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Design and Construction of a Prototype High Efficiency Air Conditioner

Prepared for:
Pacific Gas & Electric Company

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ABSTRACT

In the 1994 Report "Investigation of Peak Electric Load Impacts of High SEER Residential HVAC Units" Pacific Gas and Electric Company (PG&E) Research and Development Department and Proctor Engineering Group found that a number of techniques are effective at reducing air conditioner peak KW draw. These include: increased condenser / evaporator areas and efficiencies, increased condenser / evaporator fan efficiencies, and increased indoor / outdoor fan motor efficiencies. In that study PEG simulated multiple air conditioner designs with various parameters. Four designs were found capable of reducing peak draw by at least 500 Watts.

In this study, one of the reduced peak units was assembled from "off the shelf" components. The prototype unit was tested in the laboratory under hot dry and hot moist peak conditions. The prototype unit met the goal, showing a reduced peak watt draw equivalent to the simulation. Diversified local peak reduction was estimated at 550 watts.

The prototype unit had a higher efficiency than the typical units at SEER test conditions (82°F outside) and the difference in efficiency increased at higher outdoor temperatures. All high SEER air conditioners tested in a 1995 EPRI laboratory test of ten units (Bain et. Al., 1995) had efficiencies that deteriorated faster than the prototype unit at high temperatures. This improvement in efficiency would not only benefit the utility at peak, but also benefit the customer since the equipment sees greater use at higher temperatures. The annual customer energy savings are estimated to be between 11% and 20%.

This study also determined the effect of incorrect charge and low air flow across the inside coil at hot dry conditions. These common problems can increase peak load by over 500 watts on typical units. The Prototype is less affected (but not unaffected) by incorrect charge.

Additional peak reduction items are also discussed. Notably the indoor fan motor drew over 450 watts while producing only 44 watts of effective work. The efficiency of the indoor fan and motor assembly was 9% at standard flow conditions. An increase to 15% efficiency was accomplished merely by reducing the cabinet restriction at the blower inlet. Significant efficiency improvement is possible through optimization of fan / motor / cabinet configurations.

EXECUTIVE SUMMARY

This project built and tested a reduced peak consumption air conditioner that was designed in PG&E's "Investigation of Peak Electric Load Impacts of High SEER Residential HVAC Units". That study developed four AC designs that had simulated peak loads and annual energy consumption substantially lower than the standard units. The primary goals of this project were to assemble an air conditioner with design parameters close to one simulated design, test it in the laboratory, and calculate potential peak reduction for PG&E.

Table 1 summarizes the peak effect of the prototype unit compared to typical SEER 10 air conditioners.

Table 1. Peak Reduction Due to Prototype Design
(3 Ton Unit, PG&E Central Valley Peak Conditions 115°F Outside 80°F / 61°F Inside)

	Connected Load (KW)	**Connected Load Difference** (KW)	**Local Peak**[1] (KW Reduction)	**System Peak**[2] (KW Reduction)
Prototype	3.51			
Typical Unit[3]	4.35	0.84	**0.55**	0.39

This investigation reached the following conclusions:

1) The laboratory tests completed in this project confirm that it is possible to build reduced peak KW residential air conditioners with "off the shelf" components. A 3-ton residential air conditioner of the reduced peak design would have a diversified local peak approximately 0.55 KW less than existing SEER 10 units.

2) The projected peak reduction comes from higher efficiency at high temperatures. While the simulation overpredicted the EER of the typical and the prototype units, the percentage change in EER was very closely approximated by the simulation. The MODCON (PUREZ) simulation is a helpful tool in screening air conditioner designs for efficiency differences at peak conditions.

3) The residential customer would benefit from air conditioners designed similar to the Prototype. Such air conditioners would consume 11% to 20% less energy annually than the SEER 10 baseline units. The Prototype unit holds its energy efficiency at higher temperatures where many high SEER units loose their efficiency edge.

4) Incorrect refrigerant charge seriously affects the efficiency of both the Prototype and typical air conditioners. Units with TXV metering devices (as used in the Prototype) are less effected than units with fixed metering devices. With a fixed metering device, the EER of the air conditioner will drop by approximately 48% for as little as 30% undercharge. The effect of a 48% drop in efficiency

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[1] The highest one hour average peak draw for a defined local area. For a primarily residential climate in a hot climate it is likely to occur in the early evening (5 pm to 8 pm) on a very hot weekday.

[2] The highest one hour average peak draw for the entire PG&E utility. This peak normally occurs in the early afternoon on a hot weekday.

[3] As characterized by Carrier model 38CK036 (CC5A / CD5A / CD5BA036)

would be an almost doubling in diversified power draw if the unit could still cycle. The Prototype unit's efficiency would drop only 14% under the same 30% undercharge.[4].

5) The Prototype was tested under a variety of conditions (PG&E Peak, Hot, and Warm) with the indoor coil air flows 70% and 100% of standard. The relative effect of low air flow was more pronounced at peak conditions than at warm conditions. A 30% reduction in air flow under peak conditions results in a capacity drop of 12.8% and a drop in EER of 6.9%. The effect of the reduced EER[5] is an increase in diversified load of 7%[6].

6) The indoor fan / motor efficiency as installed was 9% compared to the goal of 30%. Elimination of the restriction caused by the close proximity of cabinet walls increased the efficiency to 15%. Significant gains in indoor fan efficiency are likely with changes in cabinet configuration.

7) The Prototype would cost approximately $190 more than a standard SEER-10 unit and approximately $265 less than a typical SEER-12 unit[7].

Based on these results, Proctor Engineering Group recommends:

1) Limited field testing of equipment with the characteristics demonstrated in the prototype design.

2) Investigation and implementation of approaches encouraging the introduction and sale of air conditioners with the characteristics demonstrated in the prototype design

3) Lab testing of fan / motor / cabinet modifications to improve efficiency.

4) Consideration of a low latent capacity air conditioner design for the PG&E service area. An air conditioner designed with higher evaporator temperatures would have lower latent capacity and a lower kVA on peak. It is sometimes argued that the latent capacity is needed in PG&E's service area on some occasions.

5) Field monitoring to determine sensible and latent load in PG&E's service area. This monitoring would answer two questions. First, the amount of load overestimation (and subsequent oversizing of air conditioners) present in Manual J and, second, the actual latent loads in PG&E's service area. Reduction in air conditioner size and air conditioners designed to higher evaporator temperatures would both reduce peak kVA.

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[4] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes incorrect charge even more detrimental than the drop in EER indicates.

[5] for air conditioners that are cycling.

[6] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes low air flow even more detrimental than the drop in EER indicates.

[7] to the contractor

I. INTRODUCTION

BACKGROUND

Residential air conditioning systems (AC) produce little utility revenue but they do produce high coincident peak load. This fact makes air conditioners a logical target for utility efforts at market transformation. It cannot be assumed however that increasing the market penetration of "high SEER" air conditioners will show peak reductions proportional to annual energy savings. Proctor Engineering Group's simulations and field studies of residential air conditioning systems have shown that peak load impacts cannot be directly correlated with energy savings (Proctor 1993). Pacific Gas and Electric Company (PG&E) is interested in equipment changes that will "lock in" peak demand reductions.

Programs designed to reduce energy use can substantially degrade the load factor for electric utilities. This results in a requirement for increased fixed assets to meet conditions that exist for only a few hours of the year, and reduced revenue from the remaining AC usage. From both utility and rate payer perspectives, it is important to reduce the peak load as well as to achieve the energy savings associated with a high efficiency air conditioning system. Reduction of the characteristic peak load from residential AC units may yield significant financial savings to all rate payers.

PROJECT HISTORY

In the study "Investigation of Peak Electric Load Impacts of High SEER Residential HVAC Units" Pacific Gas and Electric Company (PG&E) Research and Development Department and Proctor Engineering Group found combinations of "off the shelf" components that, when combined, could reduce diversified peak demand by 500 watts for a 3 ton air conditioner. One of the designs (Design C) was selected for assembly and testing under a variety of conditions that simulated actual field conditions in PG&E's service territory. These conditions included incorrect charge and low air flow across the inside coil.

Design C was selected as the best unit to build because the evaporator coil could be installed without modification to standard furnace cabinet and it did not use the more expensive Brushless Permanent Magnet motors.

MANUFACTURERS' REVIEW

The designs produced in the study were reviewed with major manufacturers and were viewed as potentially feasible designs. Prior to this phase of the project Proctor Engineering Group again discussed the design with Carrier Corporation representatives. At their suggestion, an accumulator was added to the system to further protect the compressor from the return of liquid refrigerant. With this addition, the design was considered feasible.

