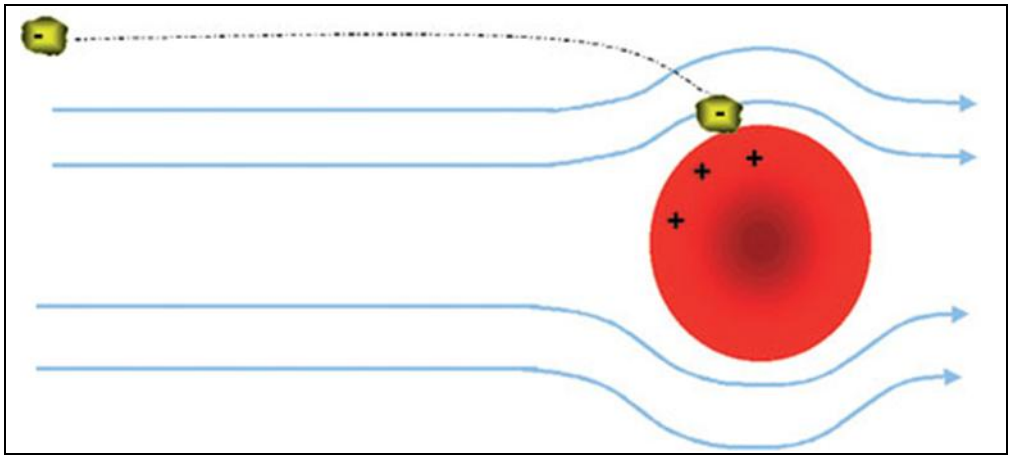
HVAC Electrostatic Filters

ET11SCE1190 Report



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EXECUTIVE SUMMARY

This study seeks to determine if electrostatic air filters provide energy savings and demand reduction to air conditioning units. Electrostatic filters are intended to replace disposable pleated media filters.

Air filtration is an integral part of air conditioning systems, but filters increase the energy consumption of these systems. They add resistance to the air stream, and this resistance increases as contaminants amass in the filter. Most commercial and residential air conditioning units use pleated media filters, which are replaced on a scheduled basis.

Electrostatic filters charge airborne particles, which attracts and attaches them to the filter media. This process increases the removal rate of contaminants. At the same time, an electrostatic filter imposes a lower pressure drop across the air stream than a pleated media filter.

This study has three parts, as follows:

1. Compare airflow pressure drop of an electrostatic filter to that of a pleated media filter, using independent test results. These tests were conducted according to the American Society of Heating, Refrigeration, and Air Conditioning Engineers Standard ASHRAE 52.2.
2. Determine the actual filter loading, in a packaged air conditioning unit serving an office building in a dirty suburban environment.
3. Measure the performance degradation, if any, of an air conditioning unit as the airflow resistance of its filter section increases. This portion of the study was conducted in a controlled environment.

Independent performance tests showed that the electrostatic filter collects ten times more airborne contaminants, at the same air pressure drop, than the baseline 2-inch(") thick pleated media filter.

During the three-month field test in an office building air conditioning unit, a baseline 2" pleated media filter loaded to a point where its final air pressure drop was only slightly more (0.02 inches water column) than its clean pressure drop. During this test period, the filter collected 0.008 pounds of contaminants.

During the controlled environment test, within the normal range of filter usage, degradation of air conditioning unit performance was observed. The air conditioning (AC) unit supply fan airflow, sensible cooling capacity, and total cooling capacity all reduced as airflow resistance was added to the system.

The lower supply airflow and sensible cooling did not result in higher room air temperature in the test chamber, nor did it result in longer run times for the air conditioning unit. The supply fan's amps reduced as external static pressure was added to the system and the airflow decreased. The compressor and condenser amps did not change during the tests. Refer to FIGURE 1 for a summary of the air conditioning power demand.

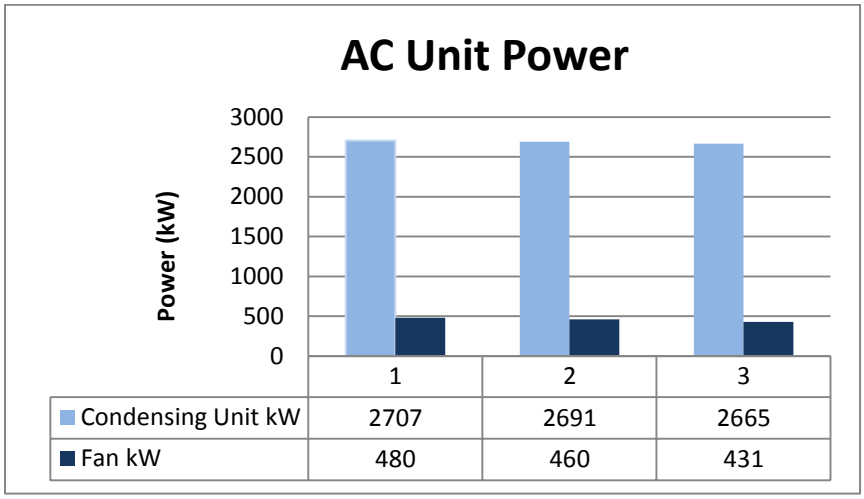


FIGURE 1. AC UNIT POWER

Based upon the findings of this study, it is not recommended that electrostatic filters be considered for future evaluation. This technology may provide better filtration performance than the incumbent pleated media filters, but it does not reduce electrical energy usage or electrical demand in a properly maintained air conditioning system.

