarely significant, and rainfall not at all. These models were then applied to evaluate the cooling energy impact of each fault over an entire year. For our

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evaluation, we applied models to the most recent Typical Meteorological Year (TMYx) annual weather data, downloaded from the climate data repository Climate.OneBuilding.org, for the nearest available location to the test facility, the Kennedy Space Center (KSC) weather station.[3]

While the prediction models showed very predictable quality, they sometimes suffered in under predicting energy use at the highest outdoor temperatures, and sometimes over predicting energy use at outdoor temperatures that fall on the curviest part of the polynomial (70-75ºF). Figure 3 plots the actual energy use during Fault 21, 30% Undercharge, versus the hourly polynomial model using outdoor temperature to the second power and solar insolation. The model falls a little short in energy use predations during the warmest days.

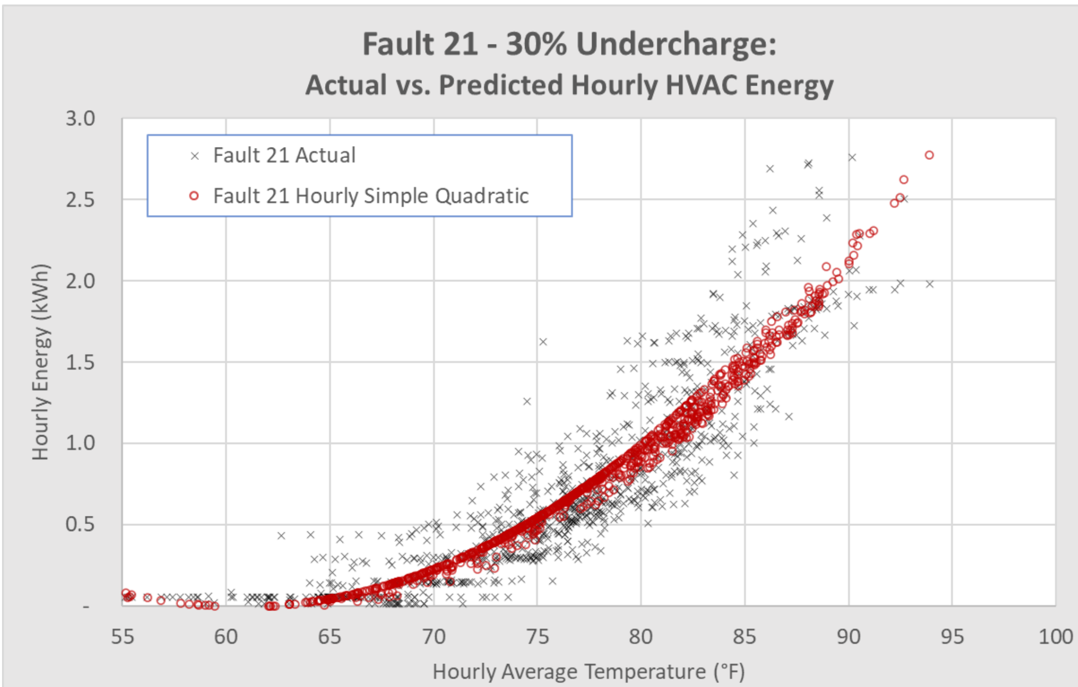

Figure 3. Fault 21 actual hourly energy use vs. polynomial prediction.

While the vast majority of runtime throughout the cooling season occurs at stage 1 (approximately >95% of the time), stage 2 runtime does occur during peak cooling loads. We speculated that the second stage energy use was not always adequately captured with this model and that the predictions might be improved by applying a different model to capture the energy use on days with longer duration stage 2 runtime. We refer to this as the hourly multi-stage model (multi-model).

Looking at 15-minute air handler energy data, we binned the data according to if the system was fully in stage 1, fully in stage 2, or somewhere between. We plotted these results for the west building baseline data in Figure 4, where the blue represents energy when the system stays in stage 1 for the full 15 minutes, orange is energy when the system stays in stage 2 for the full 15 minutes, and yellow is the energy when there is a mix of stage 1 and stage 2 runtime. The

[3] Climate.OneBuilding.Org is the repository of free climate data for building performance simulation. According to Climate.OneBuilding.Org, TMYx files are "derived from hourly weather data through 2021 in the ISD (US NOAA's Integrated Surface Database) using the TMY/ISO 15927-4:2005 methodologies." Currently not all locations have recent data.

