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Prepared by:
Proctor Engineering Group, Ltd.
San Rafael, CA 94901
(415) 451-2480

Investigation of Peak Electric Load Impacts Of High SEER Residential HV AC Units

Prepared for:
Pacific Gas & Electric Company
Department of Research and Development
3400 Crow Canyon Road
San Ramon, California 94583

Final Report
May 1994

Contributors:
John Proctor, P.E.
Zinoviy Katsnelson
George Peterson, P.E.
Ann Edminster

Creators of CheckMe!®

93.101

PG&E Research and Development

Investigation Of Peak Electric Load Impacts of High SEER Residential HVAC Units

Report 008.1-94.2

Project Manager:

Ash Agboatwala, P.E.
Senior Research Associate

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Prepared by:
Proctor Engineering Group

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ABSTRACT

The objective of this study was to collect data on and analyze the performance of presently manufactured air conditioners operating at high ambient temperatures. The design parameters that affect power draw under these conditions were investigated. Four modified air conditioner designs capable of reducing peak draw by at least 500 watts were created and tested with a computer simulation. Major air conditioner manufacturers were contacted regarding the proposed modifications. The manufacturers indicated that in principal it was possible to build units that meet the design criteria. By providing a better understanding of air conditioner performance on peak, this study will help PG&E ensure the effectiveness of residential air conditioner peak load reduction programs. This study found that SEER rating is not an accurate predictor of peak kW or kVA.

EXECUTIVE SUMMARY

In 1993, Pacific Gas and Electric Company Research and Development undertook the study entitled, "Investigation of Peak Electric Load Impacts of High SEER Residential HVAC Units". Proctor Engineering Group performed the investigation which had three primary goals:

- To determine the peak kW and kVA characteristics of standard and high efficiency air conditioners.
- To investigate the air conditioner design parameters that affect the performance at high outdoor ambient temperature.
- To determine potential future design changes in residential central air conditioners that can improve peak performance. The target improvement was a 500-watt peak draw reduction for a 3-ton residential unit operating at an outdoor ambient temperature of 115°F.

The study had four components:

- Manufacturers, distributors, utility (PG&E) program managers, and AC design experts were contacted to obtain information on market penetration and air conditioner performance. High volume models were analyzed for performance at high ambient temperatures and design features that effect performance at high temperatures.
- Industry experts were contacted to discuss the current air conditioner designs, possible future changes, their recommendations on how to lower kVA at high ambient temperatures, and cost.
- Proposed design changes were modeled with a computer simulation program. After over 300 simulations, four new air conditioner designs were modeled. The four new designs were based on existing technologies. No breakthrough technology is used in the designs.
- Experts from inside and outside the manufacturing companies were queried on the feasibility of the new designs and the cost estimates.

The investigation reached the following conclusions:

- Based on manufacturer supplied data on current high market penetration air conditioners, the SEER rating is not an accurate predictor of peak kW or kVA.
- Oversized units are likely to produce a higher kW and kVA than properly sized units (under a number of peak conditions this increase will be substantial).
- A number of techniques currently employed by the AC manufacturers to increase SEER are also effective at reducing peak draw. These include:

- Increased condenser area and efficiency
- Increased evaporator area and efficiency
- Improved condenser fan / motor efficiency (sometimes with increased air flow)
- Improved evaporator fan / motor efficiency

• A number of techniques currently employed by the AC manufacturers to increase SEER are ineffective at or detrimental to reducing peak draw. These include:

- Two speed compressor
- Variable speed compressor*
- Scroll compressor
- Evaporator fan time delay

* Current variable speed units have a low power factor and high current harmonic distortion.

• It is possible to build a 3-ton residential air conditioner with existing technology that will have a diversified local peak of .4 to .5 kW less than existing SEER 10 units. The reduced peak kW (RPK) unit would have a peak kW draw less than current SEER 12 designs. The incremental cost increase should price the RPK unit at or below the cost of current SEER 12 units. These designs would have a SEER between 12 and 14.

The incremental cost (at the contractor) of the RPK design would be:

- between $300 and $600 per local area peak kW (diversified)
- between $400 and $800 per system peak kW (diversified)

Based on this investigation, Proctor Engineering Group makes the following recommendations:

• PG&E enter into discussions with ARI on rating methods to assure peak reduction. For the utility, a direct rating of steady state kVA at a high temperature would be most effective. The cost effectiveness of one air conditioner over another is determined by both demand and energy considerations.

