Study of Algorithms for Air Conditioners

Study 3
of
Studies on Algorithm Development for Energy Performance Testing

Asia-Pacific Economic Cooperation

APEC Energy Working Group

November 2001

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Executive Summary

This report evaluates three issues related to air conditioner test standards:

1. the potential for developing algorithms that can adjust for minor differences in test conditions between test standards,
2. the role that test laboratory altitude could play in observed differences in test results,
3. a review of the history behind the development of the US SEER procedure and a discussion of how that general approach could be applied to meet the needs of other APEC economies.

The study findings are summarized below.

The US Department of Energy's Heat Pump Design Model (HPDM) was used to develop a set of generic factors to adjust cooling capacity and efficiency based on minor variations in test conditions. Different adjustment factors were developed for two major compressor types (scroll and reciprocating) as well as for different expansion devices (TXV and short orifice). The performance trends based on HPDM were confirmed by evaluating compressor curves from more than 200 products from the US market. A simple procedure was developed to use the generic adjustment factors to convert test results from one set of test conditions to another. The ability of the procedure to transform results based on minor differences – say from NAFTA's Test A to ISO T1 – look very promising. Transforming results over a wider range of conditions may also work as well, but more refinements are needed.

The impact of barometric pressure and altitude on the ISO test were evaluated. Barometric pressure is explicitly required for air flow calculations in ISO 5151. However, its use in general psychrometric calculations is implied but not explicitly stated. The importance of explicitly measuring barometric pressure for use in all psychrometric calculations was demonstrated. A 2% change (or error) in barometric pressure was shown to introduce errors of 1% in total capacity and 5% in latent capacity. Each 100 meter change in elevation results in more than a 1% change in the barometric pressure, so these calculations are especially important for test laboratories at a high altitude.

The history and development of the SEER test procedure used in NAFTA economies was reviewed to evaluate how seasonal efficiency is determined. The review focused on elements of the procedure that might be appropriate for other APEC economies that wish to develop a seasonal efficiency standard. The flexibility of the bin-based calculation approach and ways that it could be applied in other climates and conditions is presented. The potential for combining accepted international laboratory test procedures, such as ISO 5151, with economy-specific seasonal efficiency calculations is also discussed. This approach could provide each economy with estimates of seasonal efficiency tailored to local needs while retaining a single test standard for manufacturers.

Based on this study, the following recommendations are made for further work:

- validate the process of using simple adjustment factors by comparing the process to actual test results for various products and manufacturers,
- confirm that the simple adjustment factors developed in this study based on US compressors and air conditioners are representative of equipment from other APEC member economies,
- evaluate the need and potential for developing calculation procedures to predict seasonal efficiency based on the current ISO test points; consider both heating and cooling season performance; uniformly consider both variable- and constant-speed equipment.

Table of Contents

List of Figures

List of Tables

1. INTRODUCTION

Background

Test Condition Differences

Minor differences exist between the laboratory test conditions used by air conditioner (AC) test and rating standards used by the various APEC member economies. These minor differences often require that an air conditioner be tested to each individual economy's test standard. In some instances, the testing must be completed in the local test laboratory corresponding to each member economy. This process greatly increases the testing burden on manufacturers and effectively restrains trade between the member economies. Table 1 summarizes the test conditions used in the various APEC test standards.

Table 1. Summary of Test Conditions Used by National Test Standards (from Table 59, Append B, EES Report, Nov. 99)

Economy	Test procedure name	Test point name	Similarity to ISO 5151 pt T1	Stated climate type	Air temperature entering the indoor side		Air temperature entering the outdoor side		Condenser water temperature	
					Dry-bulb	Wet-bulb	Dry-bulb	Wet-bulb	Inlet	Outlet
Australia	AS/NZS 3823.1.1-98	T1	T1 except wet-bulb tolerances	Moderate	27±1(0.3)	19±0.6(0.2)	35±1(0.3)	24±0.6(0.2)	30±0.2(0.1)	35±0.2(0.1)
Canada	CAN/CSA-C368.1-M90	none	close to T1 excluding water cooled units	Not stated	26.7±0.56(0.28)	19.4±0.34(0.17)	35±0.56(0.28)	23.9±0.34(0.17)	NA	NA
	CAN/CSA-C273.3-M91	A	T1 excluding water cooled units	Steady State Wet Coil Test A	27±1(0.3)	19±0.5(0.2)	35±1(0.3)	28±0.5(0.2)	NA	NA
	CAN/CSA-C744-93	none	close to T1 excluding water cooled units	Not stated	26.7±0.56(0.28)	19.4±0.34(0.17)	35±0.56(0.28)	23.9±0.34(0.17)	NA	NA
China	GB 7725-96	T1	T1	Moderate	27±1(0.3)	19±0.5(0.2)	35±1(0.3)	24±0.5(0.2)	30±0.2(0.1)	35±0.2(0.1)
Hong Kong	ISO 5151 94(E)	T1	T1	Moderate	27±1(0.3)	19±0.5(0.2)	35±1(0.3)	24±0.5(0.2)	30±0.2(0.1)	35±0.2(0.1)
Japan	JIS C9612-94	none	T1 (except water temperature tolerances)	Note stated	27±1(0.3)	19±0.5(0.2)	35±1(0.3)	24±0.5(0.2)	30±0.3	35±0.3
	JIS B8616-93									
Mexico	NOM-073-SCFI-94	none	close to T1 except for water condenser units	Not stated	26.6±0.55(0.28)	19.4±0.33(0.17)	34.9±0.55(0.28)	23.8±0.33(0.17)	NA	NA
Intl	ISO 5151 94(E)	T1	T1	Moderate	27±1(0.3)	19±0.5(0.2)	35±1(0.3)	24±0.5(0.2)	30±0.2(0.1)	35±0.2(0.1)
Philippines	PNS 240-89	D	close to T1 except outdoor wet-bulb and differences for water condenser units		27±0.5(0.3)	19±0.3(0.2)	35±0.5(0.3)	27±0.3(0.2)	31±0.2(0.1)	37±0.2(0.1)
Chinese Taipei	CNS 3615–95	Cooling conditions	very close to T1	Not stated	27±1	19.5±0.5	35±1	24±1	30±0.5	35±0.5
	CNS 2725-95	Cooling conditions	close to T1 except for water condenser units	Not stated	27±1	19.5±0.5	35±1	24±0.5	30±0.5	35±0.6
Thailand	TIS 1155-2536 (1993)		T1 except for exclusion of arithmetic mean tolerances	Not stated	27±1	19.5±0.5	35±1	24±0.5	30±0.5	35±0.6
USA	10 CFR 430 Subpart B, Appendix F, ANSI/AHAM RAC-1-82 & ASHRAE 16-83-RA88	none	close to T1 except for water condenser units	Not stated	26.7±0.56(0.28)	19.4±0.34(0.17)	35±0.56(0.28)	23.9±0.34(0.17)	23.9±0.22(0.11)	35±0.22(0.11)
	10 CFR 430 Subpart B, Appendix M, & ARI210/240-94	A	T1 excluding water cooled units	Steady State Wet Coil Test A	26.7±1.1(0.28)	19.4±0.56(0.17) Note 2	35±1.1(0.28)	28±0.56(0.17)	29.4±0.28(0.11)	35±0.28(0.11)
	ARI 310/380-93	none	close to T1 except for water condenser units	Not stated	26.7±0.56(0.28)	19.4±0.34(0.17)	35±0.56(0.28)	23.9±0.34(0.17)	NA	NA

Tolerances are first indicated as the maximum permissible variation of individual readings from the specified value followed by the permitted variation of the arithmetical mean value from the specified value, indicated in parentheses

2-There is an apparent typographical error in the US Code of Federal Regulations, 10 CFR 430 Subpart B, Appendix M, Section 3.1.1 p 198, which states that this value (the indoor side wet-bulb temperature) should be 87°F (equivalent to 30.6°C) as opposed to the 67°F (19.4°C) stated in ARI 210/240-94 and consistent with the ISO 5151 T1 test condition. It appears that the ARI 210/240-94 values are used in practice.

Barometric Pressure

Another issue is the impact of barometric pressure on air conditioner test results. Previous APEC reports and workshops have also identified discrepancies between test results obtained in AC test laboratories located at different altitudes. There is some question as to whether current test standards properly account for the impact that barometric pressure and altitude have on calculated capacity and efficiency. It is also possible that more explicit guidance on psychrometric calculation procedures would be helpful to ensure more uniform determination of performance data by local testing laboratories.

Seasonal Efficiency

Many APEC member economies would also like to develop performance indexes, or metrics, that provide an indication the seasonal average efficiency of air conditioner systems. Currently, only the three NAFTA economies use test procedures that express the test results as a seasonal average value. The test procedures of other APEC member economies, and the ISO standards upon which most procedures are based, simply provide an efficiency that corresponds to “design” or peak conditions.

The use of peak efficiency instead of a seasonal value has historically not changed the ranking of single speed air conditioners and heat pumps (i.e., comparing the peak efficiency of single speed AC products would provide a proportional indication of their expected seasonal average efficiency). However, the introduction of two-speed and variable-speed systems has changed the situation. Multi-speed and modulating systems can have much lower efficiencies than single-speed units at peak conditions, yet have much greater seasonal efficiencies.

The SEER test and rating procedures used in the NAFTA economies account for these differences and attempt to fairly predict seasonal efficiency for all these systems. However, the testing burden of these standards is perceived as too onerous by other APEC member economies. In Japan, a simpler test procedure has been proposed to consider variable speed systems.

The Purpose and Approach of this Study

The purpose of this study is to investigate ways that simple algorithms, or mathematical models, can be used to extend or broaden the applicability of results from one test condition, or rating standard, to another. For instance, minor differences (~1°C) exist between the test conditions required by member economies’ air conditioning test standards. If standardized and broadly applicable adjustment factors can be developed, then test results determined with one test standard could conceivably be transformed to another without additional testing. This study evaluates the efficacy of this concept.

In addition, this study assesses the impact that barometric pressure could have on the psychrometric calculations using the air enthalpy or calorimetric test methods of determining capacity. Specifically the ISO standards, which are basis for many national standards, are evaluated to determine if altitude and barometric pressure are properly addressed in the explicitly stated and implied calculation procedures.

Finally, the concept of season average cooling efficiency is reviewed and the background behind the SEER calculation procedures used by the NAFTA member economies are described. The discussion generally presents the concepts of seasonal average efficiency, the type and amount of test data that are required, and how these concepts might be applied to the needs of other APEC member economies.

2. IMPACT OF OPERATING CONDITIONS

This section evaluates whether generalized or generic factors could be developed to transform the results from one test procedure to another. First, the detailed heat pump model HPDM is used to determine the impact of test conditions on capacity and efficiency for a few basic air conditioner configurations. Then a library of compressor performance data for more than 200 compressors from two manufacturers is used to explore the likely variations in these factors that might be expected based on the observed product differences.

Using HPDM to Compare Test Conditions

The Heat Pump Design Model (HPDM) has been developed by Oak Ridge National Laboratory for the US Department of Energy (USDOE) as a software tool to simulate the performance of air conditioners and heat pumps. HPDM is a hardware-based model that requires detailed information about the compressor, evaporator coil, condenser, expansion device and connecting piping that make up a system. The model was originally developed in the late 1970s and has undergone several improvements over the years to capture technology changes such as variable speed compressors and new refrigerants.

A recent ASHRAE research project (RP-857) in the US compared HPDM to measured performance data at off-design conditions from various ducted-residential air conditioners and heat pumps. Other software tools such as NIST's HPSIM tool and Purdue University's ACMODEL were also considered in the analysis. HPDM was shown to provide very good predictions of off-design performance. HPDM is now available with a well-developed user interface on the world wide web (www.ornl.gov/~wlj/hpdm/doehpdm.html). A sample of the graphical output results from the model are given in Appendix A.

Table 2 summarizes the variations in test conditions that exist in the current test standards. Outdoor and indoor temperature conditions are given as both Celsius and Fahrenheit. For each test standard, the native units are shown as bold. The relative humidity and dew point corresponding to the indoor wet bulb and dry bulb temperatures are also given for reference.

