SCE Comprehensive Mobile Home Program
HVAC QM: Data Analysis - Phase II
Volume 1: Exploratory Data Analysis and Field Inspections

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1. Introduction and Summary of Objectives

The CMHP HVAC Quality Maintenance Program seeks to optimize packaged and split system HVAC units in manufactured and mobile homes as part of a more comprehensive direct install program. The QM measure consists of multiple treatments related to duct-work and HVAC unit optimization. Air conditioning systems must be in working order to be eligible for the program; repair of non-functioning units is not covered in this program. Services are intended to improve the energy efficiency and performance of systems operating in "sub-optimal" conditions.

ADM was contracted by SCE to perform a detailed review of the data and data collection methods implemented by Synergy for the Comprehensive Mobile Home Program. The data collected on-site is entered into a database maintained by Conservation Services Group (CSG) called EM-HVAC. EM-HVAC processes the on-site measurements and uses it to validate the data, model system performance, and suggest repair services. This study therefore also reviewed some of the algorithms employed by EM-HVAC for internal consistency, for sensitivity to error in measured inputs, and for their usefulness in estimating program energy impacts.

1.1. Study Objectives

Currently the program tracks a wide range of data for each site, however; the quality of the data is paramount for an accurate assessment of system health and energy efficiency opportunities. The operating data collected by the program is done according to the guidelines presented in the "SCE Quality Maintenance Program Guide Specification and Guidelines." Since these guidelines were created in the context of HVAC system types most commonly found in single family homes, there also exists concern regarding their appropriateness for mobile homes. The objectives of this research are to:

1. Determine the reliability of the current measurement data and recommend ways to improve on-site data collection (particularly for airflow and air temperature measurements).
2. Identify inconsistencies in current program data (Phases I and II) and recommend how to improve consistency in the data through improved data collection technique or equipment.
3. Use the current program data (Phases I and II) to estimate QM/QC and Brush-less fan motor energy impacts (report overall, Phase I only, and Phase II only)

This report addresses the first two objectives in the list above. However; a separate volume was prepared to address observed energy impacts for the QM and Brush-less fan motor measures.

1.2. Executive Summary

ADM reviewed the CMHP program tracking data across both of its Phase I and Phase II data-sets (which collectively represent all mobile homes for which the program is claiming energy savings). Data were first reviewed by exploring their distributions and variances - noting any outlier data points or non-physical observations. Then fields were compared with each other to explore the internal consistency of the data set (e.g. do the various measurements taken at a particular site agree with one another to tell a similar story). In order to facilitate this analysis ADM organized the data into one of the following general "classifications":

1. Direct Measurement
2. Calculation - Measurement Based
3. Calculation - Theoretical Assumptions
4. Categorical Information
5. Site Tracking Information

The *Categorical Information* was particularly useful as it allowed ADM to analyze the data across various categorical dimensions and explore how specific factors contributed to excessive variance, outlier observations, and/or non-physical magnitudes. Factors used to categorize data included geographical regions, field technician, measurement methods, Program Year, etc. Table 1.1 provides a complete list each of the categorical data fields considered in this analysis.

Table 1.1: List of Categorical Fields

Field Name	Definition
Airflow.Method	Name of method used to measure airflow rate
Airflow.Operational.Mode	Track coil conditions at time of airflow measurements
Site.City	City in which site is located.
Tech.Id	Unique ID for technician providing service on-site.
Job.Id	Unique Identifyer for a particular Job Site.
Activity	Differentiates whether current test is pre-retrofit (Test-in) or post-retrofit (Test-out).
Test.Result	Automated value generated by EM-HVAC indicating whether or not warning/fault flags are present in the entered data. Values are "Pass" or "Fail".
ChronDate	Combines date and time fields
PHASE	Program phase as identified by program delivery year (2012="phase 1" & 2013="phase 2")
Equipment.SEER	Manufacturer's rated efficiency in SEER
Total.Capacity.in.Tons	Manufacturer's rated capacity
System.Type	Type of HVAC system. There are (5) categories represented in the data: 1) AC Split, 2) AC Package, 3) Heatpump Split, 4) Heatpump Package, 5) Ground Source Heatpump Split.
OU_Compressor.Type	Type of Refrigerant compressor. Two categories are found in the data: 1) Scroll, and 2) Reciprocating.
Multi.Stage	Flag for systems with multiple stages.
Refrigerant.Type	Identifies the name of the refrigerant used in the current system.
Metering.Device	Type of metering device (e.g. Thermal Expansion Valve or Fixed Oriface).
Liq.State	Estimated thermodynamic state of the refrigerant at the liquid line (based on measured temp and pressure). Should always be subcooled liquid.
Suc.State	Estimated thermodynamic state of the refrigerant at the suction line (based on measured temp and pressure). Should always be superheated vapor.

The set of fields classified as *Direct Measurement*, *Calculation - Measurement Based*, and *Calculation - Theoretical Assumptions* were further categorized into the following groupings: 1) Air-Side Metrics, and 2) Refrigerant-Side Metrics. These groupings organize the data according to the physical processes they represent and allows for individual analysis of these physical processes. The *Air-Side* and *Refrigerant-Side metrics* each reflect a different aspect of the overall refrigeration cycle and must therefore corroborate with one another. Once categorized, the data were explored using both graphical and statistical techniques. The following sections report ADM's findings and recommendations for the QM program. The body of this report present our findings in detail as they relate to the refrigerant-side and air-side metrics before comparing them one to the other. A detailed summary of our findings and recommendations can be found at the end of this report.

1.2.1. Study Findings: Data Reliability

This study found that the data collected from the refrigerant-side measurements do not reliably correlate with the data collected by the air-side measurements. Several observations were made which indicate that the air-side measurements of temperature and airflow are the leading cause in this discrepancy. ADM also found some instances of non-physical sets of measurements in the data (e.g. dry-bulb and wet-bulb measurements which cannot physically occur, airflow measurements at or near zero CFM, negative sub-cooling and super-heat measurements, etc.). These observations; however, represented only a small number of sites. In general the individual measurements were found to be of appropriate magnitude and within reasonable ranges. Data reliability issues were generally correlated to sensor placement not representing the intended physical parameter (temperature measurements) and large uncertainty in the measurements (Airflow).

Return air and supply air temperature measurements are currently taken inside the homes at the return and supply registers. These locations are not ideal for measuring the temperature of the air before and after the evaporator coil in mobile homes and add uncertainty to the calculated coil load. The reasons are discussed at length in Section 5.1 (Findings:Air-side Metrics).

The system airflow is currently measured at the return grill using a hand held anemometer. Measurements are only taken in homes with a down-flow configuration (in all other configurations technicians apply a default assumption of 400 cfm/ton). ADM found that in many instances there was additional airflow not accounted for in the current measurements due to leaks/infiltration into the airhandler closet and/or outside ventilation air plumbed directly into the airhandler cabinet. This is discussed in further detail in Section 5.1 (Findings:Air-side Metrics).

1.2.2. Study Findings: On-Site Data Collection

ADM found that the technicians are generally consistent in their data collection practices and follow the methods specified per program training. ADM was unable to observe/verify equipment calibration procedures during our ride-along inspections; however, in ADM's process interviews Synergy staff indicated that data collection equipment appropriately calibrated. During our ride-along inspections the data collection equipment looked to be well maintained and in good order.

While the data collection equipment was found to be of sufficient accuracy and resolution to affect the intended measurements, ADM found that the current air-side data collection practices do not sufficiently capture the intended system operating characteristics to calculate accurate air-side loads/efficiencies. This is discussed at length in Chapter 2 (Airside Measurements and Algorithms) and in Chapter 5 (Findings and Recommendations).

1.2.3. Study Recommendations

In this analysis ADM reviewed CMHP program tracking data from Phases I and II. This data included all measurements made by technicians on-site as well as data fields tracked by EM-HVAC. Based on our analysis of program data, and informed by our ride-along inspections, ADM recommends the following to improve program data reliability. Note that this list provides a high level summary our recommendations. Each recommendation is discussed at length in Section 5.3

1. Consider combining the duct-sealing and QM measures to ensure that the home's duct-work is not introducing uncertainty into the measurements and also to improve measure impacts.

2. Move the locations of the return and supply air temperature measurements to locations closer to the evaporator coil (specific locations are suggested in section 5.3). This will enable more representative measurements. In order to facilitate this recommendation we suggest that synergy's STS data acquisition system be re-implemented in enabling remote measurement.

3. Replace current airflow measurement technique/tools with the methods suggested in Section 5.3. Two different methods are suggested to accommodate different system configurations (up-flow vs. down-flow) and facilitate measurement in up-flow system configurations (airflow in up-flow systems is currently not measured).

4. Adjust current assumption of 400 CFM/Ton for system airflow down to 350 CFM/Ton when airflow measurements cannot be performed.

5. Remove select data fields from program tracking (and where present on field data collection forms) as they are not used.

2. Air-Side Measurements and Algorithms

The term *Air-Side Metrics* refers to measurements, calculations, and categorical data fields describing the thermodynamic energy balance and heat transfer processes in the the air stream as it moves across the evaporator coil. The EM-HVAC data fields which are relevant to these metrics are listed in Table 2.1. While Air-Side measurements typically present the most tractable method of directly measuring system load (and when combined with electric power data system efficiency) they can be subject to significant uncertainty. This is particularly true for any attempt to measure, in-situ, the volumetric flow rate of air. Evaporator load is calculated using Equation 2.1.

$$\dot{Q} = \dot{V} * 60(\rho_{Before} * h_{BeforeCoil} - \rho_{After} * h_{AfterCoil}) \tag{2.1}$$

where:

- $\dot{Q}$ is the Evaporator coil load [BTU/Hr]
- ρ is the density of the air $[Lb/ft^3]$ before and after the coil. This is derived from tables of the thermophysical properties of air across various temperatures and pressures.
- $\dot{V}$ is the volumetric flow rate $[CFM]$ and is measured directly (or when not possible an assumed *theoretical* value).
- h is the enthalpy of the air on either side of the coil $[BTU/Lb]$. This is calculated from Wet bulb and Dry bulb measurements of the air.

Equation 2.1 can be broken down into two essential components: 1) the change in specific energy of the air as it moves across the coil, and 2) the volumetric flow rate of the air stream through the coil. This chapter examines the field measurements and the subsequent calculated data fields from EM-HVAC as they relate to these two components of Equation 2.1. Then it explores the overall system performance as predicted by the Air-Side data.

2.1. Component 1: Specific Energy of The Air - Measurements and Calculations

Enthaply is a thermodynamic property of a substance and must be calculated (e.g. it cannot be directly measured)or looked-up in a standard thermodynamic tables. For air, standard psychometric equations can be leveraged for this purpose as well. The calculation is relatively straightforward given Dry bulb Temperature and Wet bulb Temperature are available. The accuracy of calculated enthalpy values is therefore subject to the propagation of uncertainties from these two field measurements (and is particularly sensitive to the Wet bulb Temperature measurement). Dry bulb Temperature and Wet bulb Temperature measurements are made in both the supply air and return air streams in order to calculate the change in enthalpy of the air-stream as it goes over the evaporator coil. Figure 2.1 summarizes the range and distribution of the air temperature measurements across the entire data-set.

Note that in Equation 2.1 each enthalpy *before* and *after* the coil is multiplied by the density of the air at the given conditions. While this is not tracked independently by EM-HVAC, descriptions of some data fields indicate that appropriate adjustments are made in air density for altitude as they apply to the enthalpy calculations. It is also interesting to note that the return air temperature measurements plotted in Figure 2.1 exhibit a tighter grouping (e.g. less variance) compared to the supply air temperature measurements - even though the same instruments are used to measure both. Each set of measurements contain a number of outliers. However; the supply air measurements (both wet bulb and dry bulb) show the largest range. Furthermore, the range of observations seen in these data seem to indicate that a sub-set of tests were performed when the units were not operating under valid testing conditions (as specified by the QM Program Guide) and some magnitudes, particularly in the supply air measurements, suggest that their validity is suspect. The following sections detail our findings with regards to the temperature measurements and how they relate to the overall predicted system performance.

Table 2.1: List of Air-Side Metric Fields

Field Name	Definition
Altitude	Altitude of site.
Return.Air.Dry.Bulb	Dry Bulb temperature measurement of the Return air
Return.Air.Wet.Bulb	Wet Bulb temperature measurement of the Return air
Supply.Air.Dry.Bulb	Dry Bulb temperature measurement of the Supply air
Supply.Air.Wet.Bulb	Wet Bulb temperature measurement of the Supply air
Measure.cfm	Technician reported CFM. Either physically measured, or estimated (usually nominal value) if technician is unable to properly measure airflow due to system configuration or impediments
Estimated.Airflow	Estimated from Compressor maps and measured air temperatures - adjusted for altitude. Compressor Map capacity (driven by actual measured pressures) divided by the change in evaporator air enthalpy (equals mass flow in lbs/hr) then converted to CFM
Goal.Airflow	Based on Total cfm setting above. Based on technician reported equipment setting or default nominal airflow if not provided. All values are identical to "Total.cfm.setting" except for (1) observation.
Measured.cfm.ton	Equal to division of the "Measure.cfm" field by the "Total.Capacity.in.Tons" field.
Estimated.cfm.ton	Equal to division of the "Estimated.Airflow" field by the "Total.Capacity.in.Tons" field.
Airside.Capacity	Capacity of system based on measured airflow and air temepratures. Measured or default CFM converted into Mass Flow (lbm/hr) multiplied by the change in air enthalpy (BTU/lb) - adjusted for altitude
Airside.EER	Estimate of tested efficiency based on measurements. Based on measured (or default) airflow, air temperatures and system power
Enthalpy.Actual	Measured change in evaporator air enthalpy. Actual change in enthalpy in BTU's/lbm of air across evaporator (heat absorbed by refrigerant) - adjusted for air density.
Temperature.Split.Actual	Measured sensible temperature change across evaporator. Delta from measured air temperatures across the evaporator coil

2.1.1. Return Air Temperature Measurements

The return air dry bulb and wet bulb temperature measurements were taken on-site using a pair of Testo 605-h1 thermohygrometers. Technicians were trained to place the thermohygrometer probe in the front face of the air handler cabinet return grill (See Figure 2.2). During ADM's ride-along inspections it was noted that some technicians assumed that any measurement of the surrounding indoor ambient air would be fine, and In one case the sensor was placed on top of the cabinet where it was subject to additional heating from the fan motor heat and potentially the furnace pilot flame. These observations represented a minority of site and we found that most observed measurements were consistent with the training. Some descriptive statistics for the return air temperature data can be found in Table 2.2.

Table 2.2: Descriptive Statistics of Return Air Data

Measurement	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Dry Bulb	6,630	76	4	76	61	93	32	-0.11	-0.4
Wet Bulb	6,630	60	4	60	47	76	28	-0.04	-0.4

Figure 2.3 illustrates how the return air dry-bulb temperature measurements are distributed. The wet-bulb measurements show similar structure and are therefore not shown here. Both the dry bulb and wet-bulb measurements appear to be normally distributed with a preponderance of measurements landing within a range of 70 $^{\circ}\mathrm{F}$ to 80

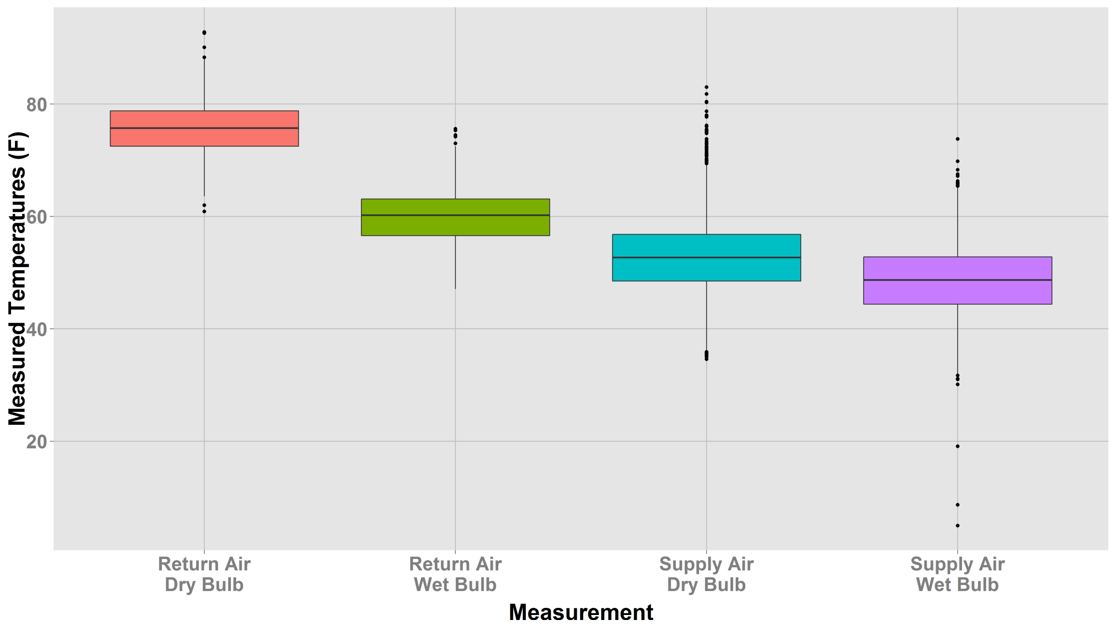

Figure 2.1: Box-Plot Summary of Air Temperature Measurement Data

Figure 2.2: Typical Return Air Temperature Measurement

°F for dry bulb measurements and 55 °F to 65 °F for the wet bulb measurements. There does appear to be some structure to the magnitudes of the measurements with respect to the time of year in which they are taken. This is demonstrated in the lower right panel in Figure 2.3. This structure is most likely due to the diurnal cycle of the average outdoor air temperature throughout the year. Some of the structure may also be due to differences in site locations as the program was implemented.