ABBREVIATIONS AND ACRONYMS

AC	Air Conditioning
AHRI	Air Conditioning, Heating, and Refrigeration Institute
AHU	Air Handling Unit
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration, and Air Conditioning Engineers
Btu/hr	British Thermal Unit/hour
CFM	Cubic Feet per Minute
DB	Degrees Dry Bulb
DPT	Dew Point Temperature
DX	Direct eXpansion
EEC	Energy Education Center
EMS	Energy Management System
°F	Degrees Fahrenheit
fpm	feet per minute
Ft^3	Cubic feet
g, gm	Grams
HEPA	High Efficiency Particulate Arrestance
HTTC	Heating, Ventilation, and Air Conditioning Technology Test Center
HVAC	Heating, Ventilating, and Air Conditioning
in. w.c	Inches water column
kW	Kilowatt
kWh	Kilowatt-Hour
lb/min	Pounds/minute
lbs	Pounds

NIST	National Institute of Standards and Technology
R	Degrees Rankine
RTD	Resistive Temperature Device
SCE	Southern California Edison
SEER	Seasonal Energy Efficiency Ratio
TC	Thermocouple
VSD	Variable Speed Drive
WB	Degrees Wet Bulb
w.c.	water column
wt.	weight

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INTRODUCTION

Heating, Ventilating, and Air Conditioning (HVAC) systems provide the important function of removing contaminants from building occupants' breathing air. Air filters are an integral part of these systems. In residential and most commercial buildings, air conditioning (AC) unit air filters are constructed of porous fibers contained in a cardboard frame. These filters are designed to slip into a rack within the AC unit and remove particles from the airstream. For the purposes of this report, this technology will be referred to as 'media filters'.

All components within the airstream of an HVAC system add resistance to the airflow. The system's fan overcame this resistance. The more resistance encountered in the system airflow, the more energy is required by the fan(s) to circulate the air. In addition, as airflow resistance increases, the rate of conditioned air delivered decreases. This decrease, in turn, can reduce the amount of cooling or heating delivered to conditioned spaces.

In media filters, airborne contaminants adhere to the fibers by coming into direct contact or by being caught between the fibers (straining). As contaminants amass in the filters, system pressure drop increases. These filters are then replaced with new, clean filters, on a set time interval. This interval varies, but good practice is to replace filters every three months.

Electrostatic filters also consist of porous fibers housed in a frame. However, the media in this filter is polarized by a high voltage current. Airborne particles are charged with an opposite polarity, which causes them to become attracted and attached to the charged media. This process increases the removal rate of contaminants. At the same time, an electrostatic filter has a lower pressure drop than an uncharged media filter.

Electrostatic air filters were first used in commercial AC systems in the 1950's. By the early 1960's, however, they had been almost completely replaced by media filters. They continue to be used in industrial applications, as well as in self-contained residential air cleaners.

Several manufacturers now offer electrostatic air filters for installation in central HVAC systems. These systems are used in most commercial and residential markets, including:

- Retail
- Small Office
- Education
- Single- and Multi-Family Residences

ASSESSMENT OBJECTIVES

The objective of this study was to determine whether the electrostatic filters reduce energy consumption of an air conditioning system, as compared to the incumbent media filters. The study scope was:

- Assess whether electrostatic air filters perform as well as media filters. Performance is defined as the ability to remove airborne particulates.
- Estimate the amount of particulates collected in a media filter in a 3-month period. This determines the increase in air pressure drop in the AC system caused by the filter as it loads.
- Measure the increase in supply fan and cooling system energy as air pressure drop increases.

The scope of this study was limited to comparing the performance of actively-charged electrostatic filters to media filters. The baseline media filters are prevalent in the residential and commercial markets. Excluded from this study were:

- Passively charged electrostatic filters. These are filters whose media is originally imparted with an electric charge, but do not receive a constant electric charge.
- Higher efficiency media filters. These filters are used in several commercial markets, including offices and medical buildings. They have, as their name states, higher filtration efficiency than standard media filters. They also impart a higher air pressure drop on the AC system.
- Carbon filters, used to remove gaseous contaminants from the airstream.

TECHNOLOGY/PRODUCT EVALUATION

This product evaluation was a combination of testing by independent laboratories, field observations of existing AC system filtration, and laboratory testing of an AC system under increasing air pressure drops.

FILTER PERFORMANCE TEST

Air filter performance is rated in accordance with American Society of Air Conditioning and Refrigeration Engineers (ASHRAE) Standard 52.2-2007 – Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size. This standard measures:

- The filter's ability to remove particles from the airstream
- The filter's resistance to airflow, or the pressure drop it imposes upon an air conditioning system

Performance testing occurred on two products:

1. American Air Filter Model 5700, 24" x 24" x 2" thick media filter
2. AspenAir Model FG242525 electronic air cleaner

In the ASHRAE 52.2 test, the filter is placed in a duct, through which a constant airflow is delivered. This air first flows through High Efficiency Particulate Air (HEPA) filters, to remove contaminants. Then, a uniform aerosol of loading dust is injected into the airstream upstream of the test filter. This dust is a mix of fine dust, powdered carbon, and cotton lint. The air velocity across the test filter face is maintained at 500 feet per minute (fpm).

The amount of dust downstream of the filter and the filter's pressure drop are measured and recorded. The difference between the upstream and downstream dust quantity determines the filter's efficiency.

Due to the specific requirements of Standard 52.2, filter tests are conducted by independent laboratories with systems specifically designed for these tests. Tests were conducted by:

1. LMS Technologies, Inc., Bloomington, MN (baseline media filter test)
2. RTI International Research, Triangle Park, NC (Electrostatic filter test)

FILTER LOADING TEST

The loading that an AC unit's air filter experiences during its life is dependent upon many factors, including:

- Level of outdoor air contaminants. Wind, weather, local topography, and industrial pollution contribute to outdoor air quality. For example, air conditioning units located near rural or desert areas may experience a higher level of contaminants.
- AC unit hours of operation
- Outside air quantity. Most airborne contaminants originate outdoors. An AC unit operating in temperate weather will be in airside economizer mode, bringing in 100% outside air. On the other hand, AC units operating during

summer months will be in minimum outside air mode much of the time. Residential systems often bring in no outside air.

- Indoor air contaminants
- Length of time in service

Due to the wide variation of conditions an air filter may experience, a field test was conducted in an AC unit under the following conditions:

- Serving a commercial office building, operating 5 days/week
- In Irwindale, California. This location is:
    - Downwind of much of the urban Los Angeles basin
    - Directly downwind of open desert landscaping of the San Gabriel River basin
    - South and west of active quarries
- In late winter/early spring - during this period of moderate weather, the airside economizer operates during the majority of the day.

The above combination would result in filter loading greater than or equal to most field conditions encountered in California.