Table 2. Test Conditions from Various Test Procedures

	Outdoor DB (°C / °F)	Indoor DB (°C / °F)	Indoor WB (°C / °F)	Indoor RH (%)	Indoor Dew Pt. (°C / °F)
ISO T1	**35** / 95	**27** / 80.6	**19** / 66.2	46.9%	14.6 / 58.2
Test A (NAFTA)	35 / **95**	26.7 / **80**	19.4 / **67**	51.1%	15.6 / 60.0
Korea	**35** / 95	**27** / 80.6	**19.5** / 67.1	49.5%	15.5 / 59.9
Test B (NAFTA)	27.8 / **82**	26.7 / **80**	19.4 / **67**	51.1%	15.6 / 60.0
Test C (NAFTA)	27.8 / **82**	26.7 / **80**	13.9 / **57**	21.5%	5.2 / 37.2
ISO T2 - mild	**27** / 80.6	**21** / 69.8	**15** / 59	52.8%	10.9 / 51.5
ISO T3 - hot	**46** / 114.8	**29** / 84.2	**19** / 66.2	38.5%	13.3 / 55.6

Note: Native units shown as bold. RH and dew point are calculated at standard barometric pressure.

The first three sets of conditions are usually referred to as the nominal or "design" rating points for AC equipment. Test A is the rating point where nominal capacity and EER are determined for the NAFTA economies. The ISO T1 condition has identical outdoor conditions and slightly warmer and drier indoor conditions. The Korean test point appears to have started with the ISO T1 condition and tried to increase the indoor humidity to be more inline with the Test A conditions.

Tests B & C are part load conditions that are used in the calculations to determine seasonal efficiency and part load performance. In NAFTA economies, SEER is determined from Test B data. The ISO T2 and T3 conditions correspond to very cool and very hot conditions, respectively.

Table 3 shows the capacity and efficiency results using the HPDM program with the various operating conditions from Table 2. The capacity and efficiency data are normalized using the ISO TI capacity and efficiency as reference. The sensible heat ratio (SHR) at each point is also given. Because of the impact that the compressor has on these temperature dependent variations, the HPDM runs were made for both a reciprocating and a scroll compressor.

Table 3. Relative Cooling Performance with Various Test Procedures

	Reciprocating Compressor			Scroll Compressor		
	Relative Efficiency	Relative Capacity	SHR	Relative Efficiency	Relative Capacity	SHR
ISO T1	100%	100%	0.74	100%	100%	0.75
Test A (NAFTA)	101.0%	101.4%	0.70	101.1%	101.3%	0.70
Korean T1	101.2%	101.7%	0.71	101.3%	101.6%	0.71
Test B	117.9%	109.9%	0.68	119.4%	106.9%	0.69
Test C (NAFTA)	104.4%	92.5%	1.00	104.6%	91.9%	1.00
ISO T2	106.5%	94.3%	0.73	106.7%	93.0%	0.74
ISO T3	79.3%	87.4%	0.89	78.2%	95.4%	0.75

Notes: Using default data from HPDM (see Appendix A). “Ideal” TXV with fixed superheat. Scroll compressor is Copeland ZR28K1-PFV. Reciprocating compressor is a Copeland CR28K6-PFV.

The data show that the three “nominal” test points yield capacity and efficiency results that are all within 1-2% of one another. The Korean and NAFTA test conditions yield nearly identical answers since the entering wet bulb temperatures are close[1]. At Test B conditions, this hypothetical AC unit has 10% more capacity and 20% better efficiency than at Test A. When going from ISO T1 to T3 conditions, efficiency decreases by more than 20%.

It is interesting to note that the capacity of the AC unit with a scroll compressor is much less sensitive to variations with outdoor temperature than the reciprocating compressor unit. This trend is one of the recognized benefits of scroll compressors. The performance benefit of the scroll becomes especially apparent at ISO T3. While the reciprocating compressor has lost nearly 13% of its capacity at T3 conditions, the scroll unit has only lost 5% of its capacity. The higher capacity also helps to keep the latent capacity fraction (as indicated by the SHR) at nearly the same level with the scroll unit. In contrast, the reciprocating compressor has lost more than half of its latent capacity due to higher suction pressures at the evaporator coil.

[1] The next section systematically evaluates the relative importance of each operating condition.

Developing Adjustment Factors

In this section we use HPDM to systematically evaluate the variation of capacity and efficiency with operating conditions, with our goal being to develop a set of simple correction factors to correct for minor variations in test conditions. Figure 1 and Figure 2 show how normalized capacity and efficiency vary with ambient or outdoor temperatures for AC units with "ideal" TXVs using both reciprocating and scroll compressors. The data are normalized based on performance at Test A conditions. Figure 3 and Figure 4 show the same trends for a system with a short-orifice expansion device. In all cases, the HPDM results are shown as points on the plots and a best-fit linear regression model is shown as a line. Nearly all the data demonstrate a highly linear trend over a wide range of ambient conditions.

The slopes for each of these capacity and efficiency trends are summarized in Table 4. The slope is expressed as the % change per each 1°C change in temperature. The slope for efficiency with outdoor temperature was a decrease of 1.9-2.0% for scroll compressors and 2.3-2.4% for reciprocating compressors. The type of expansion device had only a mild impact on the slope (i.e., the efficiency slope was slightly smaller for the short orifice).

Table 4. Impact of Test Conditions on Normalized Capacity and Efficiency

Slope units: % per °C	**Efficiency**			**Capacity**		
	Outdoor DB	**Indoor WB**	**Indoor DB**	**Outdoor DB**	**Indoor WB**	**Indoor DB**
Recip w/ TXV	-2.0%	2.2%	0.15%	-1.2%	3.3%	0.16%
Scroll w/ TXV	-2.4%	2.6%	0.19%	-0.8%	3.0%	0.11%
Recip w/ Orifice	-1.9%	1.6%	0.12%	-1.1%	2.5%	0.12%
Scroll w/ Orifice	-2.3%	1.7%	0.16%	-0.8%	2.4%	0.16%

Similar trends were observed for normalized capacity, though in this case the capacity change with temperature was lower for the scroll compressor. The capacity decreased by 1.1-1.2% for the reciprocating compressors and 0.8% for the scroll.