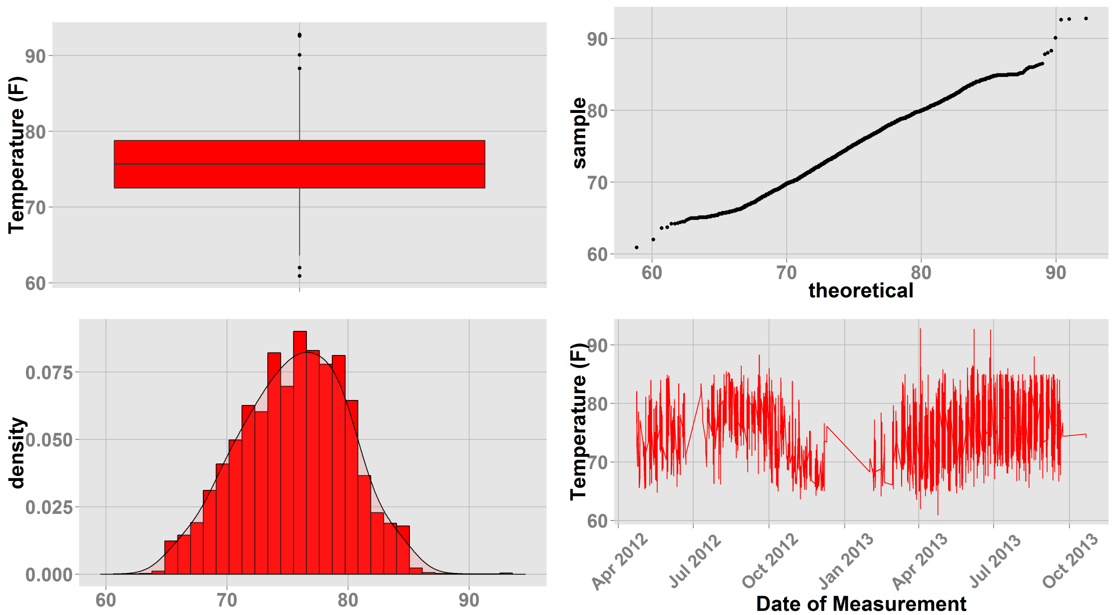

Figure 2.3: Detail of Return Air Dry Bulb Measurements

It was noted earlier that a sub-set of tests seemed to have been performed when the units were not operating under valid testing conditions. According to the Warning/Fault code matrix found in the QM program guide, testing must be performed between 70 °F to 85 °F and 50 °F to 80 °F for the dry bulb and wet bulb temperatures respectively. However, It can be seen in Table 2.2 that the dry bulb temperatures range from 60.9 °F to 92.8 °F and that the wet bulb temperatures range from 47.1 °F to 75.6 °F. ADM found that 537 (or 16 %) of the Job Id's exhibit return air temperature measurements outside of the testing criterion set forth in the QM Program Guide.

Additional information can be gleaned by sub-setting the data according to categorical factors. This allows us to test for possible influences and/or confounds which impact data quality. Examples of such factors are discussed at the beginning of this chapter and their review will not be discussed at length. Instead, we have identified a sub-set of factors which either exhibited interesting behavior or exemplified the overall findings. Figure 2.4 illustrates one such cross-section in which distributions of the return air wet-bulb and dry bulb are presented across project phase and Tech Id. These variables are particularly interesting (and important) because 1) measurement technique can improve or degrade over time, and 2) the degree of meticulousness by which measurements are performed tends to be impacted by the personalities and habits of the individuals collecting the data. Figure 2.4 exemplifies the differences in the air temperature measurements made by each technician. The distributions and means for the dry bulb measurements are more consistent than the wet-bulb measurements in Phase II. Both measurements seem to show similar variances in Phase I - though it should be noted that there are 5 times more measurements and almost twice as many technicians present in the Phase II data compared to Phase I.

When reviewing the measurements made by individual technicians ADM observed that most showed a normally distributed behavior. However; two technicians were found whose measurements exhibit a large negative skew (exemplified by Figure 2.5). In both cases there does not appear to be any dependency on the measurement with

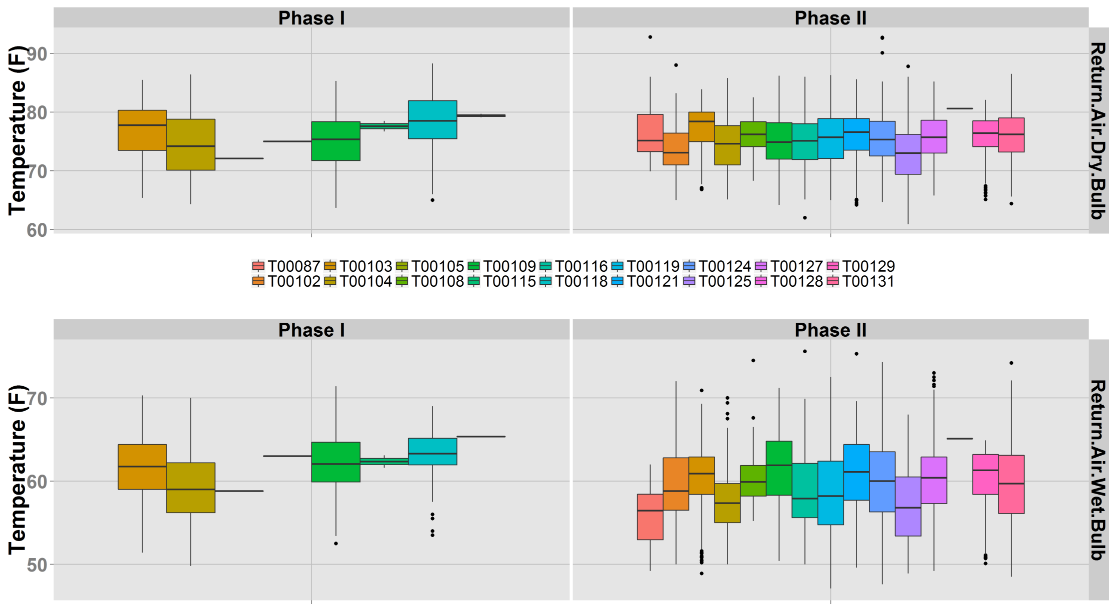

Figure 2.4: Cross-Section of Return Air Measurements by Program Phase and Tech Id

time (lower right panel), but the measurements are skewed heavily towards 80 °F. One technician was also found to have two pronounced means in their measurements which correspond to measurements made in Phase I and Phase II respectively (see Figure 2.6).

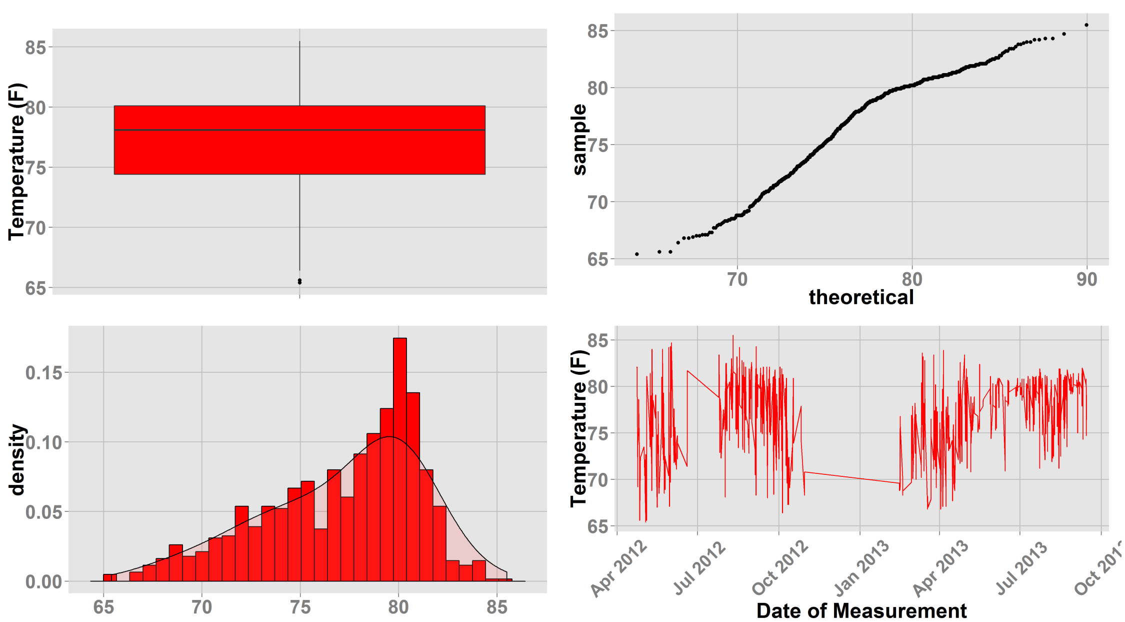

Figure 2.5: Detail of Return Air Dry Bulb Measurements Made by Technician: T00103

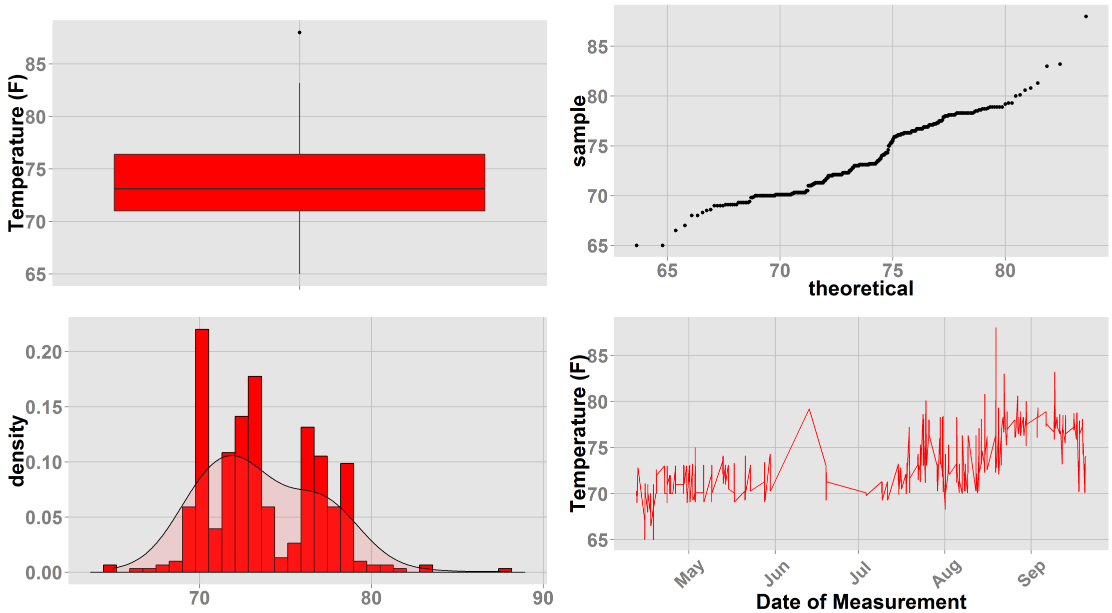

Figure 2.6: Detail of Return Air Wet Bulb Measurements Made by Technician: T00102

2.1.2. Supply Air Temperature Measurements

The supply air dry bulb and wet bulb temperature measurements were taken on-site using Testo 605-h1 thermohygrometers. Technicians were trained to place the thermohygrometer probe in the supply duct closest to the air handler. ADM observed this to be standard practice in all but a few instances where measurements were not taken from the closest supply. Figure 2.7 demonstrates a typical placement of the Testo 605-h1 thermohygrometer. Some descriptive statistics for the supply air temperature data can be found in Table 2.3.

Table 2.3: Descriptive Statistics of Supply Air Data

Measurement	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Dry Bulb	6,630	53	6	53	35	83	48	0.3	0.4
Wet Bulb	6,630	49	6	49	5	74	69	-0.1	0.4

The supply air temperature measurements behave similarly to the return air temperature measurements in that they are normally distributed and display some dependence in magnitude with time. Figure 2.8 demonstrates how the wet-bulb temperature measurements are distributed. Again, the wet-bulb and dry bulb temperature measurements show very similar structures. Dry bulb temperatures range from 34.6 $^{\circ}\mathrm{F}$ to 83 $^{\circ}\mathrm{F}$ and that the wet bulb temperatures range from 5 $^{\circ}\mathrm{F}$ to 73.8 $^{\circ}\mathrm{F}$. By comparing the standard deviations of the supply air temperature measurements in Table 2.3 with those of the return air temperature measurements (Table 2.2) one can see that there is more variance in the supply air temperature measurements. This was visually demonstrated in Figure 2.1. Furthermore, the number and magnitude of outlier measurements are exaggerated in the supply air temperature measurements. Within the supply air temperature measurements the wet-bulb data shows less variance than the dry-bulb temperature measurements.

The supply air data was sub-set according to program phase and Tech Id in Figure 2.9. It can be seen that supply air data exhibits much the same behavior as the return air data though variances are more exaggerated. In particular there are several exceedingly low observations of wet-bulb temperature measurements which spurred ADM to check all wet-bulb measurements against their theoretically lowest values (using their corresponding

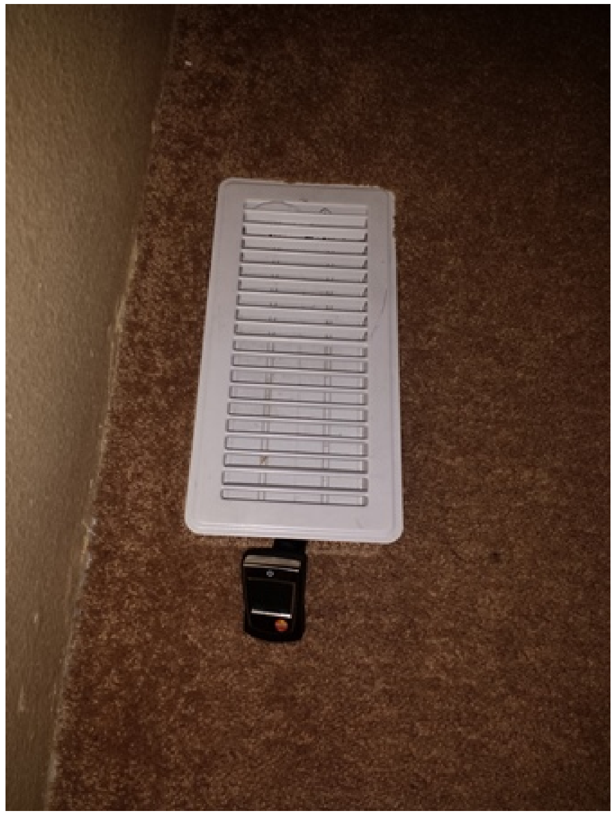

Figure 2.7: Typical Supply Air Temperature Measurement

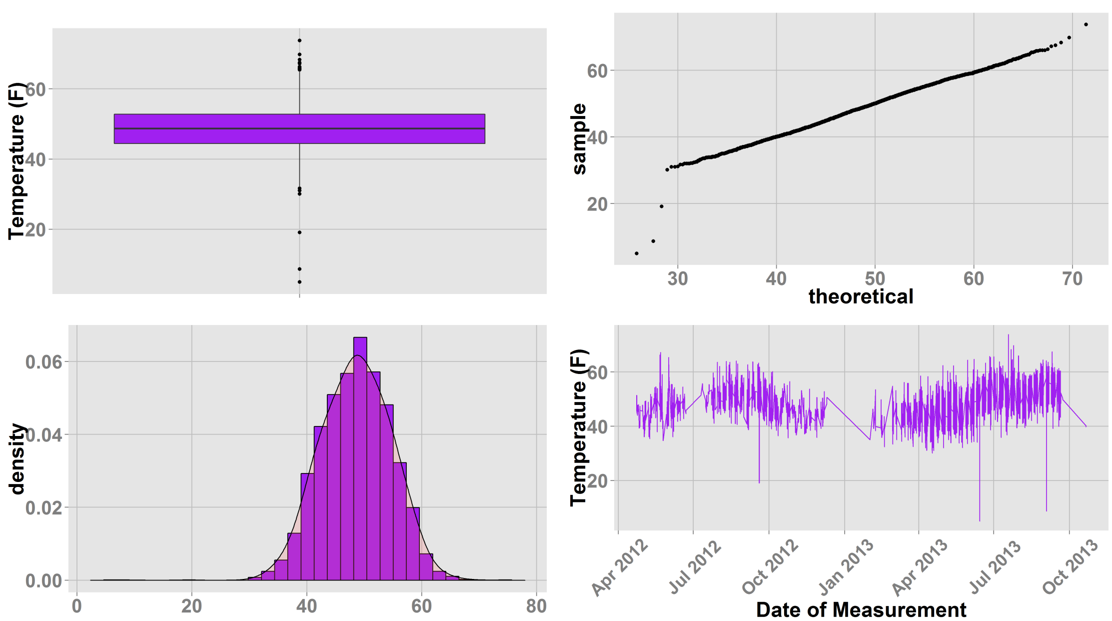

Figure 2.8: Detail of Supply Air Wet Bulb Measurements

dry-bulb measurements and assuming a relative humidity of 0%). ADM found (4) non-physical measurements for which the wet-bulb temperature was lower than physically possible for the given dry-bulb. In a similar vein, ADM also identified (2) tests in which the supply air temperature (dry-bulb) was greater that the return air temperature (dry-bulb) while the unit was reported to be in active cooling mode. If the units were indeed in cooling mode these measurements are non-physical.

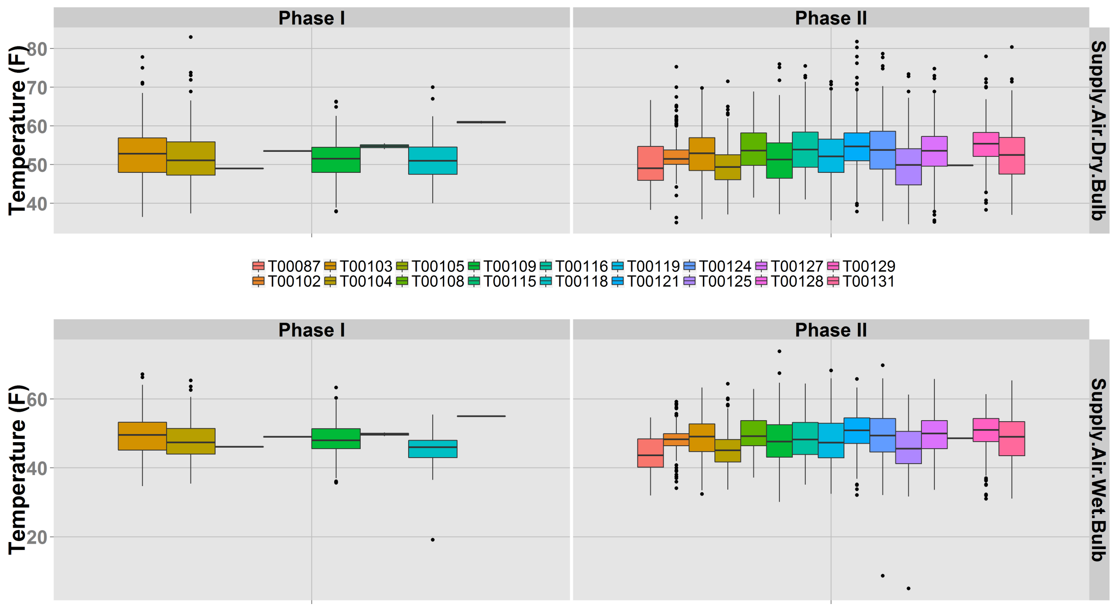

Figure 2.9: Cross-Section of Return Air Measurements by Program Phase and Tech Id

2.1.3. Enthalpy Calculations

The *enthalpy* (h), or specific energy, of the air-stream has units of $[BTU/Lb]$ and is calculated by combining the dry-bulb and wet-bulb temperature measurements. Standard psychrometric formulas and/or tables can be used to calculate enthalpy from dry-bulb and wet-bulb temperatures in which each represent the sensible and latent energy components respectively. In an air-side load measurement the load is directly proportional to the difference in enthalpies between the air preceding the evaporator coil and the air leaving the evaporator coil (See Equation 2.2). Note that the protocols in EMHVAC make the assumption that return air temperature measurements are representative of the air preceding the coil ($h_{BeforeCoil}$) and that the supply air temperature measurements are representative of the air leaving the coil ($h_{AfterCoil}$). Differences between the measured return air temperature and the actual temperature of the mixed air before the coil will introduce additional error in this calculation.

$$\Delta h = (h_{BeforeCoil} - h_{AfterCoil}) \tag{2.2}$$

Calculated enthalpy values are particularly sensitive to the wet-bulb temperature measurement. Take for example the following set of hypothetical temperature measurements: 70 $^{\circ}$F DB, and 67 $^{\circ}$F WB for which the enthalpy is calculated to be 30.7 BTU/Lb. If we assume a $\pm$2% measurement error in each temperature, there is only a negligible impact on the enthalpy when varying the dry bulb temperature ($\pm$2% results in approximately $\pm$1.5 $^{\circ}$F). However, if we account for the error in the wet-bulb measurement then the final enthalpy can vary between 29.7 and 31.7. Thus, while a $\pm$2% error in the dry-bulb measurement is negligible, the same error in a wet-bulb measurement is exaggerated into a $\pm$3% error in the enthalpy. When calculating the differential enthalpy across the evaporator the above errors are further exaggerated since two sets of measurements (and their errors) are combined. This is important to note as the above propagation of errors contribute a significant degree of uncertainty to the air-side load measurements recorded in EM-HVAC.

