Performance Evaluation of Rooftop Air Conditioning Units At High Ambient Temperatures

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ABSTRACT

The cooling capacity of packaged, air-cooled, rooftop air conditioning units (RTUs) declines as the ambient temperatures increases. Rooftop air conditioning units' performance parameters such as power consumption and efficiency rating are much less understood at temperatures exceeding 115°F. These temperatures are typically beyond those at which the RTU manufacturers test their products. Many climate regions within California achieve these high ambient temperatures on a consistent basis.

The purpose of this research is to determine the impact of high ambient temperatures on the electric demand and cooling efficiency of five-ton RTUs. This study seeks to better understand efficiency degradations and electric demand implications of high efficiency and standard efficiency units for three leading RTU manufacturers under realistic peak summer cooling conditions seen in California.

A series of laboratory tests were performed to quantify the impacts of high ambient temperatures on the energy efficiency and power usage of three standard and three high efficiency, five-ton RTUs. Performance of these units was evaluated and compared at 85°F, 95°F, 105°F, 115°F, 120°F, 125°F and 130°F air temperatures measured at the condenser air inlet. Test results revealed that in some instances the compressor of the high efficiency model experienced a more drastic increase in electric demand than the standard efficiency model, requiring up to 74% more power per degree Fahrenheit of change in ambient temperature. At an ambient temperature of 130°F, the power demand of one of the high efficiency units surpassed that of the standard model by 120 watts. The high efficiency unit from a second manufacturer would have consumed more power than its standard model had it not failed to operate beyond an ambient temperature of 127°F. Despite these instances of a high efficiency unit having a higher electric demand, all three high efficiency units proved more efficient than the standard models. Overall, the high efficiency units maintained superior efficiency and cooling output than the standard efficiency units.

Each RTU's cooling capacity deteriorated as the ambient temperature increased. The standard efficiency units used more power than the high efficiency units under almost all conditions. The rate of cooling capacity deterioration, however, varied between individual units. Generally, the high efficiency units provided superior cooling capacity to that of their standard efficiency counterparts. The superior capacity was due to improvements such as larger evaporator and condenser surface areas. Coupling the higher capacities with efficient compressors in high efficiency units also contributed to higher energy efficiency ratios (EER).

Introduction

Packaged air conditioning units are used to cool 36.5 billion square feet of commercial space in the United States with an average annual energy consumption of 16.0 kWh per square foot, which puts a heavy burden on the electric grid (EIA 1999). A large percentage of Heating, Ventilating, and Air-Conditioning (HVAC) equipment used in California's small-commercial sector consists of packaged, rooftop air conditioners of five-ton capacity (Johnson 2001).

The Seasonal Energy Efficiency Ratio (SEER) is a common cooling efficiency performance indicator of these units. The SEER provides the ratio of the total cooling of an RTU in Btus during its normal usage period for cooling to the total electric energy input in watt-hours, (ARI 210/240). While SEER provides an indication of the cooling efficiency, it fails to address the power or electric demand implications of different RTUs. The closest index commonly used in the industry that implies the electric demand efficiency of the air conditioning unit is the Energy Efficiency Ratio (EER). Manufacturers determine the EER of their equipment by dividing the net cooling capacity in Btu/hr by the total electric input measured in watts with 95^{o}F ambient air at the condenser inlet. Noteworthy, the 95^{o}F ambient temperature is much milder than the actual climate and site conditions that exist in the inland valley and desert regions of California. In this study, units with SEER of 12 and higher are referred to as "high efficiency" and those with SEERs of 10 as "standard" models.

The dependency of the EER on cooling capacity variations often obscures the variation in power requirements when either the load on the evaporator or the inlet condenser air temperature changes. At increasing ambient temperatures, the EER of the unit may remain reasonably steady because the rate at which its cooling capacity degrades is much less than the rate of increase in power consumption. The principle questions and issues that led to the investigations described in this paper were:

1. Do higher SEER RTUs require more power (kW) than standard SEER models at ambient temperatures greater than 115°F?
2. Does the cooling capacity of a high SEER unit degrade faster than a standard SEER model at high ambient temperatures?
3. Absence of reliable and independent third party performance test data at high ambient temperatures.
4. Frequent occurrence of temperatures above Air-Conditioning and Refrigeration Institute's (ARI) 115^{o}F test condition in California combined with the fact that condenser coil inlet air temperatures are typically higher than ambient dry-bulb temperatures.