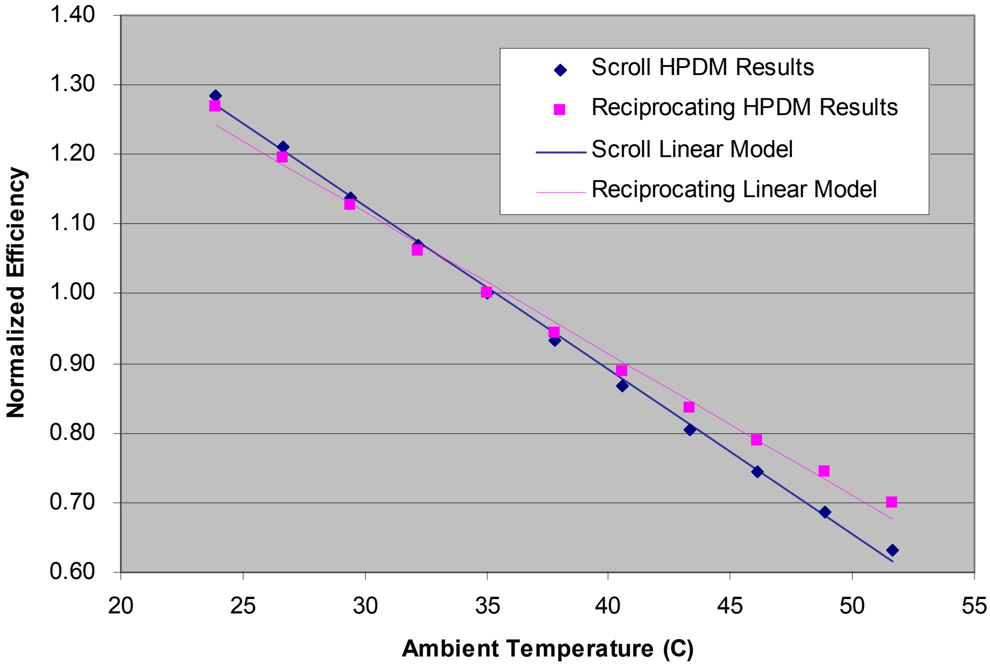


Figure 1. Variation of Efficiency with Ambient Temperature – TXV

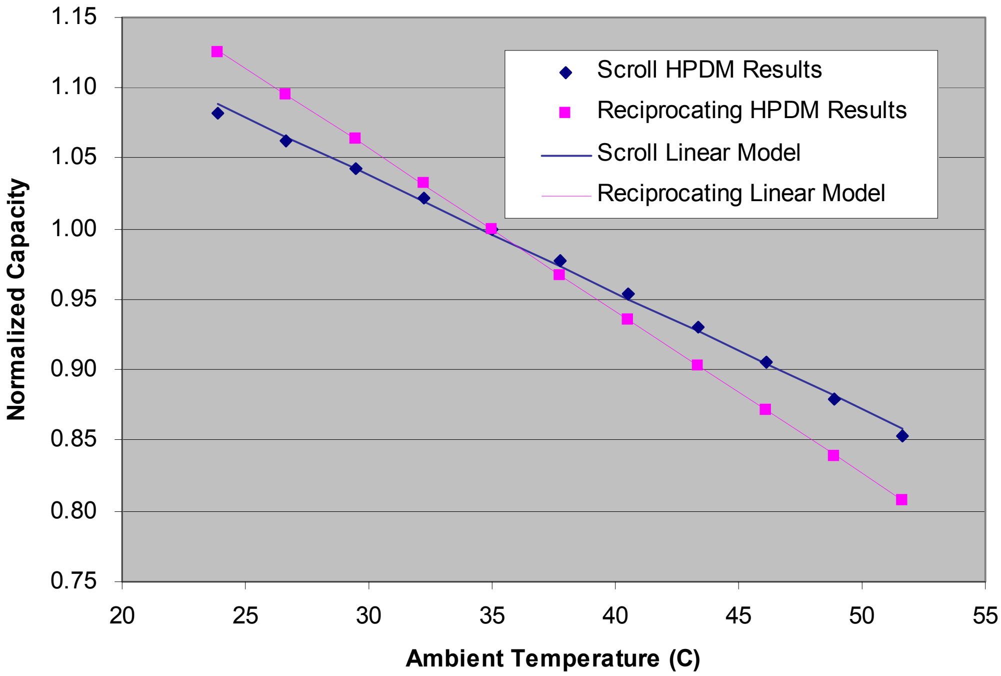


Figure 2. Variation of Capacity with Ambient Temperature – TXV

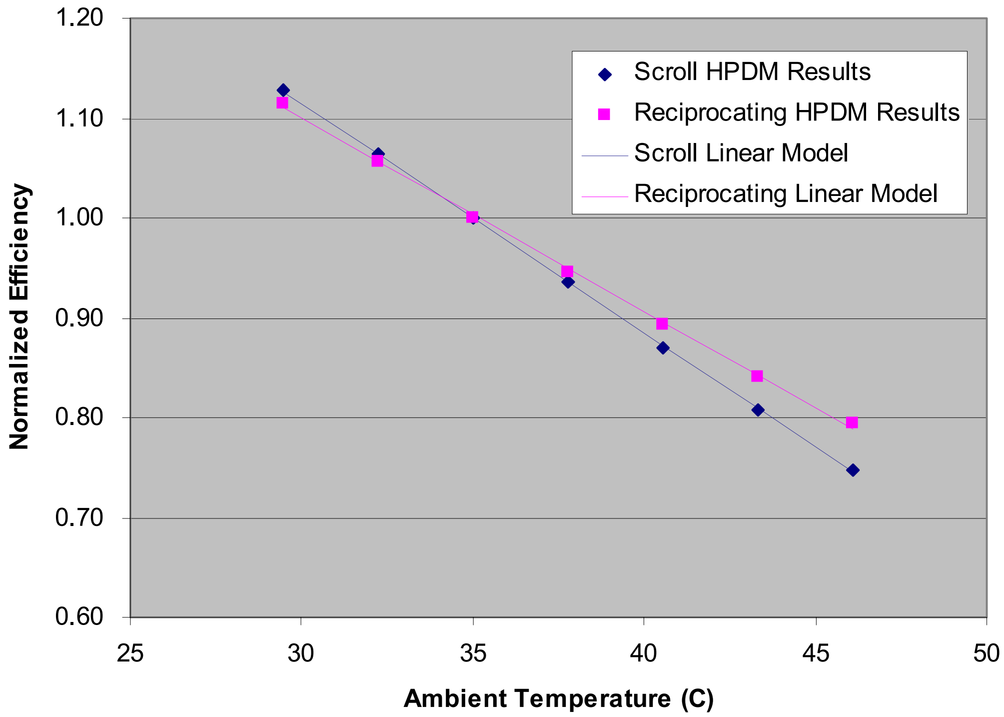


Figure 3. Variation of Efficiency with Ambient Temperature – Short Orifice

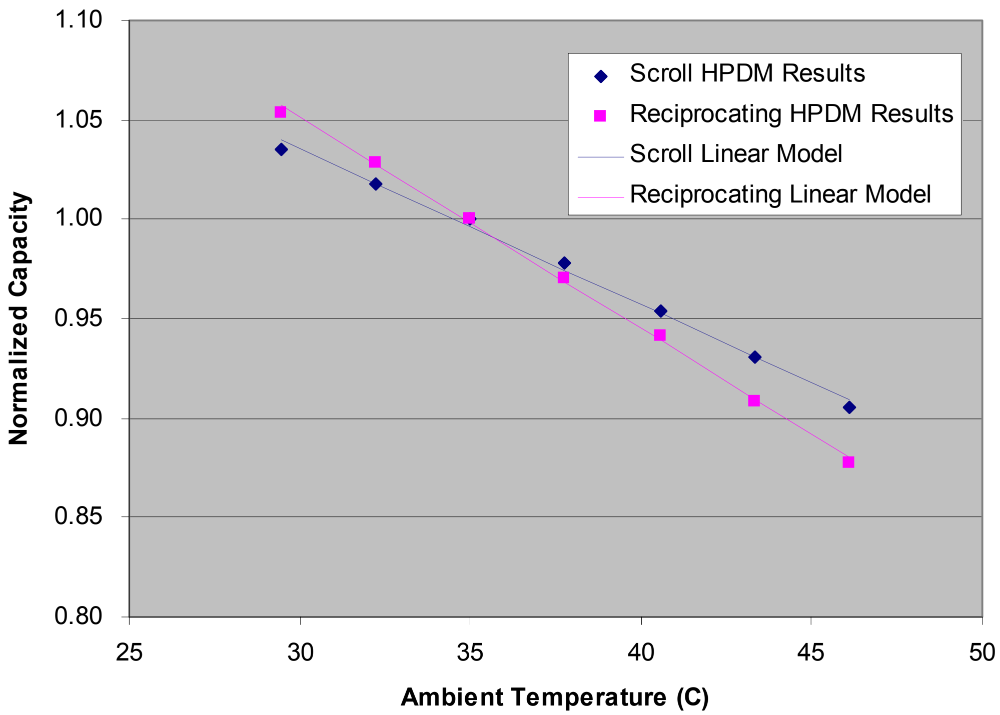


Figure 4. Variation of Capacity with Ambient Temperature – Short Orifice

Table 4 also lists data that show the impact of indoor conditions on capacity and efficiency. Some of these data are plotted in Figure 5. As expected, efficiency is a much stronger function of the wet bulb (WB) entering the indoor coil than of the dry bulb. The impact per degree is an order of magnitude greater for wet bulb than for dry bulb. In contrast to the outdoor temperature, the impact of indoor wet bulb on capacity and efficiency appears to be more dependent on the type of expansion device than compressor. As shown in Table 4, the efficiency slope for the entering wet bulb was 2.2-2.6% for the TXV system and 1.6-1.7% for the short-orifice system. The capacity slope was 3.0-3.3% for the TXV and 2.4-2.5% for the short orifice. For both capacity and efficiency the impact on indoor dry bulb is very modest, in the range 0.1-0.2% per degree.

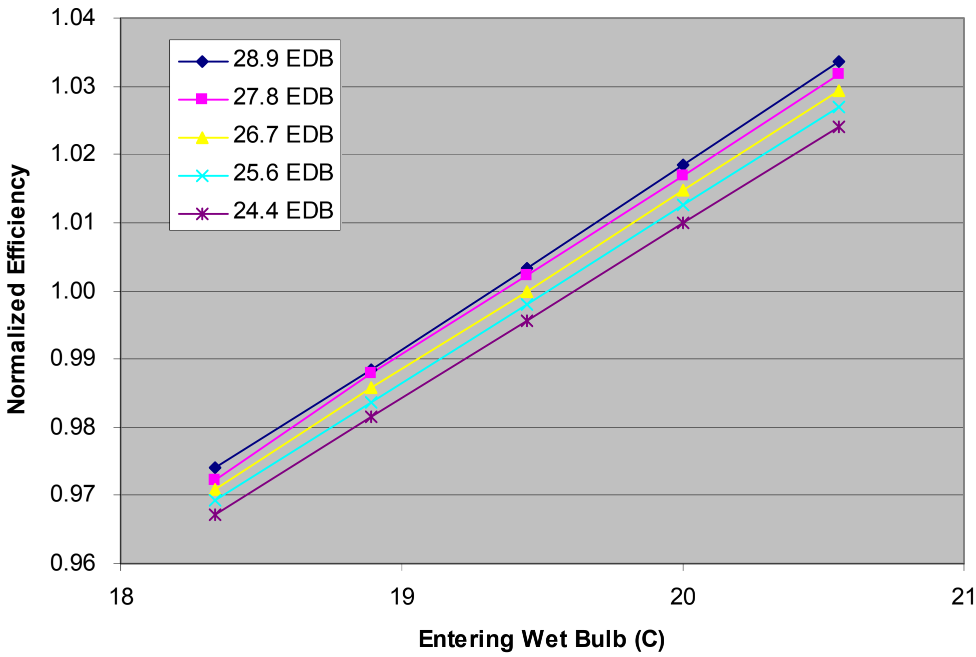


Figure 5. Variation of Efficiency with Entering (or Indoor) WB and DB Temperature – TXV

Variability in Compressor Curves

The previous section used the HPDM model to evaluate the impact of test conditions on the capacity and efficiency for a hypothetical AC system. Though two different compressors were included in the analysis, it is unclear how representative these particular models are of compressors in general.

This section compares the standard compressor performance maps, or curves, from more than 200 compressor models from two manufacturers. The compressor curves were developed as the ten-coefficient polynomial curve defined in ARI Standard 540-99. The curves provide capacity and power use as a function of the saturated condensing temperature (SCT) and the saturated suction temperature (SST). Manufacturers in North America make these coefficients available in electronic format, which made this automated analysis practical.

The focus was on compressors from Copeland and Carlyle that are intended for "medium" and "high" temperature applications and use refrigerant R-22. One premise of this analysis was that the SCT of an AC system is directly related to the outdoor temperature. While there is not a perfect one-to-one correspondence, the HPDM results in Figure 6 confirm the relationship is very close. For these runs for the TXV system, every 1°C increase in the ambient temperature resulted in a 0.96°C increase in SCT for the scroll compressor and a 0.90°C increase in SCT for the reciprocating compressor. In addition, the SST of the system was also observed to vary with ambient as well, but by a much smaller amount (typically by 0.1°C per each degree change in ambient).

These trends tend to confirm that the approximation of using SCT compressor curves (while holding SST constant) will be a good surrogate for the variation of overall air conditioner system efficiency with ambient temperature.

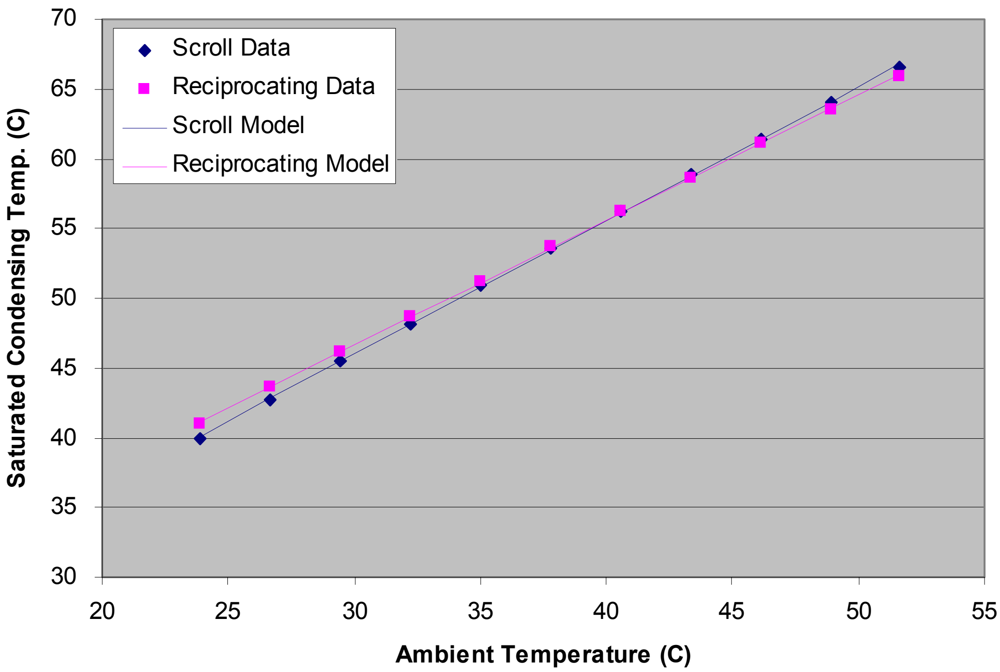


Figure 6. Variation SCT with Ambient Temperature from HPDM runs – TXV

Figure 7 shows how compressor efficiency, normalized to be 100% at 50°C SCT, varies with the saturated condensing temperature. As shown on the plot, the slope of normalized "efficiency" for this Copeland compressor at 50°C is an efficiency drop of 2.9% for each 1°C increase in condensing temperature. Repeating this process at 45°C and 55°C results in only slightly different values for the slope.

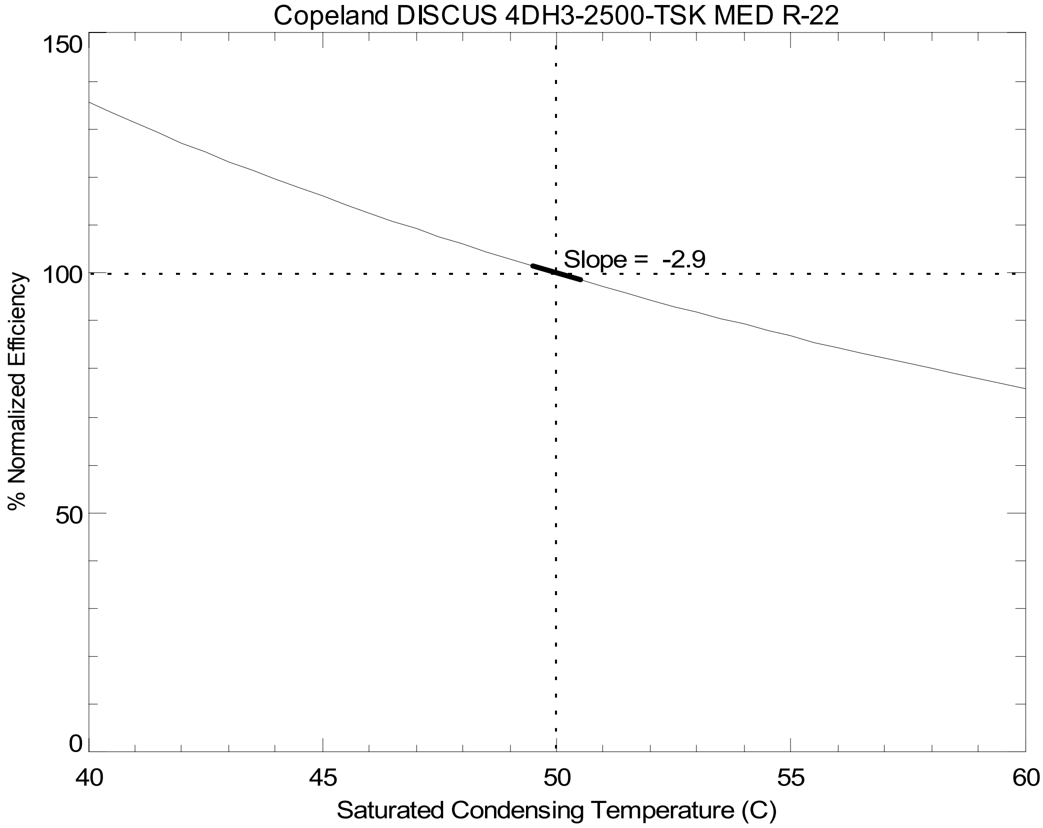


Figure 7. Trend of Normalized "Efficiency" with Sat. Condensing Temperature (SCT)

Table 5 shows the average "slope" values determined when repeating the process described above for 187 different medium and high temperature Copeland compressors that use refrigerant R22. In each case, the compressor efficiency is normalized at 3 distinct condensing temperatures (i.e., 45, 50, and 55°C). The results show that condensing temperature has less impact than other factors, such as the compressor type and the application temperature range. Figure 8 also shows the minimum and maximum values observed in each case, as well as the standard deviation of the values about the average.