DEFINITIONS AND CONVENTIONS

In this report, the following definitions and conventions apply:

- All temperatures are in °F;
- When the moisture content of the air is important (such as indoor conditions) the dry bulb temperature is listed first and the wet bulb temperature second (ex. 80/67 is 80°F dry bulb and 67°F wet bulb);
- Unit watt draw is the watt draw of the air conditioner including indoor fan, outdoor fan, and compressor;
- Indoor conditions are the air conditions prior to the indoor fan and motor (this is different from coil entering conditions);
- Capacity is the total amount of energy (sensible and latent) removed from the air stream measured from before the indoor fan to after the indoor coil;
- Normalized Capacity is the capacity at test conditions divided by the capacity at ARI conditions
- EER is the capacity divided by the unit watt draw at steady state test conditions

II. APPARATUS AND TESTS

Proctor Engineering Group built a prototype air conditioner that followed Design C, the simulated unit chosen from the four designs in the 1994 project. The prototype unit (Prototype) was assembled and tested at the Energy Systems Laboratory (ESL) of Texas A&M University, College Station, Texas. ESL's psychrometric rooms are located at the Riverside Campus Testing and Calibration facility.

DESCRIPTION OF A/C TEST UNIT

The test unit was a combination of "off the shelf" components. Compared to the baseline SEER 10 unit, it consisted of a larger condenser coil, a smaller reciprocating compressor, and a thermostatic expansion valve. While the Prototype closely followed Design C, it is not exactly the same as the simulation. A description of the components assembled into the Prototype are compared to Design C components in Table 2.

Table 2. Prototype Unit and Simulated Unit Components

	Prototype Unit	Design C
CONDENSER UNIT		
Compressor	Copeland CR28K6-PFV	Same
Accumulator	Model KH71KN160	None
Fan	Carrier 38BRB042 (Revcor 3 Blades)	Combined Fan/Motor Efficiency = 0.15
Fan Motor	Carrier 38BRB042 General Electric 5KCP93FG S071S, 1/4 hp, 208-230V, 1.4 Amps 1100 RPM, 60Hz	
Condenser Coil	Carrier 38BRB042300, Face Area - 22.4 sq. ft. 1 row, 25 fins per inch	Face Area - 23 sq. ft. 1 row, 25 fins per inch
EVAPORATOR UNIT		
Blower Motor	Dayton Wattrimmer (1)3M853, 1/3 hp, 115V	Combined Fan/Motor Efficiency = 0.30
Blower Blade	Carrier 58ZAV070-12 Centrifugal No Brand Name, Model LA22RA012B MP	
Indoor Furnace/Cabinet	Carrier Weathermaker 8000 Model 58ZAV070-12	
Evaporator Coil	Carrier CD3AA036 (Resco CD3BXA036000) Face Area: 4 sq. ft., 3 rows, 16 fins per inch	Face Area: 4 sq. ft. 3 rows, 16 fins per inch
REFRIGERANT LINES		
Liquid Line	3/8" copper	
Vapor Line	7/8" copper with 3/8" wall foam insulation	
Expansion Device	Sporlan adjustable TXV Model RIVE-3-GA	

EXPERIMENTAL SETUP

The experimental setup consisted of the psychrometric rooms, an indoor and outdoor unit, refrigerant lines, appropriate instrumentation and a data acquisition system. Instrumentation was installed for collecting pressure, temperature, and flow rate measurements for the refrigerant; temperature, humidity, dewpoint, and flow rate measurements for the air; and power measurements for the condenser unit, the compressor, and the evaporator blower.

Psychrometric Rooms

The A/C unit was tested in the two psychrometric rooms at the Energy Systems Lab at Texas A&M University Riverside Campus. These rooms provided a method for maintaining an "indoor" and "outdoor" room at a desired temperature and humidity. The psychrometric rooms were built in accordance with ASHRAE specifications and were designed for testing units with capacities up to 10 tons. In each of the psychrometric rooms, steam humidification valves, duct heaters, cooling coils, and dehumidification coils were mounted in overhead ductwork.

The control of the room temperatures was accomplished through the use of overhead air handler units containing chilled water coils and electric resistance heaters. The cooling coils were supplied with a water/ethylene glycol solution which was cooled using a 75 ton chiller. A 1000 gallon storage tank installed in the chilled water system provided additional thermal capacity and reduced chiller cycling. Heat in each room was provided to the air by four electric resistance duct heaters, each having a capacity of 12.0 KW.

Steam from an electric boiler was fed into the supply air to raise the humidity. Dehumidification coils in the supply duct removed moisture from the air and were used to lower the humidity when necessary. An electric desiccant de-humidifier was also used to achieve the low indoor room humidity specified for the PG&E conditions.

An electronic controller was used to maintain specified room conditions. It measured ambient air temperatures and humidities in each room, and provided output signals to control the cooling coil valves, the heater relays, and the steam valves. The controller was connected to a personal computer enabling the operator to display controller parameters and specify desired room setpoints. The temperatures in the rooms were maintained within ± 0.2^{o}F of the desired values.

The indoor room contained an Air Movement and Control Association (AMCA) 210 (1985) air flow chamber and an assist fan which pulled the desired flow rate of air across the evaporator coil. Four ASME nozzles of 8", 5", 5", and 3" could be used in any combination to provide air flow between 100 and 5000 cfm. A damper in the chamber allowed the adjustment of air flow through the system. Nominal air flow rates of 840 cfm and 1200 cfm were used in these tests. The flow rate varied by less than ±1% of reading for any given test.

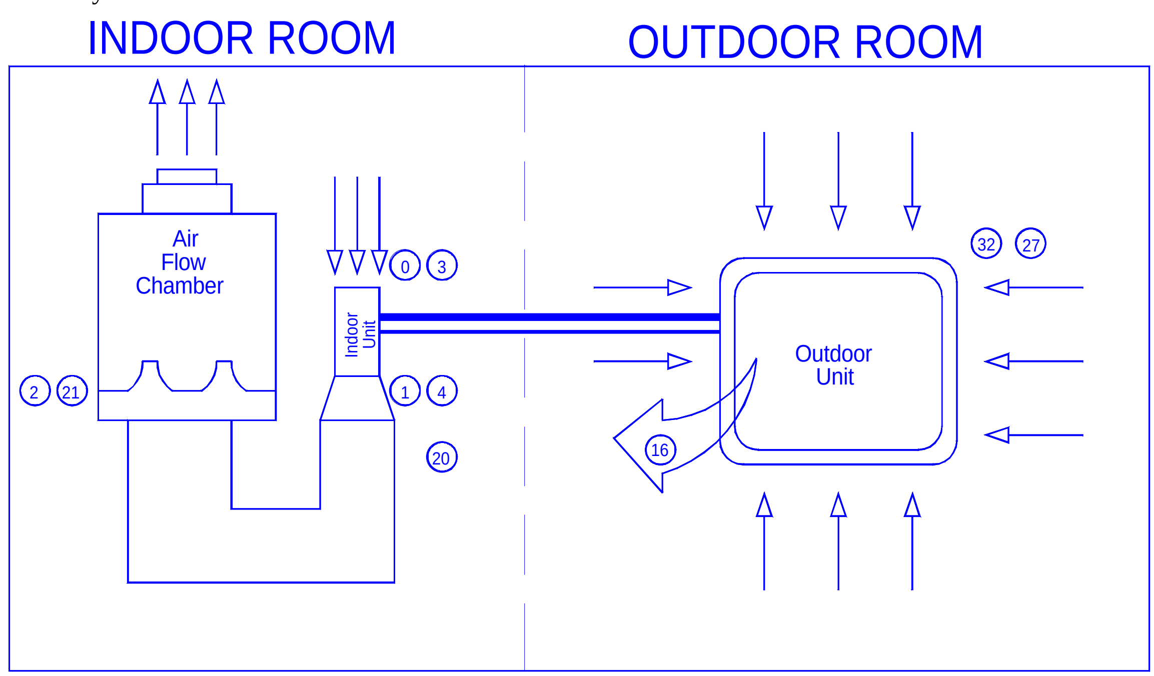
Instrumentation

Instrumentation was used on both the air-side and refrigerant-side of the system. The location of the air-side instrumentation is shown in 1.

An air sampling station was placed in the outdoor room near the condenser unit to measure outdoor room ambient temperature and relative humidity. Air was drawn from three sides of the unit to provide an average temperature of air crossing the condenser coils. A thermocouple grid was placed across the condenser unit outlet to measure the exiting air temperature.

Chilled mirror dew point sensors were used to measure the dew point temperature of the air before and after the indoor unit in the indoor room. Thermocouple grids were also placed before and after the unit for dry bulb temperature measurements. A differential pressure transducer was used to measure the pressure drop through a nozzle bank located in the Air Flow Chamber. Two 5 inch nozzles were used for all tests. A pressure transducer was also used to record the duct static pressure. A minimum static pressure of 0.15” H_20 is required on 3-ton units as specified in ARI Standard 210-B.

The air-side capacity is calculated using these measurements, and represents the net cooling effect of the air-conditioner system.