Approach

Standard and high efficiency RTUs were purchased from three leading HVAC manufacturers for testing in this project. A series of laboratory tests were performed at Southern California Edison's (SCE's) Refrigeration and Thermal Test Center (RTTC) to quantify the impacts of high ambient temperatures on the performance and power use of standard and high efficiency five-ton RTUs. Performance of these units was evaluated and compared at 85^{o}F, 95^{o}F, 105^{o}F, 115^{o}F, 120^{o}F, 125^{o}F and 130^{o}F air temperatures at the condenser air inlet.

A comprehensive data acquisition system was developed to monitor 138 channels of data every 20 seconds and log critical RTU performance parameters. The entering air conditions to the evaporator, which was one of the critical control points, was maintained at 80^{o}F dry-bulb and 67^{o}F wet-bulb for all test scenarios, regardless of changes in the ambient temperature. The collected data allows for a quantitative analysis of the operation of each RTU. Tests were carried out following a strict test protocol developed by the RTTC staff and based on ARI

210/240 (ARI 1989). Furthermore, an uncertainty analysis model was developed to establish confidence in obtained results. Preliminary results show that uncertainty in the refrigerant-side cooling capacity is less than 3%. Power measurements and EER calculations contain uncertainties of 0.5% and 4%, respectively. These uncertainty values are comparable to those found in similar studies (LeRoy 1997).

Description of Tested Units

Specifications of all six RTUs utilized for testing at the ambient conditions mentioned previously are shown in Table 1. All three high efficiency units came equipped with scroll compressors rated at higher efficiencies and in most cases contain larger coil surface areas. None of the high efficiency units was equipped with variable speed compressors.

Table 1. Manufacturer Specifications of Standard and High Efficiency 5-Ton RTUs

Manufacturer:	**A**		**B**		**C**	
Type:	**Standard**	**High Efficiency**	**Standard**	**High Efficiency**	**Standard**	**High Efficiency**
Cooling Performance						
Nominal Tonnage	5	5	5	5	5	5
Gross Cooling Capacity - Btu/hr (kW)	60,500 (17.7)	63,000 (18.5)	63,100 (18.4)	62,400 (18.3)	60,375 (17.7)	64,375 (18.9)
Net Cooling Capacity - Btu/hr (kW)	57,500 (16.9)	60,000 (17.6)	60,000 (17.6)	59,500 (17.4)	57,000 (16.7)	61,000 (17.9)
Total Unit Power (kW)	6.50	5.50	6.78	5.56	6.70	5.55
SEER (Btu/W-hr)/ EER (Btu/hr/W)	10.0 / 8.8	13.0 / 11.0	10.2 / 8.9	12.0 / 10.7	10.0 / 8.0	13.0 / 11.0
Refrigerant Charge Furnished (HCFC-22)	7.875 lbs.	10 lbs.	4.9 lbs.	8.4 lbs.	6.875 lbs.	10 lbs.
Compressor						
Type	Reciprocating	Scroll	Scroll	Scroll	Reciprocating	Scroll
Rated EER @ 95°F (based on compressor mfg data)	15.8	17.7	13.6	14.2	12.2	15.8
Condenser Coil						
Net Face Area (sq. ft.)	14.6	14.6	8.81	10.96	10.42	16.5
Number of Rows / Fins per Inch	2 / 20	2 / 20	2 / 17	3 / 17	2 / 17	2 / 17
Condenser Fan						
Motor Horsepower (W)	1/3 (248)	1/3 (248)	1/3 (248)	1/3 (248)	.25 (190)	.25 (190)
Motor speed (RPM)	1075	1075	1075	1075	1100	1100
Total Motor Power (W)	360	360	360	360	325	320
Total Air Volume (cfm)	4,200	4,200	3,470	3,370	4,000	4,100
Evaporator Coil						
Net Face Area (sq. ft.)	6.25	6.25	5.00	7.71	5.50	5.50
Number of Rows / Fins per Inch	2 / 15	3 / 15	3 / 16	4 / 16	3 / 15	4 / 15
Expansion Device Type	Balanced Port TXV	Balanced Port TXV	Standard Short Orifice TXV	Short Orifice TXV	Special Evaporator-integrated Orifice Device	Special Evaporator-integrated Orifice Device
Evaporator Blower and Drive Selection						
Nominal Motor Output (Power – voltage)	1.5 hp (1.1 kW) - 208/230V	1.5 hp (1.1 kW) - 208/230V	1 hp (0.61 kW) - 208/230V	1 hp (0.61 kW) - 208/230V	1.30 hp (.989 kW) - 208/230V	1.20 hp (.895 kW) - 208/230V
Wheel Nominal Diameter x Width (in.)	11 ½ x 9	11-1/2 x 9	11 x 11	11 x 11	10 x 10	10 x 10

Test Design

Test Protocols

The determination of all test units' capacities and performance characteristics closely followed test methods specified in ARI 210/240-89, which adopts the American Society of Heating, Refrigeration, and Air-conditioning Engineers (ASHRAE 1988) Standard 37-1988. Cooling capacities were determined by applying the air-enthalpy and the refrigerant-enthalpy methods to test results. All following performance comparison discussions rely on results obtained from the refrigerant-enthalpy method.