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separation according to energy threshold is not perfect but attempts to achieve the best separation among operating scenarios with minimal overlap. We also examined the time of day and temperatures at which the stages were operating. For this dataset, stage 2 operation is observed to be isolated to the hours from 1200 to 1900 and stage 2 runtime was found not to occur below an hourly temperature of 83°F. Significantly, we found that stage 2 operation also does not occur below a daily average temperature of 80°F.

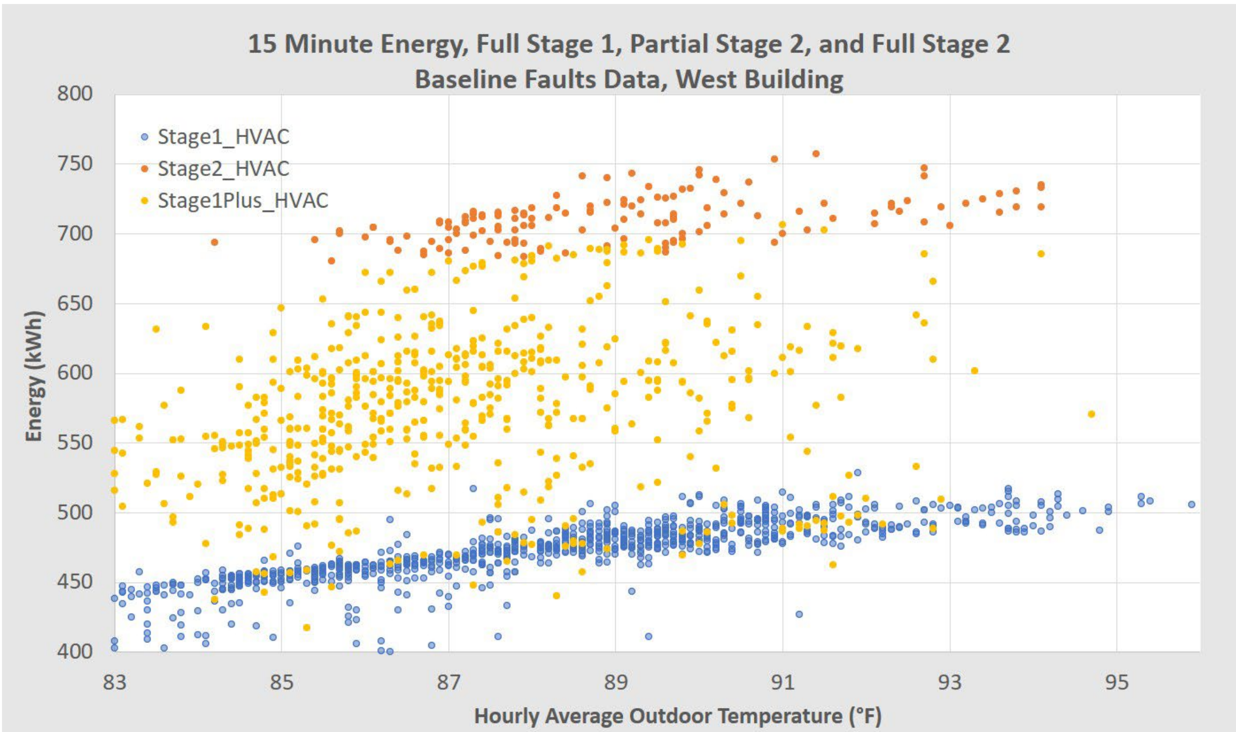

Figure 4. Energy by stage across hourly outdoor temperature for west building baseline data.

The information gleaned from this evaluation was useful in refining our prediction models. It enabled us to bin the data into three groups and develop separate models for each:

1. Hours when stage 2 might occur and the hourly temperature is above a threshold (86F - indicating stage 2 system energy)
2. Hours when stage 2 might occur and an hourly temperature below is set threshold
3. Hours when stage 2 does not occur (indicating stage 1 system energy only)

We examined models for data in each of these bins with varied versions of the two variables that are consistently significant in predicting cooling energy use – outdoor temperature and solar insolation. Table 3 provides a list of the variables tested for the separate model development.

Table 3. Model variables considered for hourly multi-model development.