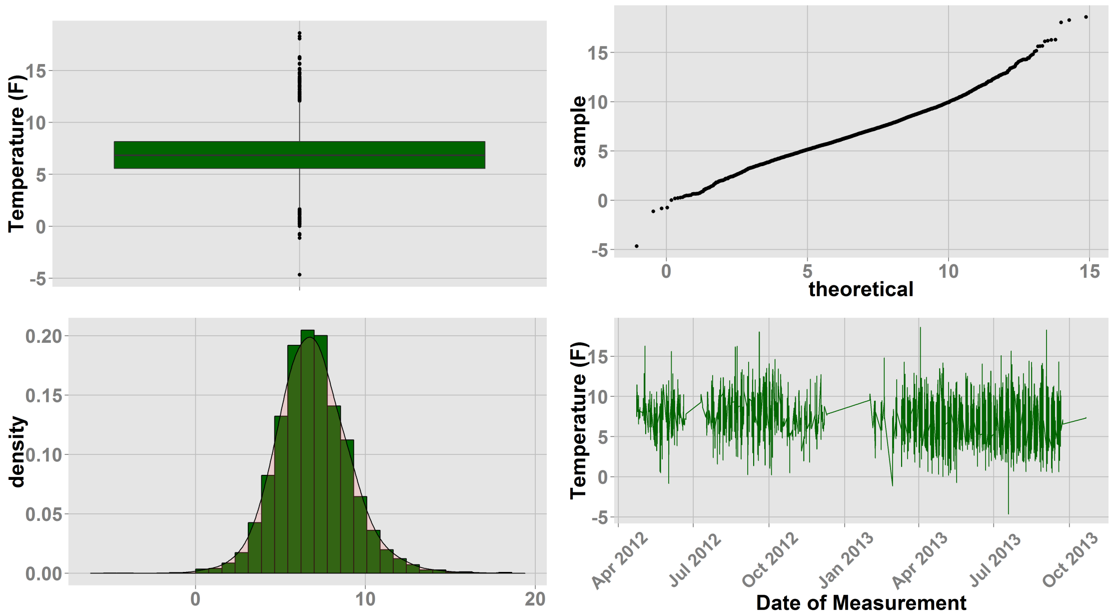

Figure 2.10: Detail of Delta Enthalpy Calculations in EM-HVAC

ADM reviewed the enthalpy data in EM-HVAC by independently calculating the enthalpy using dry-bulb and wet-bulb temperature measurements using psychrometric formulas which can be found in the ASHRAE Fundamentals Handbook. EM-HVAC data does not maintain fields for return air and supply air enthalpies individually, though it does record the change in enthalpy (or delta enthalpy) across the coil. ADM could therefore only compare the delta enthaply calculated in EM-HVAC against our own calculations. Figure 2.10 demonstrates the distribution of delta enthalpy calculations in EM-HVAC, and the results of this comparison are shown in Figure 2.11. It can be seen there that the majority of EM-HVAC observations can be reproduced within a reasonable range though EM-HVAC data is missing the exceedingly large delta enthalpy values observed in ADM's calculations (see the left panel in Figure 2.11. Upon closer review, these unreasonably large delta enthalpy values (as per ADM's calculations) result from non-physical supply air temperature measurements (discussed in the previous section). These enthalpy observations are not present in the EMVAC data and represent the key discrepancies between ADM's calculated enthalpies and those recorded in EM-HVAC. Since there are no fields in EM-HVAC recording enthalpy individually for the supply and return stream it is difficult to pin-point an exact cause for this discrepancy though it likely stems from the algorithms used in EM-HVAC to calculate enthalpy from the dry-bulb and wet-bulb measurements.

Both ADM's calculations and EM-HVAC records indicate 4 observations in which the delta enthalpy is negative. For such observations to be correct, the system would either need to be: 1) in heating mode or 2) humidifying the air. Of the (4) negative observations, there was only one case in which the supply air temperature was hotter than the return air temperature. In this instance the return air was 79 $^{\circ}\mathrm{F}$ and the supply air 83 $^{\circ}\mathrm{F}$. Given the temperature magnitudes, it is unlikely that the unit was heating. Instead it is more likely that the values were transposed during data entry, or that there is significant error in these temperature readings. Table 2.4 lists the supply and return air temperature measurements for each case of negative delta enthalpy.

Reviewing Table 2.4 it can be seen that the condenser unit was operating (per values in columns *Cond. Ph1* and *Cond. Ph2*) while the dry-bulb temperature data shows a marginal temperature difference. In each case the supply air wet-bulb temperature was higher than the return air wet-bulb temperature which (if accurate) indicates that the units were humidifying the air-stream. ADM reviewed this assumption by calculating the return air dew

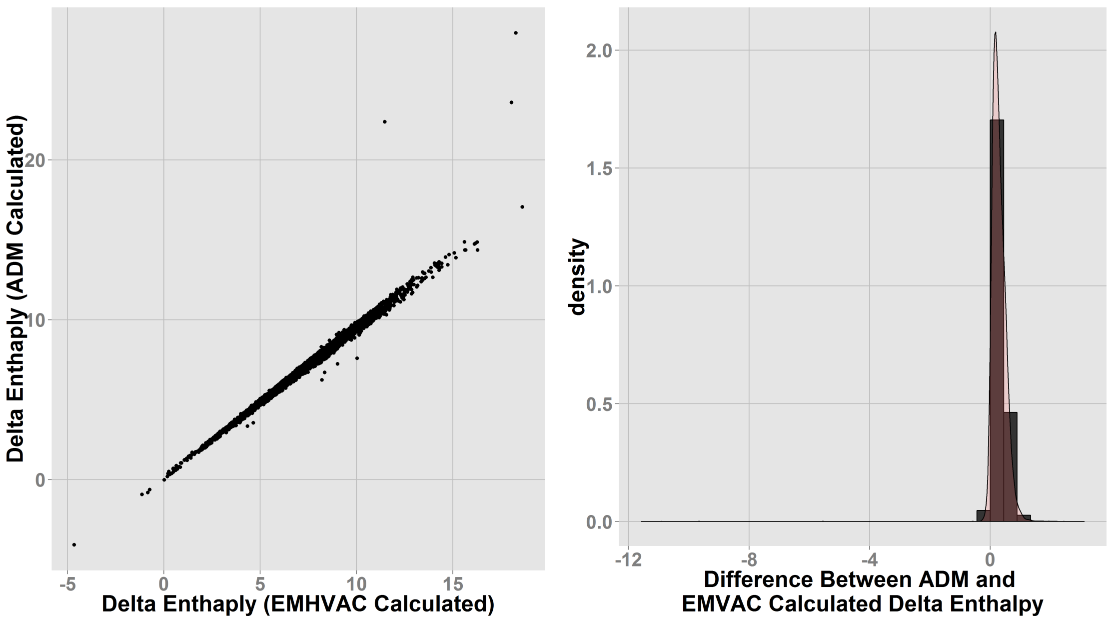

Figure 2.11: Review of Delta Enthaply Calculations

Table 2.4: Listing of Negative Delta Enthalpy Mesurements

Case	Return DB [F]	Return WB [F]	Supply DB [F]	Supply WB [F]	Cond. Ph1 [Amps]	Cond. Ph2 [Amps]	dh [BTU/Lb]
Case 1	67	55	65	56	12	13	-0.74
Case 2	66	53	57	55	10	11	-1.14
Case 3	79	69	76	74	12	12	-4.66
Case 4	79	64	83	65	10	10	-0.83

point temperatures and compared them against the corresponding apparatus dew point temperatures (estimated per field *ET.Goal*). In all instances the evaporator coil is colder than the dew point of the return air. Thus the units should be de-humidifying the air. It is concluded therefor that these observations are likely an artifact of error in the placement of supply and return air temperature measurements.

ADM worked with the developed to understand the algorithms used by EM-HVAC at a high level, though the specifics of their application was not reviewed. ADM was able to re-produce the delta-enthalpy values recorded in EM-HVAC with reasonable consistency and found that most observations in the EM-HVAC data are slightly higher than what ADM calculated for the same inputs (though there is some scatter). This may be due to differences in the formulas/constants applied by ADM and EM-HVAC to calculate enthalpy from dry-bulb and wet-bulb temperatures. It is also likely that EM-HVAC pre-processes some measurements which is evidenced by the lack of outlier delta-enthalpy observations in EM-HVAC.

2.2. Component 2: Volumetric Air Flow Measurements

Accurate airflow measurements are very important when calculating an air-side load and system efficiency. However, they are also very difficult to perform well. This is because the airflow profiles being measured are non-laminar, non-steady state, and characterized by significant velocity gradients. Currently, Indoor fan airflow is measured with a Testo 417 vane anemometer. Technicians were trained to measure the return air grill area and input it into the digital anemometer. They then perform a traverse with the digital meter to make multiple measurements across the return grill. The meter averages the air velocities and converts it to volumetric flow using the measured grill area. A typical airflow measurement is shown in Figure 2.12.

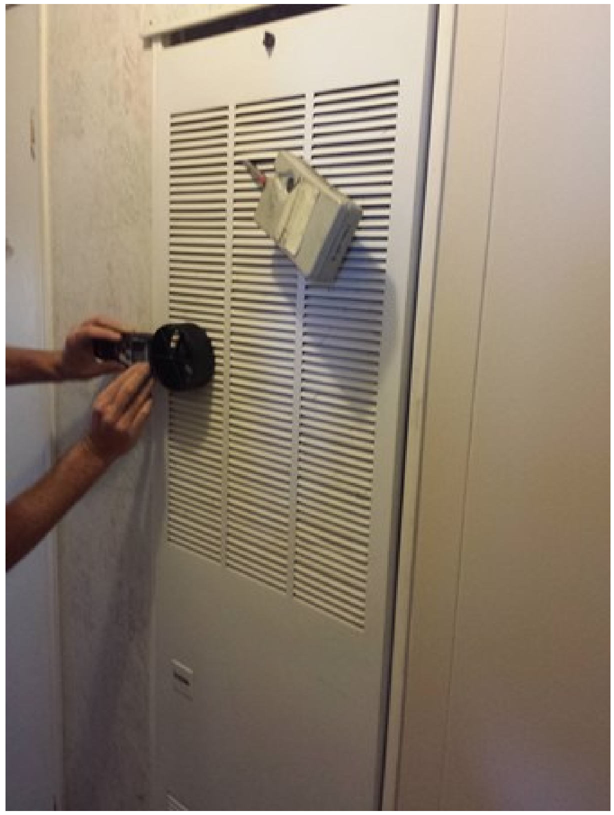

Figure 2.12: Typical Volumetric Air-Flow Measurement

This measurement was only possible on the *down-flow* style air handler units. Actual airflow measurement did not occur on up-flow units due to the fact the return air is coming from under the house and access to the airflow to be measured by vane anemometer is not possible. ADM observed that airflow measurements were made consistently according to the technicians' training. As a notable exception, at one site an additional return grate had been cut into the closest above the door which was not included in the area measurement. Several sources of error were identified which included:

1. Instances in which air leakage around the closet door itself would likely be non-negligible
2. Several systems visited by ADM had outdoor air ducted into the fan cabinet that was not taken into account
3. Systems with outdoor access doors whose leakage is not accounted for

When technicians were unable to directly measure air-flow (e.g. up-flow configured systems) it was estimated by assuming the system will flow 400 CFM/ton. This assumption accounts for a significant portion of observations in the EM-HVAC data and is based on expected flow rates in systems common to Single Family residences. Figure 2.13 illustrates the air-flow measurement data found in EMHVAC while Table 2.5 provides some descriptive statistics.

The *Measure.cfm* and *Goal.Airflow* fields both show significantly less variation than the *Estimated.Airflow* field. This is largely due to how these fields are populated in EM-HVAC. The *Measure.cfm* field represents data collected on-site and entered by the technician. Its value is either a measurement made with a wind-vane anemometer traverse, or defaulted to 400 CFM/Ton at the technician's discretion. The *Goal.Airflow* field is identical to the *Total.cfm.setting* field in all observations except for one. This field represents the expected CFM indicated by the fan speed settings. 93% of these observations are defaulted to 400 CFM/Ton. The *Estimated.Airflow* is quite different than either of the others in that it is a calculated CFM based on refrigerant side measurements and applied

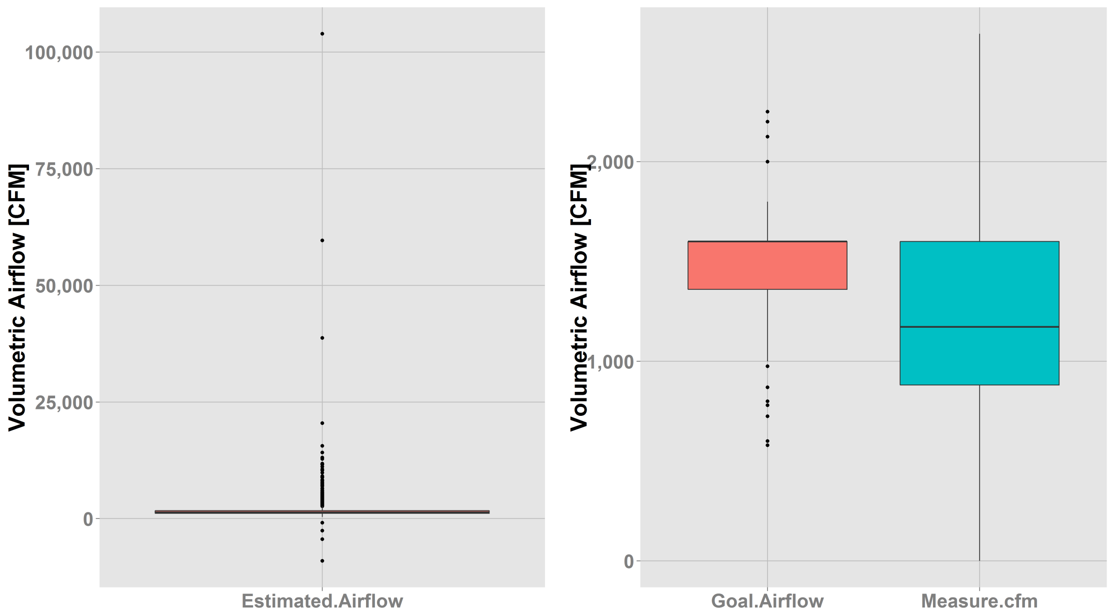

Figure 2.13: Box-Plot Summary of Air Flow Measurement Data

Table 2.5: Descriptive Statistics of Volumetric Air Flow Data

Field	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Measure CFM	6,630	1,181	396	1,172	0	3e+03	3e+03	0.04	-0.5
Estimated Airflow	6,630	1,560	1,745	1,370	-9,065	1e+05	1e+05	38.26	2,006.4
Goal Airflow	6,630	1,482	239	1,600	580	2e+03	2e+03	-0.06	0.4

compressor map data. Thus, while defaulted values of 400 CFM/Ton homogenize the data in both the *Measure.cfm* and *Goal.Airflow* fields (effectively minimizing variance and eliminating outliers), the *Estimated.Airflow* is subject to variation in testing conditions, propagation of multiple measurement errors, and errors introduced in extrapolating measurements using prototypical compressor maps.

Since the *Measure.cfm* field is used in EM-HVAC for the air-side performance calculations it is explored in further detail here. Figure 2.14 demonstrates the distribution of the data which has a large concentration of readings around 1,600 CFM. Most of these measurements correspond to default values of 400 CFM/Ton and, when removed, the remaining data is normally distributed with a mean of 976 CFM (or 271 CFM/Ton). While the mean of measurements looks to be reasonable in magnitude, there are a number of measurements which are unreasonably low. For example, (3) observations report 0 CFM and and additional (10) report less than 100 CFM.

Two additional fields relating to air flow are maintained in EMVAC which normalize the measured (or estimated) volumetric air flow rate to the rated system capacity. These fields are *Measured.cfm.ton* and *Estimated.cfm.ton*. Since these fields are predicated on the *Measure.cfm* and *Goal.Airflow* fields discussed above, their data demonstrate similar behavior. Looking at the *Measured.cfm.ton* specifically, one can see the 400 CFM/Ton default assumption stands out from the rest of the data whose mean is around 250 CFM/Ton (see Figure 2.15). It should be noted that the *Estimated.cfm* and corresponding *Estimated.cfm.ton* fields contain outliers of significant magnitude which indicates that their calculations are very sensitive to error in the refrigerant side measurements.