All tests were performed at constant conditions for a period of one hour. Each rooftop unit to be tested was installed in the test room in accordance with the manufacturer's installation instructions. Ambient air velocity in the vicinity of the condenser section was monitored closely and maintained below 500 fpm. The ten foot high ceiling in the outdoor control environment room provided more than six feet of clearance for condenser discharge air flow. A distance of at least three feet was provided between the test room's walls and the RTU case.

ARI 210/240 stipulates that the cooling capacity rating includes the effects of blower fan heat, but excludes supplementary heaters. Power measurements include input to the compressor and fans, control power, and any other items required as part of the RTU's normal operation. Ambient temperatures of 95°F and 115°F, two of the standard ARI test conditions, were performed in this project to benchmark test results against the manufacturer's published data.

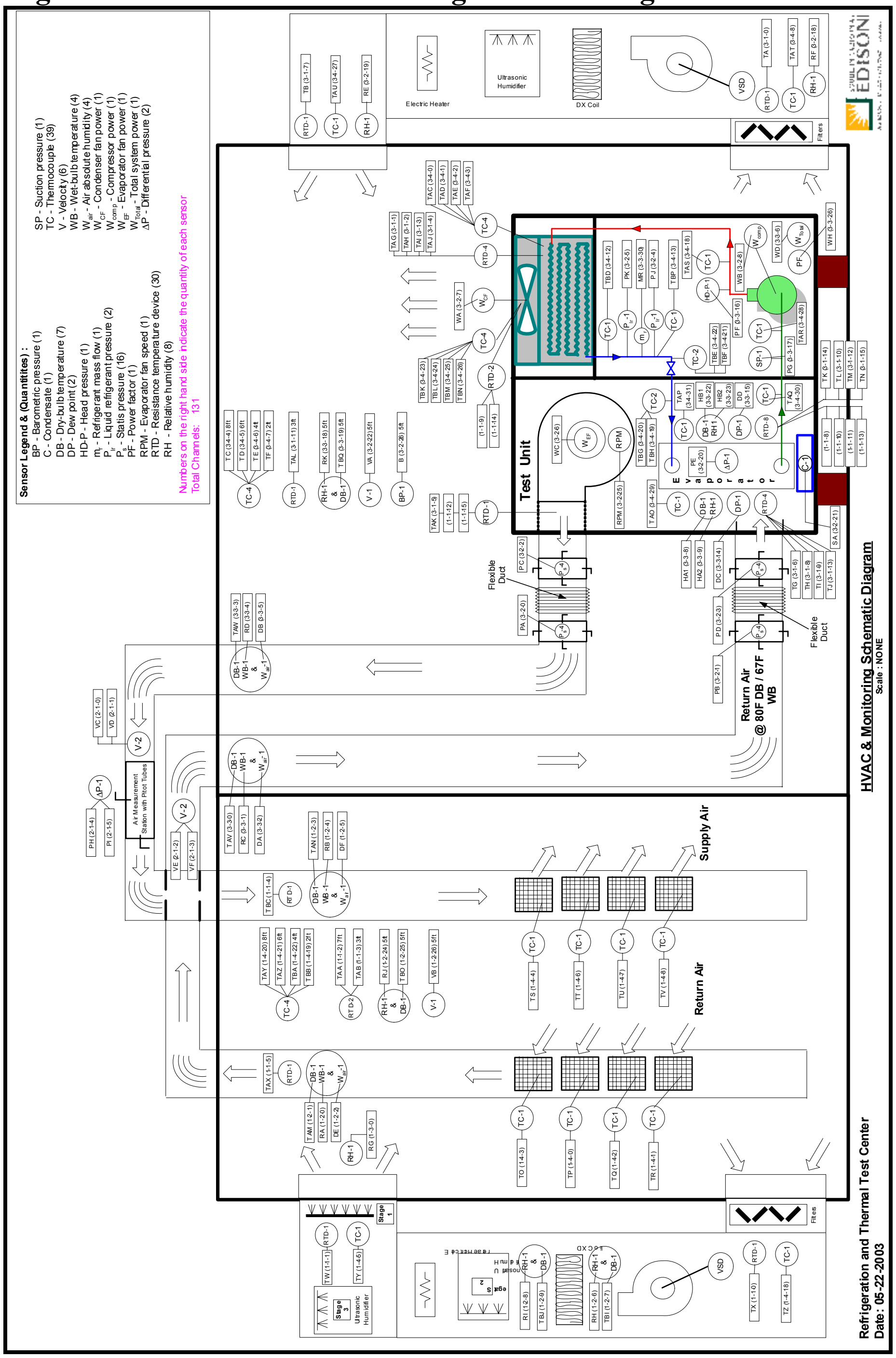
ARI 210/240 requires that, when multiple analyses are used, the total cooling capacity shall be the evaporator-side capacity of two simultaneously conducted analyses, which shall agree within 6.0%. Gross cooling capacity was determined by the product of refrigerant mass flow rate and change in refrigerant enthalpy across the evaporator coil. The sensible capacity was obtained by subtracting the latent load, which was based on mass of condensate, from the gross cooling capacity.