0 Dry Bulb Temperature Into Indoor Unit

3 Dew Point Temperature into Indoor Unit

1 Dry Bulb Temperature Exiting Indoor Unit

4 Dew Point Temperature Exiting Indoor Unit

20 Duct Static Pressure

21 Differential Pressure Across Nozzle Bank

2 Air Temperature At Entrance to Nozzle Bank

32 Dry Bulb Temperature Entering Outdoor Unit

27 Relative Humidity Entering Outdoor Unit

16 Dry Bulb Temperature Exiting Outdoor Unit

Figure 1. Air-Side Instrumentation

The refrigerant-side sensor arrangement is shown in Figure 2. Refrigerant pressure and temperatures were measured at the compressor inlet and outlet, and the indoor coil inlet and outlet. The refrigerant mass flow rate was metered in the liquid line. The flow meter could not function properly during two-phase conditions. All liquid line measurements were taken before the TXV expansion device.

Additional thermocouples were installed on the individual refrigerant circuits of the indoor coil. The indoor coil branched into six individual refrigerant circuits just after the expansion device. The thermocouples were surface mounted on the copper lines approximately midway through the coil circuits. Insulation was installed over the thermocouples to provide accurate readings.

The refrigerant-side capacity was calculated using the measured mass flow rate and calculated refrigerant enthalpies based on the indoor coil temperatures and pressures. A refrigerant-side capacity was not calculated when two-phase conditions were encountered.

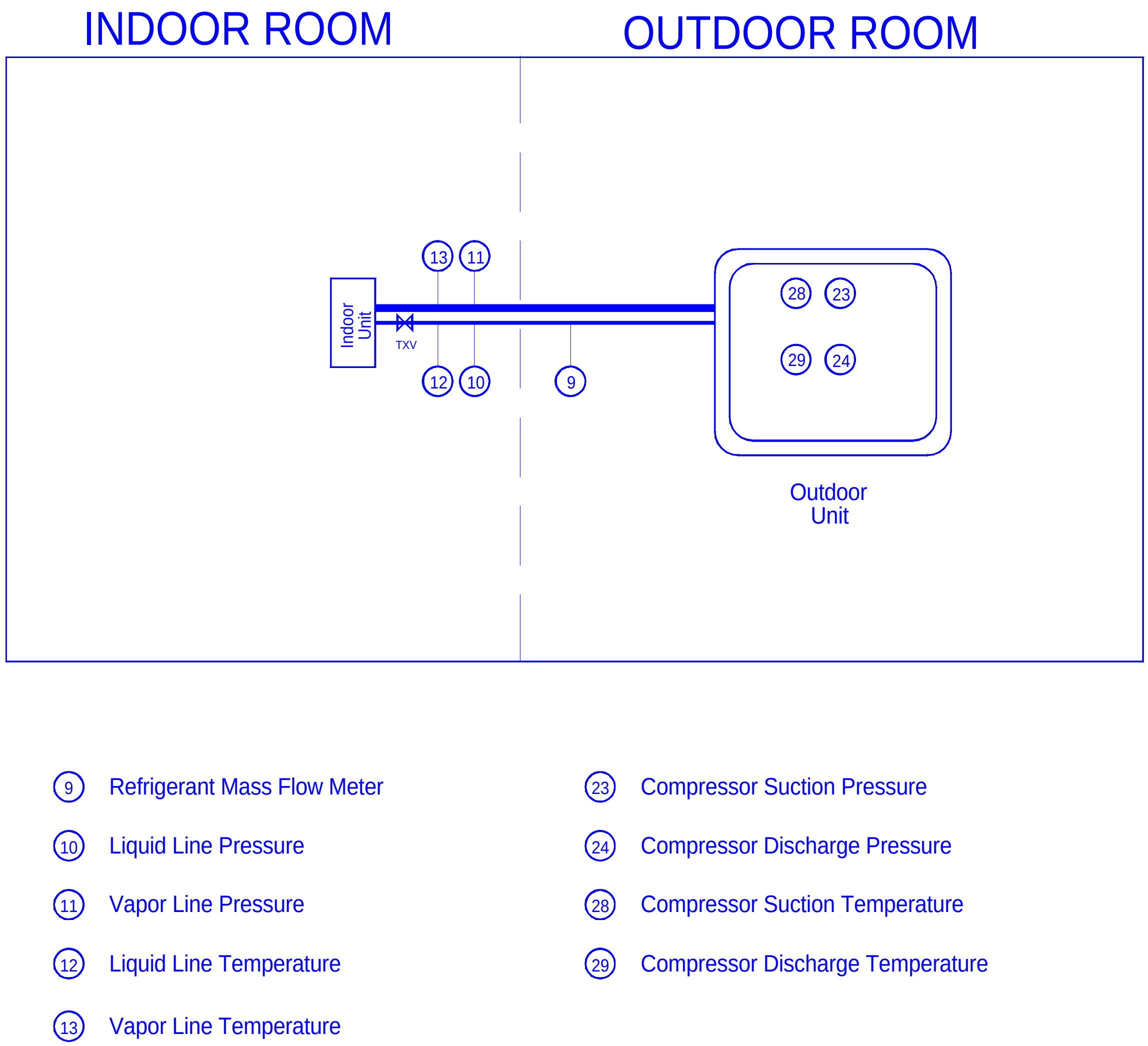


Figure 2. Refrigerant-Side Instrumentation

Power measurements were taken using electrical watt transducers. These instruments measured current draw and voltage, calculated true power, and supplied an output voltage signal to the data acquisition system. Power measurements were made for the compressor and outdoor fan, the compressor alone, and the indoor fan. The electrical metering instrumentation is shown in Figure 3.

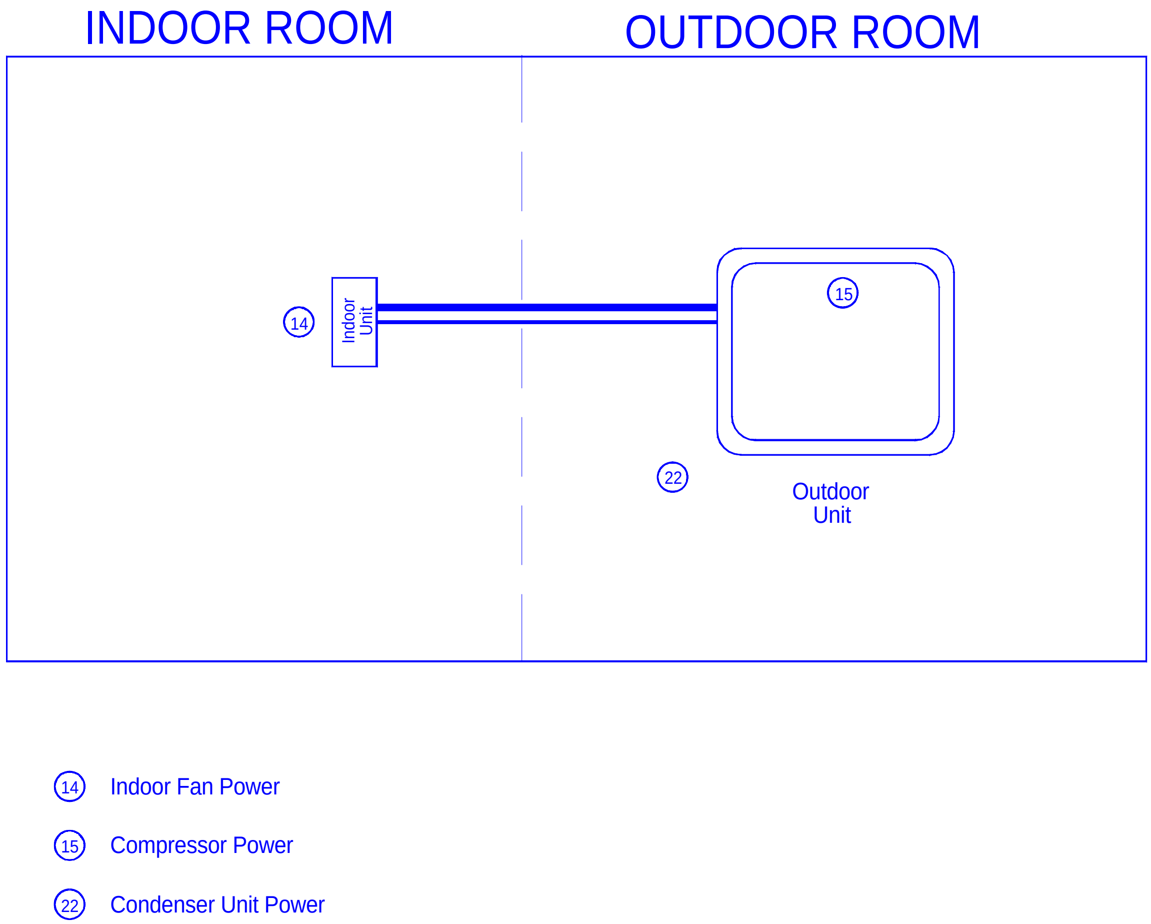


Figure 3. Electrical Metering Instrumentation

Data Acquisition System

The voltage signals from the sensors were collected using an electronic data logger. The input boards on the data logger accepted thermocouple inputs and voltage inputs from the sensors attached to the A/C unit. The data logger was connected to a personal computer where the signals were converted into engineering units and displayed in real-time on the screen. These values were also written to the computer hard drive at 15 second intervals.