• Properly size new and replacement units to reduce peak kVA.

• Monitor the actual indoor conditions (temperature and humidity) of a sample of air conditioned residences in PG&E's service territory. This would substantially add to the current information and assist development of a low peak kVA unit. Little is known about the latent capacity needs in hot dry climates. Some effective design changes to reduce peak may result in reduced latent capacity. Whether a reduction in latent capacity would effect comfort or energy use in hot dry climates has not been tested.

- Through an alliance with the manufacturers, build and lab test the most promising RPK designs. Based on the results undertake a limited field test of the designs.
- Lab test alternative cabinet designs matched to more efficient blowers. This could result in substantial efficiency gains at a low cost.

I. INTRODUCTION

BACKGROUND

Thirty percent of PG&E’s residential customers have central air conditioners or heat pumps for cooling (over 3 times that of all evaporative cooling) (XENERGY 1990). Residential air conditioning systems (AC) produce relatively little utility revenue but they do produce high coincident peak load. They are a common target for utility Demand Side Management (DSM) programs under the presumption that the programs will achieve reduced customer energy bills and a proportional peak reduction for the utility. It cannot be assumed however that current air conditioner DSM programs capable of producing significant energy savings will show proportional (or even any) savings under peak conditions. Proctor Engineering GroupÕs simulations and field studies of residential air conditioning systems have shown that peak load impacts cannot be directly correlated with energy savings (Proctor 1993). Additionally, Pacific Gas and Electric Company’s (PG&E) analysis of manufacturersÕ data shows that high Seasonal Energy Efficiency Ratio (SEER) units may sometimes be ineffective in reducing peak. Residential AC DSM programs to reduce energy use can substantially degrade the load factor for electric utilities. This results in a requirement for increased fixed assets (generation, transmission, and distribution) 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.

GOALS

This study had three primary goals:

- To determine the peak kW and kVA characteristics of standard and high efficiency air conditioners.
- To investigate the air conditioner design parameters that affect the performance at high outdoor ambient temperature.
- To determine potential future changes in central Direct Expansion (DX) residential air conditioners that can improve peak performance. The target improvement was a 500-watt peak draw reduction for a 3-ton residential unit operating at 115°F outdoors.

APPROACH

Proctor Engineering Group carried out this project in four phases:

- analysis of manufacturer's data,
- expert interviews,
- analysis of potential changes, and
- a final feasibility check of four simulated designs and their costs.

In the first phase manufacturers, distributors, utility (PG&E) program managers, and AC design experts were contacted to obtain information on market penetration and air conditioner performance. Based on market penetration and variety, high volume air conditioners were selected and available performance data was gathered. Each high volume model was analyzed for its performance at high ambient temperatures and design features that effect performance.

The second phase of the project was built on the information gathered in phase one. Industry experts were contacted to discuss the current air conditioner designs, possible future changes, their recommendations on how to lower kVA at high ambient temperatures, and cost. These interviews opened new areas of investigation leading into phase three.

In the third phase, changes in air conditioner design were evaluated by various calculations and simulations. Most of the proposed design changes were modeled with a computer simulation program, Oak Ridge National Laboratory's MODCON. After over 300 simulations, four new air conditioner designs were developed and modeled. These designs use existing technology and meet the goal of a 500 watt reduction in peak.

Proctor Engineering Group (PEG) contacted a wide variety of experts in the fourth phase of the project. Experts from inside and outside the manufacturing companies were queried on the feasibility of the new designs, the cost estimates, and for other suggestions that they might have in producing a unit that would draw less on peak.

The results of this investigation are detailed in this report.

DEFINITIONS

Peak

Electrical Demand Peak (peak) is a concept that is widely discussed, often with substantially differing definitions. PEG has identified three different concepts of peak, system peak, local area peak, and regulatory peak. For this report the following definitions apply:

- System Peak - The highest one hour average power draw for the entire PG&E utility. This peak normally occurs in the early afternoon on a summer weekday.
- Local Area Peak - The highest one hour average power draw for a defined local area. The timing of this peak will be dependent on the makeup and climate of the local area. For a primarily residential area in a hot climate it is likely to occur in the early evening (5 pm to 8 pm) on a very hot weekday.
- Regulatory Peak - The average power draw over a period defined by regulation. For PG&E, the regulatory peak is defined as the average power draw over all the weekday hours between noon and 6 PM in the summer months.

In this report only system and local area peak are examined. These numbers should not be applied to regulatory peak.