Airflow over the evaporator coil was maintained at 1,875 (± 5%) cfm against a total static pressure of 0.55 (± 0.10) in.wg. for all test conditions. Each unit was charged with the weight of refrigerant prescribed by the manufacturer plus an additional amount to compensate for the extra volume of the mass flow meter and associated piping.

Data Acquisition

The RTTC test facility is equipped with a National Instruments SCXI computer-based data acquisition system to acquire and log test data. It includes a high-performance signal conditioning and instrumentation system for PC-based data acquisition and control. The data acquisition system processes and averages 100 records from 138 data points every 20 seconds. This factory calibrated system is traceable to the National Institute of Standards and Technology's (NIST) standards. Prior to each test, all instruments are calibrated to minimize test uncertainties. Furthermore, collected and stored data for each sensor was rechecked for precision and accuracy at the end of each test scenario. The key parameters were consistently screened to ensure tests had been performed within acceptable limits. Figure 1 is a schematic diagram showing the locations of all sensors used to monitor test parameters.

Figure 1. Detailed Test Monitoring Plan Showing Sensor Locations



Refrigerant temperatures and pressures were monitored at critical points of the refrigeration cycle including suction and discharge of the compressor, condenser outlet, evaporator outlet, and expansion device inlet. Air temperatures were measured at the evaporator inlet and outlet, the condenser inlet and outlet, as well as several locations within th air distribution system. Airflow was determined using a Wilson flow grid mounted inside the ducting between the indoor and ambient test chambers. Condensate from the evaporator coil was piped to a digital scale where the mass could be recorded. Room conditions such as temperature, dew point, and air velocity were monitored in both environmental control rooms. Power demand of the evaporator fan, condenser fan, and compressor were recorded as well. All data was acquired through LabView 6.02 software, which included a graphical data acquisition environment with a data logging and supervisory control add on.

Critical raw data was continuously screened for validation prior to importing the data into the RTTC's engineering model. The collected data points from the 20-second intervals were averaged into one-minute intervals where necessary and used for further screening of the test data. One-minute averages allowed data trends to be displayed with a high resolution, thus enabling the engineering model to generate calculated hourly results such as cooling loads. The primary data points used for comparative analysis were taken from refrigerant-enthalpy results. Based on test design and instrumentation utilized for this project, a higher confidence was placed on refrigerant-enthalpy data.

The one-minute averages were used to produce tabular and graphical representations of various correlations within the engineering model. Several graphs were created to initially screen the calculated data. After careful examination and validation of the initial screening plots, the informational plots were produced. This final set of data plots provided relationships between the calculated quantities. In cases where data flaws were detected, a series of diagnostic investigations were carried out, corrections were implemented, and tests were repeated.

Test Facility

The RTTC's HVAC testing area was used in this project. The HVAC testing area includes both indoor and outdoor controlled environment chambers. Both chambers are served by dedicated heating, cooling, dehumidification, humidification, and filtration systems. An ultrasonic humidifier, controlled by a sophisticated central processing unit-based energy management system, injected precise moisture quantities into the indoor chamber. Due to high temperature conditions, a centrifugal humidification system was used in place of the ultrasonic humidifier in the outdoor environment room. Each room is served by an air handling unit equipped with split, direct expansion refrigerant coils, variable speed drive-controlled variable air volume fans, variable supply air temperature control, and an electric heating system controlled by pulse modulation of the power supply. The AHUs circulate air through supply and return air plenums. A multiplex compressor rack system, consisting of two 15 horsepower (hp) scroll compressors equipped with variable speed drive and variable suction control, serve the two AHUs. The tested RTU was positioned inside the outdoor ambient controlled environment room and connected to an insulated air distribution system. This air distribution system circulated conditioned air from the test unit into the indoor controlled environment room.