The unit was allowed to run 15 minutes to reach steady state, then data were collected from each test for a minimum of 20 minutes. The data file was then processed using an analysis program developed by the Energy Systems Laboratory. The output from this analysis program was a summary report file, and a comma separated variable (CSV) file that can be imported into any modern spreadsheet program. The summary reports for each test are listed in Appendix B.

TEST CONDITIONS

In order to determine the prototype AC performance parameters for PG&E's service territory and to compare the prototype with currently manufactured units the tests were performed at the following conditions:

- Expected peak thermal conditions for the PG&E service territory: dry coil performance at 115 outdoor coil entering air and 80/61 indoor (Peak);
- Two sets of conditions which are representative of PG&E service territory climate zones

  Hot: 100 outdoor, 75/59 indoor (this is the same absolute moisture content as 80/61)

  Warm: 90 outdoor, 75/59 indoor
- One set of conditions for direct comparison to the EPRI sponsored laboratory tests conducted on representative air conditioners at high temperatures 115 outdoor, 80/67 indoor (EPRItest);
- ARI standard rating conditions: 95 outdoor, 80/67 indoor (ARI);
- DOE-B conditions used in the SEER rating calculation 82 outdoor, 80/67 indoor (DOE-B)

In order to determine the prototype AC performance under the installation and service conditions repeatedly found in the field, the equipment was tested under conditions of refrigerant charge and indoor coil air flow that span the range observed in field testing. Tests were performed under the following conditions:

- Two levels of air flow across the inside coil

  Standard, 400 cfm per ton

  Low, 280 cfm per ton [this is approximately one standard deviation below the mean (345 cfm per ton) measured air flow in field tests of new air conditioners];
- Three levels of refrigerant charge:

  Standard, 10 lb. of refrigerant (determined by testing to establish the maximum EER at ARI conditions)

  Undercharged, 30% less than the standard [this is approximately one standard deviation below the mean measured charge (84% of standard) in a field test of new air conditioners]

  Overcharged, 30% more than the standard.

In order to obtain an indication of the effect of the cabinet on the indoor fan and motor efficiency, one test was performed with the sides of the cabinet removed to allow free air flow into the fan.

Twenty-three performance tests are reported, each test consisting of a minimum of 20 minutes of steady-state operation. Test conditions for each of the tests are summarized in Table 3.

Table 3. Summary of Test Conditions

Name	Conditions	Refrigerant Charge (lb.)	Indoor Air Flow (cfm)
Test 1	ARI 95, 80/67	10.0	1199
Test 2	PG&E Peak 115, 80/61	13.0	1208
Test 3	PG&E Peak 115, 80/61	13.0	846
Test 4	PG&E Peak 115, 80/61	10.0	844
Test 5	PG&E Peak 115, 80/61	10.0	1207
Test 6	PG&E Peak 115, 80/61	7.0	1209
Test 7	PG&E Peak 115, 80/61	7.0	851
Test 7A	PG&E Peak 115, 80/61	9.0	1195
Test 7B	EPRItest 115, 80/67	10.0	1204
Test 8	PG&E Hot 100, 75/59	13.0	1201
Test 9	PG&E Hot 100, 75/59	13.0	836
Test 10	PG&E Hot 100, 75/59	10.0	834
Test 11	PG&E Hot 100, 75/59	10.0	1205
Test 12	PG&E Hot 100, 75/59	7.0	1204
Test 13	PG&E Hot 100, 75/59	7.0	845
Test 14D	PG&E Warm 90, 75/59	13.0	1211
Test 15D	PG&E Warm 90, 75/59	13.0	845
Test 16C	PG&E Warm 90, 75/59	10.0	846
Test 17C	PG&E Warm 90, 75/59	10.0	1204
Test 18C	PG&E Warm 90, 75/59	7.0	1207
Test 19C	PG&E Warm 90, 75/59	7.0	848
Test 1A-C	DOE-B 82, 80/67	10.0	1211
Cabinet Effect	ARI 95, 80/67	10.0	1209

III. RESULTS

PERFORMANCE OF THE PROTOTYPE UNIT

The performance (capacity, true power consumption, and efficiency) of the prototype unit was determined through the laboratory tests. These tests were completed under a range of climate, charge, and indoor coil air flow conditions.

Climate

The prototype unit was designed to be a nominal 3 ton air conditioner. The total net capacity of the prototype was 34,062 Btuh at ARI conditions (95 out, 80/67 in). The capacity of a unit drops as the outdoor temperature rises and as the indoor wet bulb temperature falls. Tested under PG&E peak conditions (115 out, 80/61 in) the total capacity was 27,883 Btuh with a sensible heat ratio of 0.98.

It is convenient to look at the capacities under any given set of conditions normalized to standard ARI conditions. The climate conditions of interest to PG&E are dry. This results in a lower capacity than ARI conditions (which is based on higher humidity). The normalized net capacities for the prototype under PG&E conditions are shown in Figure 4.

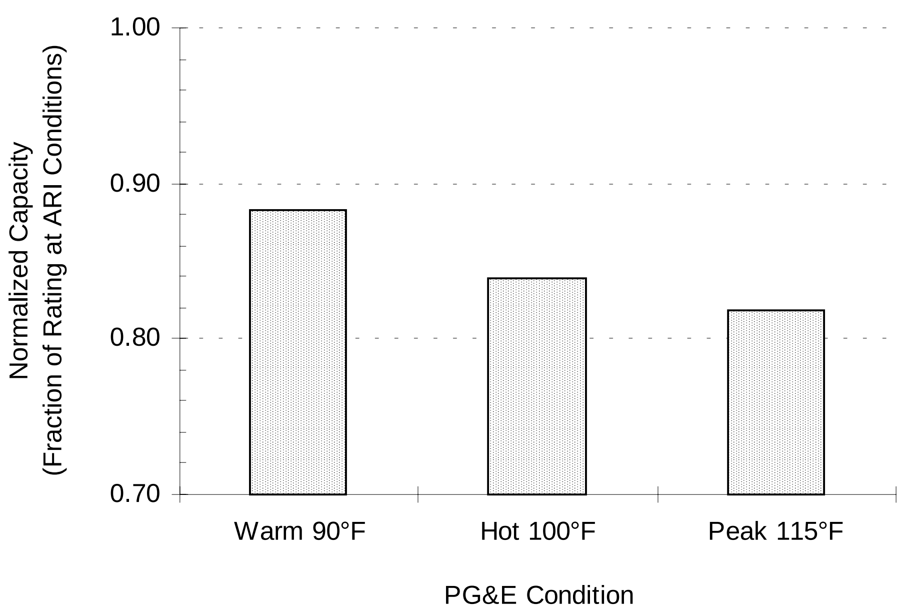


Figure 4. Climate Effect on Prototype Capacity

The primary focus of the prototype design is the steady state efficiency at PG&E Peak conditions. The Energy Efficiency Ratio (EER) was chosen as the primary analytical variable since it avoids confusion due to differences in capacity that are particular to each unit. Essentially, once it is established that a particular EER can be accomplished, units with a range of capacities can be built having that efficiency.

The EER of the prototype was 10.71 at ARI conditions and 7.86 at PG&E Peak conditions. The normalized EERs for the prototype under PG&E conditions are shown in Figure 5.