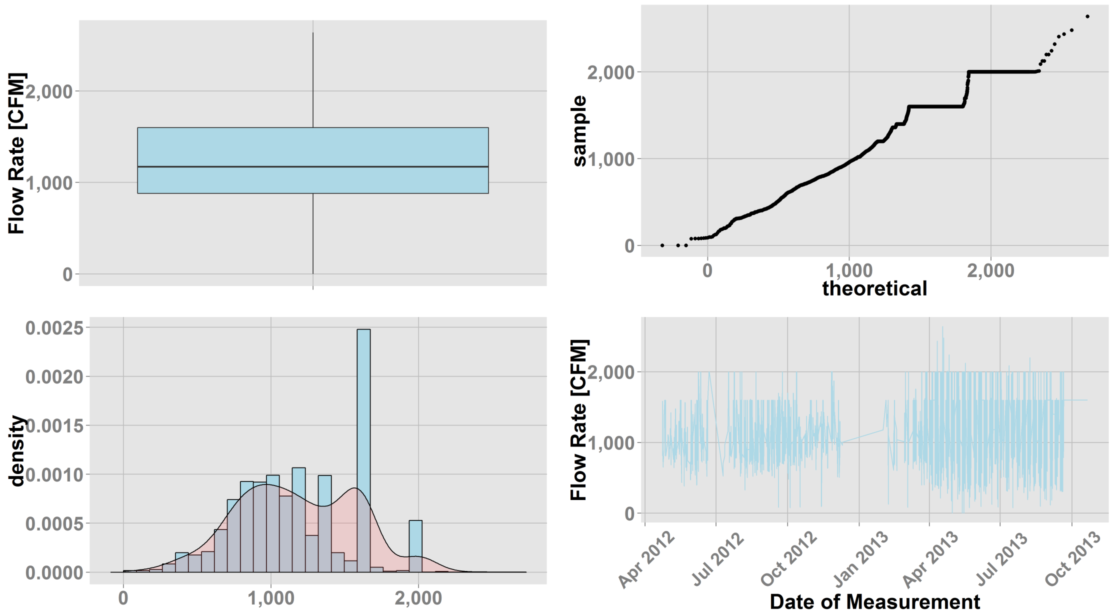

Figure 2.14: Detail of On-Site Air Flow Measurement Data

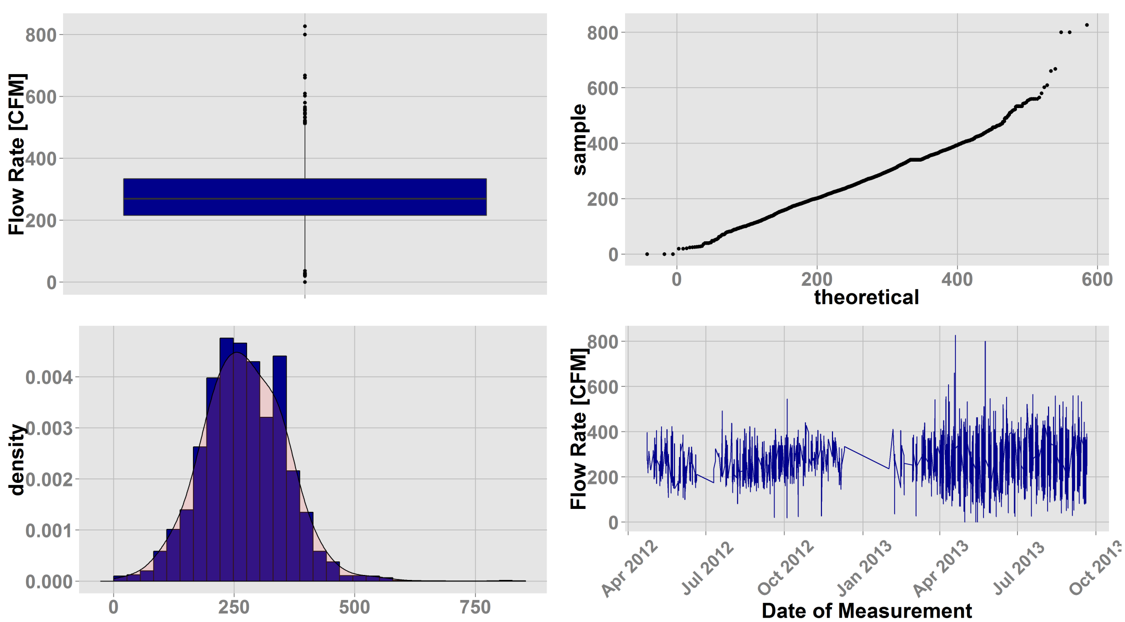

Figure 2.15: Distributions of "Measured" Air Flow Rate per Nominal Compressor Capacity (Default Observations Excluded)

2.3. Air-side Performance Calculations

This section discusses the calculated coil loads (predicated on the air-side measurements discussed in the sections above) and also reviews the resulting *in-situ* system efficiencies as calculated by EM-HVAC and ADM when the electrical measurements are applied to coil load calculations.

2.3.1. Evaporator Coil Load Calculations

ADM used equation 2.1 to calculate evaporator coil load based on the air-side data fields discussed previously in this section. The *Airside.Capacity* field in EM-HVAC calculates this for each site as well and its data are shown in Figure 2.16. The *Airside.Capacity* data shows a mean measured air-side capacity of 33,511 BTU/Hr, though this number is likely skewed due to the significant number of sites which leverage an assumed 400 CFM/Ton. If all tests are removed which assume 400 CFM/Ton then the mean capacity drops to 29,696 BTU/Hr. Values of air-side capacity range from -17,713 BTU/Hr to 126,145 BTU/Hr and inherit all of the inconsistencies observed in both the enthalpy and airflow measurements.

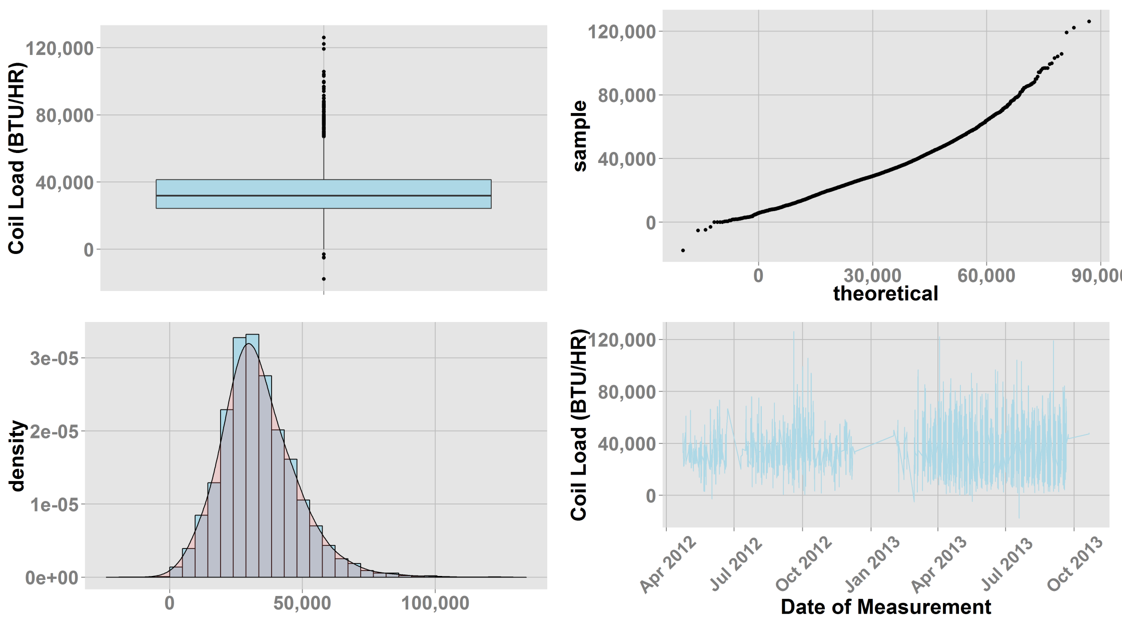

Figure 2.16: Detail of "Measured" Air-Side Capacity (Evaporator Coil Load)

ADM reviewed the air-side load calculation in EM-HVAC against our own calculations which leverage EM-HVAC volumetric air flow measurements (in the *Measure.cfm* field) and the delta enthalpy values we derived using standard psychrometric equations. The differences between ADM's calculated loads and those recorded in EM-HVAC are shown in Figure 2.17. Since ADM had to rely on the the airflow measurement data in EM-HVAC, the differences in Figure 2.17 are driven entirely by differences already discussed in the enthalpy calculations.

2.3.2. In-Situ System Efficiency

System efficiency can be calculated by dividing the Air-Side load by the system's electrical power to derive an in-situ EER value (though this should not be confused with the Rated EER at AHRI test conditions). In-situ EER values were calculated for each Job Id in order to compare differences between the Test-In and Test-Out conditions. The distributions of these data are presented in Figure 2.18 where it can be seen that there is significant overlap between the in-situ EER measured at Test-In and Test-Out. For reasons discussed later in this report, sites whose outdoor air temperatures differ significantly from Test-In to Test-Out were removed from the observations plotted in Figure 2.18. Specifically, all sites whose outdoor temperature differed by greater than 5 F between Test-In and Test-Out were excluded from these calculations. Additional observation were removed where ADM found the data to represent non-physical conditions or system operation outside of the appropriate testing window. The data indicate that in the population there was an average increase in the EER by approximately 0.7 (or 9%). Table 2.6 provides some additional detail regarding the in-situ EER data. Note that the study observed EER values below 4 and above 20 in some sites. However; such observations were removed from the data presented in Figure 2.18 as they are considered non-physical. This manifests in the "missing" left tail in the distribution of EERs.

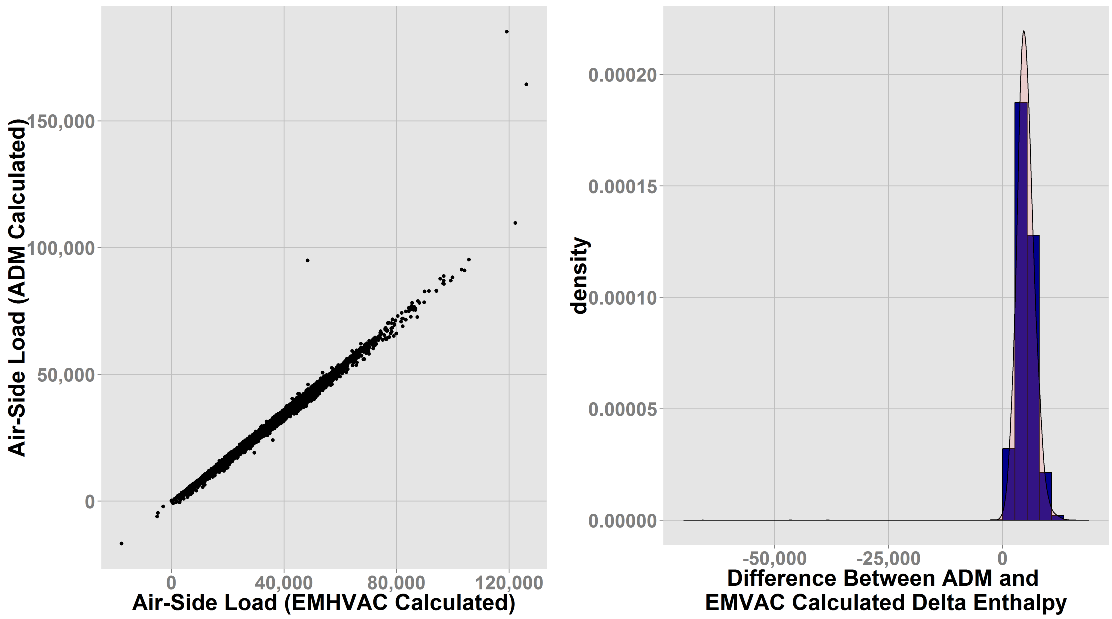

Figure 2.17: Review of Air-side Load Calculations

Figure 2.18: Comparison of In-Situ EER Measurements Between Test-In and Test-Out

Table 2.6: Summary of Air-side EER Data

Activity	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Test-In	1,627	7.7	3	7.4	0.1	25	25	0.9	2
Test-Out	1,627	8.4	3	8.0	0.8	31	30	0.8	2
Change	1,627	0.7	2	0.5	-15.9	20	36	0.6	10

3. Refrigerant-Side Measurements and Algorithms

Refrigerant-Side Metrics refer to measurements, calculations, and categorical data fields describing the thermodynamic energy balance and heat transfer processes in the the refrigerant as it circulates through the vapor compression cycle. The EM-HVAC data fields which are relevant to these metrics are listed in Table 3.1. Refrigerant-side metrics must be combined with compressor performance curves/data in order to estimate system performance. The performance curves represent refrigerant mass flow rate, compressor capacity, and compressor input power as a function of saturated suction and discharge temperatures in a polynomial whose form is defined by AHRI Standard 540. However; this data is typically not made available to the public for the residential air conditioner systems impacted by the Comprehensive Mobile-Home Program and often generic curves are substituted it their place. Currently EM-HVAC applies generic curves based on data publicly available on reciprocating and scroll compressors deemed representative of what Synergy technicians encounter in the field.

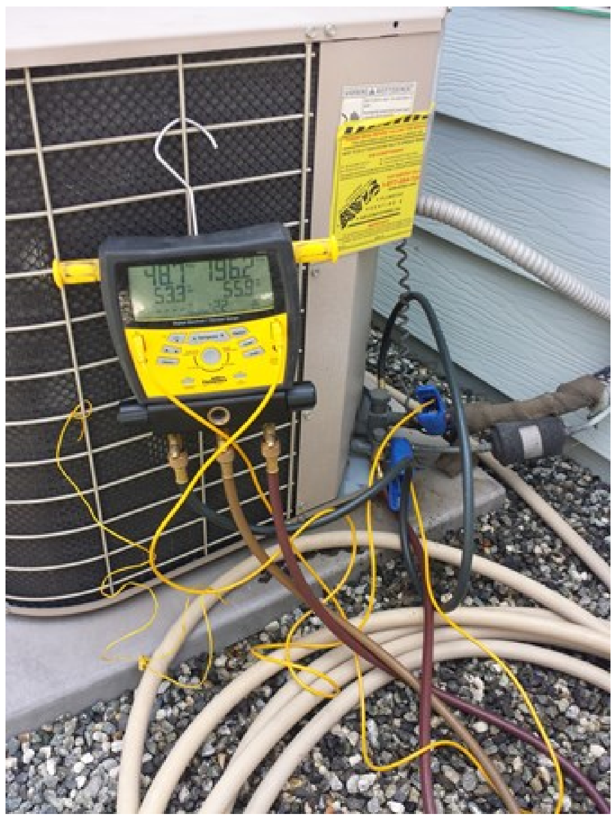

Figure 3.1: Typical Installation of the Digital Manifold Gauge Set

Refrigeration system measurements were observed to be taken using a Fieldpiece SMAN3 digital manifold gauge set. Measurements were observed to be taken once the system had been operating for approximately 10 minutes and as close to steady state as possible. Figure 3.1 demonstrates a typical implementation of the digital manifold gauge set to measure refrigerant properties. Equation 3.1 illustrates the general form of the formula used to calculate evaporator load using refrigerant-side measurements.

$$\dot{Q} = \Big(h_{Liquid}(LT, LP) - h_{Suction}(ST, SP)\Big) * \dot{m}(ST, DT) \tag{3.1}$$

where:

h	is the enthalpy of the refrigerant $[BTU/Lb]$ before and after the evaporator coil. This is derived from tables of the thermophysical properties of refrigerants R22 and R410A.
LT	is the measured temperature of the liquid line.
ST	is the measured temperature of the suction line.
LP	is the measured pressure of the liquid line.
SP	is the measured pressure of the suction line.
$\dot{m}$	is the mass flow rate of refrigerant - estimated by applying generic compressor performance curves.

Like the air-side measurements, the formula can be broken down into two terms: *delta enthalpy* and *mass flow rate*. The enthalpy difference is measured between the liquid line and suction line where each is a function of the measured refrigerant temperature and pressure. Mass flow rate is estimated using compressor performance curves and with observed saturation temperatures (adjusted for differences in super-heat and sub-cooling). Since calculated results for each component (both the delta enthalpy and mass flow rate) are predicated on a common set of measurements, the refrigerant pressure and temperature measurement data fields are explored in this chapter at depth. While the outdoor air temperature measurement does not play a direct role in the system performance calculations, it does impact system performance and must be considered in order to compare "Test-In" and "Test-Out" results for a given system. As such, some time is also spent exploring the outdoor air data as it relates to system performance.

Table 3.1: List of Refrigerant-Side Metric Fields

Field Name	Definition
Condensing.Air.Entering.Temperature	Outside air temperature. Measured at the entrance to the condenser coil.
Liquid.Pressure.Discharge.Pressure	Refrigerant pressure measurement at liquid line.
Suction.Pressure	Refrigerant pressure measurement at suction line.
Liquid.Line.Temprature	Surface temperature measurement of the Liquid line
Suction.Line.Temprature	Surface temperature measurement of the Suction line
Compressor.Capacity	Capacity of system based on Compressor map (AHRI 540 polynomial). Capacity of system based on measured system pressures plugged into a Compressor map representing reported system.
Compressor.EER	Estimate of system efficiency based on compressor map and measured power. Based on compressor map capacity divided by measured power
ET.Actual	Evaporating Temperature (ET) Actual. Evaporator saturation temeprature from measured suction pressure.
SH.Actual	Superheat (SH) Actual . Measured suction line superheat entering condenser assembly
SC.Actual	Subcooling (SC) Actual. Measured liquid temperature cooling (below saturation) on high-side upon leaving condenser coil assembly

3.1. Liquid and Suction Line Temperatures

The Fieldpiece SMAN3 digital manifold gauge set had integrated clamp on thermocouples which were used to read liquid line and suction line temperatures. Technicians were trained to use sandpaper to clean the copper lines near the service valves in order for the thermocouples to have good contact with the lines. Generally the technicians cleaned and sanded the lines well and placed the temperature probes near the service valves. There were a few cases of where the technician needed to troubleshoot bad temperature measurements, usually resulting from poor contact of the thermocouple with the copper line. Technicians were trained to try to keep temperature measurements out of direct sunlight and this was generally observed to be the case.

Data from the temperature measurements made at the condensing units are shown in Figure 3.2. Measured outdoor air temperature is only used to determine whether or not ambient conditions are appropriate for testing and are therefore not discussed here at length. No tests were identified to have been performed at outdoor air temperatures less than the minimum threshold for the refrigerant metering device. A number of outliers are present in the liquid and suction line temperature measurements. The liquid and suction line temperatures also exhibit significant ranges - to the extent that their data overlap. There are 10 instances in which the recorded liquid line temperatures are greater than the suction line temperatures. Only 2 of these observations appear to have been due to transposition error (e.g. Suction temperature recorded as liquid and visa versa). The remaining measurements are very close in magnitude indicating that the units were either non in steady-state operation, the temperature probes were not installed correctly, or temperature measurements were influenced by external factors (for example radiant heat from the sun). Note that units in the fields cannot be expected to operate at a true steady-state (where the loads on the evaporator and condenser coils are constant for the duration of the tests/measurements). However; the units in the field will eventually stabilize at a load which should change at a rate much slower than the duration of the test measurements. If the measurements are made before the system loads stabilize then each measurement would reflect a completely difference operating point.