Seasonal Energy Efficiency Ratio (SEER)

SEER is a method of rating the efficiency of residential air conditioners. For single speed units, it is based on three tests. These tests are:

- DOE “B” - A steady state test of efficiency with an outside temperature of 82°F, an inside temperature of 80°F, and 50% relative humidity.
- DOE “C” - A steady state test similar to DOE “B” with low indoor relative humidity.
- DOE “D” - A cycling test run under DOE “C” conditions with 6 minutes on and 24 minutes off.

These tests include all input energy (including indoor and outdoor fans) and total cooling capacity (including latent and sensible). The loss due to cycling is calculated from tests “C” and “D”, and reported in a Coefficient of Degradation (C_D). The results are used to calculate the SEER: (for single speed units)

$$SEER = \text{DOE “B” Efficiency} \times (1 - .5 \times C_D)$$

A default C_D of .25 can be used for any single speed unit. Using that figure the above equation evaluates to DOE "B" Efficiency x .875. The result of this definition of SEER is that a steady state test at an outdoor temperature of 82°F accounts for at least 87.5% of the rating.

Contractor Cost

Both component and whole air conditioner costs are the estimated costs to a HVAC contractor who buys 10 air conditioners or less at a time. The prices may vary depending on the annual sale volume from the distributor to the particular contractor. Most distributors also offer a volume discount for purchases bigger than 25 or 50 units at a time. These are based on quotes from wholesalers and equipment distributors in Northern California, as well as information gathered from manufacturers.

These prices are the result of not only material, manufacturing, transportation, and marketing costs, but also marketing considerations. It is conceivable that the "mature market" costs for some of these items may be lower.

II. PHASE ONE - ANALYSIS OF MANUFACTURERS' DATA

Analysis of peak performance was examined first with a simplified model of air conditioners running continuously under peak conditions. The analysis was expanded to encompass units that were operating in other modes with Model P, a proprietary model of Proctor Engineering Group. Analysis concentrated on air conditioners that had the highest sales volumes and their high efficiency analogs of the same capacity. These were nominal 3-ton split air conditioners manufactured by Carrier, Lennox, Rheem, Trane, and York. Features and typical cost to the contractor for the analyzed units are listed in Table 1.

Table 1. Sample Cost and Design Features for Units with High Market Share (at various efficiency levels)

Unit	Design Features	Cost (to contractor)
Manufacturer A		
10 SEER	Single speed reciprocating compressor 9 sq.ft. 1 row condenser with 25 fins per inch (FPI) Orifice flow control	$747[1]
12 SEER	Single speed scroll compressor 19 sq.ft. 1 row condenser with 25 FPI Orifice flow control	$1077[1]
16.6 SEER	Two speed reciprocating compressor 18 sq.ft. 2 row condenser with 20 FPI two speed ICM condenser fan motor 7.4 sq.ft. 3 row evaporator with 14 FPI multi-speed ICM evaporator fan motor Thermostatic Expansion Valve (TXV) flow control	$2431[2] ($1531[4]+$900[5])

Manufacturer B		
10 SEER	Single speed reciprocating compressor 12.22 sq.ft. 1 row condenser with 24 FPI 4.13 sq.ft. 3 row evaporator with 12 FPI Orifice flow control	$719[1]
12.3 SEER	Single speed reciprocating compressor 25.5 sq.ft. 1 row, condenser with 24 FPI 3.44 sq.ft. 3 row evaporator with 12 FPI multi-speed evaporator fan motor TXV flow control	$1058[1]
14 SEER	Single speed reciprocating compressor 24.08 sq.ft. 2 row condenser with 22 FPI dual-speed condenser fan motor 3.44 sq.ft. 3 row evaporator with 12 FPI multi-speed evaporator fan motor TXV flow control	$1400[1]
15.5 SEER	Variable speed reciprocating compressor 26.92 sq.ft. 1 row condenser with 20 FPI variable-speed condenser fan motor 6.19 sq.ft. 3 row evaporator with 12 FPI variable-speed ICM evaporator fan motor TXV flow control	$2800[3]
Manufacturer C		
10.65 SEER	Single speed reciprocating compressor 14.7 sq.ft. 1 row condenser with 20 FPI	$958[1]
12 SEER	Single speed Scroll compressor 15.9 sq.ft. outer coil / 5.5 sq.ft. inner coil 1.36 row condenser with 20 FPI	$1326[1]
13.3 SEER	Two-speed reciprocating compressor 18.2 sq.ft. outer coil / 13.1 sq.ft. inner coil 1.75 row condenser with 20 FPI Evap. fan motor is field set to one speed	$2128[1]
15.5 SEER	Two-speed reciprocating compressor 18.2 sq.ft. outer coil / 13.1 sq.ft. inner coil 1.75 row condenser with 20 FPI Evap. fan motor is switched to low speed when compressor runs on low speed	$2208[1]