Table 5. Normalized Efficiency Slope Determined for Various Copeland Compressors

		Compressor Curve *Efficiency* Slope (% EER per °C)				
Compressor Application	**SCT (°C)**	**COPELAMATIC N_{High}=45 N_{Med}=23**	**COPELAWELD N_{High}=49 N_{Med}=7**	**DISCUS N_{High}=0 N_{Med}=29**	**SCROLL N_{High}=6 N_{Med}=28**	**ALL N_{High}=100 N_{Med}=87**
HIGH	45	-2.81	-2.91	-	-3.23	**-2.89**
	50	-2.79	-2.88	-	-3.40	**-2.87**
	55	-2.82	-2.88	-	-3.60	**-2.89**
MED	45	-2.56	-2.56	-3.01	-2.05	**-2.55**
	50	-2.48	-2.58	-2.89	-2.22	**-2.54**
	55	-2.41	-2.66	-2.82	-2.41	**-2.57**

Notes: Using default compressor curves with SST = 7°C. Using R-22 only.

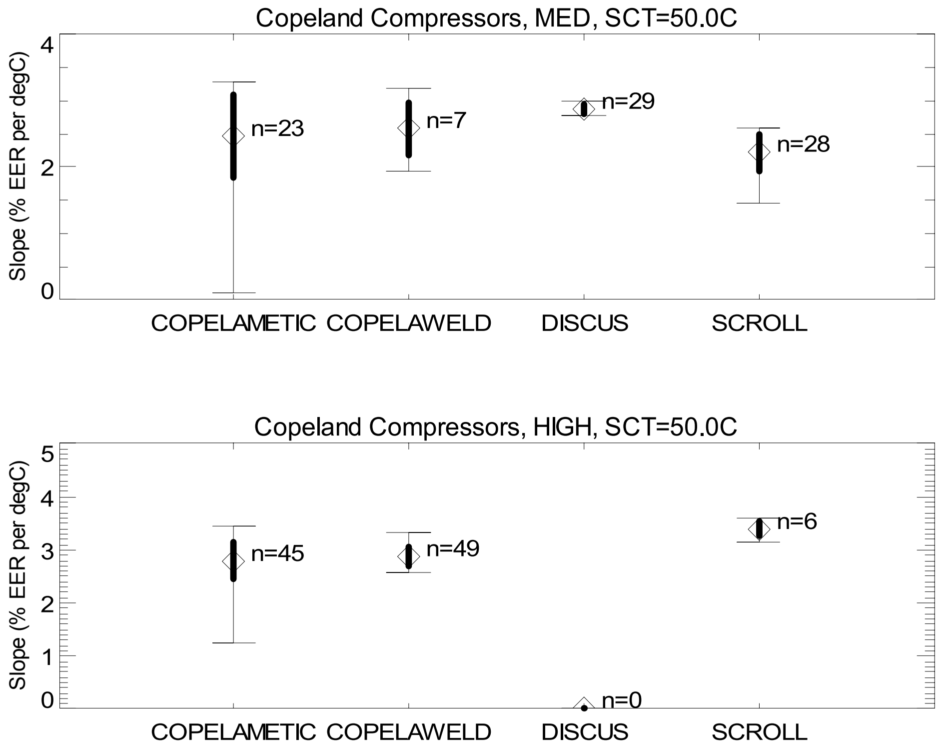


Figure 8. Variations in Efficiency Slope Determined for Various Copeland Compressors

The overall average change for the high temperature compressors – which are most representative of air conditioning duty applications – is a 2.9% decrease in efficiency for each 1°C increase in condensing temperatures. For the medium temperature compressors this value drops to 2.5%. The Scroll and Discus compressors show the most variation from that average trend. Though more variation may be apparent for medium temperature units.

Table 6 and Figure 9 show the same data for 33 Carlyle semi-hermetic compressors. Compressors from this manufacturer also averaged about 2.9% per °C for both medium and high temperature models. One surprising note for the Carlyle compressors was the greater degree of temperature dependence (which means the curves are less linear with SCT). The sensitivity of the slope change with temperature is about 5 times greater for the Carlyle models than was observed for the Copeland compressors.

Table 6. Normalized "Slope" Values Determined for Various Carlyle Compressors

		Compressor Curve *Efficiency* Slope (% EER per °C)		
Compressor Application	**Sat Cond. Temp (°C)**	**Carlyle 06D N_{High}=6 N_{Med}=12**	**Carlyle 06E N_{High}=7 N_{Med}=8**	**All N_{High}=13 N_{Med}=20**
HIGH	45	-3.21	-2.86	**-3.02**
	50	-3.04	-2.73	**-2.88**
	55	-2.93	-2.66	**-2.79**
MED	45	-3.13	-2.88	**-3.03**
	50	-2.94	-2.79	**-2.90**
	55	-2.86	-2.73	**-2.81**

Notes: Using default compressor curves with SST = 7°C. Using R-22 only.

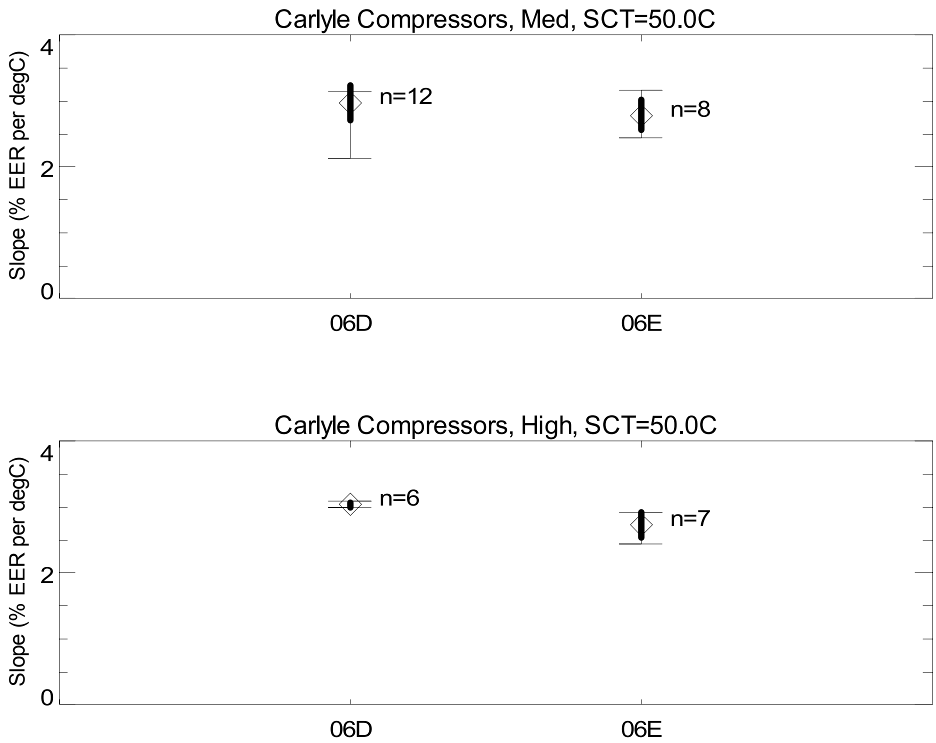


Figure 9. Variations in Efficiency Slope Determined for Various Carlyle Compressors

While the results of this analysis are not exhaustive, they do imply than many compressors have similar values for the efficiency slope. In fact, as shown in Figure 10, an analysis of the efficiency slope based on the Carnot COP of cooling machine at similar conditions yields a similar answer, of about –2.3% per °C.

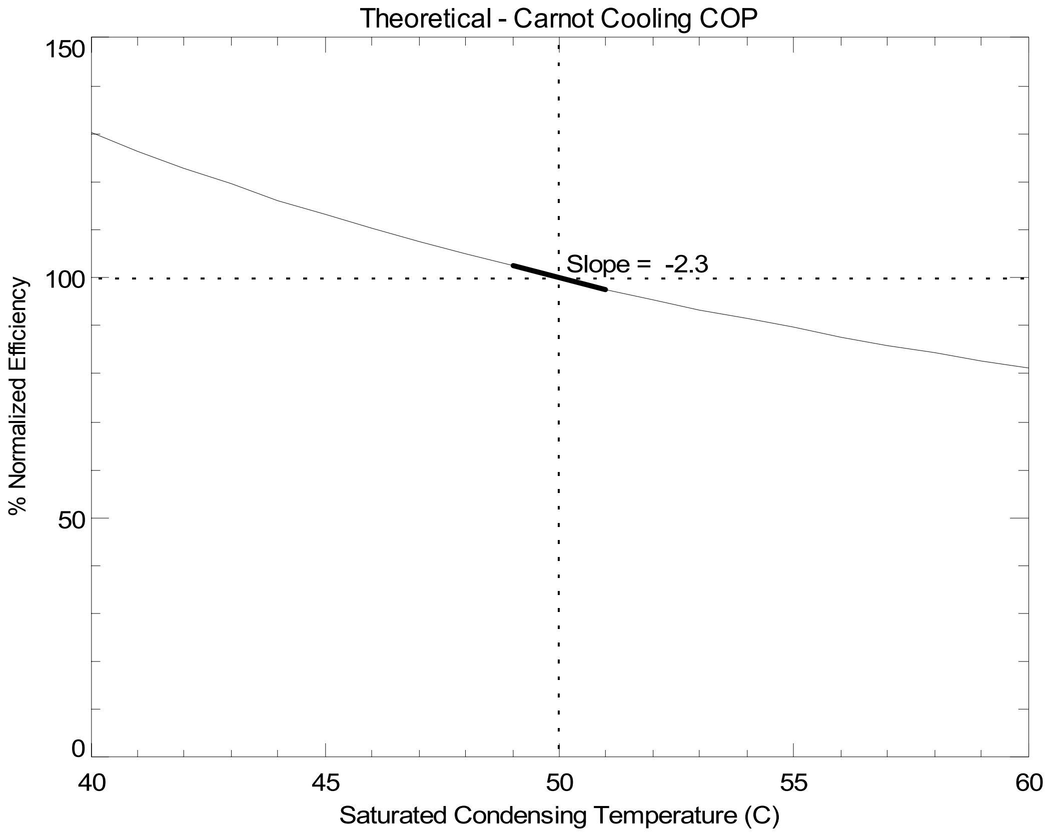


Figure 10. Variations in Efficiency Slope Based on the Carnot COP for a Theoretical Cooling Machine (SST=7°C)

The same process was repeated from the Copeland and Carlyle compressor curves to look at the impact of SCT on capacity. Table 7 and Figure 11 show the results for the Copeland compressors and Table 8 and Figure 12 show the results for the Carlyle compressors. The trends in the capacity slope were similar to those observed for the efficiency slope values. The Copeland compressors on average had slopes of -1.6% per °C for compressors rated for high temperature applications and -1.2% per °C for medium temperature compressors. The slope for scroll compressors were consistently lower than for reciprocating compressors (as had also been observed with HPDM). The Carlyle compressors averaged –1.5% per °C for high and –1.4% per °C for medium temperature units.

Overall, the detailed evaluation with empirical compressor curves tend to confirm that the performance characteristics of the two compressors used for the HPDM runs are typical of most other compressors. While some variations between the compressor types were observed, most of the variations are explained by compressor type (scroll or reciprocating) or temperature application (medium or high). The performance variations within a given category were typically on the order of 10%.

Table 7. Normalized Capacity Slope Determined for Various Copeland Compressors

		Compressor Curve *Capacity* Slope (% EER per °C)				
Compressor Application	**SCT (°C)**	**COPELAMATIC** N_{High}=45 N_{Med}=23	**COPELAWELD** N_{High}=49 N_{Med}=7	**DISCUS** N_{High}=0 N_{Med}=29	**SCROLL** N_{High}=6 N_{Med}=28	**ALL** N_{High}=100 N_{Med}=87
HIGH	45	-1.45	-1.64	-	-1.05	**-1.52**
	50	-1.57	-1.79	-	-1.17	**-1.65**
	55	-1.71	-1.96	-	-1.32	**-1.81**
MED	45	-1.12	-1.55	-1.26	-0.86	**-1.12**
	50	-1.19	-1.71	-1.34	-0.96	**-1.21**
	55	-1.27	-1.88	-1.45	-1.07	**-1.32**

Notes: Using default compressor curves with SST = 7°C. Using R-22 only.