Base Independent Variable	Derived Independent Variables
Hourly outdoor temperature	Current and prior averages for hours: 1, 2, 3, 5, 11, and 23
	Prior hour averages for hours: 3, 12, and 24
	Prior night average for hours: 3, 5, and 7
	Daily average
Solar Insolation	Current and prior averages for hours: 1 - 8
	Prior hour averages for hours: 1 - 11

Working with the largest set of baseline data, we identified the strongest models to be:

1. For hours between 1200 and 1900 and outdoor temperature equal to or above 86°F:
   Energy = $A + A^2 + B + C^2$, where:
   A = Outdoor temperature
   B = Average outdoor temperature for the prior 12 hours
   C = Average solar insolation for the prior 9 hours
2. For hours between 1200 and 1900 and outdoor temperature below 86°F:
   Energy = $A + A^2 + B + C^2$, where:
   A = Outdoor temperature
   B = Average outdoor temperature for the prior 24 hours
   C = Average solar insolation for the prior 9 hours
3. For hours from 0 to 1200 and 1900 to 2300:
   Energy = $A + A^2 + B + B^2$, where:
   A = Outdoor temperature
   B = Average solar insolation for the prior 9 hours

These basic structures were applied to all fault data, however the threshold temperature between regressions 1 and 2 were modified by fit and ability to predict the highest cooling energy use hours.

Results from the hourly multi-model fault penalty estimations in comparison to those of the side-by-side evaluation results are provided in Table 4. Composite RMSE for the hourly predictions ranges from 0.13-0.18 kWh, with corresponding composite coefficient of variance (CV-RMSE) ranging from 13.2%-25.0%. According to ASHRAE Guideline 14 (ASHRAE 2023), models with a CV-RMSE less than 30% are declared calibrated. For Faults 20 and 25, the limited temperature range over which data was collected did not permit evaluation of the impact of these faults over an entire year. Instead, these faults were evaluated for a partial year by only tabulating cooling energy for hours with outdoor temperatures that fell within the range supported by the data. For Fault 20, this limits evaluation to hotter summer months of approximately June-September resulting in an impact that is higher than what can be expected for an annual evaluation. For Fault 25, the evaluation is representative of most of the year's cooling weather, from approximately March – November. For all other faults the lowest overall temperature with observed cooling, 56.5°F, was the lowest temperature for a TMYx hour with predicted cooling.

Table 4. Hourly multi-model and side-by-side fault penalty estimates.

Fault Name	Side-by-Side Daily Model Regressions @ 80°F	Hourly Multi-Model Regressions - TMYx KSC						
			Composite		Annual/ Partial Year Baseline Cooling Energy (kWh)	Annual/ Partial Year Fault Cooling Energy (kWh)	Annual / Partial Year Cooling Energy Penalty	
	Daily Cooling Energy Penalty (%)	n (days)	n (hours)	Adj R^2			kWh	%
Baseline	N/A	320	7680	0.90	5,509	N/A	N/A	N/A
Duct leak[1]	16.8%	11	264	0.90	2,616	3,280	664	25.4%
Undercharge[2]	3.3%	39	936	0.92	5,509	5,697	189	3.4%
Undercharge, Duct leak[2]	22.4%	48	1,152	0.92	5,509	6,622	1,114	20.2%
Overcharge	8.8%	29	696	0.89	5,509	5,996	488	8.8%
Overcharge, Duct leak	30.4%	40	960	0.91	5,509	7,236	1,727	31.4%
Undercharge, Non-cond.[1]	9.7%	14	336	0.91	4,815	5,527	712	14.8%
Undercharge, Non-cond., Duct leak	38.4%	42	1,008	0.92	5,509	7,745	2,236	40.6%
Non cond., Duct leak	33.8%	53	1,272	0.92	5,509	6,735	1,227	22.3%
Non cond.	12.4%	49	1,176	0.91	5,509	5,997	488	8.9%

[1]Hourly results modified. Annual results are not supported by data due to limited temperature range. Partial year evaluated using temperature range supported by data.
[2]Daily results modified per footnote in Table 2.