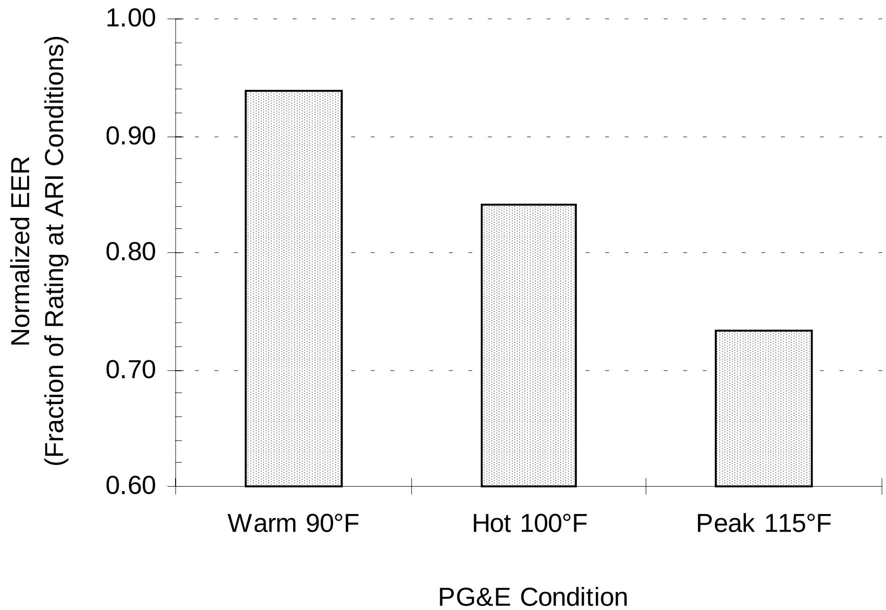


Figure 5. Climate Effect on Prototype EER

Charge

Undercharged and overcharged air conditioners are very common in the field. Various studies have shown that over half of the air conditioners are either overcharged or undercharged more than 5% (Blasnik et. Al., 1996). The prototype unit was tested with a TXV metering device since previous tests had proven that TXV units are influenced less by incorrect charge. The unit was tested under a variety of conditions (PG&E Peak, Hot, and Warm) with the refrigerant charge varied from 70% to 130% of standard charge. The effect of undercharge was more pronounced than the effect of overcharge on this unit. A 30% undercharge under peak conditions results in a capacity drop of 19.3% and a drop in EER of 17.1%. The effect of this drop in EER is an increase in diversified load by cycling air conditioners of 20%[8].

[8] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes incorrect charge even more detrimental than the drop in EER indicates.

The normalized capacities and EERs for the prototype under PG&E conditions are shown in Figures 6 and 7.

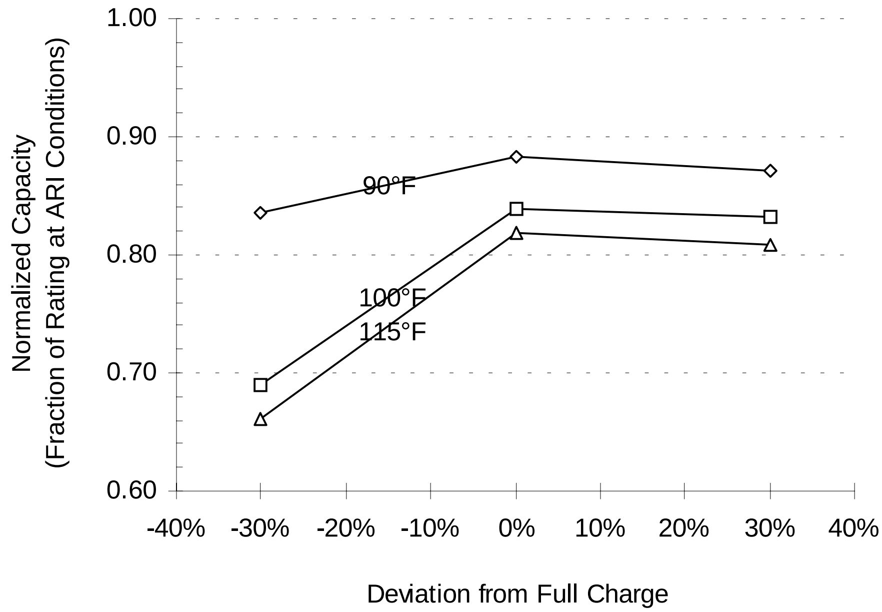


Figure 6. Charge Effect on Prototype Capacity

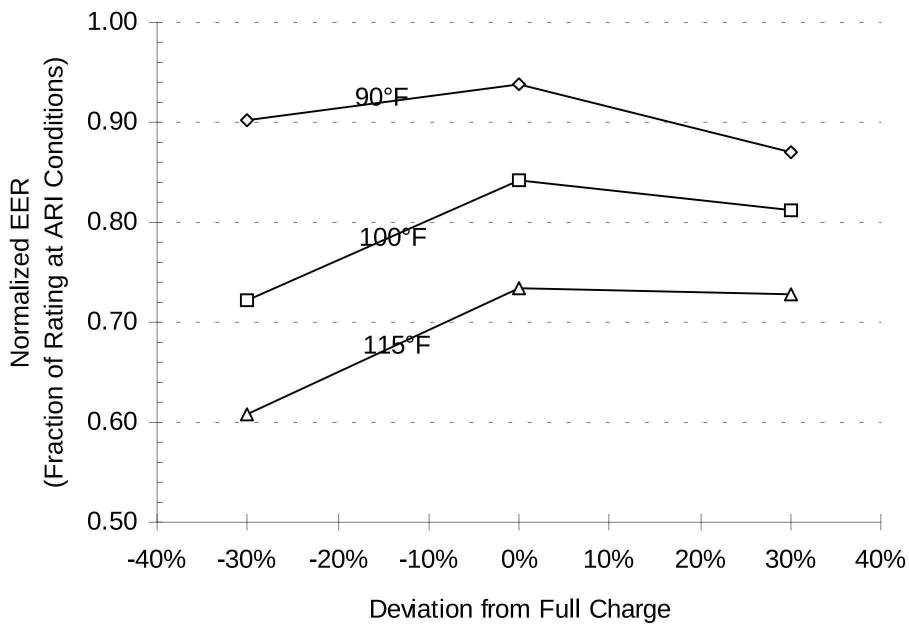


Figure 7. Charge Effect on Prototype EER

COMPARISON TO FIXED ORIFICE

Fixed orifice refrigerant metering is commonly found in field installations of minimum SEER air conditioners. Orifice metering is more sensitive to incorrect charge than TXV metering. The efficacy of a TXV metering device is confirmed by comparing the prototype's performance to fixed metering systems tested previously at Texas A&M (Farazad and O'Neal, 1993; Rodriguez et al., 1995). The prototype is far less sensitive to incorrect charge than the fixed metering systems as shown in Figure 8. At 70% of full charge the prototype retains 86% of its full charge efficiency, while the EPRI tested fixed metering unit's efficiency has dropped to 52% of its full charge efficiency. The prototype unit retains 86% of its correct charge efficiency even when it is overcharged by 30%.

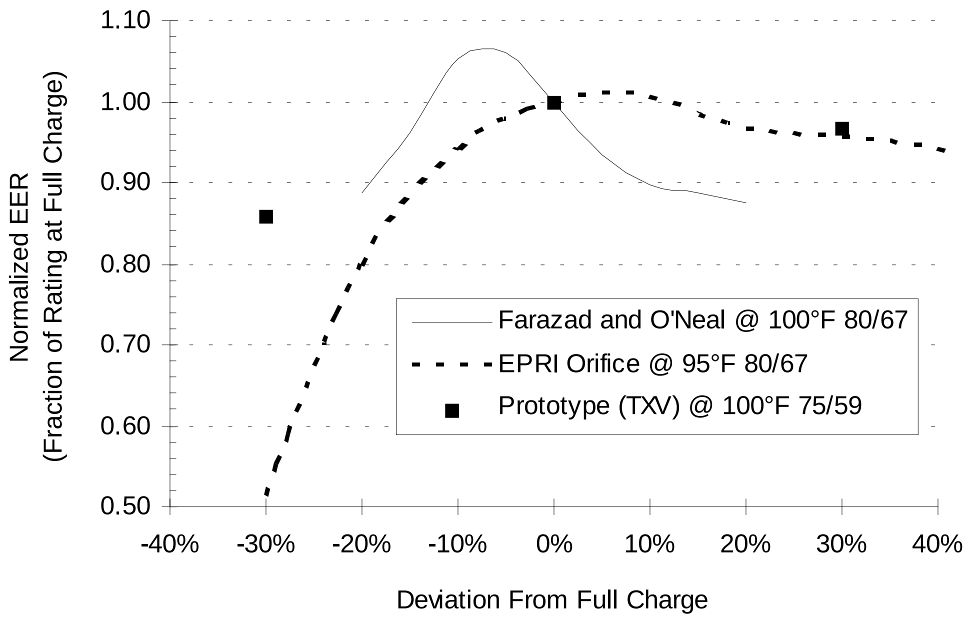


Figure 8. Charge Effect on EER - Prototype and Common Fixed Orifice

COMPARISON TO OTHER TXV

The effect of charge on the prototype's performance is similar to the effect of charge on the performance of previously tested TXV units (Rodriguez et al., 1995 and Farazad and O'Neal, 1993). This similarity is shown in Figure 9.