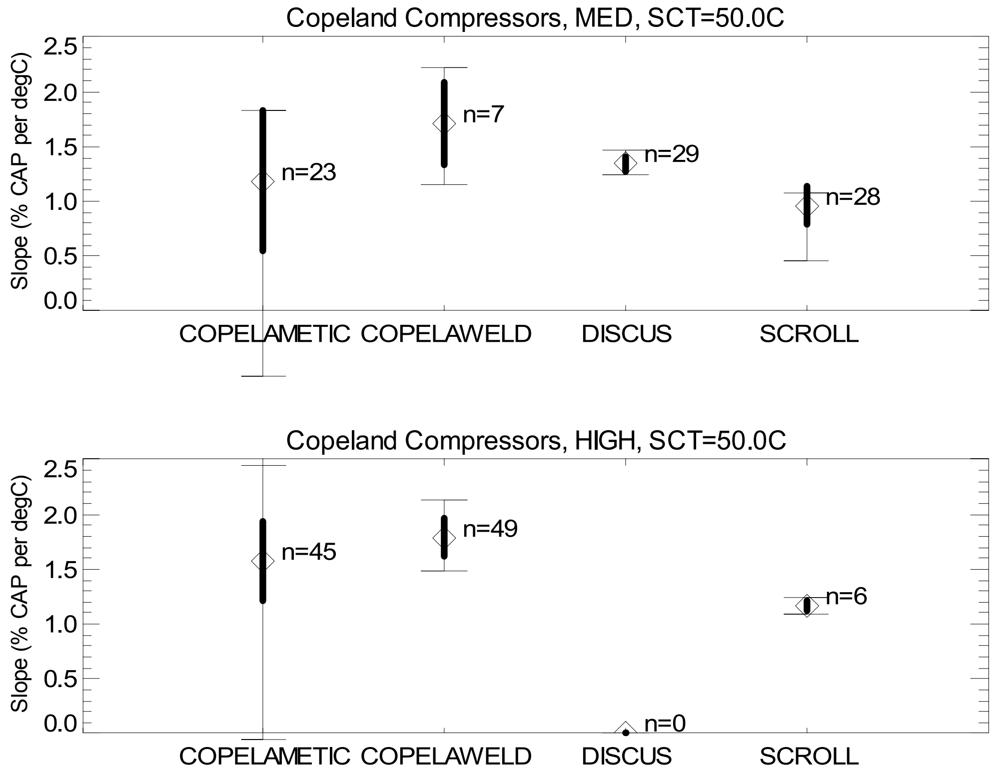
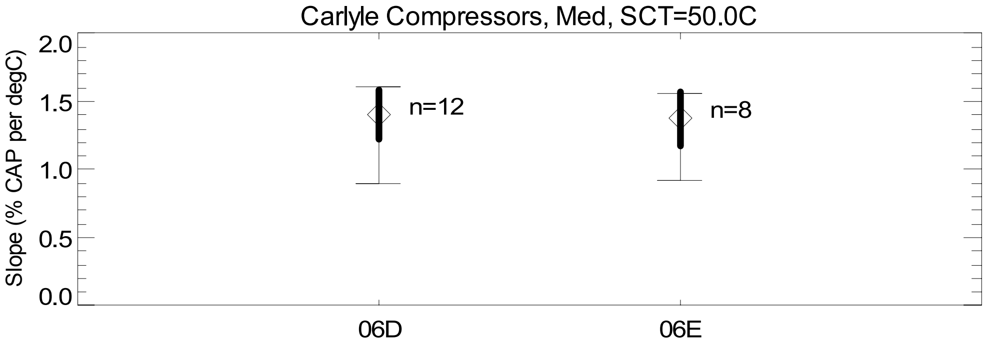


Figure 11. Variations in Capacity Slope Determined for Various Copeland Compressors

Table 8. Normalized Capacity Slope Determined for Various Carlyle Compressors

		Compressor Curve *Capacity* Slope (% EER per °C)		
Compressor Application	**Sat Cond. Temp (°C)**	**Carlyle 06D N_{High}=6 N_{Med}=12**	**Carlyle 06E N_{High}=7 N_{Med}=8**	**All N_{High}=13 N_{Med}=20**
HIGH	45	-1.41	-1.34	**-1.38**
	50	-1.51	-1.44	**-1.47**
	55	-1.61	-1.55	**-1.58**
MED	45	-1.32	-1.28	**-1.31**
	50	-1.40	-1.37	**-1.39**
	55	-1.50	-1.48	**-1.49**

Notes: Using default compressor curves with SST = 7°C. Using R-22 only.



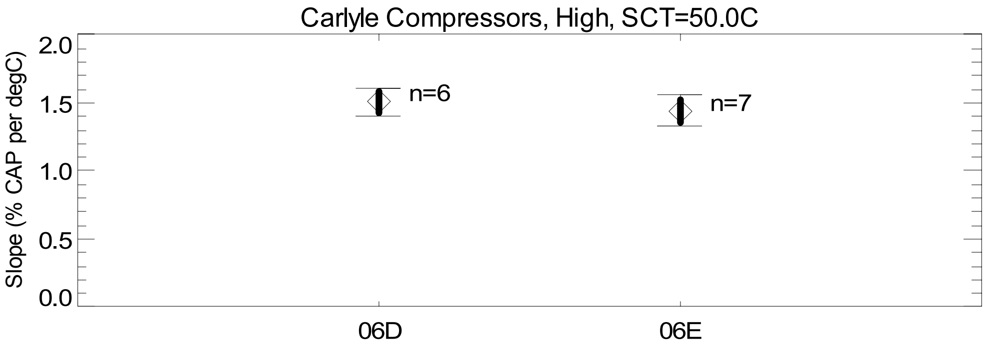


Figure 12. Variations in Capacity Slope Determined for Various Carlyle Compressors

Recommended Adjustment Factors

The concept of expressing adjustment factors in terms of a percent change in capacity or efficiency per degree Celsius appears to be technically sound. These factors should be consistently able to adjust for the minor differences between the three major "design" test points (i.e., ISO T1, NAFTA Test A, and Korean T1). Adjustments between substantially different test points (say T1 to T3) may also be possible, though more work is needed to confirm the approach can be extended over that wide a range.

Table 9 and Table 10 list our recommendations for factors to correct for minor variations in test conditions. These factors are based the HPDM results shown in Table 4 and Figure 1 through Figure 5. Of course, the specific test points used to normalize the adjustment factors will have an impact. In the case of these HPDM runs, the results were normalized to a percentage basis based on Test A conditions. However, these factors should still be appropriate when using either the ISO T1 or the Korean test results as the base condition.

Table 9. Recommended Factors to Adjust for Outdoor Conditions (% Change per °C)

	Efficiency	Capacity
Reciprocating Compressor	-2.0%	-1.2%
Scroll Compressor	-2.4%	-0.8%

Table 10. Recommended Factors to Adjust for Indoor Conditions (% Change per °C)

	Efficiency		Capacity	
	WB	DB	WB	DB
TXV	2.4%	0.16%	3.2%	0.14%
Short Orifice	1.6%	0.14%	2.4%	0.14%

The following example shows how the factors might be applied. We start with test results for a hypothetical air conditioner at NAFTA Test A conditions:

Cooling Capacity: 36,200 Btu/h
Cooling Efficiency: 9.75 Btu/Wh
(assume scroll compressor and TXV)

To convert this data to ISO T1 conditions, we first convert from English to SI units and express the results to the required number of significant digits[2]:

Cooling Capacity: (36,200 Btu/h) / (3.413 Btu/Wh) = 10,610 W
Cooling Efficiency: (9.75 Btu/Wh) / (3.413 Btu/Wh) = 2.86 W/W

Then we apply the appropriate adjustment factors from Table 9 for outdoor conditions and Table 10 for indoor conditions. The change in test conditions are taken from Table 2 (i.e., 0°C for ODB, -0.4°C for IWB, and +0.3°C for IDB).

Test A Cooling Capacity =		10,610 W	
ODB Capacity Adjustment =	(10,610 W) x (-0.008 x 0 °C) =	0 W	
IWB Capacity Adjustment =	(10,610 W) x (0.032 x -0.4°C) =	-136 W	
IDB Capacity Adjustment =	(10,610 W) x (0.0014 x +0.3°C) =	+5 W	
Calculated ISO T1 Capacity =		10,480 W	(-1.2%)

Test A Cooling Efficiency =		2.86 W/W	
ODB Efficiency Adjustment =	(2.86 W/W) x (-0.024 x 0°C) =	0 W/W	
IWB Efficiency Adjustment =	(2.86 W/W) x (0.024 x -0.4°C) =	-0.027 W/W	
IDB Efficiency Adjustment =	(2.86 W/W) x (0.0016 x +0.3°C) =	+0.001 W/W	
Calculated ISO T1 Efficiency =		2.83 W/W	(-0.9%)

2 ISO 5151 requires capacity to be reported to 4 significant digits and efficiency to significant 3 digits. The NAFTA standards require capacity and EER with slightly less precision: to the nearest 200 Btu/h and 0.05 Btu/Wh in this size range.

The changes in capacity and efficiency calculated with these factors are in line with the variations predicted in Table 3. However, the approach of using simple addition of the factors is probably not valid over a wider range of conditions.

One questions is: how certain we are about the recommended adjustment factors, and what impact that certainty would have on the resulting answer? As an example, if the adjustment factors are off by 10% for sample calculations above, then the resulting capacity and efficiency change would be off by 13 Watts and 0.003 W/W, respectively. An error of 13 Watts in the conversion is the same magnitude as the reporting requirements for capacity under ISO 5151 (i.e, to the nearest 10 Watts). A similar error of 0.003 W/W for efficiency is actually lower than the ISO reporting requirements, so would have no effect on the reported efficiency.

The variations in these adjustment factors due to compressors and other system issues are on the order of 10%. Therefore, we have some confidence that capacity and efficiency adjustment factors can be determined to within a certainty level that is less than or equal to the degree of precision required for reporting these values under ISO 5151. This is especially true when converting between the traditional "design" test conditions such as NAFTA Test A, ISO T1, and Korean TI.

Using the same simplified procedure to transform ISO T1 test results to ISO T3 conditions has a less satisfying outcome. The T3/T1 capacity and efficiency ratios (from Table 3) are 78.2% and 95.4%, respectively. The ratios calculated with the simplified procedure were 73.9% and 91.5%. The simplified procedure predicted capacity and efficiency changes that were 4% lower than had been predicted by HPDM.

3. IMPACT OF ELEVATION & BAROMETRIC PRESSURE

This section evaluates the impact that barometric pressure – and therefore test laboratory elevation – can have on the determination of capacity and efficiency. The focus of this analysis is to determine if the explicit and implied calculations in ISO Standard 5151 properly consider the impact barometric pressure. First we review the calculations to determine the air flow rate. Then, the impact of barometric pressure on other psychrometric calculations to determine cooling capacity are reviewed.

Air Flow Rate Calculations

The measurements and calculations to determine the air flow rate based on a standard flow nozzle from Annex B of ISO 5151 are given below (the calculations reflect the proposed revisions to the standard as of October 2000).

The volume flow rate of air is calculated by the equations outlined in Table 11 below. These calculations do explicitly require the barometric pressure, which is used to calculate the specific volume of air. Therefore barometric pressure is explicitly accounted for in the determination of the airflow by these procedures.

Table 11. Summary of Airflow Calculations from ISO 5151, Annex B

$$q_v = C_d A \sqrt{2 p_v v'_n}$$

where:

q_v	-	volume flow rate air-water mixture (m3/s)
C_d	-	Nozzle Discharge Coefficient
A	-	Nozzle area (m^2)
P_v	-	differential pressure across the nozzle (Pa)
v'_n	-	specific volume of moisture air at nozzle inlet (m^3/kg of moist air)

which is calculated as:

$$v'_n = \frac{v_n}{(1+W_n)} = \frac{p_A \, v_{n,sp}}{p_n (1+W_n)}$$

v_n	-	specific volume of dry air at nozzle inlet (m^3/kg of dry air)
p_A	-	standard barometric pressure (101,325 Pa)
p_n	-	barometric pressure at nozzle inlet (Pa)
W_n	-	humidity at nozzle inlet (kg water / kg dry air)

Other Psychrometric Calculations

Other psychrometric calculations are required to determine the latent and sensible cooling capacity. For instance, if the total capacity is determined by the air-enthalpy method (using the calculations from Annex D), the total, sensible and latent cooling capacities are calculated as shown in Table 12 below. While ISO standard shows how to calculate capacity, explicit guidance on the method of calculating enthalpy and humidity ratio from the measured data (i.e., dry bulb and wet bulb temperatures) is not given. These psychrometric calculations should include barometric pressure to be correct.