The test AC unit was a Carrier Model #50TFQ005 rooftop packaged heat pump. The unit was located on the roof of an office building located at 6042A North Irwindale Avenue, Irwindale, CA. This AC unit supplies a nominal 1,600 Cubic Feet per Minute (CFM) supply air and uses two 16” x 25” x 2” thick Tri Dim pleated media filters.

Two new air filters were weighed, using a Sartorius Model# Combics 1 Plus scale. The filters were then installed in the AC unit. The combined static pressure drop across the clean filters and the downstream cooling coil was then measured, using a The Energy Laboratory digital pressure gauge. Due to the configuration of the AC unit, a static pressure sensor could not be inserted between the filter and the downstream cooling coil.

The filter remained in the operating AC unit for 90 days. At the end of this test, the combined static pressure drop across these filters and cooling coil was again measured and recorded. The filters were then removed and reweighed.

AIR CONDITIONING UNIT PERFORMANCE TEST

TEST UNIT

The test unit installed in the test chambers of the Technology Test Centers (TTC) was a 3-ton split system equipped with an air-cooled condenser and a single-speed compressor. Prior to any tests, the AC system was charged with the proper amount of refrigerant (R-410A) following the manufacturer’s instructions. Figure 2 shows a picture of the indoor section of the AC unit installed in the indoor test chamber. The following lists the model numbers for the main component of the AC unit under test.

Outdoor condensing unit:	Trane 4TTB3036D1000A, refrigerant R-410A, nominal 3-ton
Indoor coil:	Trane 4TXCB042B3, nominal 3-ton
Gas furnace (blower unit):	Trane TUD1B080A9361A, 4-speed direct blower drive

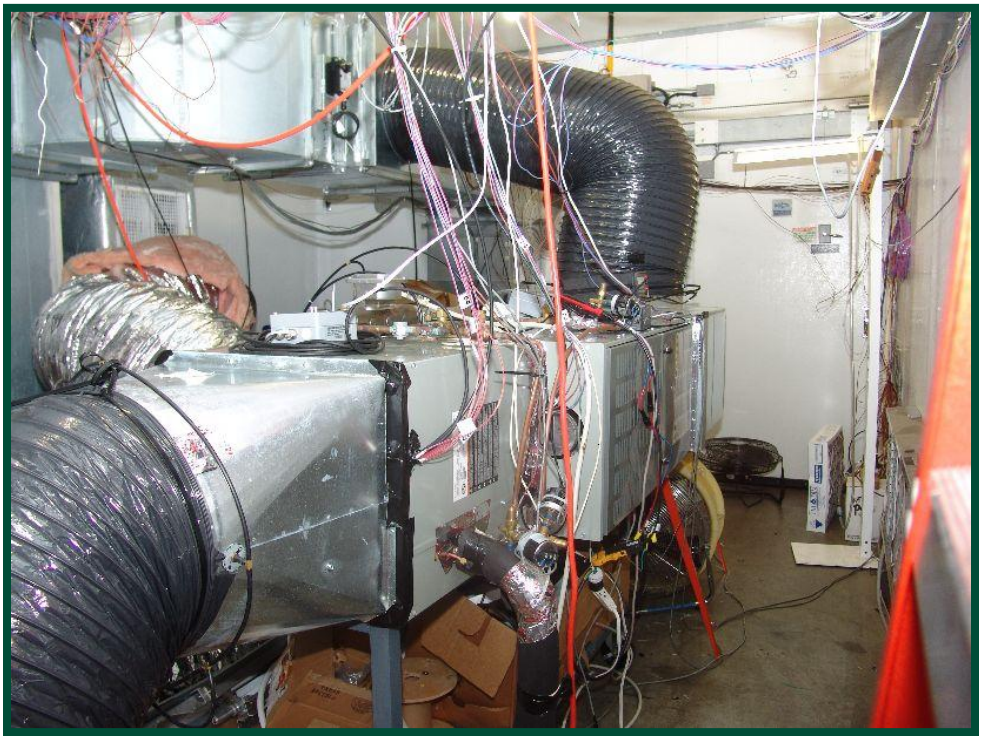


FIGURE 2. THE INDOOR SECTION OF THE AIR CONDITIONING UNIT IN THE INDOOR TEST CHAMBER

TEST METHODOLOGY

Generally, the test unit's capacities and performance characteristics were determined following the ASHRAE Standard 37-2009,[i] and the Air Conditioning, Heating, and Refrigeration Institute (AHRI) Standard 210/240.[ii] However, several modifications were made to the methodology to meet the needs of this project.

Instead of conventional tests, performance testing occurred at several prescribed increments of indoor unit filter section pressure drop. To accurately control the pressure drop across the filter section, an opposed blade manual volume damper was installed in this section, in lieu of actual filters. The damper could be adjusted to simulate different air pressure drops required for the test.

Rather than maintaining full-load conditions in the indoor test chamber, a constant, lower heat load was maintained and the AC unit was allowed to cycle according to its normal thermostat-controlled operation. The indoor chamber's thermostat was set to 75°F dry bulb (DB) and electric heaters were used to impose a constant sensible cooling load on the unit. This load was calculated to be 10% less than the unit's sensible cooling capacity at 95°F outdoor air temperature, to simulate operation of a typically oversized unit. Latent heat was added to the conditioned space with the ultrasonic humidifier.

At the test onset, the AC unit's actual supply flow, Cubic Feet per Minute (CFM), was measured to be 970 CFM, with 0.1" w.c. pressure drop across the filter section. The manufacturer's published capacity for the AC unit is 24,000 British thermal units (Btu) per hour (/hr) sensible and 7,525 Btu/hr latent at the following operating conditions:

- 970 CFM supply
- 95°F entering condenser air temperature
- 75°F DB entering supply air temperature
- 63°F WB entering supply air temperature

Ninety percent of this capacity was calculated to be 21,600 Btu/hr sensible, and 6,770 Btu/hr latent. To maintain the sensible load throughout the tests, 6,000 Watt (W) (20,450 Btu/hr) resistance heat was added to the indoor chamber. To maintain this latent load throughout the tests, 0.13 lbs/minute steam was added to the indoor chamber by the humidifier. Table 1 summarizes the test points for the AC unit performance test.