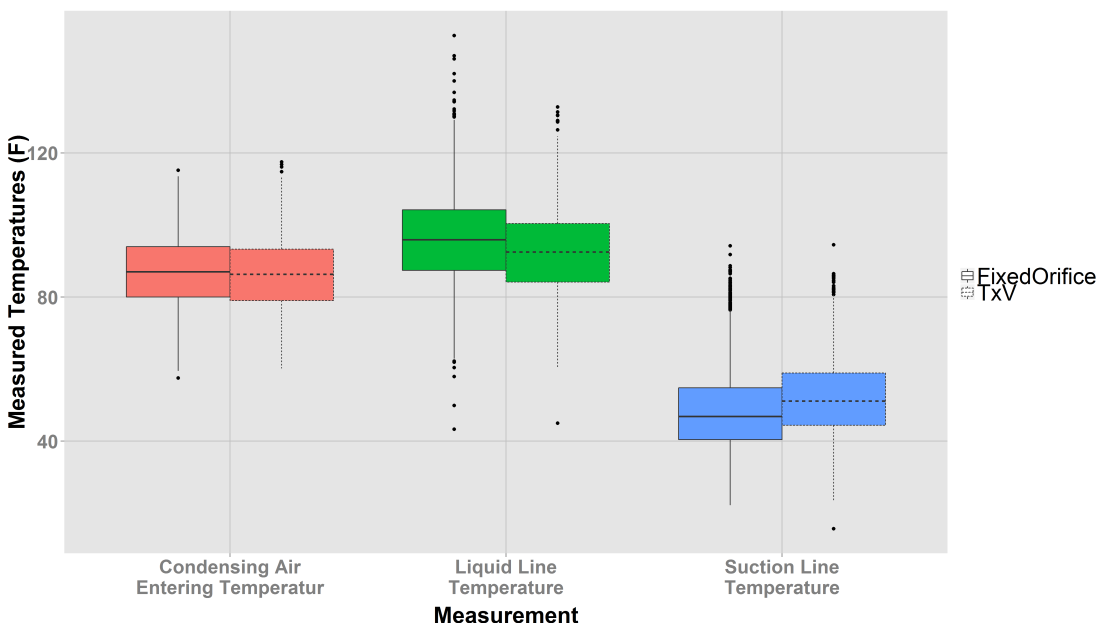

Figure 3.2: Box-Plot Summary of Refrigerant Temperature Measurement Data

A closer look at the liquid line temperature measurements indicates that they are normally distributed with a mean of 95. Figure 3.3 illustrates the distribution of liquid line temperatures. The suction temperature measurements are similarly distributed though they show much more positive skew. Table 3.2 provides some descriptive statistics comparing the two sets of data.

Table 3.2: Descriptive Statistics of Refrigerant Line Temperature Measurements

Field	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Liquid Line Temperature	6,630	95	12	95	43	153	109	0.1	0.08
Suction Line Temperature	6,630	50	12	48	16	94	79	0.7	0.15

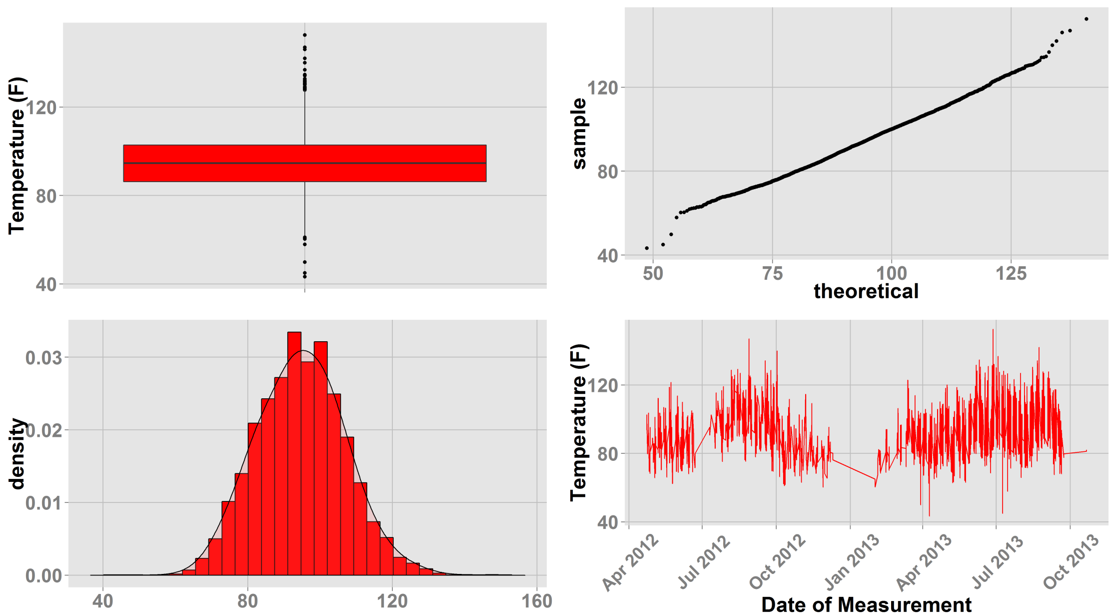

Figure 3.3: Detail of Measured Liquid Line Temperatures

Superheat and sub-cooling are two parameters used to gauge refrigerant charge levels and must be within an appropriate range for an air conditioner to operate efficiently. Superheat is defined as the difference between the temperature of the refrigerant in the suction line and its saturation temperature (dew point) corresponding to its pressure. Similarly, sub-cooling is the difference in temperature of the refrigerant after it passes through the condenser (liquid line) and it saturation temperature. While these temperatures are not used directly in the calculation of system performance, they are used to determine whether or not the current refrigerant charge is appropriate and are thus tracked by EM-HVAC as an indicator for system health. Figure 3.4 compares the super-heat and sub-cooling data calculated in EM-HVAC to values calculated by ADM using the recorded refrigerant temperatures and pressures (corrected for site altitude). Note that ADM was able to replicate the same values for sub-cooling and super-heat as recorded in EM-HVAC. From Figure 3.4 it can be seen that the sub-cooling data show less variance than the super-heat data, though there are a number of significant outliers.

A closer look at the super-heat and sub-cooling data shows that variances are generally lower for *Test-Out* observations than at *Test-In*. This is expected since one objective of the QM measure is charge adjustment which seeks to add or remove charge until a particular super-heat and/or sub-cooling value is achieved. However; it is notable the super-heat data for fixed orifice metering devices shows less variance in the Test-Out observations than data for the TxV (see lower two panels in Figure 3.5). This is counter-intuitive given the differences in how the two metering devices function. Typically, super-heat data for a fixed orifice will vary based on the indoor and outdoor ambient conditions. While system charge for such metering devices is assessed by comparing the measured super-heat to a particular goal - the goal can vary considerably based on prevalent conditions. TxV metering devices are designed to maintain super-heat within a narrow range and thus use a sub-cooling goal to assess system charge. The narrow band of super-heat measurement in Test-Out conditions may indicate improper use of super-heat tables.

Negative values are present in both *Test-In* and *Test-Out* observations, as well as in both Thermal Expansion Valve (TxV) and Fixed Orifice metering devices. A negative value indicates that the refrigerant is in an opposite thermodynamic state to what is expected (e.g. negative sub-cooling indicates the refrigerant is a vapor rather than liquid) and do not represent a physical possibility in a functioning air-conditioner. The negative observations fall into one of two groupings: 1) *Significant Outlier*, or 2) *Near Zero Value*. Figures 3.5 and 3.6 illustrate the distributions of super-heat and sub-cooling data. Negative values of super-heat and sub-cooling are likely due to

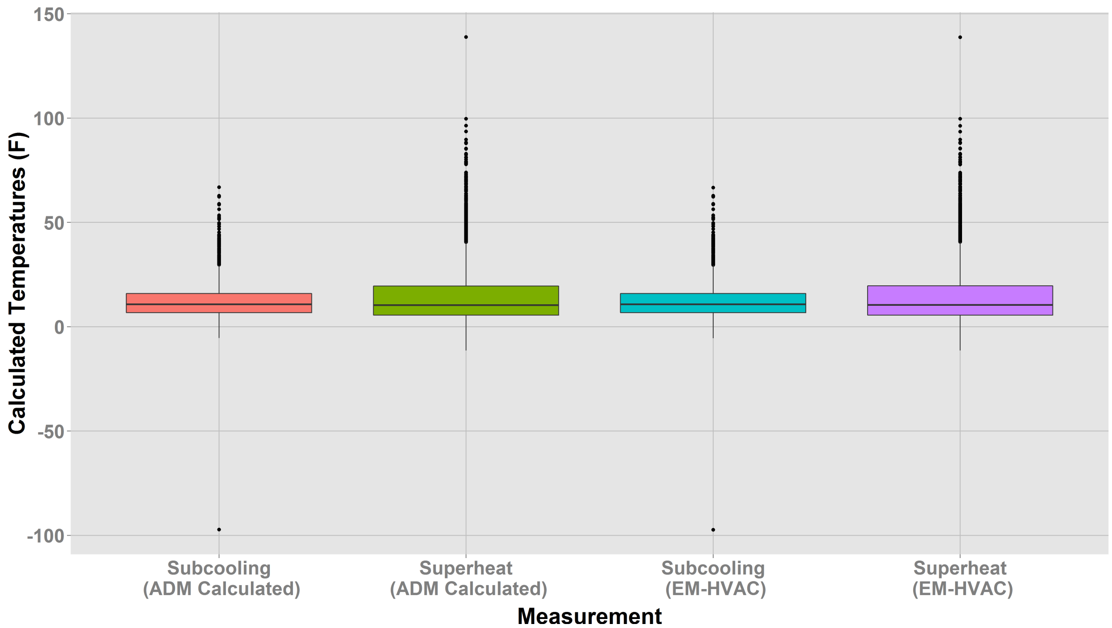

Figure 3.4: Box-Plot Summary of Superheat and Subcooling Data

measurement error (e.g. sunlight influencing thermocouple measurements, incorrectly placed temperature probe, clogged schrader valves, etc.).

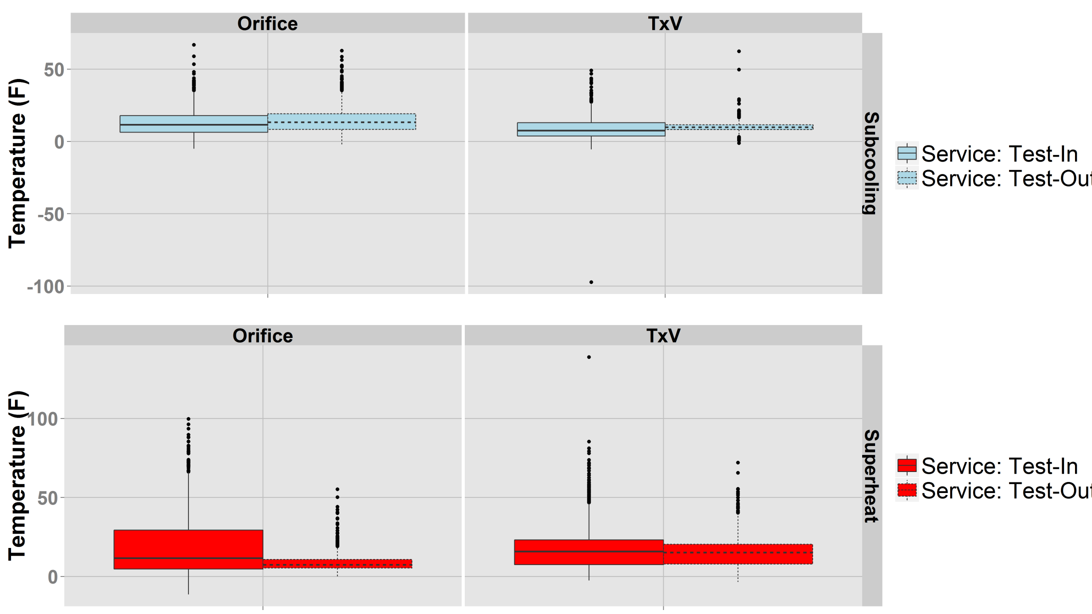

Figure 3.5: Comparison of Superheat and Subcooling Data by Metering Device and Test Conditions

The sub-cooling and super-heat data are subject to the propagation of error in two measurements - refrigerant pressure and temperature. Furthermore, the refrigerant pressure measurement is not used directly. Instead it is applied to analytically determine the refrigerant saturation temperature at the measured pressure value - intro-

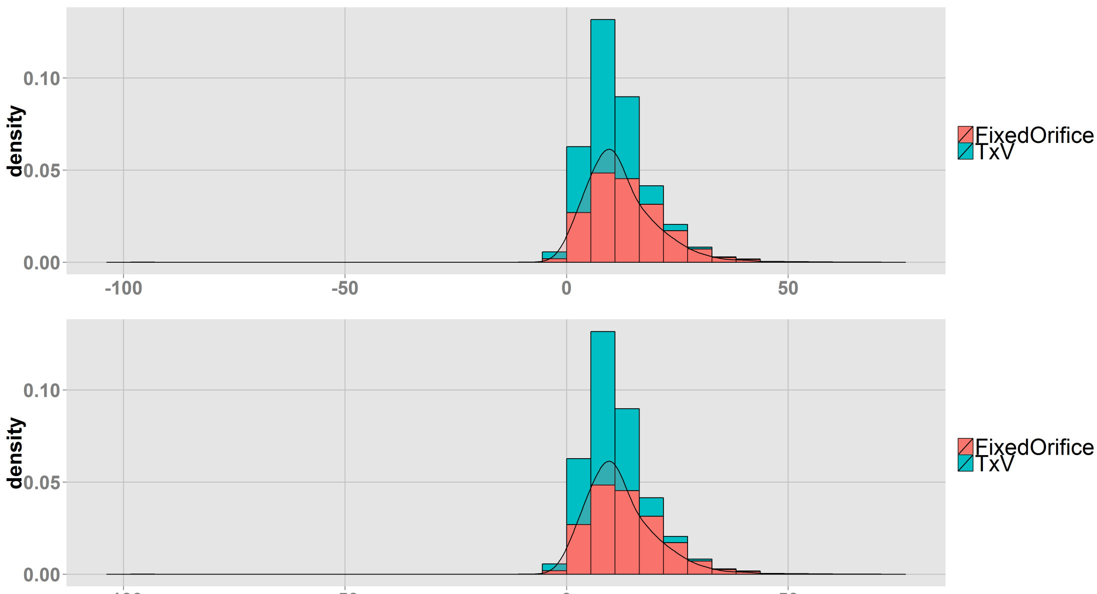

Figure 3.6: Distributions of Superheat and Subcooling Data

Table 3.3: Descriptive Statistics of Superheat and Subcooling Data on TxV Equiped Units

Field	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Superheat (Test-In)	1,240	18	15	16	-3	139	141	2	5
Superheat (Test-Out)	1,240	15	10	15	-4	72	76	1	3
Subcooling (Test-In)	1,240	9	8	8	-97	49	146	-1	29
Subcooling (Test-Out)	1,240	10	4	10	-1	62	63	4	46

ducing additional uncertainty into the final values of super-heat/sub-cooling. The following example illustrates the propagation of error in a super-heat and sub-cooling calculation:

> In this example of error propagation we assume that each refrigerant temperature and pressure measurement is subject to a 3% measurement error. Let's assume the Liquid and Suction line temperature and pressure measurements are 95 F/230 PSI and 50 F/76 PSI respectively, and that the refrigerant is R22. No additional error is added to the calculation of refrigerant saturation temperature from the pressure measurement; however, when the $\pm$3% measurement errors are combined to calculate super-heat and sub-cooling the results are subject to $\pm$21.1 % and $\pm$45.2 % in the super-heat and sub-cooling calculations respectively. Tables 3.6 and 3.5 illustrate the upper and lower error bounds of each property in this example.

Given the exaggerated error propagation in super-heat and sub-cooling values, it is likely that many of the negative observations which fall *near zero* are a result of standard measurement and instrumentation error. However; observations of *Significant Outliers* are likely introduced by the technician through improper set-up of their instrumentation.

Table 3.4: Descriptive Statistics of Superheat and Subcooling Data on Fixed Orifice Equiped Units

Field	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Superheat (Test-In)	2,075	19	18	12	-11.4	100	111	1	1
Superheat (Test-Out)	2,075	9	5	7	0.1	55	55	2	9
Subcooling (Test-In)	2,075	13	9	12	-5.1	67	72	1	2
Subcooling (Test-Out)	2,075	14	8	13	-2.5	63	65	1	2

Table 3.5: Example of Error Propigation in Subcooling Calculations (3% Measurement Error)

Property	Lower.Bound	Measured	Upper.Bound
Pressure	223	230	237
Temperature	92	95	98
Saturation Temperature	104	106	109
Subcooling	6	12	17

3.2. Liquid and Suction Line Pressures

The Fieldpiece SMAN3 digital manifold gauge set was used to measure liquid line and suction line pressures. Technicians were trained to attach the liquid and vapor lines to the correct service valves on the condensing unit and to inspect the service valve for operability. Liquid and Suction line pressures are used to estimate the refrigerant saturation temperature using table of thermodynamic properties for the refrigerant of interest. Figure 3.7 illustrates the distribution of refrigerant pressure data. The data are further grouped into the type of refrigerant within the system as well as the style of metering device. Significantly more variance is present in the Liquid line pressure data compared to the suction line pressure data. The observed differences between pressures due to refrigerant type is expected given their different thermodynamic properties (systems with R410A operated at higher liquid line pressures than systems with R22). Also, the data show that liquid line pressures in TxV metered systems are slightly lower than in fixed orifice systems.

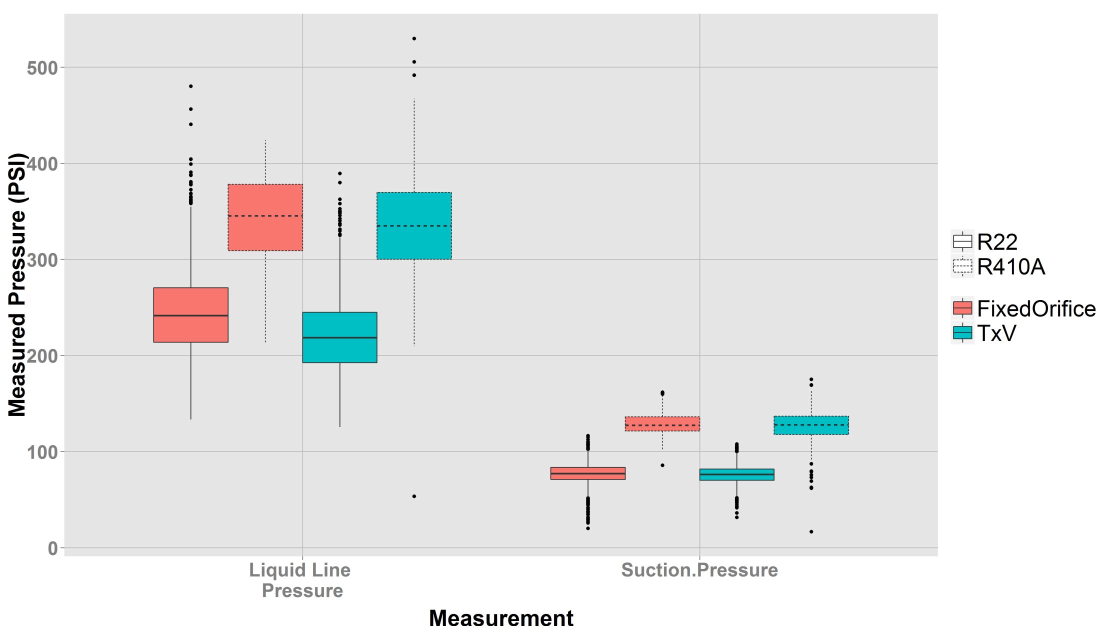

Figure 3.7: Box-Plot Summary of Refrigerant Pressure Measurements

Table 3.6: Example of Error Propigation in Superheat Calculations (3% Measurement Error)

Property	Lower.Bound	Measured	Upper.Bound
Pressure	74	76	78
Temperature	48	50	52
Saturation Temperature	33	35	36
Superheat	12	15	18

Table 3.7: Descriptive Statistics of Refrigerant Pressure Measurements

Field	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Liquid Pressure (R22)	5,932	236	42	234	126	480	355	0.42	0.4
Suction Pressure (R22)	5,932	77	10	77	20	116	96	-0.25	1.5
Liquid Pressure (R410A)	698	336	52	336	54	530	476	-0.06	0.9
Suction Pressure (R410A)	698	127	15	128	17	175	158	-0.87	4.7

ADM reviewed the data for additional dependencies and, as should be expected, found significant correlation in liquid line pressure with outside air temperature. It can be seen by Figure 3.8 that the much of the range present in the Liquid Line pressure data is actually driven by a wide range in outdoor ambient conditions at the time of the measurements. The data show that this trend is consistent for each refrigerant type (e.g. parallel trends of similar slope), and that there are significantly more systems charged with R22 than R410A.