Table 12. Summary of Capacity Calculations from ISO 5151, Annex B

Total Capacity: $\phi_{tci} = \frac{q_{vi}(h_{a1} - h_{a2})}{v_n} = \frac{q_{vi}(h_{a1} - h_{a2})}{v'_n(1+W_n)}$

Sensible Capacity: $\phi_{sci} = \frac{q_{vi}\left(c_{pa1}t_{a1} - c_{pa2}t_{a2}\right)}{v_n} = \frac{q_{vi}\left(c_{pa1}t_{a1} - c_{pa2}t_{a2}\right)}{v'_n(1+W_n)}$

Latent Capacity: $\phi_{lci} = \frac{K_1 q_{vi}(W_{i1} - W_{i2})}{v_n} = \frac{K_1 q_{vi}(W_{i1} - W_{i2})}{v'_n(1+W_n)}$

Where:

q_{vi}	-	volume flow rate air-water mixture from indoor coil (m^3/s)
v_n	-	specific volume of dry air at nozzle inlet (m^3/kg of dry air)
t_{a1}	-	temperature of air entering indoor coil (°C)
t_{a2}	-	temperature of air leaving indoor coil (°C)
c_{pa1}	-	specific heat of air at entering conditions (kJ/kg-°C)
c_{pa2}	-	specific heat of air at leaving conditions (kJ/kg-°C)
h_{a1}	-	enthalpy of air entering indoor coil (kJ/kg of dry air)
h_{a2}	-	enthalpy of air leaving indoor coil (kJ/kg of dry air)
W_{i1}	-	humidity of air entering indoor coil (kg water/kg dry air)
W_{i2}	-	humidity of air leaving indoor coil (kg water/kg dry air)

Figure 13 shows the impact that barometric pressure (and therefore elevation) can have on the calculated humidity ratio. A 2% change in barometric pressure can change the entering humidity by 0.006 kg/kg.

To get an idea of the impact barometric capacity has on latent capacity, we can evaluate the impact that pressure would have on the humidity difference across the cooling coil (i.e., W_{i1} - W_{i2}). The humidity difference is directly related to the latent capacity of the unit. Figure 14 shows how the "latent capacity" would be affected by barometric pressure. If the barometric pressure is 2% different from standard pressure, then psychrometric calculations that ignore barometric pressure can result in a 5% error in the determination of latent cooling capacity.

Figure 15 repeats this process for the enthalpy difference across the cooling coil (h_{a1} - h_{a2}), which is proportional to the total capacity of the unit. Again, a 2% change in the barometric pressure can result in a 0.7% change in total capacity. Therefore, psychrometric calculations that determine enthalpy but do not consider barometric pressure are expected to introduce errors of this magnitude.

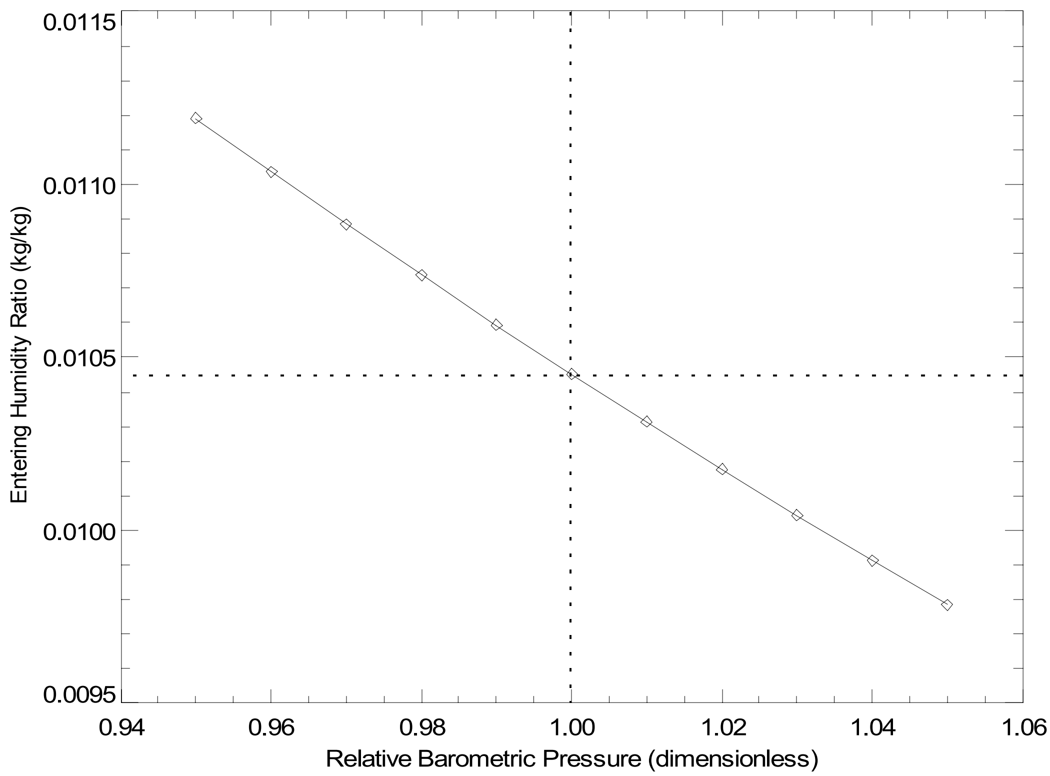


Figure 13. Impact of Barometric Pressure on Calculated Humidity Ratio

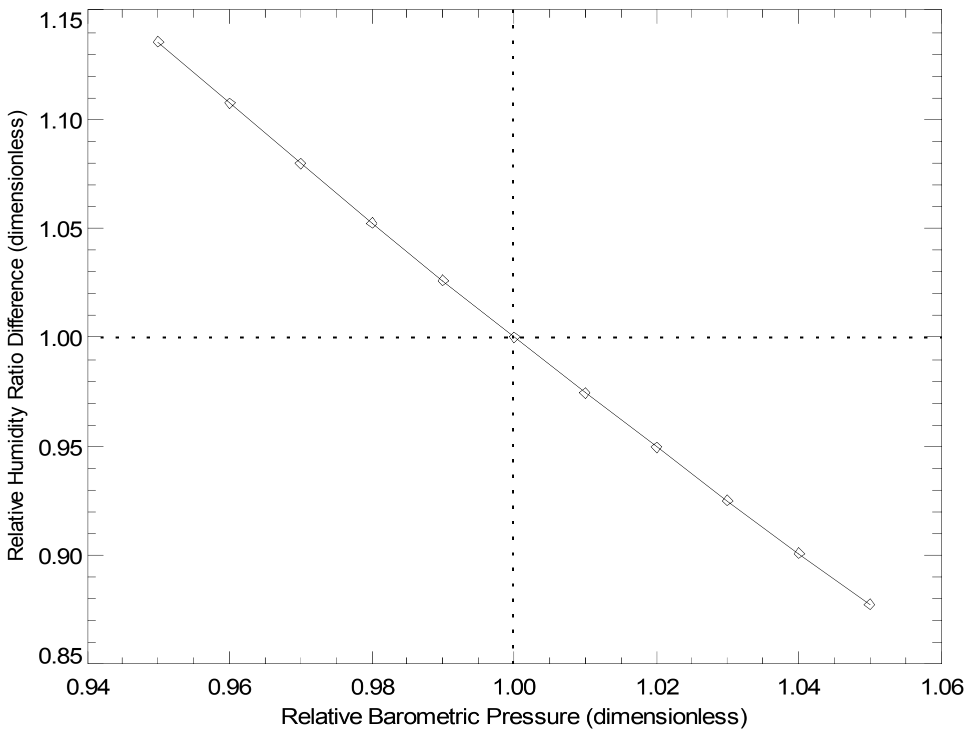


Figure 14. Impact of Barometric Pressure on "Latent Capacity" Calculations (W_{i1} - W_{i2})

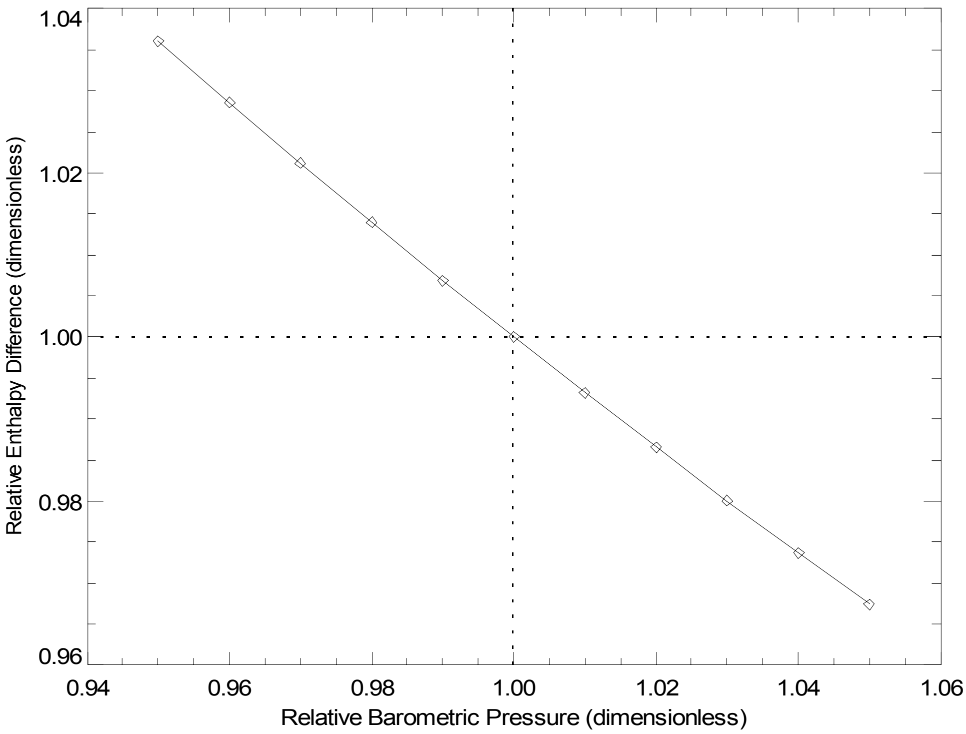


Figure 15. Impact of Barometric Pressure on "Total Capacity" Calculations (h_{a1} - h_{a2})

Barometric pressure can normally fluctuate by 2-3% do to changes in weather. Elevation above sea level, can also have a significant impact. For each 100 meter increase in elevation, the nominal barometric pressure decreases by 1.16%.

4. SEASONAL EFFICIENCY CALCULATIONS

The stated goal of many minimum energy performance standards (MEPS) is to provide an indication of annual energy use for the appliance or system. All current air conditioner test standards, with the exception of the American SEER procedure, provide an efficiency value that is representative of performance at peak conditions. Therefore, these nominal efficiency values are not a good predictor of the seasonal average cooling efficiency.

While nominal efficiency values such as ISO T1 are not representative of seasonal performance, many within the industry have historically argued that they still do provide a representative ranking of the expected seasonal energy use for most air conditioners. Previous comparisons of nominal and seasonal efficiency values for air conditioner products in the American market demonstrated the strong correlation between these seasonal and nominal efficiencies (Thomas, Tree and Goldschmidt 1980).

A disruption to this correlation has resulted by the introduction of variable-speed and two-speed systems into the market. Modulating systems provide high efficiency at part load conditions. As a result, a modulating system may have a modest efficiency at T1 conditions, yet have much better seasonal efficiency than a single speed system.

The American SEER procedure takes a uniform approach in its treatment single and multi-speed systems. The next section reviews its calculation procedures and the history behind them in an effort to highlight the features that other APEC economies might find useful in their efforts to predict seasonal efficiencies appropriate for local conditions.

NAFTA's SEER Test and Rating Procedure

The U.S. procedure to determine the seasonal energy efficiency ratio (SEER) was developed by the National Bureau of Standards (NBS) which is now known as the National Institute of Standards and Technology (NIST) by a team of researchers (Parken et al 1977; Kelly & Parken 1978; Parken et al 1985). The SEER calculation procedures were originally developed based upon a bin analysis that calculated the cooling load, capacity and efficiency over a range of ambient temperatures. Temperature bin data were used to assign the number of hours to each temperature bin and effectively weight the AC operating hours based on the time spent at each operating condition.