TABLE 1. AC UNIT PERFORMANCE TEST CONDITIONS

Test Condition	Outdoor Air Temp (◦F DB)	Room Air Temp (◦F DB)	Room Air Temp (◦F WB)	Filter Static Pressure (in. w.c.)	Sensible Load Imposed (Btu/hr)	Latent Load Imposed (Btu/hr)
A1	95	75	63	0.1	20,450	6,770
A2	95	75	63	0.25	20,450	6,770
A3	95	75	63	0.5	20,450	6,770

For each test, the AC unit operation was brought to steady-state conditions, with constant sensible and latent cooling loads, intake condenser air temperature, and room temperature. The supply fan and condensing unit were allowed to cycle on and off in response to the indoor test chamber's thermostat. Then, these conditions were maintained for two hours, and data was measured every 20 seconds.

Power measurements included input to compressor, condenser, and evaporator fan, and controls and other items required as part of the system for normal operation. The outdoor control environment room's 10-foot high ceiling provided sufficient clearance (more than the required 6 feet) from condenser discharge. The required distance of at least 3 feet was provided between the test room's walls and the equipment side surfaces. The following highlights the key aspects of air- and refrigerant-enthalpy methods.

The air- and refrigerant-enthalpy methods described by test standards AHRI 210/240 and ASHRAE 37 were used to measure psychrometric properties of the air and refrigerant properties. This involved installing instrumentation at specified locations as outlined by these standards. The air and refrigerant properties were used to determine the cooling capacities and efficiencies of the AC unit at full load.

AIR-ENTHALPY METHOD

The air-enthalpy method used measured psychrometric properties of air flowing across the AC unit's evaporator coil. These measurements included dry-bulb temperature (DBT), wet-bulb temperature (WBT), and the relative humidity (RH) of air in the upstream and downstream of the coil at measured airflow rates. Accordingly, the air enthalpy change was used to determine the gross cooling capacity of the unit. Using the measured evaporator fan power input, the net cooling capacity of the unit was determined. This was done by subtracting evaporator fan heat from the gross cooling capacity.

REFRIGERANT-ENTHALPY METHOD

The refrigerant-enthalpy method used measured refrigerant properties at the inlet and outlet of the evaporator coil. The gross cooling capacity of the unit was determined using the refrigerant enthalpy change, and mass flow rate. Refrigerant enthalpy changes were determined from pressure and temperature measurements entering and leaving evaporator coil. A corriolis mass flow meter was installed in the liquid line to measure the liquid refrigerant flow rate. To obtain reliable refrigerant flow rate, the refrigerant must be in a 100% liquid state. Pressure transducers were installed before and after the mass flow meter to measure and record the pressure drop across the flow meter. Monitoring the pressure drop across the meter ensured that liquid refrigerant did not flash and undergo a saturation temperature change of larger than $3^{o}F$, as prescribed by the test standards. In addition, installation of two sight glasses immediately upstream and downstream of the flow meter confirmed the refrigerant was in a 100% liquid state at the inlet and outlet of the flow meter. Refrigerant temperature and pressure measurements were taken of the refrigerant vapor entering and leaving the compressor in the refrigerant lines approximately 10 inches from the compressor shell.

MONITORING POINTS

The monitoring plan included 94 points. The following list captures the core monitoring points. Figure 3, not to scale, depicts the schematic diagram of all sensor locations used in this project. As depicted, for critical temperature measurements including air entering and leaving the indoor unit, a temperature grid was assembled.

1. **Refrigerant side**
   - Compressor discharge temperature and pressure
   - Compressor suction temperature and pressure
   - Liquid line temperature and pressure before and after the mass flow meter
   - Refrigerant mass flow rate
2. **Indoor air**
   - Dry Bulb (DB) Temperature
   - Wet Bulb (WB) Temperature
   - RH
3. **Outdoor air**
   - DB
   - RH

4. **Indoor unit**
   - Air DB at the inlet of evaporator fan
   - Air DB and Dew Point Temperature (DPT) at the inlet of evaporator coil
   - Air DB, DP, and RH at the outlet of evaporator fan
   - Evaporator airflow
5. **Condensate mass (using digital scale)**
6. **Power**
   - Compressor
   - Condenser fan
   - Evaporator fan
   - Total indoor unit
   - Total condensing (outdoor) unit
   - Auxiliary heaters for indoor test chamber sensible load