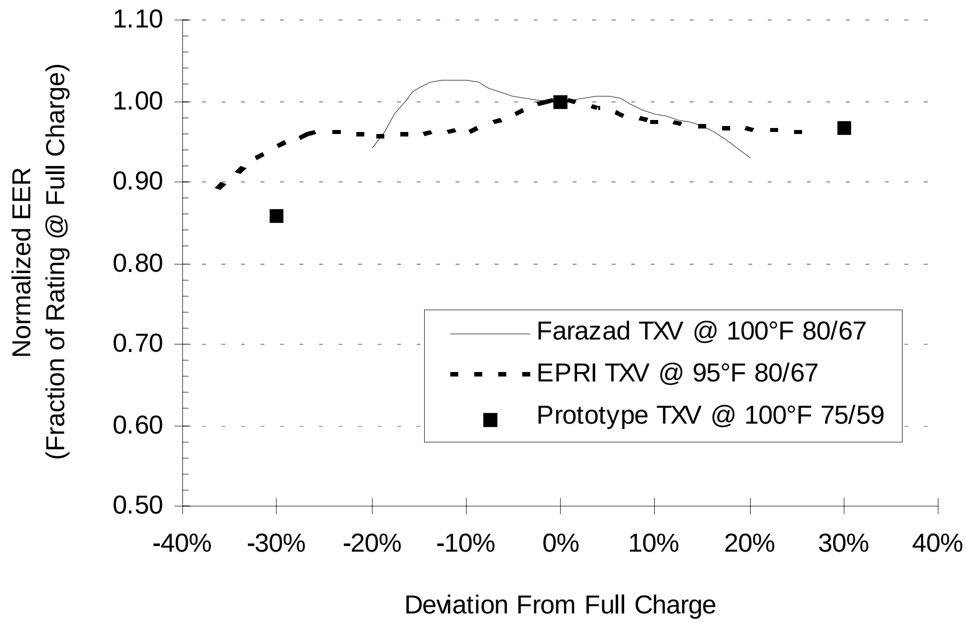


Figure 9. Charge Effect on EER - Prototype and Other TXV Units

Air Flow

Field studies of residential air conditioners have shown that low air flow across the inside coil is common (Proctor: 1991a, 1991b, 1991c). In one EPRI study (Blasnik et al., 1995) the air flow across the inside coil on newly constructed homes was 86% of standard (with a standard deviation of 20%). The prototype was tested under a variety of conditions (PG&E Peak, Hot, and Warm) with the indoor coil air flows 70% and 100% of standard. The relative effect of low air flow was more pronounced at peak conditions than at warm conditions. A 30% reduction in air flow under peak conditions results in a capacity drop of 12.8% and a drop in EER of 6.9%. The effect of the reduced EER[9] is an increase in diversified load of 7%[10].

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[9] for air conditioners that are cycling.

[10] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes low air flow even more detrimental than the drop in EER indicates.

The normalized capacities and EERs for the prototype under PG&E conditions are shown in Figures 10 and 11.

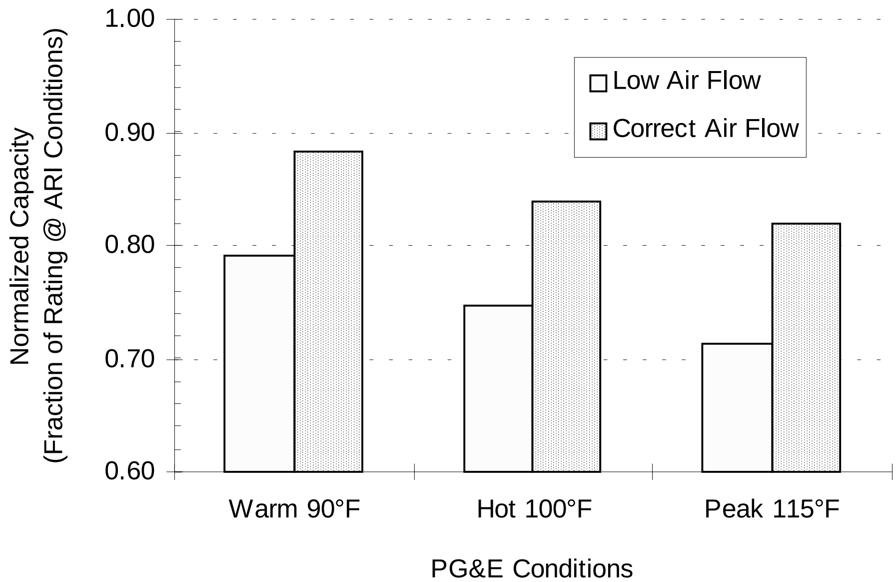

Figure 10. Indoor Coil Air Flow Effect on Prototype Capacity

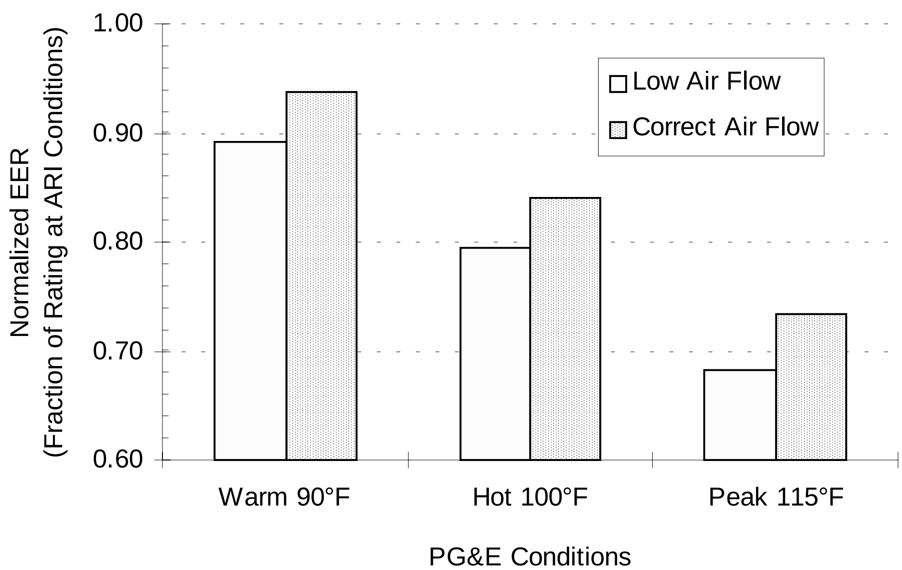

Figure 11. Indoor Coil Air Flow Effect on Prototype EER

COMPARISON TO OTHER UNITS

The sensitivity of the prototype to low air flow across the inside coil is better than that of a typical fixed orifice unit. The prototype unit is compared to two typical units (Rodriguez et al., 1995) in Figure 12.

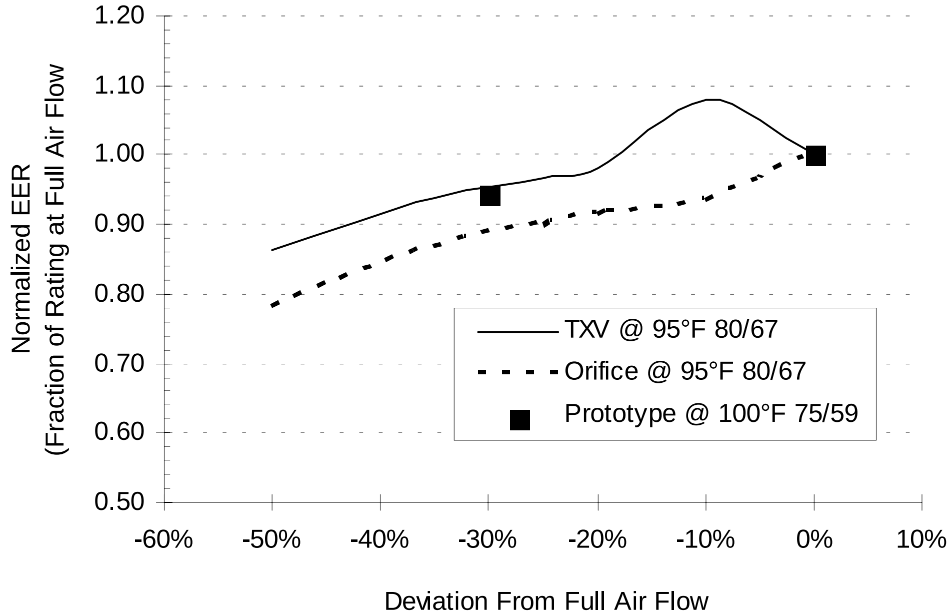

Figure 12. Indoor Coil Air Flow Effect on EER - Prototype and Other Units

Performance Summary

A summary of results from 23 tests is listed in Table 4. EER and air-side capacity are listed in addition to the conditions for each test. Detailed data sheets for each test are in Appendix B.

An energy balance between the air-side capacity and the refrigerant-side capacity was made to ensure that the measured data were accurate. The indoor fan power was added to the air-side capacity, and compared to the refrigerant side capacity. ARI requires an energy balance within 6% for a test to be considered accurate. The energy balance is shown for each test on the data summary sheets in the appendix.

The refrigerant-side capacity was not available on some tests due to two-phase refrigerant conditions at the evaporator sensors or the refrigerant mass flow meter. The refrigerant side capacity was not reported when the refrigerant was in the two-phase state for more than 80% of the test.