Discussion of cooling energy impacts of faults. The modeled results based on monitored data are roughly in line with the simulated results informed by steady state lab testing in NIST's study (Domanski 2014). The simulated heating/cooling results for the Houston climate are most closely comparable to the central FL climate. The individual overcharge and undercharge faults in the FRTF show impacts ranging from 4% - 9% increase in cooling energy use. Of all faults tested, the impacts of the 30% undercharge fault deviate most from the NIST simulated result. While not specifically tested, it appears likely that the 30% undercharge fault is muted in a two-stage heat pump compared to a single stage heat pump, which was used in the NIST study. As previously mentioned, the two-stage heat pump operates at first stage the vast majority of the time, while the full recommended refrigerant charge is designed for full capacity at second stage. While the lack of refrigerant will, at times, result in less cycling at first stage, the larger impact is seen during second stage, with earlier switching from first to second stage compared to the system with full charge, and remaining in second stage for a longer duration. Due to the capacity limitation experienced by the undercharged system, during peak conditions the desired indoor set point cannot be maintained, resulting in loss of comfort. The impact of the overcharge fault is seen relatively constantly, at both first and second stage, as the extra refrigerant places extra load on the compressor resulting in high power consumption. Capacity was not markedly affected by the overcharge fault within the range of test conditions evaluated.

The impact of the non-condensables fault is in the range of 9% - 12% increase in cooling energy use. This fault has some characteristics of both under and overcharge faults, resulting in some loss of capacity, and hence indoor temperature control at peak, and some additional power draw on the compressor.

The duct leakage fault results in the largest impacts of any singular fault, with cooling energy increases ranging from 17%-25%. It is important to note that this energy penalty is representative of a seasonal impact, as the data were collected over a limited temperature range that did not support a full annual evaluation. The duct leakage fault effectively limits the cooling capacity of the heat pump, and results in loss of indoor temperature control during peak conditions. This fault has the largest impact on indoor relative humidity, further adding to a loss of comfort. Duct leakage was balanced on the supply and return sides of system, however, the entering air condition at the evaporator coil has the largest bearing from duct leak impacts. As the duct leak air conditions vary, the cooling performance will also vary. The return leaks were intentionally pulled from the floor area of the attic near a vented soffit. During mid-day summer, this air would be very hot and humid having the largest impact compared to drier air during early spring morning when leakage may reduce cooling load for a few hours. It is worth noting that during several hours of the cooling season, there would be little to no sensible load, but the latent load could still be significant. It is under these conditions where cooling energy impact is small, but comfort control is poorer.

The effects of combination faults are roughly additive. In particular Fault 26 with 30% undercharge, duct leakage, and 50% non-condensables results in a 38% - 40% increase in cooling energy use. The existence of these faults in combination could easily result in many residential systems being replaced with larger systems as the losses to cooling capacity and inability to maintain comfort may be interpreted as the heat pump being undersized.

Conclusions

Results from the controlled lab study show that common installation faults can increase annual cooling energy anywhere from 5-10% for an individual fault, and up to 30-40% when multiple faults are involved including significant duct leakage. While magnitudes of fault induced energy penalties will vary among homes in differing locations with differing envelope, system, and occupancy characteristics, impacts remain significant. Results from the field study show that the prevalence of installation faults is commonplace, even among a sample of systems mostly permitted and installed as part of utility programs. Combined, these results reinforce that systems in most homes are likely consuming more energy than necessary.

Fault intensities measured in the field study were found to be similar to what has been seen in past studies dating back to the 1980's. Only 55% of the RS systems were correctly charged. Most of the systems were undercharged, preventing systems from delivering rated

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cooling capacity and efficiency. The median amount of undercharge for the 10 systems where charge was able to be corrected was 18% based on weight. Evaporator coil air flow was found to be 15% lower than needed to achieve nominal rated capacity. Ducts were found to be nearly 100% leakier than certain new construction benchmarks that consider cfm25/100sqft, severely hindering achievement of rated capacity. Results look a bit better when comparing cfm25 as a % of nominal rated air flow, however, this is in part due to the systems being oversized by an average of 35%. Three quarters of the oversized systems were oversized by more than the half a ton a contractor would need to round up if the cooling load exceeded the cooling capacity of the next smaller system.

QI processes and programs can assist contractors to design and install a system according to industry standard guidance. They have been found to be successful in ensuring at least some critical installation parameters are achieved, specifically proper refrigerant charge. This has required a combination of classroom and field training, availability of real-time technical assistance, and random quality assurance checks that require the contractor to make a return visit if the charge is not set correctly in the first place. However, program administrators have found that there are limits on what QI programs can require of contractors, while still enabling them to be cost competitive in markets where QI is the exception, rather than the norm. To transform the market towards QI, requirements for QI need to be built into code and rebate programs, along with the training, jobsite support, and quality control needed to support contractors. In jurisdictions that currently require some levels of QI, program evaluations may highlight where improvements can be made.

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