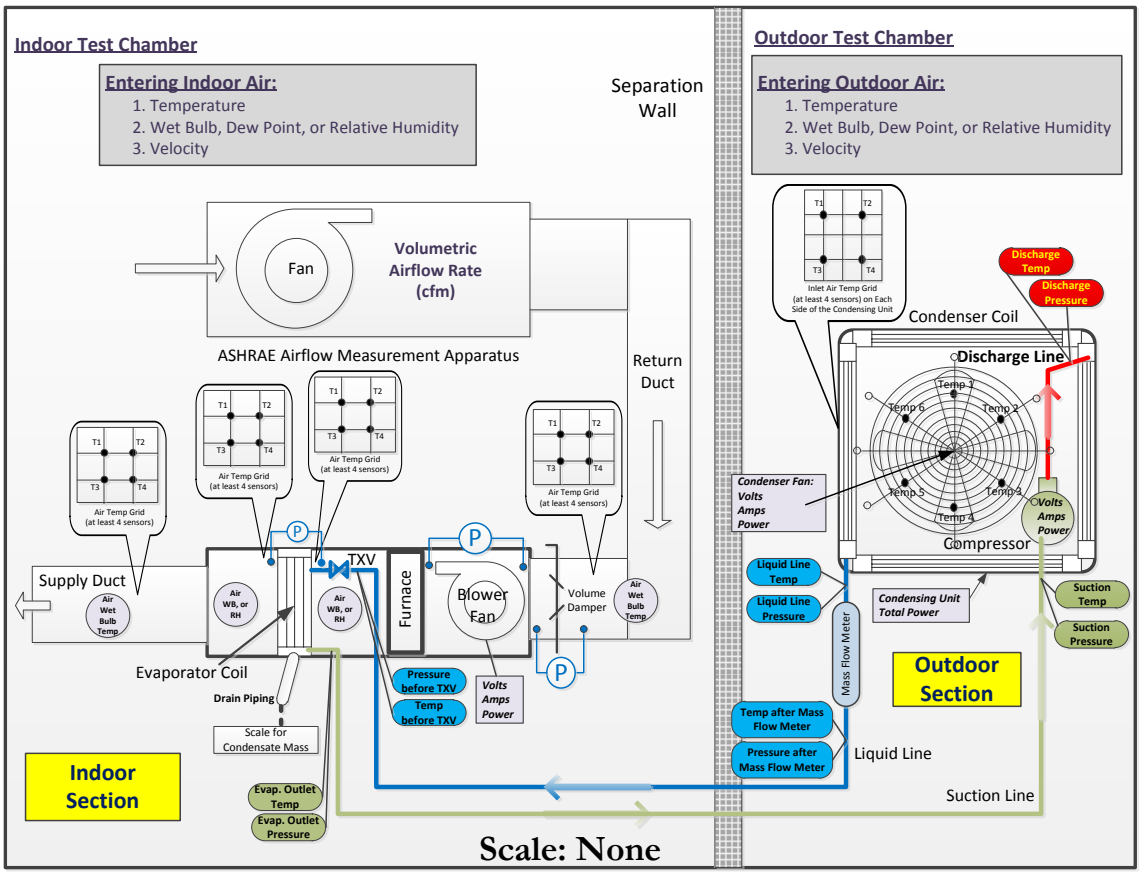


FIGURE 3. SCHEMATIC DIAGRAM OF TEST SETUP WITH MONITORING POINTS

DATA ACQUISITION

The logging of test data occurred by using the National Instruments' SCXI data acquisition system. The data acquisition system was set up to scan 94 data channels in 20-second intervals and log data in one-minute intervals. As part of TTCs quality control protocol, the design of the data acquisition system is completely independent of the supervisory control computer. This approach eliminated compromising the data collection by the control sequence's priority over data acquisition.

Screening of collected data ensured key control parameters were within the acceptable ranges. In the event that any of the control parameters fell outside the acceptable limits, the problem was flagged and a series of diagnostic investigations were carried out. Corrections were then made and tests were repeated, as necessary. After the data passed the initial screening process, data were imported to a customized refrigeration analysis model, where detailed calculations were performed. Appendix A lists the specifications for the instruments.

RESULTS

FILTER PERFORMANCE TEST

Table 2 and Table 3 contain data read from the results of the ASHRAE 62.2-2007 tests performed on the baseline pleated media filter and the electrostatic filter, respectively. Copies of the test results appear in Appendix C.

TABLE 2. FILTER PERFORMANCE, AMERICAN AIR FILTER MODEL 5700 MERV-6 24" X 24" X2" PLEATED MEDIA FILTER

Filter Loading (grams (gm))	Pressure Drop (in. water column (w.c.))
0 (Initial)	0.25
50	0.35
100	0.47
150	0.6
200	0.8
225	1.0

TABLE 3. FILTER PERFORMANCE, ASPENAIR MODEL FG242525 ELECTRONIC AIR CLEANER

Filter Loading (grams (gm))	Pressure Drop (in. water column (w.c.))
0 (Initial)	0.25
240	0.37
628	0.50
1216	0.62
1893	0.75
2373	1.02

Filter Loading Test

Table 4 contains measured data from the field test of the baseline pleated media filter, after three months of operation.

Table 4. Filter Loading

Stage	Date	Filter Wt (lbs)	Pressure Drop (in. w.c.)*
Initial	February 21, 2012	1.344	0.25
Final	May 15, 2012	1.352	0.27

* - Filter/cooling coil pressure drop. Filter pressure drop could not be isolated, due to AC unit configuration.

Air Conditioning Unit Performance Test

For each test, data points were measured every 20 seconds for a 2-hour period, under steady-state conditions. Table 5 summarizes the supply fan airflow for each test.

Table 5 . Supply Fan Airflow

Test Condition	Filter Static Pressure (in. w.c.)	Average Supply Fan Flow (CFM)	Maximum Supply Fan Flow (CFM)	Minimum Supply Fan Flow (CFM)*
A1	0.09	970	976	954
A2	0.25	914	920	903
A3	0.49	793	805	542

*Minimum airflow was observed during the first 20-second reading upon fan start-up.

Table 6 summarizes the measured electrical current draw of the supply fan and condensing unit for each test.

Table 6. Air Conditioning Unit Power

Test Condition	Filter Static Pressure (in. w.c.)	Average Supply Fan Volts	Average Supply Fan Power (Watt)	Average Condensing Unit Volts	Average Compressor Power (Watt)	Average Condenser Power (Watt)
A1	0.09	114	479	207	2585	122
A2	0.25	115	460	208	2570	123
A3	0.49	115	431	209	2540	124

Table 7 summarizes the AC unit's measured test conditions, and Table 8 summarizes the average DX cooling coil's measured inlet and outlet conditions during these tests.

TABLE 7. AIR CONDITIONING UNIT TEST CONDITIONS

Test Condition	Supply Fan Flow (CFM)	Space Temperature (ºF DB)	Space Temperature (ºF WB)	Room Input Sensible Load (W)	Room Input Latent Load (lb/min)
A1	970	75	64	5,948	0.13
A2	914	75	64	5,882	0.12
A3	793	75	64	5,866	0.13

TABLE 8. AIR CONDITIONING UNIT COOLING COIL TEMPERATURES

Test Condition	Supply Fan Flow (CFM)	Coil Inlet Temp (ºF DB)	Coil Inlet Temp (ºF WB)	Coil Outlet Temp (ºF DB)	Coil Outlet Temp (ºF WB)
A1	970	76.0	57.9	57.4	53.4
A2	914	75.9	57.2	56.8	52.7
A3	793	75.9	57.0	55.2	51.4

These data were used to calculate sensible and latent cooling capacities. Equation 1 calculates coil sensible cooling output (Q_{sens}):

EQUATION 1. SENSIBLE HEAT

$$Q_{sens} = 1.08 \times CFM \times (T_{DBin} - T_{DBout}) \quad (1)$$

Water Vapor Partial Pressure at DBT (Pws) is calculated using measured DPT in Equation 2:

EQUATION 2. WATER VAPOR PARTIAL PRESSURE

$$P_{ws} = exp\left(\frac{C_1}{R} + C_2 + C_3 \times R + C_4 \times R^2 + C_5 \times R^3 + C_6 \times ln\ R\right)^1 \quad (2)$$