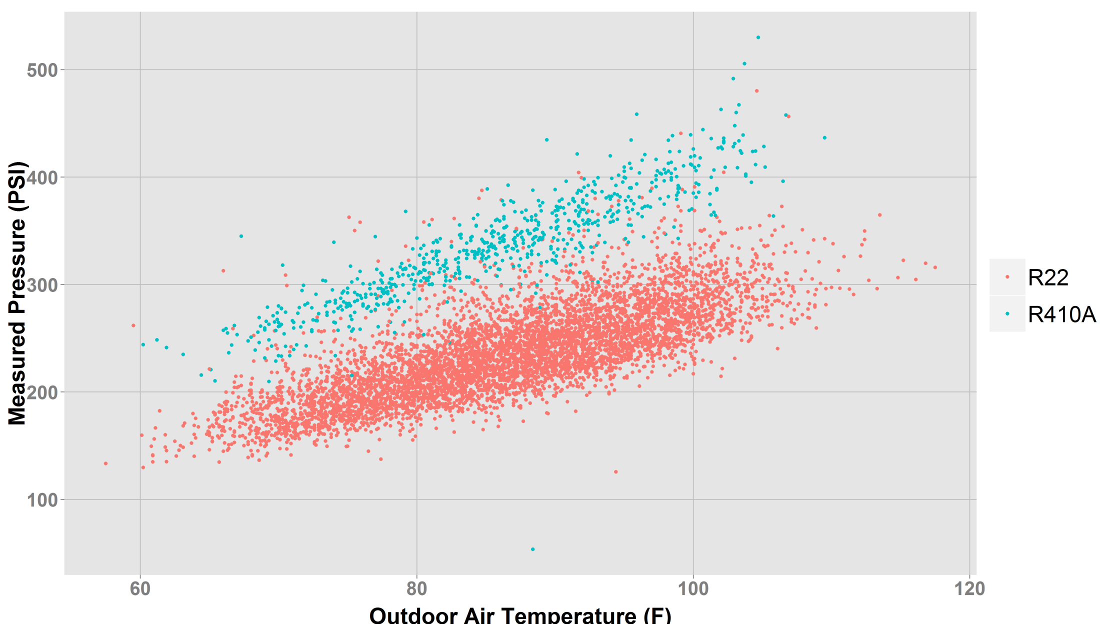

Figure 3.8: Correlation Between Refrigerant Liquid Line Pressure and Outdoor Air Temperature

3.3. Outdoor Air Temperatures

Ambient outdoor air temperature was measured with a type K thermocouple at the inlet to the condenser. Note that this report uses the term *outdoor air temperature* in place of the more technically correct term for this data - the *entering air temperature* (EAT). EAT describes the temperature of the air entering the condenser coil which may differ from the local ambient outdoor air temperature due to condensing unit placement (e.g. is it surrounded by radiating surfaces or in a shaded space). The technicians were trained to place the thermocouple around the protective cage around the condenser fins and keep it out of direct sunlight (seen in Figure 3.9). This measurement was made consistently by the technicians. The technician would plug the thermocouple into one of the temperature reading slots on the Fieldpiece SMAN3 to get the reading. There was an incident of bad ambient temp measurements due to a bad thermocouple. The ambient temperature measurement was checked with one of the Testo hygrometers and the thermocouple was plugged into the thermocouple port on the digital Amp probe. It was noted that there was some variance in readings between the Amp probe and the SMAN3. It was unclear if the Amp probe had been calibrated or needed to be calibrated since the temperature reading was not typically made there.

Figure 3.9: Typical Measurement of Outside Air Temperature

It can be seen in Figure 3.10 that the outdoor air temperatures are normally distributed with a mean of 87 $^{\circ}\mathrm{F}$. The QM Program guide prescribes the range of acceptable outdoor air temperatures for testing. Testing should not be performed at outdoor air temperatures less than 55 $^{\circ}\mathrm{F}$ for fixed orifice metering devices or 60 $^{\circ}\mathrm{F}$ for TxV metering devices. The upper boundary is 130 $^{\circ}\mathrm{F}$. It can be seen by the summary statistics listed in Table 3.8 that all outdoor temperature observations fall within this range.

Table 3.8: Summary of Outdoor Air Measurements by Metering Device

Metering Device	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Fixed Orifice	4,150	87	10	87	58	115	58	-0.12	-0.5
TxV	2,480	86	10	86	60	118	57	0.03	-0.5

Outdoor air temperature also plays a role in system efficiency. As the outside air temperature increases the system must work harder to achieve the same capacity. Thus the efficiency of the system (e.g. the EER) decreases as outdoor air temperature increases. This impact must be considered when trying to compare Test-In to Test-Out data since the measurements are taken at different times and in some cases with very different outdoor air temperatures.Table 3.9 demonstrates that while the average difference between Test-In and Test-Out is negligible (about 1.7 $^{\circ}\mathrm{F}$) the range spans from -28 $^{\circ}\mathrm{F}$ to +30 $^{\circ}\mathrm{F}$. System performance results become more difficult to compare as the difference in outdoor air temperature between the Test-In and Test-Out measurements increases.

ADM also looked at the time lapse between Test-In and Test-Out across all sites in the program data (the results of which are included in Table 3.9). Note that while the average recorded interval between tests was 2 hours, the time differences show a considerable range (with some tests taken days apart). There are 73 observations showing a negative difference (e.g. the Test-Out is recorded as being taken before the Test-In.). ADM expects

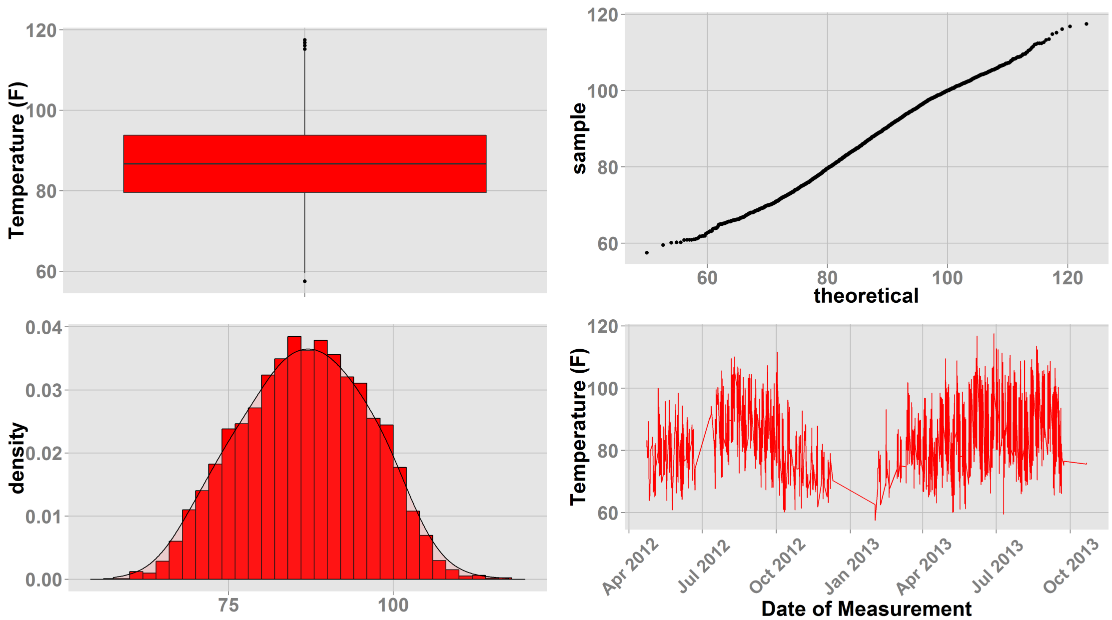

Figure 3.10: Detail of Measured Outdoor Air Temperatures

Table 3.9: Differences in Outdoor Air Temperatures between Test-In and Test-Out Measurements

	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Test In Temp (F)	3,315	86	10	86	60	118	57	1e-04	-0.6
Test Out Temp (F)	3,315	87	9	88	58	117	59	-1e-01	-0.5
OA Diff Between Tests (F)	3,315	2	5	1	-28	30	58	-2e-01	4.1
Time Lapse (Hours)	3,315	2	14	1	-68	340	408	2e+01	300.8

that these are due to transcription errors on the part of the technician filling out the form, or during data entry into EM-HVAC. During ADM's ride-along inspections we noted that the interval between tests was on the order of 1 to 2 hours, which is consistent with the average interval seen in the EM-HVAC data.

3.4. Refrigerant-side Performance Calculations

ADM obtained the compressor map data used in EM-HVAC in order to model compressor performance characteristics - specifically refrigerant mass flow rate. Model sensitivity to curve data was reviewed by comparing results between the curves used in EM-HVAC to alternative compressor performance curves for scroll and reciprocating compressors produced by other manufacturers. ADM found that while there was some sensitivity, the curves themselves were relatively robust. ADM also noted that the manufacturer and model numbers chosen for use in EM-HVAC seemed appropriate based on observations made during ADM's ride-along visits. It was therefore determined that once normalized, the curves used by EM-HVAC sufficiently represent *generic compressor performance* for the mobile homes being modeled.

The in-situ system performance was estimated using refrigerant-side measurements in two approaches: 1) application of Equation 3.1 with the refrigerant-side measurements (adjusted for super-heat), and 2) application of compressor capacity curve data (adjusted for both sub-cooling and super-heat). Figure 3.11 compares the two methods by graphing their results against each other. It can be seen that the two methods produce very similar results and much of the scatter is likely due to propagation of errors in the measurements used as inputs. Note though that all observations of negative super-heat and sub-cooling (discussed in the previous section) are removed as they are expected to be errant measurements.

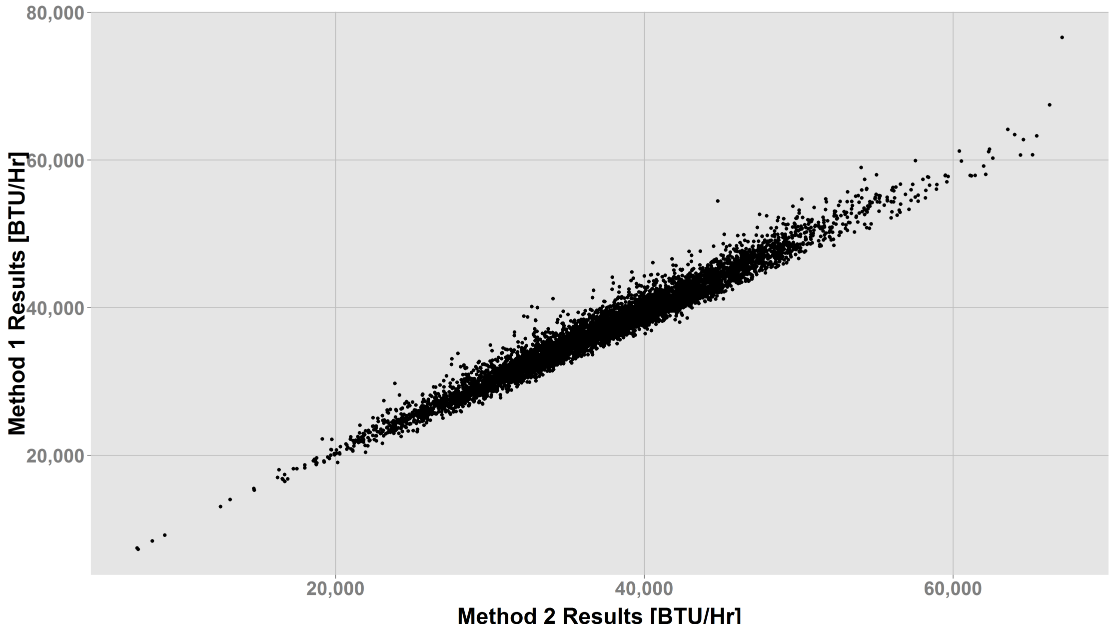

Figure 3.11: Comparison of Calculated System Load Between Equation 3.1 and Capacity Curves

Next, ADM reviewed the system load as calculated by EM-HVAC to the two methods above. It can be seen in Figure 3.12 that again the results are very similar though with some scatter. It is less obvious why scatter is present when comparing against ADM's capacity calculations from the compressor curves (*Method 2*) given the use of identical curves. Furthermore, ADM worked with Roltay Inc. Energy Services to reproduce the same capacity calculation algorithms used by EM-HVAC. It should also be noted that all negative sub-cooling and super-heat observations are removed as they result in unrealistic capacity calculations per Equation 3.1. This is made evident when comparing the the distributions of each field to one another as is done in Figure 3.13 where one observes numerous negative outlier observations in the variable titled *ADM: Enthalpy and Mass Flow Equation (Method 1)*.

System efficiency is again calculated by dividing the system load (in this case based on refrigerant-side measurements) by the system's electrical power. in-situ EER values were calculated for the refrigerant-side measurements in the same way as it was done for the air-side and their distributions plotted in Figure 3.14. Again, there is significant overlap between the Test-In and Test-Out measurements, though there is a slight difference in the mean in-situ EER. The data indicate that in the population there was an average increase in the EER by approximately 0.38 (or 4%). Table 3.10 provides some additional detail regarding the in-situ EER data.

Table 3.10: Summary of Refrigerant Side EER Measurements

Activity	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Test-In	1,627	9.4	2	9.2	4	20	15	0.7	1
Test-Out	1,627	9.7	2	9.6	4	19	15	0.7	1
Change	1,627	0.4	1	0.3	-7	8	15	1.0	12

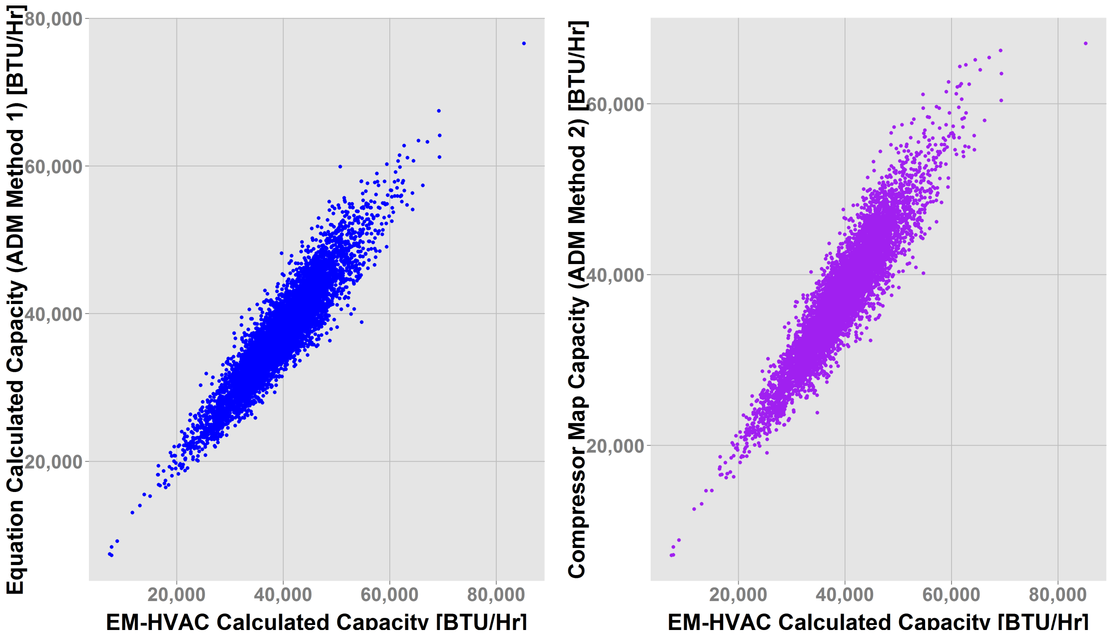

Figure 3.12: Comparison of System Load as Calculated By EM-HVAC to ADM's Calculations

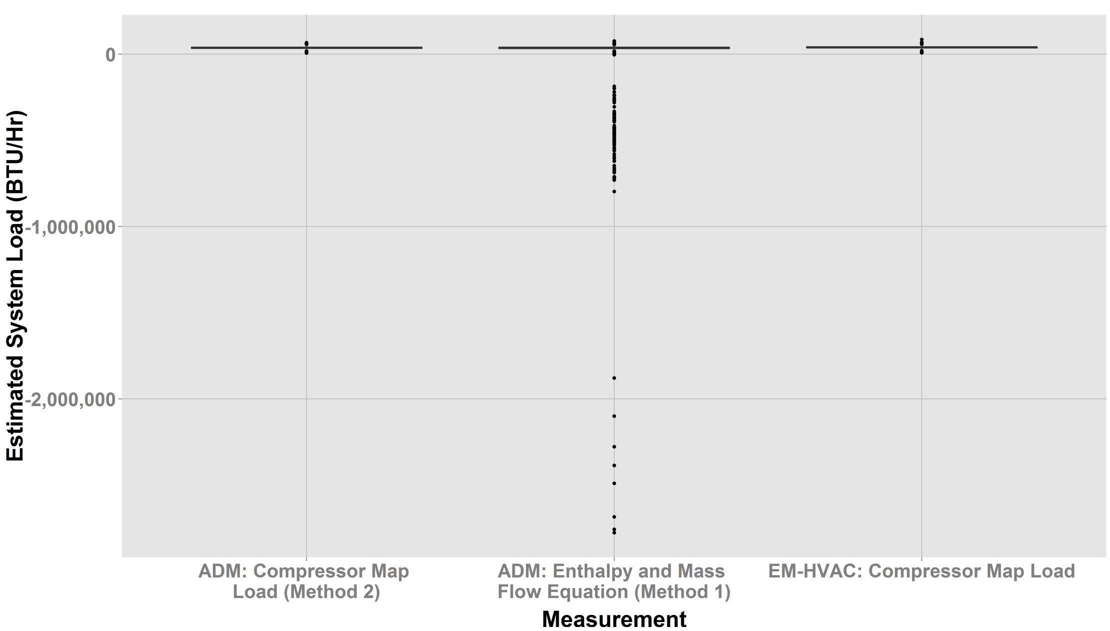

Figure 3.13: Box-Plot Summary of Refrigerant-Side Performance Calculations

Figure 3.14: Comparison of In-Situ EER Measurements Between Test-In and Test-Out (Refrigerant Side)

4. Comparing Refrigerant and Air-side Measurements

It was mentioned earlier that the air-side metrics and refrigerant-side metrics effectively take two different approaches to measuring the same phenomenon - *system load*. While their results should be theoretically identical, room must be made for propagation of measurement error in their derivation. Some scatter is expected, and corroboration is tested by the degree to which results from one metric correlate with those of the other. However; it can be seed in Figure 4.1 that the EM-HVAC data show significant scatter.