Test Scenarios

All units were tested at ambient air dry-bulb temperatures of 85°F, 95°F, 105°F, 115°F, 120°F, 125°F, and 130°F.[1] Air entering the evaporator coil of all units was kept at a constant condition of 80°F dry-bulb / 60.7°F dew point, corresponding to 67°F wet-bulb, as required by ARI 210/240.

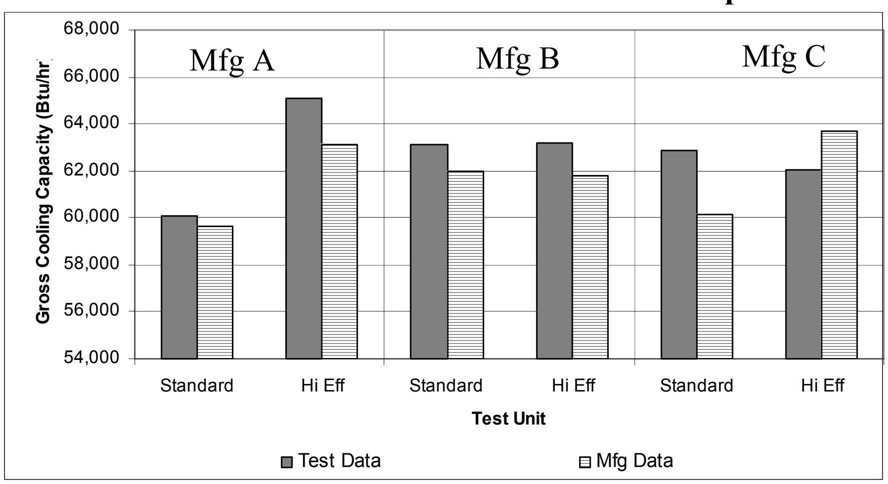
[1] The standard efficiency unit from Manufacturer C failed to operate under 130°F conditions. The high head protection mechanism of this unit constrained the test conditions to a maximum temperature of 127°F.

Discussion of Results

Confidence Tests

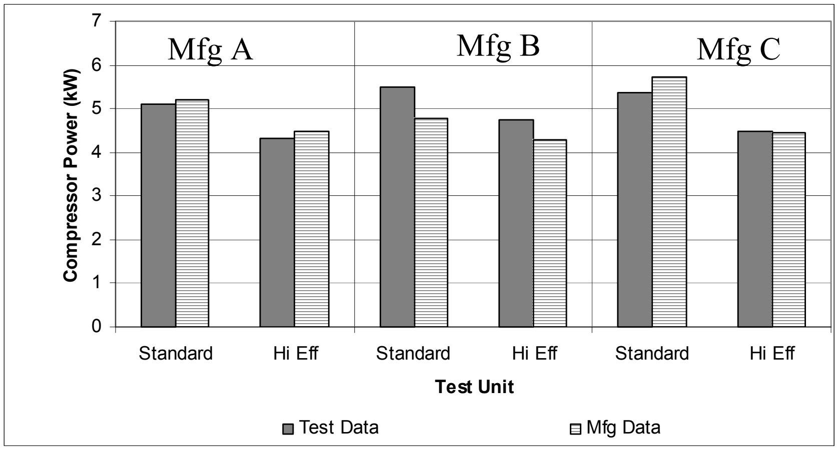
Manufacturers typically publish unit performance data of their products collected at 95^{o}F ambient conditions, as required by ARI. Consequently, to gain confidence in test data, lab results collected at 95°F ambient were compared with published data. The RTTC test data yielded slightly higher gross cooling capacity than the manufacturers published data for five of the six units (Figure 2). The average and maximum discrepancies between the manufacturer data and test results were found to be 1,534 Btu/hr or 2.5% and 2,700 Btu/hr or 4.5 %, respectively. It is not clear, however, whether the manufacturer's published data used in this comparison depends on actual ARI 210/240 tests or ARI certified simulations. Furthermore, the errors associated with the manufacturer's results are unknown. Hence, it is difficult to discern the exact sources of these discrepancies.