Where:

R = DPT in Rankine
C_1 = – 1.0440397 E + 04
C_2 = – 1.1294650 E + 01
C_3 = – 2.7022355 E – 02
C_4 = 1.2890360 E – 05
C_5 = 2.4780681 E – 09
C_6 = 6.5459673

From the partial pressure, Equation 3 calculates the air's humidity ratio (*ω)*:

EQUATION 3. AIR HUMIDITY RATIO

$$\omega = 0.621945 \times P_{ws} \div \ 14.696 - P_{ws} \quad (3)$$

Equation 4 calculates air enthalpy (*h)*:

EQUATION 4 . AIR ENTHALPY

$$h = (0.24 \times T_{DB)} + \omega \times (1061 + 0.44 \times T_{DB}) \quad (4)$$

Where ω = air humidity ratio

Finally, Equation 5 calculates the total cooling capacity ($Q_{TOTAL)}$ of the DX coil:

EQUATION 5. TOTAL COOLING CAPACITY

$$Q_{TOTAL} = 4.5 \times CFM \times (h_{in} - h_{out}) \quad (5)$$

Table 9 presents the calculated sensible and total cooling capacity of the air conditioning unit, under the three test conditions.

TABLE 9. AIR CONDITIONING UNIT COOLING COIL OUTPUT

Test Condition	Supply Fan Flow (CFM)	Sensible Cooling (Btu/hr)	Total Cooling (Btu/hr)
A1	970	19,490	26,890
A2	914	18,980	25,980
A3	793	17,670	24,250

Finally, the run time of the AC unit during the two-hour steady-state tests were recorded. The unit total kWh was calculated for this period by multiplying the measured AC unit's kW by the time period, then summing the kWh. Equation 6 provides the kWh calculation:

EQUATION 6. AC UNIT ENERGY USAGE

$$kWh = \quad Unit\ kW \times \frac{20\ minute}{60\ minute} \quad (6)$$

Total run time was tabulated from the 20-second results of each test. Table 10 presents these results.

TABLE 10. AIR CONDITIONING UNIT RUN TIME – 2 HOUR TEST

Test Condition	Supply Fan Flow (CFM)	Energy Input (kWh)	Test Condition (hours)
A1	970	4.99	1.56
A2	914	4.85	1.53
A3	793	4.96	1.60

DATA ANALYSIS

The results of the ASHRAE 62.2 test indicate that the electro-static filter has approximately 10 times the dust holding capacity as the baseline 2" thick pleated media filter. Figure 4 compares the filter capacities.

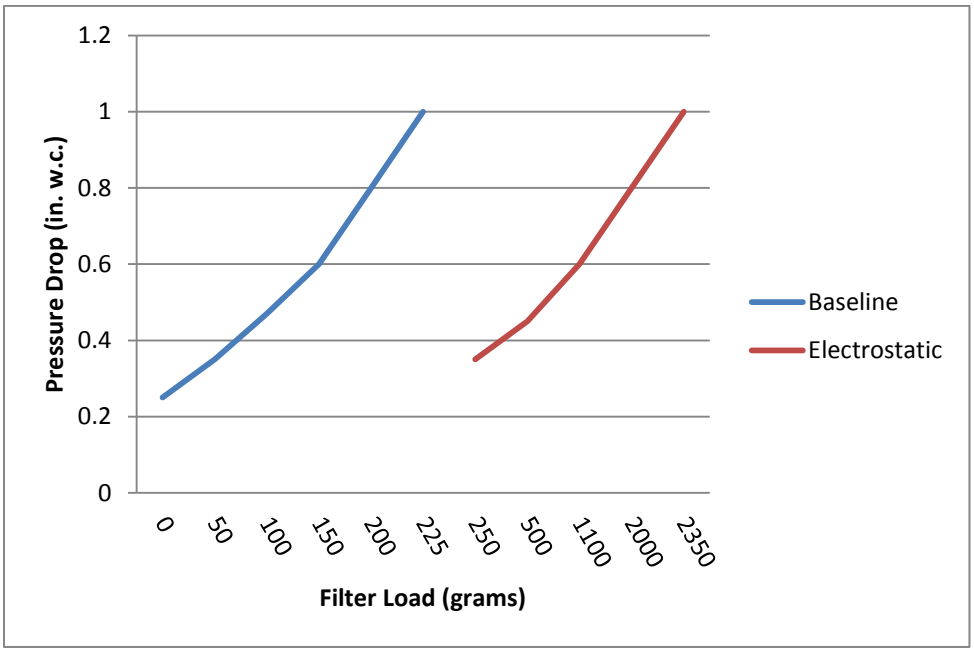


FIGURE 4. FILTER PERFORMANCE

Over the 3-month field test period, the pleated media filter collected 0.008 pounds of airborne contaminants. The measured pressure drop across the filter/cooling coil increased during this period from 0.25" w.c. to 0.27" w.c., an increase of 0.02" w.c.

As the controlled environment AC unit's filter section static pressure drop was increased from 0.10" to 0.50" w.c., the supply fan CFM decreased from 970 CFM to 793 CFM, an 18% reduction (Figure 5). The sensible cooling capacity of the test AC unit decreased from 19,490 Btu/hr to 17,670 Btu/hr, a 9% decrease (Figure 6). Total cooling capacity decreased 10%, from 26,890 Btu/hr to 24,250 Btu/hr.