Table 13. Summary of Original Bin-Based Methods to Calculate SEER

$$SEER = \frac{Annual\_Cooling}{Annual\_Energy\_Use} = \frac{\Sigma q(T_j) \cdot n_j \cdot CLF}{\Sigma e(T_j) \cdot n_j \cdot \frac{CLF}{PLF}}$$

where:

$q(T_j)$ - trend for AC cooling capacity as a function of ambient temperature
$e(T_j)$ - trend for AC cooling energy use as a function of ambient temperature
T_j - ambient temperature in the j^{th} bin
n_j - number of hours in the j^{th} bin

and where:

$$CLF = \frac{BL(T_j)}{q(T_j)}$$

$$BL(T_j) = \frac{q(95)}{1.1} \cdot \frac{(T_j - 65)}{(95-65)}$$

$$PLF = 1 - C_d \cdot (1 - CLF)$$

CLF - cooling load fraction
PLF - part load fraction (degrades efficiency at part load)
$BL(T_j)$ - building cooling load line
(assuming the AC unit is 10% oversized at 95°F and the load goes to zero at 65°F)
C_d - cooling degradation factor (assumed to be 0.25 by default)

Temperature Bin Data
"1000 Hour" Cooling Season

Temp - T_j (°F)	67.5	72.5	77.5	82.5	87.5	92.5	97.5	102.5
Hours - n_j	214	231	216	161	104	52	18	4

This bin-method was computationally intensive and therefore the following simplified method was developed as an approximation:

$$SEER_0 = EER(82) \cdot (1 - C_d \cdot (0.5))$$

This much simpler approximation was found yield very similar values for SEER and therefore was adopted into all current versions of the standards the describe the SEER calculation procedure (ARI 210/240-1994; ASHRAE 116-1995; DOE 1979). While, the bin-based method has been dropped for single speed units, it has been retained for the SEER calculations for two-speed and variable speed units. The greater complexity of analyzing these units has justified retaining the more complex bin-based calculation methods.

Figure 16 graphically shows the results of bin calculation procedure using a hypothetical single speed unit with an EER of 10 Btu/Wh at 95°F and 12 Btu/Wh at 82°F (or a cooling COP of 2.93 W/W at 35°C and 3.52 W/W at 27.8°C). Using the temperature data from the typical "1000 hour cooling season" specified in the U.S. standards, the bin calculation method predicts the SEER to be 11.29 Btu/Wh, which is within 1% of the SEER of 11.4 Btu/Wh predicted by the simplified SEER calculation method.

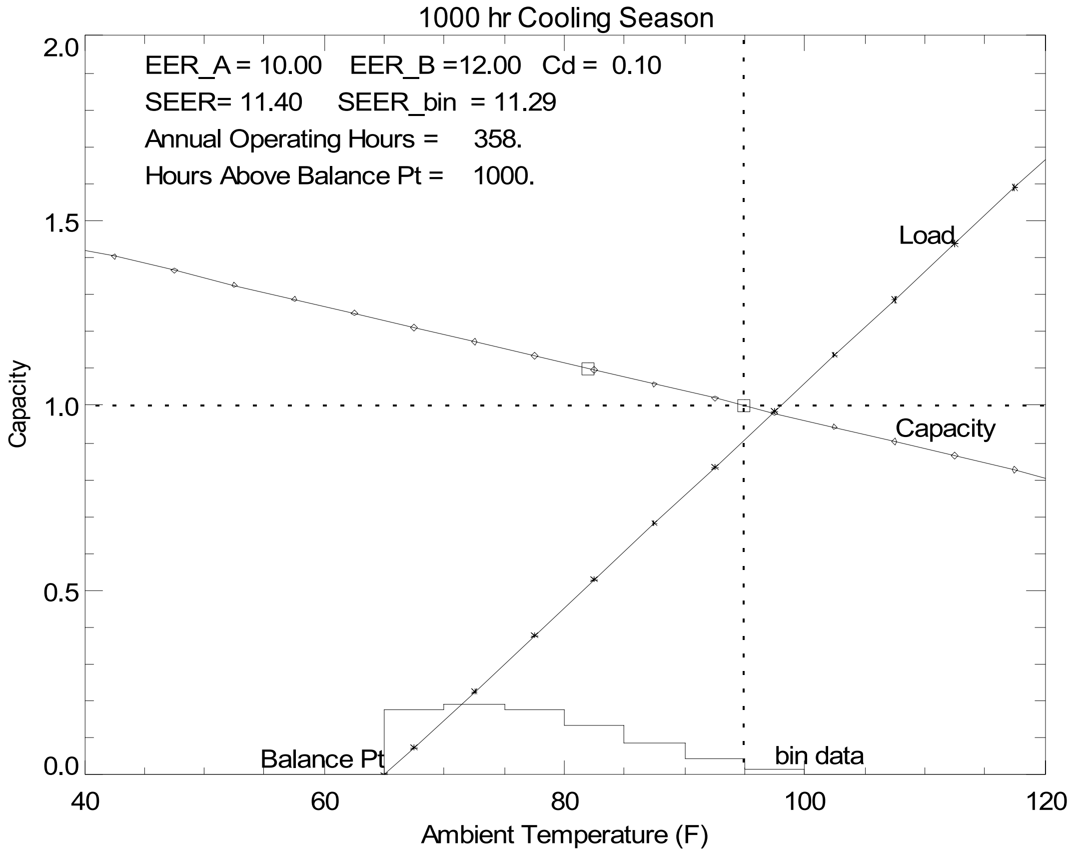


Figure 16. Graphical Description of Original Bin-Based Calculation Procedures for SEER

The original SEER development work recognized that different temperature bin data for different climates would yield slightly different seasonal efficiency values. However they found that the variations were small for most U.S. locations. Figure 17 and Figure 18 confirm that the bin-calculated SEER does not change substantially when different weather data are used. Both plots used typical meteorological year (TMY) data for 232 locations. The sites include locations in the 50 US states as well as selected military installations around world.

The distribution of bin-calculated SEER values for all 232 sites is given in Figure 17. The average of sites is 11.65 Btu/Wh, compared to the nominal SEER value of 11.4 Btu/Wh for this hypothetical unit. Many individual locations were much higher than that average, reaching values as high as 13.4 Btu/Wh. Figure 18 shows that many of the outlying points were caused by extremely light cooling loads. Locations with fewer than 200-300 full load cooling hours[3] accounted for nearly all of the SEER values over 12 Btu/Wh. The points with less than 50 full load hours generally correspond to locations in Alaska. Most locations conformed fairly well to the nominal SEER value of 11.4 (which is shown on both plots as a dotted line). Outlier values on the plot at high cooling loads generally correspond to locations outside of North America. These are points are identified on the plot, including locations in Hawaii (HI), Puerto Rico (PR), Cuba (CU), and the Philippines (PN). Desert locations such as Arizona (AZ) appeared to account for the below average seasonal values

[3] Full load cooling hours (FLC) are defined as the total annual cooling load, divided by the cooling capacity of the unit at Test A conditions.

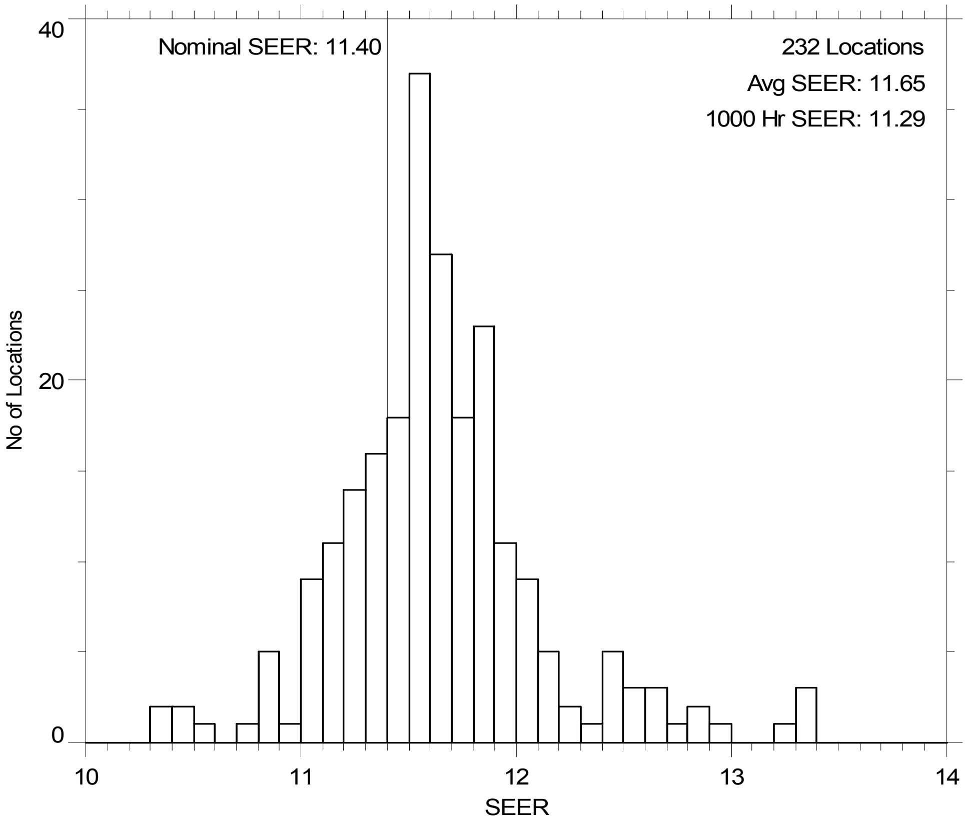


Figure 17. Variation in Bin-Calculated SEER for 232 TMY Locations

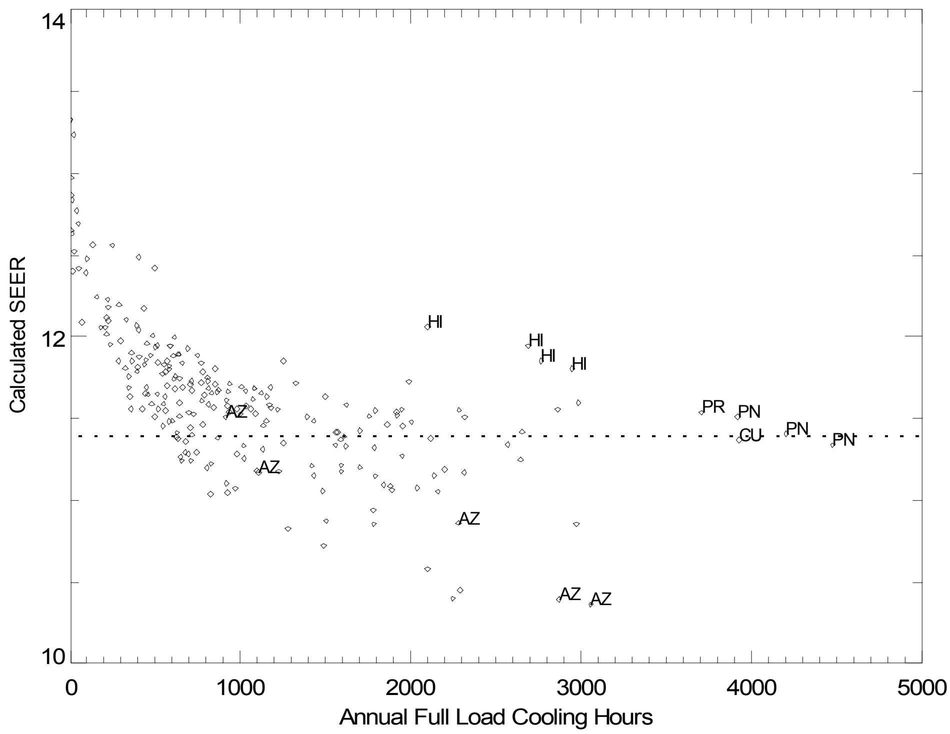


Figure 18. Trend of Bin-Calculated SEER with Cooling Load for 232 TMY Locations

Considering Variable Speed Systems

The seasonal efficiency of variable- and two-speed systems are also be determined by this bin approach in the SEER procedure. For two-speed systems, a second capacity line is added to Figure 16 corresponding to the lower compressor speed (this requires two additional test points). Then the system is assumed to operate at low speed in temperature bins where the low speed capacity is equal to or greater than the cooling load line. For variable-speed systems, the low-speed line corresponds to the minimum operating speed for the system and another intermediate speed test point is required to the predict the performance of the system as it modulates from low to high speed in response to the load.