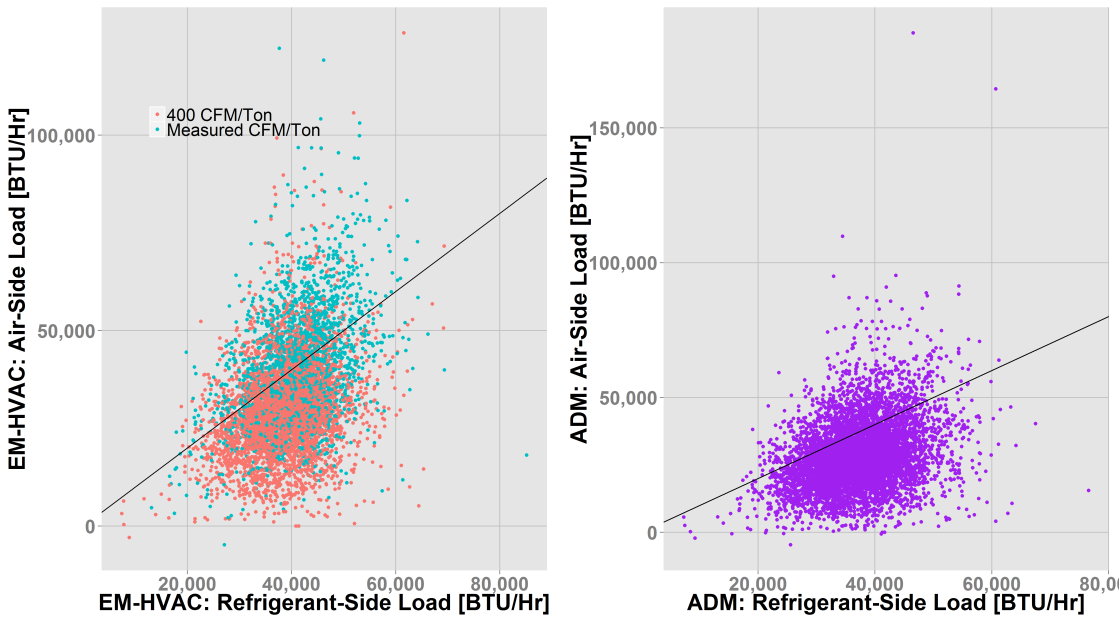

Figure 4.1: Comparison of Air-Side to Refrigerant Side System Load Calculations

Comparing Air-Side and Refrigerant-Side System Load Estimates

Figure 4.1 graphs the system load as predicted by the refrigerant and air-side measurements against each other in two plots. The left panel compares the two sets of load calculations derived by EM-HVAC and the right panel compares ADM's load calculations. The best-fit lines indicate that the general trend is as expected, but the scatter indicates that they are very weakly correlated. Correlation coefficients were calculated with magnitudes of 0.36 for the EM-HVAC data sets and 0.29 for ADM's reproductions. Note that both of these coefficients are quite low, indicating only a weak positive correlation.

The left panel in Figure 4.1 is sub-set into observations where the airflow measurements are defaulted to 400 CFM/Ton and those for which actual airflow measurements are made. Loads calculated using a defaulted CFM/Ton value are consistently higher than those using actual flow measurements. It can be seen in Table 4.1 that when observations using default airflow rate assumptions are removed, the average difference between the Refrigerant-side and Air-side load calculations increases substantially.

The scatter (or *disturbance*) in the Air-Side/Refrigerant-Side comparisons is reviewed in Figure 4.2 by taking the difference between the loads estimated by each data-set (e.g. Refrigerant-Side Load minus Air-Side Load). A negative value indicates that the Air-Side calculations resulted in a higher estimate for system load than the Refrigerant-Side calculation for a given test. The data in Figure 4.2 correspond to the row titled *All Observations* in Table 4.1.

Table 4.1: Impact of Airflow Assumptions on Differences in Air-side and Refrigerant-side Load Calculations

Airflow	n	Mean	Std. Dev.	Median	Min	Max	Range	Skew	Kurtosis
Direct Measurement	4,332	8,939	12,065	9,148	-64,579	59,242	1e+05	-0.5	2
Default Assumption	2,298	658	13,906	1,489	-84,534	67,009	2e+05	-0.5	2
All Observations	6,630	6,069	13,328	6,915	-84,534	67,009	2e+05	-0.5	2

The differences seen in between the Air-Side and Refrigerant-Side calculations appear to be normally distributed with a mean of 6,069 BTU/Hr. Thus the refrigerant-side load calculations predict a higher load on average than the air-side calculations. It can also be seen that there are a similar number of negative observations as there are positive. This implies that despite the significant uncertainty in system performance estimates, the measurements themselves were performed consistently. This conclusion is consistent with ADM's observations during ride along inspections.

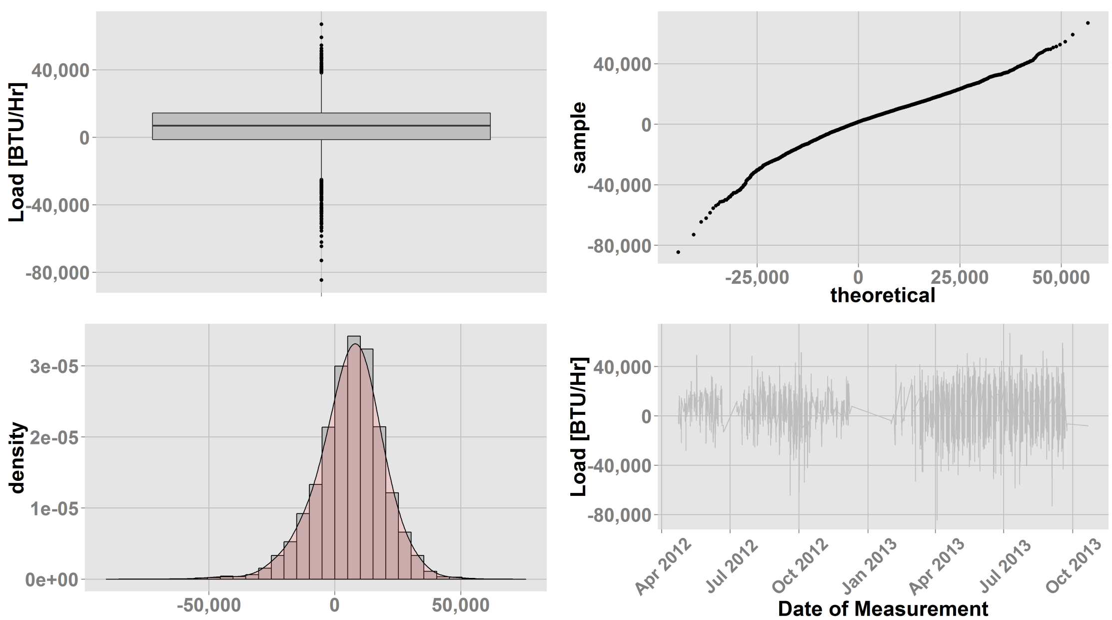

Figure 4.2: Difference Between Air-Side and Refrigerant-Side Load Calculations

Comparing Measured System Efficiency Improvements as Estimated by Air-Side and Refrigerant-Side Measurements

System efficiency calculations (in-situ EER) are compared in Figure 4.3. Again it can be seen that there is very little correlation between the air-side and refrigerant-side measurements with correlation coefficients of 0.36 and 0.37 for the Test-In and Test-Out data respectively. Finally, in Figure 4.4 the measured improvement in in-situ EER is compared between the air-side and refrigerant-side metrics. It can be seen that the means for bot data-sets are very similar, though there is less variance in the refrigerant-side calculations. The average in-situ EER improvement across both metrics is 0.54 or 6.5%.

Conclusions Regarding Refrigerant-Side and Air-Side System Performance Calculations

The Refrigerant-Side estimates of system loads, in-situ efficiencies, and performance improvements are expected to be more reliable, currently, than those based on the Air-Side metrics. This conclusion is based on the issues identified in the air-side measurement data, not the fundamental metric itself(e.g. formula and application of measurement data). If the Air-Side measurements are improved such that they are more representative of the

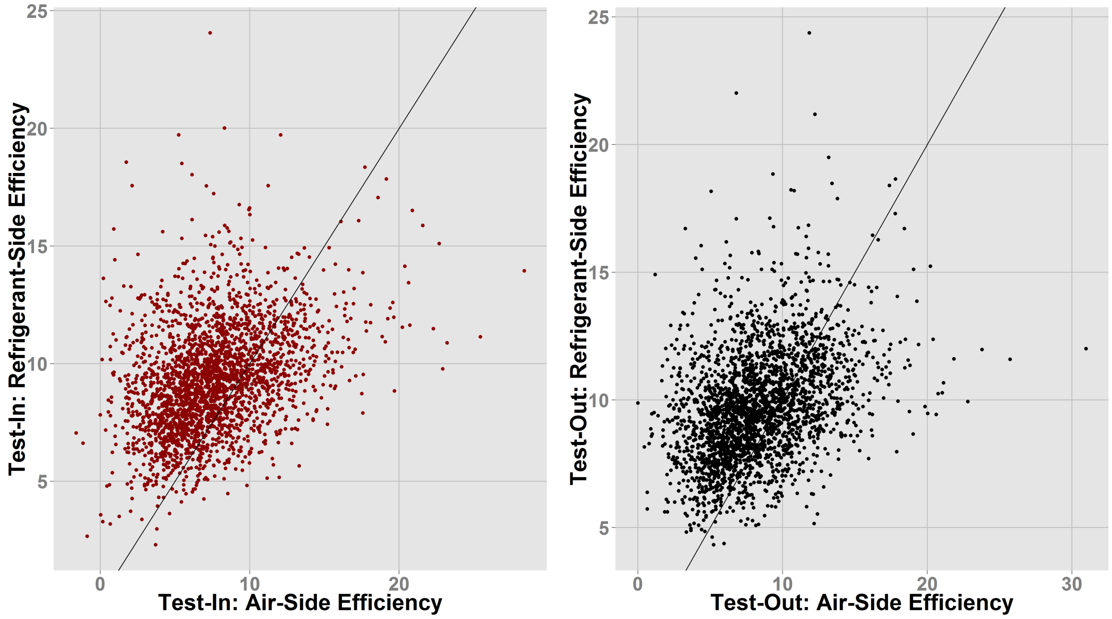

Figure 4.3: Comparison of Air-Side to Refrigerant Side In-Situ Efficiency Calculations

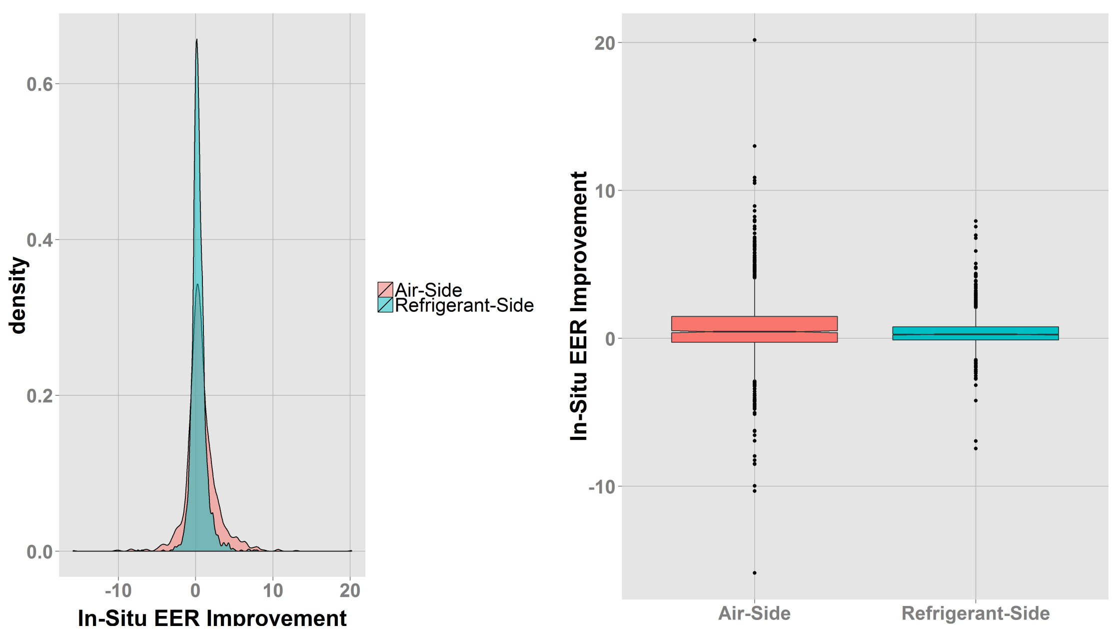

Figure 4.4: Comparison of In-Situ EER Improvement Between Refrigerant Side and Air-Side Metrics

conditions before and after the evaporator coil the correlation between Air-Side and Refrigerant-Side estimates will improve and the Air-Side estimates will become more reliable than the Refrigerant-Side which must rely on generic compressor performance curves. While the program impacts as predicted by the Refrigerant-Side measurements represent (currently) the most reliable empirical measurement of the QM Measure's performance, their reliance on these generic curves introduces additional uncertainty which would not be present in a more direct measurements (e.g. an Air-Side).

5. Findings and Recommendations

In this study ADM reviewed the data collected on-site by Synergy technicians order to assess their consistency and reliability in tracking program energy impacts. This report addresses two of the three objectives outlined in Section 1:

1. Determine the reliability of the current measurement data and recommend ways to improve on-site data collection (particularly for airflow measurements).

2. Identify inconsistencies and recommend how to improve consistency in the data.

This section summarizes our findings and recommendations as they relate to the data and data collection process.

5.1. Findings: Air-side Metrics

ADM found considerable uncertainty in the air-side system load and system performance calculations. This uncertainty stems from the location of or method used to collect both temperature and airflow data. Also, ADM noted that measurements must currently be taken at different points in time as the technician moves around the home to read their instruments. While it is unlikely that there are significant changes in the systems' operating conditions occurring between measurements, some variation is expected. Even small changes in system load or operating conditions between measurements introduces error into the system performance calculations used by EM-HVAC which assumes the measurements are taken simultaneously (effectively a *snap-shot* in time). Our findings as they relate to the air-side data and data collection methods are presented below.

Return Air Temperature

While ADM found some variance in the Return Air temperature measurements across technicians, they were generally made consistently and in accordance with current Synergy training standards. The measurements are a reasonably accurate representation of the indoor ambient conditions before entering the air-handler, though they are not representative of the mixed air directly preceding the coil. The current measurement location contributes significant uncertainty to the air-side measurements of system performance as does not accurately capture these conditions due to the following:

1. Many systems introduce outside air downstream of the measurement location. Some systems were observed to have dedicated outside air inlets, while others were expected to have a non-negligible degree of infiltration. In most cases the outside air adds heat to the air-stream before the coil and thus generates additional system load not accounted for by the current measurements.

2. Additional heat is added to the air-stream by the fan which is currently not captured by return air temperature measurements.

3. In up-flow systems the return air is drawn under the house (and in some instances mixed with the standing air under the mobile home) before returning to the air-handler. The current measurements do not account for heat addition/removal in this process.

In order to facilitate an accurate assessment of system load and refrigerant charge, the return air temperature measurement must (as closely as possible) represent the state of the air directly preceding the cooling coil. Given the variances in system configurations between homes, it would be more tractable to move the measurement to a more ideal location than to apply correction factors to the adjust the temperatures measured at this location. Note that this study did not attempt to quantify the impact of this finding (e.g. the difference in air temperature between the current measurement point and the point recommended in the following Sections).

Supply Air Temperature

The supply air temperature measurements were observed to be taken consistently and in accordance with the current Synergy training standards. While there were a few non-physical observations (e.g. the combination of wet-bulb and dry-bulb temperatures cannot physically be achieved) the observations were largely within appropriate ranges.

As is the case for the return air measurements, the supply air temperature measurement must (as closely as possible) represent the state of the air directly following the cooling coil. The current measurement location is the closest supply register to the air-handler. This does not provide an accurate representation of the conditions immediately following the cooling coil for the following reasons:

1. While the register may be the "closest," air must still travel through the duct-work and often absorbs heat along the way (This is more so the case for up-flow systems). In down-flow systems the air flows through ducts made out of the floor joists themselves which can further cool the air (when the air under the home is cooler than the supply stream) or impact the measured moisture content in the air-stream.

2. In several instances ADM noticed that the grates at which supply temps were made were in the fully closed position. In these instances it was unclear how much airflow the temperature sensor actually received.

Volumetric Airflow

Volumetric airflow measurements proved particularly challenging in up-flow system configurations as there is no way to reliably access the return air inlet with a wind vane anemometer. When Synergy technicians encountered such systems they used and assumption of 400 CFM/Ton in order to estimate the airflow. Where measurements were taken (e.g. most down-flow configurations) the technicians were consistent and followed the current program standards. There were a wide range of airflow values in EM-HVAC, including observations of Zero CFM. ADM noted the following factors in the current airflow measurement practices which contribute a significant degree of uncertainties to the system load/efficiency calculations:

1. 35% of the airflow observations were defaulted to 400 CFM/Ton. When these observations were removed the remaining observations are normally distributed with a mean of 271 CFM/Ton. This is considerably less than the current default assumption and indicates that the current default may be an inaccurate assumption for mobile homes.

2. Airflow measurements are currently taken at the return air intake grill inside the home. As discussed in the findings for return air measurements, systems were observed to have dedicated vents for outdoor air and/or infiltration from exterior doors. These sources of outside air represent additional volumetric airflow not captured by in the current measurement practices and likely underestimate the actual airflow across the coil.

While the airflow measurements data indicates that a more appropriate default value may be closer to 271 CFM/Ton ADM feels this figure is biased low due to factor 2 in the list above. As such, in our recommendations we suggest a default value in between this and the current with the caveat that the default value be re-visited once additional airflow data is collected using the methods recommended in Section 5.3.

5.2. Findings: Refrigerant-side Metrics

Only (5) measurements are made of the refrigerant, and access to these measurement points is generally straight-forward. ADM found found that Synergy technicians were consistent in there processes and performed these measurements in accordance with the current Synergy training standards. Currently it is unclear how reliable the refrigerant side estimates of load and performance are as they rely heavily on generic compressor performance data and could not be corroborated with the air-side data (thought the lack of corroboration may simply be due to significant error in the air-side calculations). The following are findings which relate the refrigerant-side measurements and/or calculated fields.