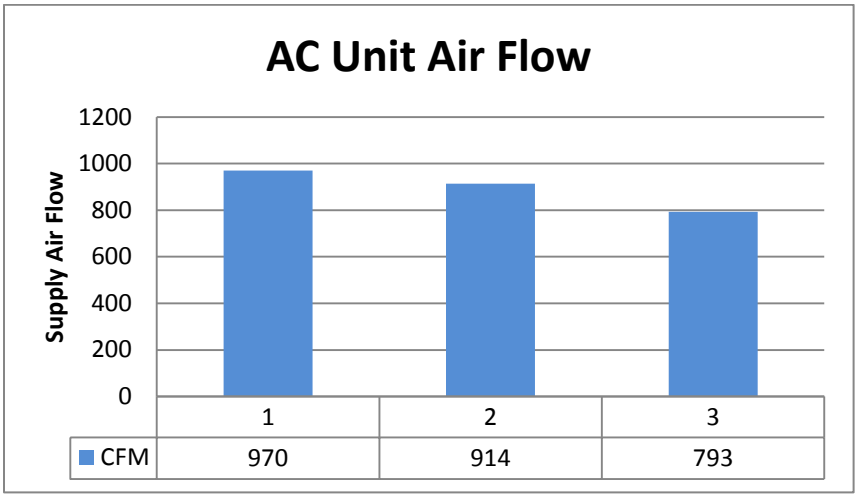


FIGURE 5. AC UNIT AIRFLOW

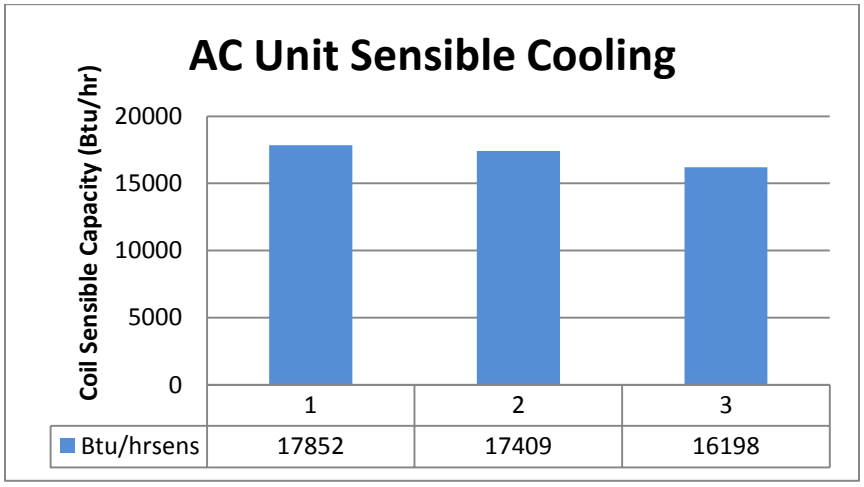


FIGURE 6. AC UNIT SENSIBLE COOLING

Under the same test conditions, the fan energy decreased from 480 to 431W, a 10% decrease. The condenser power did not change, and the compressor power decreased from 2,585 to 2,540W, a 1.7% decrease (Figure 7).

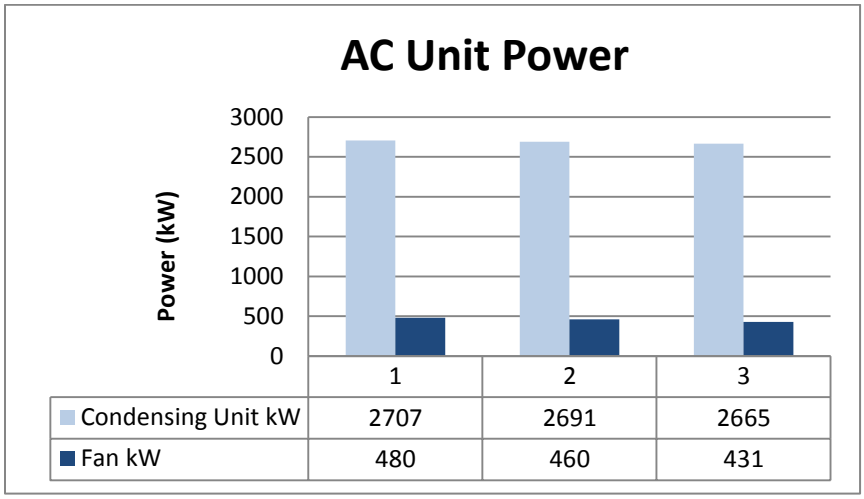


FIGURE 7. AC UNIT POWER

To maintain the same sensible and total cooling load within the test chamber space, the system runtime increased 2.5%, from 1.56 to 1.6 hours, from initial pressure drop (0.1" w.c.) to final static pressure drop (0.5" w.c.). The overall energy consumed did not change, from 4.99 to 4.96 kWh.

EVALUATIONS

Independent tests performed to ASHRAE Standard 52.2 specifications confirmed electrostatic filter manufacturer's performance claims. Electrostatic filters were able to collect ten (10) times more airborne contaminants, at the same air pressure drop, than the baseline 2" thick pleated media filters.

Under actual field operating conditions, the final pressure drop of baseline 2" pleated media filters, after three months of use, was only slightly higher than its original, clean pressure drop. During this test period, the filter collected 0.008 pounds of contaminants. The 3-month test period was selected to simulate actual filter life of a normally maintained commercial AC unit. The scope of this test did not include longer-term filter usage.

Within the normal range of filter usage, the expected degradation of AC unit performance was observed. The AC unit supply fan airflow reduced 18% as 0.5" static pressure was added to the system. The system's measured sensible and total cooling capacities reduced in response to this lower supply airflow. The unit's supply air temperature lowered in response to lower airflow, as the DX coil adjusted to changed operating conditions.

The lower supply airflow and sensible cooling did not result in higher room air temperature in the test chamber, nor did it result in significantly longer run times for the AC unit. The supply fan's amps reduced as external static pressure was added to the system and the airflow decreased. The compressor and condenser amperage did not change during the tests.

Based upon the results, the electrostatic filter has the potential to provide higher capacity air filtration than the incumbent technology. Within the normal range of filter operation, in a properly maintained AC system, this may result in better airflow and sensible cooling capacity. Actual results will vary according to hours of system operation and the level of incoming airborne contaminants.

However, within the limits of operation of well-maintained AC unit operation, the electrostatic filter did not provide electrical energy savings or a reduction in system electrical demand.

RECOMMENDATIONS

Based upon the findings of this study, it is not recommended that electrostatic filters be considered for future evaluation at this time. This technology may provide better filtration performance than the incumbent pleated media filters, but it does not reduce electrical energy usage or electrical demand in a properly maintained AC system at the time of this study.

APPENDIX A – INSTRUMENTATION

Table 11 provides the specifications and calibration dates for all sensors used in this project. Calibration of all instruments occurred prior to conducting any tests.