Table 4. Performance Test Results

Name	Conditions	Refrigerant Charge (lb.)	Indoor Air Flow (cfm)	EER	Capacity (Btu/hr)
Test 1	ARI 95, 80/67	10.0	1199	10.71	34062
Test 2	PG&E Peak 115, 80/61	13.0	1208	7.79	27560
Test 3	PG&E Peak 115, 80/61	13.0	846	7.27	24115
Test 4	PG&E Peak 115, 80/61	10.0	844	7.32	24308
Test 5	PG&E Peak 115, 80/61	10.0	1207	7.86	27883
Test 6	PG&E Peak 115, 80/61	7.0	1209	6.52	22503
Test 7	PG&E Peak 115, 80/61	7.0	851	6.38	21034
Test 7A	PG&E Peak 115, 80/61	9.0	1195	7.73	27269
Test 7B	EPRItest 115, 80/67	10.0	1204	8.54	30514
Test 8	PG&E Hot 100, 75/59	13.0	1201	8.70	28364
Test 9	PG&E Hot 100, 75/59	13.0	836	8.50	25589
Test 10	PG&E Hot 100, 75/59	10.0	834	8.50	25450
Test 11	PG&E Hot 100, 75/59	10.0	1205	9.01	28589
Test 12	PG&E Hot 100, 75/59	7.0	1204	7.73	23490
Test 13	PG&E Hot 100, 75/59	7.0	845	7.55	21935
Test 14D	PG&E Warm 90, 75/59	13.0	1211	9.32	29701
Test 15D	PG&E Warm 90, 75/59	13.0	845	9.05	27031
Test 16C	PG&E Warm 90, 75/59	10.0	846	9.56	26931
Test 17C	PG&E Warm 90, 75/59	10.0	1204	10.05	30060
Test 18C	PG&E Warm 90, 75/59	7.0	1207	9.65	28456
Test 19C	PG&E Warm 90, 75/59	7.0	848	9.29	25655
Test 1A-C	DOE-B 82, 80/67	10.0	1211	12.50	36807
Cabinet Effect	ARI 95, 80/67	10.0	1209	10.75	34832

COMPARISON TO TYPICAL SEER-10 AIR CONDITIONERS

A study supported by EPRI and a number of member utilities, Bain et al. (1995) tested ten air conditioners in the laboratory used in this study. Two units were selected by the participating utilities as representative of typical SEER-10 units. One of the units (designated B42SOR1H) was a nominal 3.5 ton split system with a reciprocating compressor and an orifice metering device. The indoor unit included an A-coil and a fan. This unit is referred to in this report as the “Typical Unit A”. The other typical SEER-10 unit (designated as G48SOR1C) was a nominal 4 ton split system with a reciprocating compressor and an orifice metering device. The indoor unit consisted of an A-coil evaporator with no air handler. This unit is referred to in this report as the “Typical Unit B”.

Previous PG&E / PEG studies (Proctor et.al., 1994; CADMAC, 1996) have identified the most commonly installed residential air conditioners in PG&E’s service territory as reported by the distributors. The Carrier unit identified was 38CK036 with indoor section CC5A / CD5A / CD5BA036. This comparison unit is a nominal 3 ton split system with a reciprocating compressor and an orifice metering device. The indoor unit consists of an A-coil but no air handler (it, just like the prototype unit, is typically installed on a furnace). The SEER of this unit is 10. This unit is referred to in this report as the “Typical Unit C”. The capacity and efficiency estimates for Typical Unit C were derived from the manufacturer’s published data. Those data are computer projections at peak conditions based on tests at lower temperatures. The EPRI report (Bain et al., 1995) found that the capacity at 115°F outside averaged 4% less than the manufacturers’ listed capacity and the EER averaged 4.3% less than listings under the same conditions. These adjustments were made to the manufacturer’s listings for comparison to the Prototype.

Relative Efficiency and Connected Load Reduction

The EER and capacity of air conditioners drop as the outside temperature increases. Since the percentage drop in EER is greater than the drop in capacity, the connected load increases with increasing outdoor temperatures. The EPRI study found that split system efficiency dropped 1.12% for each °F increase in outdoor temperature. The Prototype showed less efficiency drop (0.96% per °F). The EERs of the typical units and the Prototype are compared in Figure 13.

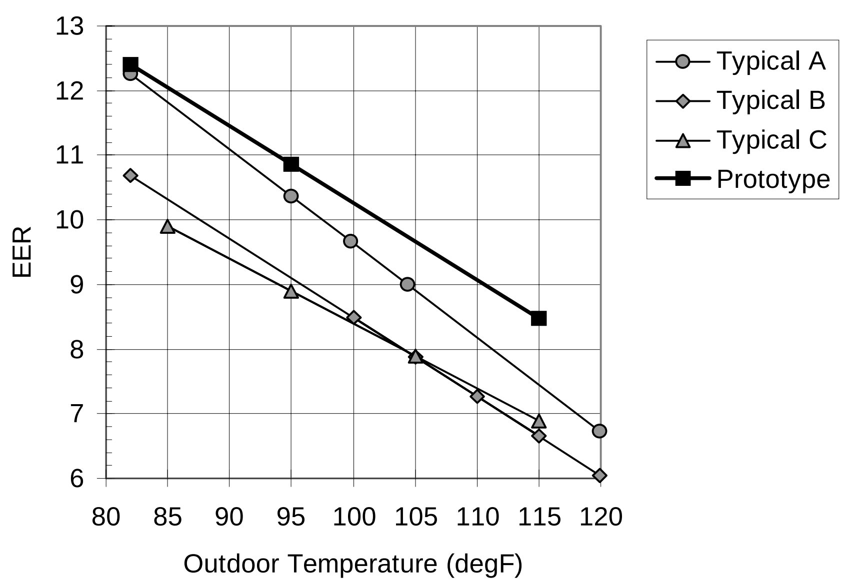

Figure 13. EER Comparison - Typical Units to Prototype (80/67°F)

The EERs, connected loads[11], and connected load reductions are summarized in Table 5.

Table 5. Comparison of Prototype EER and Connected Load to Typical Units (80/67 Inside)						
	EER 82	**EER 95**	**EER 115**	**EER drop (% drop/°F)**	**Connected Load 115 (KW)**	**Connected Load Reduction (KW)**
Prototype	12.50	10.71	8.54	0.96	3.51	
Typical Unit A	12.26	10.37	7.45	1.19	4.03	0.52
Typical Unit B	10.69	9.10	6.66	1.34	4.51	1.00
Typical Unit C	10.20	8.90	6.88	1.02	4.35	0.84

The Prototype unit had an EER of 8.5 at 115°F outside, while the typical units EER ranged from 6.7 to 7.4. If units of these designs were built to provide an identical 30,000 Btuh, the power draw of typical units would range from 4.0 to 4.5 KW. The Prototype would have a power draw of 3.5 KW.

At 115°F, the Prototype would have a connected load 500 to 1000 watts less than the typical units[12].

COMPARISON TO GOAL

The purpose of this prototype is to prove the concept that low peak power air conditioners can be built with common “off the shelf” parts. The goal of this project has been the “proof of concept” for a diversified peak reduction of 500 watts compared to a common SEER-10 unit.

While it is informative to investigate the performance of a single air conditioner, the primary item of importance to the utility is what the population of air conditioners is doing at peak. To project the effect of the Prototype on a diversified (population wide) basis a peak model must be applied that accounts for units that will not be running, units that will be running continuously, and others that will be cycling. Proctor Engineering Group has developed Model P, a dynamic model based on submetered AC data. Model P takes into account:

- The condition of the unit
- The demographics of the population
- The size of the unit compared to the house cooling load
- The control mechanism applied
- The time of day

[11] Based on a unit of this design delivering 30,000 Btuh capacity at 115°F.

[12] The connected load for a typical air conditioner is identical within measurement error between 80/67,115 and 80/61,115 (see Tests 5 and 7A). For this reason there is no distinction made in connected load between ARI indoor conditions (80/67) and PG&E indoor conditions (80/61). The summer weather conditions in the California Central Valley are hot and dry. This results in a low moisture content in the air inside the house and typical conditions much drier than the ARI test conditions.

Four customer classes have been identified and subclasses established. With this model, peak reduction is estimated based on customer / home class, subclass, and proposed changes in design, thermal distribution, or heat gain. For the California Central Valley population, for the design changes in the Prototype, with no change in size relative to load, the multiplier to account for diversity at system peak was 0.46 and for local peak 0.65[13].

The resultant peak reduction due to the Prototype design is detailed in Table 6. The PG&E Research and Development Report 008.1-94.2 concluded: "It is possible to build a 3-ton residential air conditioner with existing technology that will have a diversified local peak of 0.4 to 0.5 KW less than existing SEER 10 units." The laboratory tests completed in this project confirm that conclusion. A local diversified peak reduction of 0.34 to 0.65 KW is expected under typical PG&E peak conditions. If the prototype unit were used as a replacement for the common air conditioner used in PG&E's service territory[14], the result would be a diversified local peak reduction of 550 watts.