Figure 2. Comparison of Gross Cooling Capacity from Manufacturers Published Data and RTTC Test Data Based on 95^{o}F Ambient Temperature



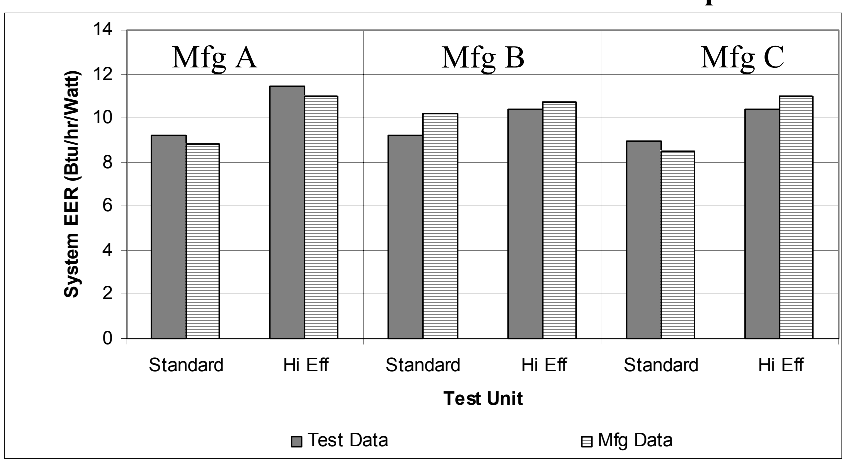
Similar comparisons were made between test units' and manufacturers' compressor power consumption data (Figure 3). The average and maximum discrepancies between the manufacturer data and test results were 299 watts or 6.2% and 716 watts or 15%, respectively.

Figure 3. Comparison of Compressor Power Consumption from Manufacturers Published Data and RTTC Test Data Based on 95°F Ambient Temperature



Lastly, EERs for the 95°F ambient condition were compared in the same fashion and are depicted in Figure 4. The average and maximum discrepancies between the manufacturer data and test results were 0.52 Btu/hr/watt or 5.2% and 0.96 Btu/hr/watt or 9.4%, respectively.

Figure 4. Comparison of System EER from Manufacturers Published Data and RTTC Test Data Based on 95°F Ambient Temperature



These comparisons revealed an average difference of 6.2% or less for gross cooling capacity, system EER, and compressor power between the test results and manufacturers' published data. As a result of these comparisons, a reasonable degree of confidence in test design was established.

Comparison of Test Data

The main objective of this project is to quantitatively compare the operation of standard and high efficiency units under varying ambient temperature conditions. The graphs in the following section illustrate the quantitative test results obtained for standard and high efficiency units of the three manufacturers. The heavier lines in all graphs represent data pertaining to the high efficiency models.

Under ambient temperatures below 125°F the standard RTUs demanded more power than high efficiency units (Figure 5). As temperatures increased, the power demand of high efficiency units from manufacturers A and C grew at a much higher rate than the standard model. As a result, the power curves of the standard and high efficiency units of manufacturers A and C tended to converge rapidly. At an ambient temperature of approximately 130°F manufacturer A's high efficiency unit actually consumed 120 watts more than its standard efficiency model. Results from Manufacturer B do not indicate any power convergence. Figure 6 intends to depict the convergence of power use seen only in manufacturer A & C's products. It also predicts the power usage of manufacturer C's standard unit at 130°F. This prediction was made by curve fitting manufacturer C's standard unit power usage based on experimental results as a function of ambient temperature and extrapolating to 130°F. Figure 6 suggests that if manufacturer C's standard efficiency unit had operated at the 130°F ambient condition, it also may have been surpassed by the high efficiency unit's power usage.

Figure 5. Total System Power Consumption Based on RTTC Test Data for All Six Standard and High Efficiency Units Subject to Various Ambient Temperatures