Superheat Goals for Fixed Orifice Systems

ADM noted that the super-heat data for fixed orifice metering devices showed a significant reduction in variance between Test-In and Test-Out. This is counter-intuitive for this type of metering device which is expected to show variance in the super-heat which is highly dependent on outdoor ambient conditions and the conditions of indoor air-stream across the evaporator coil. While not definitive, this observation implies that when system charges are performed on fixed orifice devices the super-heat tables may not be properly applied.

During ADM's ride-along inspections there were several instances in which the technicians measurements of return air wet-bulb and outdoor dry-bulb temperatures resulted in an invalid super-heat goal when looked-up on the super-heat table found in the QM Program Guide. In each case the technician reported this to the call center and was directed to use a default goal value of 5 degrees F. This often led to a reduction in the system charge and a likely less than optimum final charge. While such a practice is conservative in that it protects the compressor/system from potential overcharging, it is not ideal when attempting to optimize system performance. There are many variables which impact an accurate assessment of the super heat goal. These include:

1. Accurate measurement of the wet-bulb temperature entering the evaporator coil
2. Accurate measurement of the dry-bulb temperature entering the condenser coil
3. How close the actual system follows the assumed 400 CFM/Ton assumptions used in the goal super-heat tables.
4. How close the generic goal super heat tables match the super-heats specified by the manufacturer

It has been pointed out that the current return air wet-bulb measurements are in many cases not representative of the actual wet-bulb temperature of the air entering the evaporator coil. Also, the assumption of 400 CFM/Ton for these systems appears inaccurate given the current set of airflow measurements. Thus items (1) and (3) likely play a considerable role in the errant super-heat goals, though it is also possible that in some instances the load on the system is too low to accurately charge the system.

Given the number of variable which play a part in establishing super-heat goals, and the uncertainty in which they currently contribute to the invalid goal values, no specific recommendations are provided to address this aspect. As data (particularly on airflow and return air wet-bulb) become more reliable this issue should be re-evaluated by program staff to ensure that systems with fixed orifice metering devices are receiving a proper charge adjustment.

System Performance

Both the Air-side and refrigerant-side system performance and efficiency calculations were found to have significant uncertainty due to the factors mentioned above. Very little correlation was found between the two metrics which limits their current usefulness in measuring program impacts to looking at the relative impacts on system efficiency (e.g. percent improvement rather than an absolute EER). Both metrics do agree that there is some relative improvement in system efficiency between from Test-In to Test-Out though they disagree as to the magnitude. The refrigerant-side metrics predict a 4% improvement while the Air-side metrics predict a 9% improvement. It is ADM's opinion that the refrigerant-side calculations currently represent the most reliable set of results.

5.3. Recommendations

The following are ADM's recommendations for the CMHP in order to improve data reliability and its usefulness to tracking program impacts:

Recommendation 1: Consider combining the duct-sealing and QM measures

Many of the measurements made to facilitate the QM measure are dependent upon functioning duct-work. This includes the supply and return air temperature measurements required to make accurate charge adjustments in systems with fixed orifice metering devices. ADM recommends that the duct-sealing measure be combined with the QM measure in order to facilitate more accurate and consistent air-side measurements.

Recommendation 2: Move the locations of the return and supply air temperature measurements to enable more representative measurements.

Currently the return and supply air temperature measurements are not entirely representative of the air-stream before and after the coil. ADM recommends that the location of these measurements be moved to more representative areas in the system. While the current Testo 605-h1 thermohygrometer units have sufficient accuracy and resolution to perform these measurements, their form is not conducive to accurate sensor placement. In order to facilitate better placement, ADM recommends that Synergy re-implement the *Synergy Technician System* (STS) which communicates wirelessly to the technician's lap-top enabling remote installation of the sensors. Specific recommendations are provided separately for the supply and return air below (with differences pointed out for the up-flow and down-flow system configurations). Note that ADM's recommendations are predicated on the STS data acquisition system's remote sensor placement capability without which some recommendations may prove logistically intractable.

Return Air Temperature

For reasons discussed earlier in this report, it is important that the return air temperature measurement are made in the air-stream as close to the fan inlet as possible. In both up-flow and down-flow systems technicians have good access to the air-handler fan for return air temperature measurements. ADM recommends that Synergy technicians install the temperature and humidity probes from the STS data acquisition system inside the fan cabinet with the probe section inserted into the air-stream as it is pulled into the face of the fan. The suggested locations are shown in Figure 5.1 for down-flow (right) and up-flow (left) configurations.

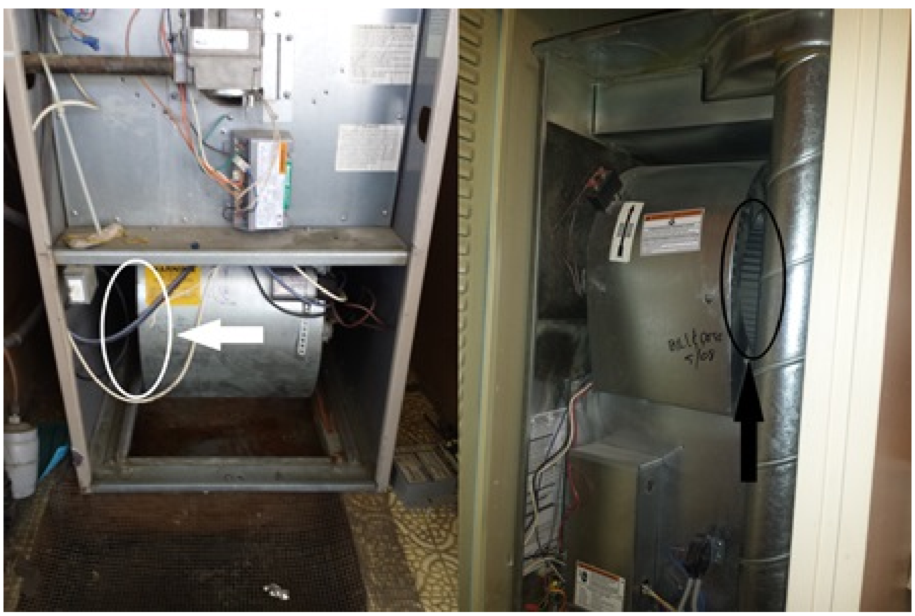

Figure 5.1: Recommended Locations for Return Air Temperature/humidity Sensor Placement

Supply Air Temperature

The supply air temperature measurements should be taken as close to the coil as possible. In both the up-flow and down-flow configurations the technicians have good access to the cooling coils; however, due to the coil geometry the supply side of the coil is difficult to access in down-flow configured units. ADM recommends that the supply air temperature/humidity sensor is placed in the duct-work immediately following the cooling coil. While the location of this measurement within the distribution system is the same for both up-flow and down-flow system configurations, access is made more difficult in down-flow configurations. Note that in both cases ADM recommends drilling a small access hole into the supply air duct-work. This hole can be easily patched by the technician afterwards using UL 181 Approved aluminum sealing tape.

Table 5.1: Recommended Supply Air Temperature/Humidity Locations

System Type	Location Description
Down-flow	It is recommended that the technician access the supply duct section immediately following the air handler under the home. The technician would be required to enter the crawlspace under the mobile home and drill a small hole into the duct facilitating sensor placement directly into the supply stream as it leaves the coil.
Up-flow	In up-flow configurations the technician can place the sensor in the duct-work above the cooling coil through the panel used to access the coil for cleaning/maintenance. Alternatively, where a sufficient length of duct-work exists above the air handle, the technician can drill a small hole in which the sensor probe can be placed directly into the supply stream as it leaves the coil.

Figure 5.2 illustrates the recommended access point for an up-flow system configuration. The preferred location is to drill a hole in the length of duct-work above the coil (shown on the right in Figure 5.2). If such a length is not accessible (e.g. the coils terminates at the ceiling of the closet) then a placement inside the duct-work, accessed through the coil housing, is the suggested alternative. Note that the second approach is only feasible if a wireless data acquisition system, like the STS, is employed. Because the technicians do not currently enter the crawlspace under the home to access/inspect system duct-work, a similar photograph was not taken of the suggested placement in down-flow systems.

Figure 5.2: Recommended Locations for Supply Air Temperature/humidity Sensor Placement in Upflow Systems

Recommendation 3: Change airflow measurement technique - using different methods as necessitated by system configuration

ADM recommends that two different methods be used to measure volumetric system airflow specific to the configuration of the system (up-flow vs. down-flow). Our recommendations for each system configuration are as follows:

Down-flow System Configuration

For down-flow systems ADM recommends that Synergy construct a duct-work transition which can be superimposed over (or replace entirely) the air handler's top access panel. The duct-work should make a smooth transition to a rectangular opening designed to fit a TrueFlow® air handler flow meter. In ADM's ride-along inspections we observed that there should be sufficient space for the proposed measurement and the air-handling equipment is of a common for factor which would facilitate use of a pre-fabricated duct transition. This system would improve airflow measurements in (3) ways:

1. The proposed method is expected to be more time efficient. A prefabricated panel and TrueFlow meter would eliminate the time required to make multiple measurements across the grill. It would also eliminate the time required to measure and input grill area(s) into the anemometer.
2. The TrueFlow meter will take airflow measurements at consistent locations in the air-stream making the results more consistent across tests (Test-In and Test-Out) as well as across sites.
3. Using duct-work with the TrueFlow panel enables the technician to control the air-stream entering the air-handler - eliminating the concern of uncaptured airflow through infiltration, cracks in the closet door, etc.

In some instances ADM identified down-flow systems with a dedicated outside air duct plumbed directly into the air-handler. ADM recommends that in such instances the outdoor duct-work be taped off for the duration of the tests. One concern with this method is that technicians may forget to remove tape from the outside air duct. ADM recommends that this be added as an item to the site checklist. If concerns persist then it may be of interest to require photographic evidence that the tape was removed. Technicians should also receive additional training on the use of the TrueFlow measurement tools.

Up-flow System Configuration

Currently no flow measurements are being taken for up-flow system configurations. For these systems ADM recommends that the TrueFlow® air handler flow meter be used - located below the supply fan in the slot used to retain the air filter. In up-flow systems for which the TrueFlow is too large for the recommended location, ADM recommends that a flow hood type measurement be used. Flow hood measurements should taken at each supply register within the home and the individual flow measurements added together to calculate the total system flow rate. An example of such a tool is the Alnor capture hood produced by TSI. Note that it is considered best practice to take several one-time measurements of a particular parameter (e.g. flow through a particular register) and average the observations.

Recommendation 4: Adjust the Current Default System Airflow Per Ton Value Downwards Based on Measurements

When defaulted observations are removed from the airflow measurement data the average measured CFM/Ton is 271. Currently this value has significant uncertainty and for reasons discussed in the section on airflow measurements is likely a low estimate. However; it does indicate that mobile home systems are designed at less than 400 CFM/Ton. ADM recommends that the current default assumption be revised downwards to 350 CFM/Ton and that the data collected by the recommended changes to airflow measurement be evaluated concurrently to substantiate or revise this default assumption.

Recommendation 5: Consider removing select data fields from EM-HVAC and field forms

Upon review of the data fields in EM-HVAC, ADM found several fields in which no data were entered and several fields with identical values. ADM recommends that these fields be removed from the field data collection forms as they are not in use:

Table 5.2: Data Fields Recommended for Removal

Field Name	Description
QC Test Status	No description provided. All values read "none".
Return Dimension	No description provided. All values read "IWC".
Return Value	No description provided. All values read Zero.
Return measurement Location	No description provided. Values read either NA or "NotSet".
Supply Dimension	No description provided. All values read "IWC".
Supply Value	No description provided. All values read Zero.
Supply measurement Location	No description provided. Values read NA.
Supply Ductwork	No description provided. All values read Zero.
Supply Discharge	No description provided. All values read Zero.
Supply TESP	No description provided. All values read Zero.
Approach used for System Charge	No description provided. All values read "No".

6. Appendix

The following are equipment cut-sheets for the recommended air-flow measurement equipment. Since the recommended Synergy data acquisition system is proprietary to Synergy, data sheets cannot be provided here. For data sheets describing the data collection equipment currently in use see the QM Program Guide document.

TrueFlow® Air Handler Flow Meter

The Energy Conservatory's **TrueFlow® Air Handler Flow Meter** provides a simple and accurate measurement of air flow through residential air handlers and filter grills.

The TrueFlow® Meter temporarily replaces the filter in a typical air handler system when measuring air flow. If the filter location is directly adjacent to the air handler, the TrueFlow® Meter will measure the total air handler flow. If the filter is located remotely at a single central return, it will measure the air flow through the central return.

Other methods for estimating the air handler flow rate, such as the temperature rise method, static pressure and fan curve method, and the Duct Blaster® isolated return method, have been found to be problematic or time-consuming.

Extensive field testing has shown that the TrueFlow® Meter:

- Fast and easy to use in the field. Direct cubic feet per minute values are delivered in 2 to 3 minutes without extensive calculations.
- Measures flow through air handlers rated from 1 ton to 5 tons, 400 to 2,000 CFM.
- Works in a wide range of return plenum and air handler fan configurations.
- Changes size quickly to fit standard and custom filter slots.
- Four times more accurate than the single point temperature rise method.
- Works with any manometer having a resolution of 1 Pa or 0.005 inches of water.
- Flow accuracy of ±7% for most applications when used with a 1% accurate pressure gauge.
- Works with our DG–700 Digital Pressure and Flow Gauge to directly display air flow in cubic feet per minute through the metering plate.

TrueFlow® Air Handler Flow Meter Specifications

Accuracy* of flow using a DG–700 Digital Pressure and Flow Gauge or equivalent (+/- 1% of reading)	**± 7% of indicated reading**
Accuracy of flow using analog pressure gauges	**± 9% of indicated reading**
Range of flow: #14 Metering Plate	**365 to 1,565 CFM (620 to 2,600 cmh, 172 to 740 l/s)**
Range of flow: #20 Metering Plate	**485 to 2,100 CFM (825 to 3,570 cmh, 225 to 990 l/s)**
Storage and operating temperature range	**–40°F to +150°F (–40°C to +65°C)**
Nominal sizes of plates with gasket material connected: #14 Metering plate	**14.5 in. by 20.5 in. (37 cm by 52 cm)**
Nominal sizes of plates with gasket material connected: #20 Metering plate	**20.5 in. x 20.5 in. (52 cm by 52 cm)**
Weight—Metering plates, spacers and carrying case	**13 lbs. (5.9 kg)**
The TrueFlow® Meter will fit in most standard size filter slots.	
The compatible filter sizes with #14 Metering plate #1 are	**14 x 20, 14 x 25, 16 x 20, 16 x 24, 16 x 25, 18 x 20**
The compatible filter sizes with #20 Metering plate #2 are	**20 x 20, 20 x 22, 20 x 24, 20 x 25, 20 x 30, 24 x 24**

Specifications subject to change without notice.

*Accuracy is installation dependent. Obstructions within 6 inches (15 cm) upstream or 2 inches (5 cm) downstream of the metering plate that block air flow through any of the metering holes may reduce the flow accuracy.

TrueFlow® Plate #14 with 2x20 spacer at a central return..

TrueFlow® Plate #14 with 2x20 spacer at filter slot between the return and the air handler.

TrueFlow® Air Handler Flow Meter kit includes: #14 and #20 Plates, 8 spacers, carrying case, hose, static pressure probe and manual.

Complete service and technical support is built in.

All of our products come with a full two-year warranty on parts and labor, and access to the most knowledgeable customer service staff in the industry. If you have questions on the use of our products or how to handle unusual situations, you can count on us to give dependable answers. We always stock a complete line of replacement parts and can respond quickly to any service or equipment problem.

Our nearly 30 years of expertise goes beyond simply knowing about equipment. The Energy Conservatory's on-going research, active participation with technical associations, and close working relationships with the world's leading building scientists keeps us involved in the development and field testing of many of the performance testing industry's techniques. This means you always have the most up-to-date information and testing procedures.

To order, or for more information contact:

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Air Volume Instruments

Shown: LoFlo Balometer® Capture Hood Model 6200D

LoFlo Balometer® Capture Hoods

Models 6200, 6200D, 6200E, and 6200F

The LoFlo Balometer® Capture Hood is the ideal way to measure very low volumetric flow. Measure confidently and accurately supply or return flows from 10 to 500 cfm (17 to 850 m^3/h). This light weight instrument is great for residential or light commercial use.

Features and Benefits

- Uses 4 C-size alkaline batteries
- Weighs only 6.5 lb (3 kg) with 2 ft x 2 ft (610 mm x 610 mm) hood attached
- Simulated analog display shows air trends and digital readings
- Use with or without a hood

Easily observed trend values and fast meter response make the LoFlo Balometer® Capture Hood the preferred tool of residential air balancers.

Rugged. Reliable. Professional.

TSI | ALNOR®

Balometer®

Capture Hoods

LoFlo Models

Specifications

Range
10 to 500 cfm (17 to 850 m³/h) (4.7 to 236 l/s)

Accuracy
±(3% + 5 cfm) [±(3% + 8,5 m³/h, 2,4 l/s)]

Height

Model 6200	22 in. (559 mm)
Model 6200D	34.5 in. (876 mm)
Model 6200E or base only	15.5 in. (394 mm)
Model 6200F	32 in. (813 mm)

Weight
about 6 lbs (2.7 kg) with hood
4.6 lbs (2.1 kg) base only

Base Diameter

Opening	13.3 in. (338 mm) diameter
Hood sizes	16 in. x 16 in., 2 ft x 2 ft, or 26 in. x 26 in. (406 mm x 406 mm, 610 mm x 610 mm, or 650 mm x 650 mm)
Display	3.5 digit, .44 in. (11 mm) high, digital display with 26 segment simulated analog display
Resolution	1 cfm from 10 to 500 cfm (0.1 l/s from 4.7 to 9.9) (1 l/s from 10 to 236 l/s)

Power Source
4C 1.5V alkaline batteries (optional Nickel Cadmium)

Battery Life
10 hrs. minimum with continuous use

Model Description

Model 6200	with 16 in. x 16 in. (406 mm x 406 mm), 8 in. (200 mm) tall hood
Model 6200D	with 2 ft x 2 ft (610 mm x 610 mm) hood
Model 6200E	with base only, metric
Model 6200F	with 16 in. x 16 in. (406 mm x 406 mm), 18 in. (457 mm) tall hood

Optional Accessories

634620110	2 ft x 2 ft (610 mm x 610 mm) hood and frame kit
634620085	16 in. x 16 in. (406 mm x 406 mm), 18 in. (457 mm) tall hood and frame kit
634620120	16 in. x 16 in. (406 mm x 406 mm), 8 in. (200 mm) tall hood and frame kit
634620130	26 in. x 26 in. (650 mm x 650 mm) hood and frame kit

The LoFlo Balometer® Capture Hood is mainly used in residential or light commercial applications for taking measurements from 10 to 500 cfm (17 to 850 m³/h). The compact size allows them to be used where full size hoods would not fit such as over bathroom stalls or filing cabinets.

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