Developing Seasonal Efficiency Ratings for Other Economies

As demonstrated by the NAFTA SEER procedure, a prediction of seasonal air conditioner efficiency requires that capacity and efficiency be measured at a minimum of two test conditions for a constant speed system. Multiple speed systems will require even more test points. Then – as the HPDM runs in Section 2 had shown – there is sound technical basis for using a linear model to extend those results across a wide range of ambient conditions. This type of capacity functions should be the basis for any prediction of seasonal efficiency.

Some argument could even be made for using a single test point with the adjustment factors given in Section 2 to develop a linear capacity function. However, this “single test point” approach would still drive manufacturers to create products that are optimized at a single test condition, effectively defeating purpose of developing a seasonal efficiency rating procedure.

The next step is to select weather data that are appropriate for determining seasonal performance. As shown in Figure 17 and Figure 18, two test points can be used to calculate the seasonal efficiency for any set of weather conditions. Therefore, a single set of test conditions can be used in economy-specific calculation procedures of seasonal efficiency.

The ISO T1, T2 and T3 test conditions could serve this purpose. Even though indoor conditions do vary slightly between the three test points, it can be argued that the variations are consistent with the corresponding changes in ambient. Hot climates could use T1 and T3 conditions to develop an appropriate linear function. Cooler climates could use T1 and T2 test conditions to develop a linear model. The choice of balance point and the over-sizing assumptions included in the SEER procedure could also be tailored to the needs and characteristics of the local conditions.

5. CONCLUSIONS AND RECOMMENDATIONS

Conclusions

This study has evaluated the test procedures for air conditioners and heat pumps to understand how algorithms could be used to:

1. reduce the testing burden on manufacturers by correcting for small differences between common test points,
2. improve the consistency of laboratory tests conducted at various altitudes,
3. develop more realistic predictions of seasonal efficiency and energy use.

This study has shown that simple, straightforward algorithms can be used to adjust for the minor differences between the common "design" rating points, including ISO T1, Korean T1, and NAFTA's Test A. A procedure was developed in this study using "generalized" adjustment factors to correct for these minor differences in operating conditions. An evaluation of numerous compressors demonstrated that the amount of variation in these factors was small compared the reporting precision required by each test standard. The sample calculations presented in this study show that the adjustment factors can be used with confidence to correct for the small differences between the operating conditions used by these common rating points. Corrections over wider differences might be possible as well, though more refinement would be necessary to accurately and consistently make these corrections.

This study reaffirmed the importance of barometric pressure measurements in the proper determination of air conditioner cooling capacity and efficiency. Barometric pressure is explicitly required for air flow calculations in ISO 5151. However, its use in the general psychrometric calculations of that standard is implied but not explicitly stated. The importance of explicitly including barometric pressure in all psychrometric calculations was demonstrated through sensitivity analysis. A 2% change (or error) in barometric pressure was shown to introduce errors of 1% in total capacity and 5% in latent capacity. Each 100 meter change in elevation can result in more than a 1% change in the barometric pressure, so these calculations are especially important for test laboratories located at a high altitude.

Most air conditioner test and rating procedures used by APEC member economies – with the exception of NAFTA's SEER procedure – provide an efficiency measure that is only representative of performance at nominal or design conditions. In order for a test standard to be representative of seasonal energy use, it must also consider system performance at part load conditions. The history and development of the US SEER test procedure was reviewed to understand how the seasonal average efficiency is determined. The review focused on elements of the procedure that might be appropriate for other APEC economies that wish to develop a seasonal efficiency standard, including:

- the accuracy of linearly extrapolating measured performance over a range of ambient conditions based on only two test points,
- the need for additional test points when determining seasonal efficiency for two-speed and variable-speed systems,
- the flexibility of the bin-based calculation approach and ways that it could be applied in other climates and conditions,

The potential for combining accepted international laboratory test procedures, such as ISO 5151, with economy-specific seasonal efficiency calculations is also discussed. Even though the T1, T2, and T3 test points include slight variations in indoor conditions, the slight changes between these points are consistent with the change in ambient temperature. Therefore, the ISO test points could easily be used to develop simple linear models of performance as a function of ambient temperature that could be used in seasonal efficiency calculations. Cooler climates might choose to use the T1 and T2 test points while hot climates could use T1 and T3 test points. This approach could provide each economy with estimates of seasonal efficiency tailored to local needs while retaining a common test standard for manufacturers. Variable- and two-speed systems would require additional tests at the lower speeds, though the same test conditions could most-likely be used.

Recommendations

Based on the results of this study, we recommend that further work be completed in the following areas:

1. The validity of using adjustment factors to account for minor differences between test conditions should be verified by using actual test data for various products. It seems likely that one or more manufacturers may have test data for the same product tested at different conditions (for example: ISO T1, Korean T1, and Test A). If a significant sample of test results is available from a mix of manufacturers, a statistical analysis could be used to confirm the accuracy and validity of the using the simplified adjustment procedures for a wide array of equipment.

2. Further study is required to confirm that the adjustment factors developed in this study based on US semi-hermetic compressor and air conditioner products are also representative products from other APEC member economies. One important segment for consideration would be Japanese manufacturers who supply most of the smaller hermetic compressors used in room air conditioners and heat pumps in Pacific Rim economies. Further work is also necessary to determine if other system issues – beyond expansion device and compressor type – are pertinent factors for categorizing these adjustment factors.

3. The need and potential for developing a seasonal cooling efficiency standard based on the current ISO5151 standard should be further considered. Generalized seasonal efficiency calculation procedures could be added as an addendum or as a new standard. The specific parameters and assumptions about load and climate could be selected based on the economy-specific needs.

4. Methods to determine seasonal efficiency for the heat pumps in the heating mode should also be developed. Again, the current test conditions in ISO standards should be used as the basis for developing procedures to predict seasonal efficiency and energy use. Then economy-specific parameters can tailor the seasonal calculation procedures to represent local needs.

5. Further work is needed to develop test procedures to handle variable- and two-speed systems. The ISO Working Group 6 overseeing ISO 5151 (SC6/WG) was reportedly working to incorporate the Japanese approach to handling variable-speed systems (JIS B 8616 1999). However, recent communications with at least one committee member indicates that this process of incorporating the Japanese variable speed procedures into ISO 5151 is making slow process. Now may be a good opportunity to revisit the issue of how to develop a consistent and uniform means of predicting seasonal efficiency for both variable- and constant-speed air conditioner systems.

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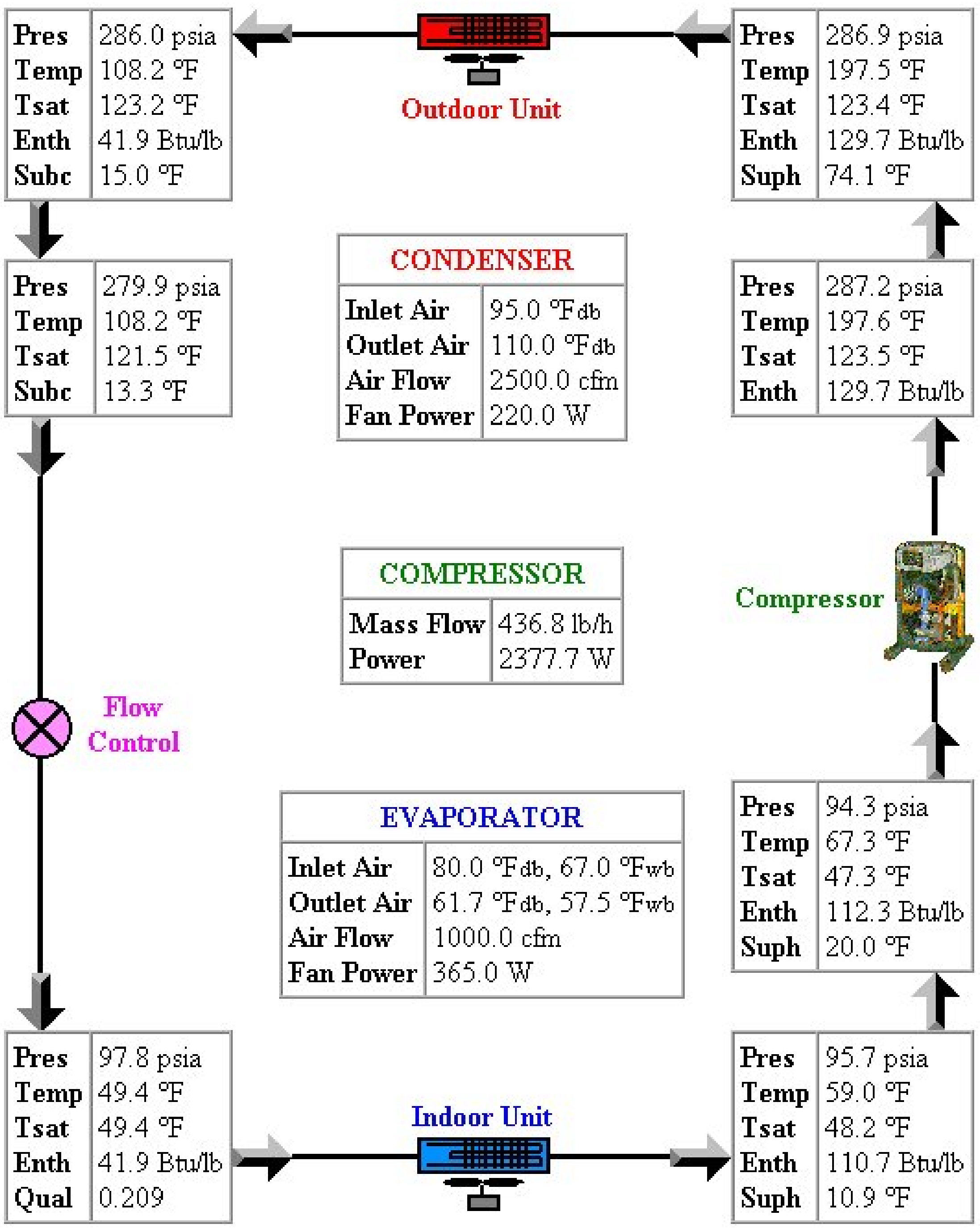
APPENDIX A

Results from Base Case HPDM Runs (Test A, Recip, TXV) - SCREEN 1

COOLING MODE	
Refrigerant	R-22
System EER	9.739 Btu/Wh
System Capacity	28855 Btu/hr
Sensible Heat Ratio	0.701
Total Power	2962.7 W



Results from Base Case HPDM Runs (Test A, Recip, TXV) – SCREEN 2

COOLING MODE	
Refrigerant	R-22
System EER	9.739 Btu/Wh
System Capacity	28855 Btu/h
Outdoor Ambient	95.0 °F
Indoor Ambient	80.0 °F

Outdoor Unit

Flow Control Size	
Cap. Tube ID	0.105 in
Orifice ID	0.066 in
TXV Capac.	1.861 tons

Flow Control Specs	
Cap. Tube	
#	1
Length	80.00 in
Orifice	
#	1
Length	0.500 in
Chamfer	0.000 in
TXV	
Nozzle Size	2.5 tons
Distrib. ID	1/4 in
Distrib. L	30.00 in

Flow Control

CONDENSER	
Face Area	14.713 ft^2
Face Velocity	169.9 fpm
Air-Side U	7.689 Btu/h-ft^2-°F
Air-Side A	450.7 ft^2
Total UA	1905 Btu/h-°F
Air-to-Tot R	61.6 %

Charge Inventory	
Condenser	62.1 %
Liquid Line	10.8 %
High-Side	73.3 %
Compressor	5.6 %
Accumulator	0.0 %
Evaporator	18.8 %
Low-Side	26.7 %
Total Charge	6.572 lbm

EVAPORATOR	
Face Area	3.802 ft^2
Face Velocity	263.0 fpm
Air-Side U	6.731 Btu/h-ft^2-°F
Air-Side A	223.2 ft^2
Total UA	1161 Btu/h-°F
Air-to-Tot R	83.4 %

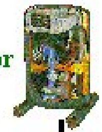
Compressor Size	
Displacement	2.366 in^3
Swept Volume	281.4 cfh
Motor Size	2.250 hp

Compressor Rating	
EER	10.90 Btu/Wh
Capacity	28200 Btu/h

Compressor Map Multipliers	
Power	1.000
Mass Flow	1.000

Compressor Operation	
Pressure Ratio	3.047
% Nom. Load	123.5 %
Speed	3425 rpm
Volum. Eff.	95.9 %
Motor Eff.	86.5 %
Isen. Eff.	68.2 %

Compressor



Indoor Unit

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