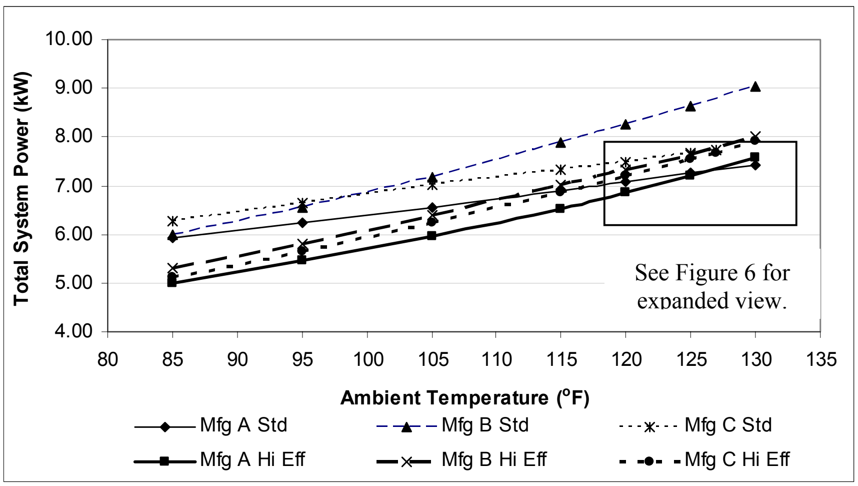
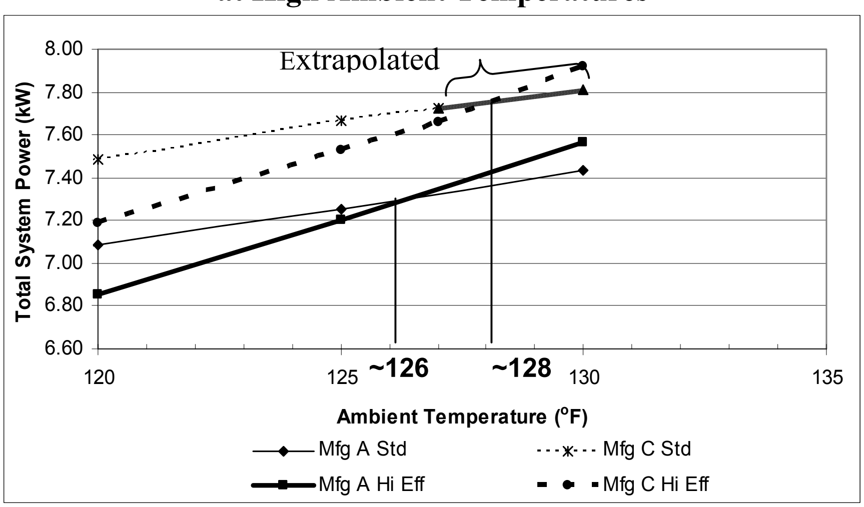
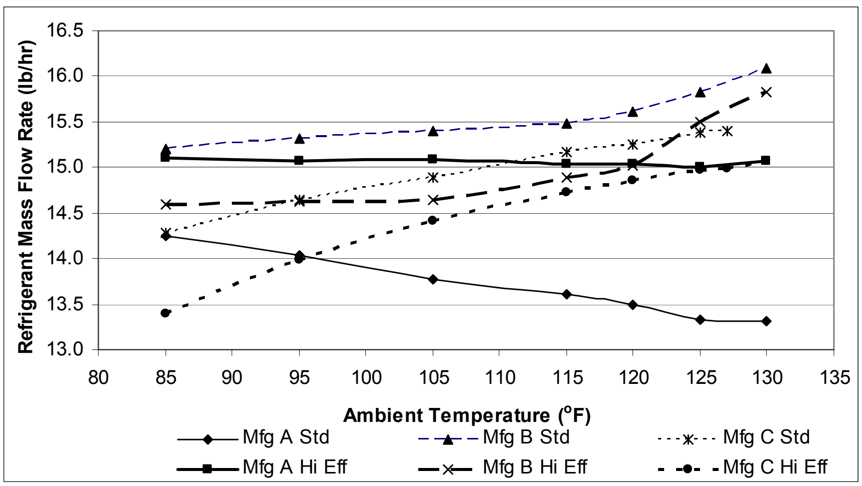


Figure 6. Expanded View of Figure 5 Showing the Intersection of Power Curves at High Ambient Temperatures



The increase in power usage as ambient temperature increased is attributed to the increase in compressor work, which is a function of the compression ratio and mass of circulating refrigerant. At high ambient temperatures, the heat rejection ability of the RTU degrades, which results in high head pressures. Therefore, the compressor must work against a greater pressure difference between the evaporating and condensing pressures. The high head pressure also causes a slight rise in suction pressure. At higher suction pressures, refrigerant becomes denser and the compressor has to compress a larger mass of vapor. Figure 7 demonstrates that, of the six units tested, only those from manufacturer A did not experience an increase in mass flow rate at high ambient conditions. It is not clear why these units behaved differently and this observation is being investigated.