TABLE 11. SPECIFICATIONS, CALIBRATION DATES, LOCATIONS, AND CORRESPONDING MONITORING POINTS FOR SENSORS

SENSOR TYPE	MAKE/MODEL	ACCURACY (NIST TRACEABLE)	CALIBRATION DATE (LOCATION)	CORRESPONDING KEY MONITORING POINTS
Temperature (type-T thermocouples)	Masy Systems, Ultra-Premium Probe	± 0.18°C [at 0°C] (± 0.32°F)	5-4-2011 (In-house)	• Inlet of evap fan • Inlet of evap • Outlet of evap • Indoor room • Outdoor room • All refrigerant temps
Relative Humidity (RH)	Vaisala, HMP 233	± 1% (0-90% RH) ± 2% (90-100% RH)	5-5-2011 (SCE's Metrology Lab)	• Outlet of evap
Wet Bulb	Vaisala, HMP 247	± 0.013% of reading	5-9-2011 (SCE's Metrology Lab)	• Indoor room
Relative Humidity (RH)	Vaisala, HMP 247	± (0.5 + 2.5% of reading)% RH	5-9-2011 (SCE's Metrology Lab)	• Indoor room
Dew Point	Edgetech, Dew Prime DF Dew Point Hygrometer	± 0.2°C (± 0.36°F)	5-5-2011 (SCE's Metrology Lab)	• Inlet of evap • Outlet of evap
Pressure (0-1000 psi)	Setra, C207	± 0.13% of full scale	4-14-2011 (In-house)	• Discharge • Inlet TXV
Pressure (0-500 psi)	Setra, C207	± 0.13% of full scale	4-14-2011 (In-house)	• Suction • Outlet evap
Pressure (0-10 inches of water, in-wg)	Ashcroft, AQS-28304	± 0.06% of full scale	4-14-2011 (Tektronix Calibration Lab)	• Across indoor unit • Across filter section
Power	Ohio Semitronics, GW5-002C	± 0.2% of reading ± 0.04% of full scale (cond: 1,000W FS) (comp: 5,000W FS)	5-11-2011 (In-house)	• Condensing unit • Compressor • Condenser fan
Power	HIOKI 3169-21	± 0.5% of reading	5-10-11 (In-house)	• Indoor unit • Evap fan
Refrigerant Mass Flow Meter	Endress-Hauser, (Corriolis meter) 80F08-AFTSAAACB4AA	For liquids, ± 0.15% of reading For gases, ± 0.35% of reading	7-22-2010 (Homer R. Dulin Co.)	• Refrigerant flow rate
Scale	HP-30K	± 0.1 gram (± 0.0035 ounces)	11-29-2010 (In-house)	• Mass of condensate

APPENDIX B – TECHNOLOGY TEST CENTERS

Southern California Edison's (SCE) Technology Test Centers (TTC) are a collection of technology assessment laboratories specializing in testing the performance of integrated demand side management (IDSM) strategies for SCE's energy efficiency (EE), demand response (DR), and Codes and Standards (C&S) programs. Located in Irwindale, CA, TTC is comprised of three centers focused on distinct energy end uses: Heating, Ventilating, and Air Conditioning Technology Test Center (HTTC), Refrigeration Technology Test Center (RTTC), and the Lighting Technology Test Center (LTTC.

By conducting independent lab testing and analysis, TTC widens the scope of available IDSM solutions with verified performance and efficiency. TTC tests are thorough and repeatable, and conducted in realistic, impartial, and consistent laboratory environments to ensure the best quality results and recommendations.

The Design and Engineering Services (DES) group of SCE's Customer Service Business Unit manages TTC as a sub-element of the Emerging Technologies program.

HEATING, VENTILATION, AND AIR CONDITIONING TECHNOLOGY TEST CENTER

Heating, Ventilation, and Air Conditioning Technology Test Center (HTTC) evaluates the latest residential and commercial heating, ventilation, and air conditioning equipment. By testing systems and strategies in controlled environment chambers capable of surpassing industry standards and producing realistic climatic conditions, the HTTC can help EE program designers, customers, and the industry make informed HVAC design and specification decisions.

RESPONSIBILITIES

Key responsibilities include:

- **Testing:** HTTC tests HVAC equipment in support of California's statewide Emerging Technologies, Codes and Standards, and Demand Response. Testing capabilities include:
  - Packaged units (up to 7.5 tons)
  - Split systems
  - Control systems
  - Fault detection and diagnostic systems (FDD)
- **Evaluation:** HTTC evaluates the latest residential and commercial heating, ventilation, and air conditioning equipment to provide customers with the information necessary to make informed equipment purchasing decisions.
- **Equipment Efficiency Enhancement:** With funding support from statewide programs and research grants, HTTC works with manufacturers, state, and federal agencies to improve EE regulations addressing HVAC equipment.

TEST CHAMBERS AND EQUIPMENT

Test chambers and equipment include:

- **HVAC Indoor Test Chamber:** This 292 square foot test chamber provides thermal conditions typically found in air-conditioned spaces of residential and commercial buildings, where maintaining desirable human comfort is critical. It is used to collect precise data on temperature, airflow, and humidity in order to test various cooling strategies.
- **HVAC Outdoor Test Chamber:** This 250 square foot test chamber is used to replicate outdoor weather conditions, and to examine how air conditioning units respond under realistic climatic conditions. Temperatures can be maintained as high as 130°F.

APPENDIX – C AAF_5700 FILTER TEST RESULTS



AAF_5700_Filter_Test_results.pdf

APPENDIX – D B03290701 PRESSURE DROP WITH DUST FEED FOR 24 X 24 UNIT



B03290701 Pressure drop with dust feed f

REFERENCES

[i] ASHRAE 2009 Fundamentals, Chapter 1, Equation (6)

[ii] American Society of Heating, Air Conditioning, and Refrigeration Engineers (ASHRAE) Standard 37.2009, Methods of Testing for Rating Electrically-Driven Air Conditioning and Heat Pump Equipment, Atlanta, GA.

Air Conditioning, Heating, and Refrigeration Institute AHRI 210/240 – Performance Rating of Unitary Air Conditioning and Air Source Heat Pump Equipment, Arlington, VA