Table 6. Peak Reduction Due to Prototype Design

(115°F Outside 80°F / 61°F Inside)

	Connected Load	Connected Load Difference	Local Peak Reduction	System Peak Reduction
Prototype	3.51		0	0
Typical Unit A	4.03	0.52	**0.34**	0.24
Typical Unit B	4.51	1.00	**0.65**	0.46
Typical Unit C	4.35	0.84	**0.55**	0.39

[13] This application of Model P has been developed over the last six years for PG&E. The primary source of submetered air conditionser data used to develop this application was from Fresno, California.

[14] As characterized by Carrier model 38CK036 (CC5A / CD5A / CD5BA036)

COMPARISON TO SIMULATION

The simulation projected an EER 115 of 9.15 for "Design C". The Prototype built to approximate Design C had an actual EER 115 of 8.54. A number of items contribute to this difference. The first is the difference between a simulation and a piece of hardware. The typical units also did not perform as well as the simulation projections. Three components of the Prototype did not perform up to their published values, the compressor, the inside coil, and the inside fan and motor.

Relative Change in EER Maintained Between Simulation and Actual Equipment

The projected peak reduction comes from an increase in efficiency at high temperatures. While the simulation overpredicted the EER of both the typical and the prototype units, the percentage change in EER was closely approximated by the simulation. This data is displayed in Table 7.

Table 7. Relative Change in EER, Comparison of Simulation to Hardware

	Prototype EER 115	Typical EER 115	Prototype Improvement over Typical Unit
Simulation	9.15	7.70	19%
Hardware	8.54[15]	7.00[16]	22%

Hardware Problems

The original smaller compressor purchased for installation in this unit was more than 5% low on capacity and EER. A replacement compressor was used in all the tests reported. This compressor still did not perform up to manufacturer's specifications, but was within 5% of those specifications.

The inside coil is listed as having a bypass factor of 0.13 at 1200 cfm. The actual bypass factor for the coil was over 0.30 in tests at 1200 cfm[17]. Diagnostics were completed to determine the cause of the high measured bypass factor. Static pressure drop across the coil was measured at 0.24 inches of water column at 1200 cfm. This measurement exactly matched the manufacturer's specification, indicating that there was not a "shortcut" around the coil. Temperature probes were added to the coil circuits to detect any problem with blockage in any circuit. The resultant data however indicated that all circuits were working properly. The cause of the higher bypass factor was not determined.

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[15] Test 7B

[16] Average for Typical units A, B, and C.

[17] Bypass factor is a measure of coil effectiveness

Design C used in the simulation had a indoor fan / motor assembly with a combined efficiency of 0.30, however the actual efficiency of the indoor fan and motor assembly was 0.09 at a flow of 1200 cfm. In the cabinet effect test, removal of the side panels increased the efficiency to 0.15, a significant improvement, but far short of the 0.30 targeted. More study is needed of improved fan / motor / cabinet combinations.

Compressor, coil, and fan / motor performance up to manufacturers' specifications would improve EER and reduce power draw beyond that achieved in these tests.

CABINET EFFECT TEST

In addition to the capacity and EER ratings, the static pressure of the indoor fan was measured during several tests using an inclined manometer. The static pressure was measured between the indoor blower unit and the evaporator coil and was measured at nominal air flow rates of 840 and 1200 cfm. The cabinet sides of the indoor blower unit were cut and removed for the Cabinet Effect Test. The fan static pressure was recorded for this configuration at a nominal air flow rate of 1200 cfm. The measured flows, fan static pressures, and fan efficiencies are listed in Table 8.

The indoor fan / motor efficiency as installed was 9% compared to the 30% goal established in the simulation. Simple elimination of the restriction caused by the close proximity of cabinet walls increased the efficiency to 15%. Significant gains in indoor fan efficiency are likely with changes in cabinet configuration.

Table 8. Fan/Motor Efficiencies

	Air Flow Rate (cfm)	Fan Static Pressure (in H_2O)	Indoor Fan Power (watts)	Fan/Motor Efficiency
Test 14	1201	0.31	477	**9%**
Test 15	840	0.73	356	20%
Cabinet Effect	1209	0.54	507	**15%**

Cabinet size and design effect the efficiency of the unit in at least two ways. Small cabinets make the use of larger more effective evaporator coils difficult. They also limit the size, inlet and outlet conditions of the blower. As the size of the evaporator coil is constrained the efficiency of the unit is also constrained. Higher blower efficiencies are possible with increased blower size. Attention to cabinet design and return ducting configuration could improve the inlet and outlet conditions of the blower and overall fan efficiency.

CUSTOMER BENEFIT

The residential customer would benefit from air conditioners designed similar to the Prototype. Such air conditioners would consume 11% to 20% less energy annually than the SEER 10 baseline units. The Prototype unit holds its energy efficiency at higher temperatures where many high SEER units have a reduced efficiency edge.

IV. CONCLUSIONS AND RECOMMENDATIONS

This study tested a prototype residential air conditioner designed to reduce power consumption under peak conditions. Conclusions and recommendations are discussed in the various sections of the report. Below is a summary of these conclusions and recommendations.

CONCLUSIONS

1) The laboratory tests completed in this project confirm that it is possible to build reduced peak KW residential air conditioners with "off the shelf" components. A 3-ton residential air conditioner of the reduced peak design would have a diversified local peak of 0.34 to 0.65 KW less than existing SEER 10 units. If the prototype unit were used as a replacement for the common air conditioner used in PG&E's service territory[18], the result would be a diversified local peak reduction of 550 watts.

2) The projected peak reduction comes from higher efficiency at high temperatures. While the simulation overpredicted the EER of the typical and the prototype units, the percentage change in EER was closely approximated by the simulation. The MODCON (PUREZ) simulation is a helpful tool in screening air conditioner designs for efficiency differences at peak conditions.

3) The residential customer would benefit from air conditioners designed similar to the Prototype. Such air conditioners would consume 11% to 20% less energy annually than the SEER 10 baseline units. The Prototype unit holds its energy efficiency at higher temperatures where many high SEER units loose their efficiency edge.

4) Incorrect refrigerant charge seriously affects the efficiency of both the Prototype and typical air conditioners. Units with TXV metering devices (as used in the Prototype) are less effected than units with fixed metering devices. With a fixed metering device, the EER of the air conditioner will drop by approximately 48% for as little as 30% undercharge. The effect of a 48% drop in efficiency would be an almost doubling in diversified power draw if the unit could still cycle. The Prototype unit's efficiency would drop only 14% under the same 30% undercharge.[19].

5) The Prototype was tested under a variety of conditions (PG&E Peak, Hot, and Warm) with the indoor coil air flows 70% and 100% of standard. The relative effect of low air flow was more pronounced at peak conditions than at warm conditions. A 30% reduction in air flow under peak conditions results in a capacity drop of 12.8% and a drop in EER of 6.9%. The effect of the reduced EER[20] is an increase in diversified load of 7%[21].

6) The indoor fan / motor efficiency as installed was 9% compared to the goal of 30%. Elimination of the restriction caused by the close proximity of cabinet walls increased the efficiency to 15%. Significant gains in indoor fan efficiency are likely with changes in cabinet configuration.

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[18] As characterized by Carrier model 38CK036 (CC5A / CD5A / CD5BA036)

[19] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes incorrect charge even more detrimental than the drop in EER indicates.

[20] for air conditioners that are cycling.

[21] Excluding interactions with the distribution system. Distribution system losses generally increase as the capacity of the system drops. This effect makes low air flow even more detrimental than the drop in EER indicates.

RECOMMENDATIONS

Based on the results of this study, Proctor Engineering Group recommends:

1) Limited field testing of equipment with the characteristics demonstrated in the prototype design.
2) Investigation and implementation of approaches encouraging the introduction and sale of air conditioners with the characteristics demonstrated in the prototype design
3) Lab testing of fan / motor / cabinet modifications to improve efficiency. This could result in substantial efficiency gains at a low cost. Some changes in fan / motor / cabinet design can be accomplished without increasing the cabinet size, other changes might involve market transformation to overcome resistance.
4) Consideration of a low latent capacity air conditioner design for the PG&E service area. An air conditioner designed with higher evaporator temperatures would have lower latent capacity and a lower kVA on peak. It is sometimes argued that the latent capacity is needed in PG&E's service area on some occasions.
5) Field monitoring to determine sensible and latent load in PG&E's service area. This monitoring would answer two questions. First, the amount of load overestimation (and subsequent oversizing of air conditioners) present in Manual J and, second, the actual latent loads in PG&E's service area. Reduction in air conditioner size and air conditioners designed to higher evaporator temperatures would both reduce peak kVA.

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APPENDIX B INDIVIDUAL TEST SUMMARIES

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