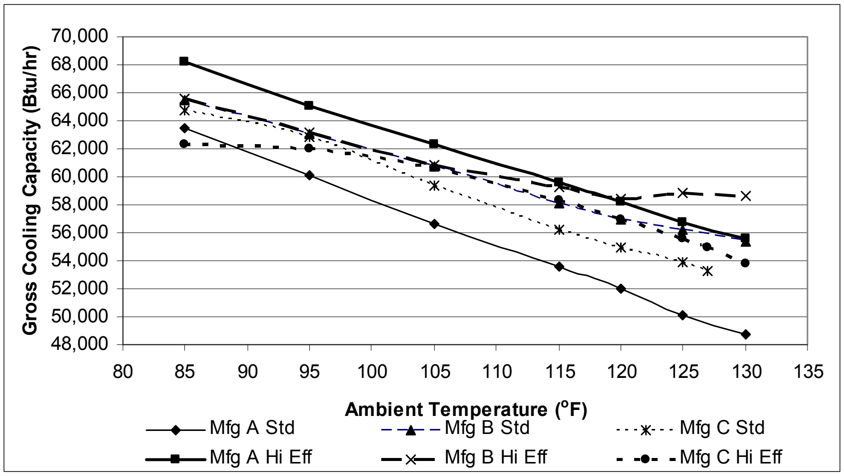
Figure 7. Refrigerant Mass Flow Rate Based on RTTC Test Data for All Six Standard and High Efficiency Units Subject to Various Ambient Temperatures



Each RTU's gross cooling capacity curve dropped as the ambient temperature increased and there was no recurring relationship between the standard and high efficiency units (Figure 8). The two manufacturer A models' capacity decreased steadily, although the standard unit's performance deteriorated at a slightly higher rate than the high efficiency model.

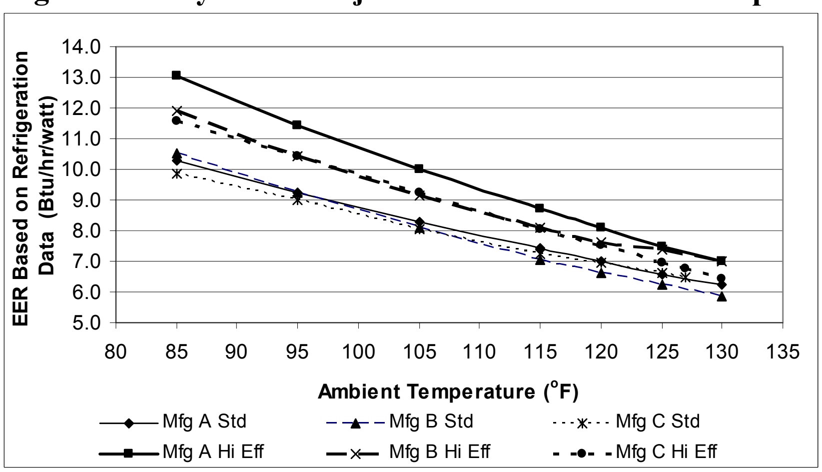
Between ambient temperatures of 85°F and 105°F the units from manufacturer B provided nearly equal cooling capacities. Beyond ambient temperatures of 105°F the high efficiency unit began to outperform the standard model. The cooling capacity of manufacturer C's high efficiency model lagged behind standard unit when temperatures were below 100°F. The high efficiency unit, however, maintained higher cooling capacity than the standard model as ambient temperature increased beyond 100°F. The capacity degradation profile of both of manufacturer A's products stayed similar to that of manufacturer C's standard efficiency unit.

Figure 8. Gross Cooling Capacity Based on RTTC Refrigeration Side Test Data for All Six Standard and High Efficiency Units Subject to Various Ambient Temperatures



The energy efficiency ratio (EER) represents the relationship between net cooling capacity and total power consumption of the units. Despite the fluctuations in cooling capacity and power consumption, all three high efficiency units performed more efficiently than their standard efficiency counterparts over the entire range of ambient temperatures (Figure 8). However, as the temperature became more extreme, the EER values of the standard and high efficiency units approached convergence.

Figure 8. EERs Based on RTTC Refrigeration Side Test Data for All Six Standard and High Efficiency Units Subject to Various Ambient Temperatures



Conclusions

As ambient temperatures approached extreme conditions the performance of all units' compressors began to degrade. The compressor power demand of the high efficiency units increased at a higher rate than that of the standard units as the ambient temperature increased. Power consumption of the standard and high efficiency units from two of the manufacturers began to converge as the ambient temperature increased. The power consumption and cooling capacity of the third manufacturer's units showed similar and non-converging rates of degradation. The compressor of the high efficiency unit of manufacturer A used 120 more watts than the standard unit at the condenser inlet air temperature of 130°F. This excess power use is greater than the uncertainty in the experiment's power measurements. Most likely, this phenomenon would have been repeated in manufacturer C's products as well if its standard unit had not shut off due to high head pressure past 127°F. Despite the convergence of power consumption at high temperatures, the high SEER units operated at higher EERs than standard units due to greater cooling capacity under extreme ambient conditions. Under ambient temperatures ranging from 105°F to 125°F, higher SEER units were able to provide greater cooling capacity while using less power than the standard units.

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