Manufacturer D		
10.1 SEER	Single speed reciprocating compressor 11 sq.ft. 1 row condenser 3.8 sq.ft. 4 row evaporator with 13 FPI Orifice flow control	$1147[1]
12 SEER	Single speed reciprocating compressor 15.8 sq.ft. 1 row condenser 7.6 sq.ft. 6 slab evaporator with 18 FPI Orifice flow control	$1560[1]
13.1 SEER	Single speed scroll compressor 16 sq.ft. 2 row condenser 3.8 sq.ft. 4 row evaporator with 13 FPI TXV flow control	$2100[1]
Manufacturer E		
10.2 SEER	Single speed reciprocating compressor 14.1 sq.ft. 1 row condenser with 18 FPI Orifice flow control	$1085[1]
12 SEER	Single speed reciprocating compressor 23.5 sq.ft. 1 row condenser with 16 FPI Orifice flow control	$1422[1]

1. Condensing unit and indoor coil only
2. Condensing unit and fan coil
3. Condensing unit and furnace
4. Condensing unit
5. Fan Coil

Because the AC does not exist in isolation, the installation and maintenance of the unit, the structure it cools, and how it is controlled must be considered. Model P takes into account these environmental variables. Three items should be noted because of their particular significance. First, oversized air conditioners are likely to draw more kW at peak than properly sized units, second, overcharged units will draw more at peak, and third, faulty thermal distribution systems (ducts) have their most detrimental effect at peak temperatures.

This phase of the investigation addressed questions of peak performance of residential split system air conditioners including kVA, and SEER as a predictor of peak kVA.

PEAK PERFORMANCE - CAPACITY, EFFICIENCY, KW, AND KVA

Air conditioner performance at high ambient temperatures can be measured in a number of ways: total capacity is a measure of the total amount of energy removal (both sensible and latent cooling), kW is the total unit power (including compressor and all fans), and Energy Efficiency Ratio (EER) is a measure of how efficiently the power is converted into cooling capacity. The relationship between these three can be expressed as:

$$\text{EER (Btu / Wh)} = \frac{\text{Total Cooling Capacity (Btu / hr)}}{1000 \text{ x kW}}$$

Air conditioners with high EER produce the cooling capacity with less kW. A high efficiency air conditioner is an advantage to the customer, who will have lower cooling costs.

For customer comfort, the total capacity of the unit should meet the cooling load of the home under design conditions. By definition a “properly sized” air conditioner is somewhat smaller than the largest likely cooling load. Such a unit would run continuously during the hottest hours of the hottest day. An air conditioner with higher capacity will run shorter, less efficient, cycles. In moist climates (unlike PG&E’s service area in the California Central Valley) shorter cycles will lower the latent capacity of the unit.

On the utility side of the meter, kVA is the most important measure (kVA is equal to kW / Power Factor). For the utility, lower kVA will reduce the size of the generating plant and the T&D system. A high efficiency air conditioner is an advantage to the utility because it will draw less kVA for a given capacity.

An oversized air conditioner is a disadvantage to the utility for it will draw a higher kVA than a properly sized one.

For the initial simplified analysis two assumptions were made. First to eliminate the effects of differences in capacity, all units from a manufacturer were compared at the same capacity, that of the lowest capacity unit (LC). The Normalized kW draw is calculated by dividing LC by the EER of the unit being analyzed. The second assumption was that units were properly sized to run continuously at an outdoor temperature of 95°F.

Single-Speed Units

Presently manufactured single-speed 3-ton split units sold in California have a SEER range from 10 and 15.3. The highest SEER’s occur when the unit includes an air handler with an efficient Brushless Permanent Magnet variable speed blower.

Figure 1 shows the average hourly kW draw at various outdoor temperatures and indoor conditions of 80°F dry bulb and 67°F wet bulb with 1200 cfm supply air flow. Below 95°F the unit was assumed to have a duty cycle just sufficient to meet a cooling load that drops linearly with outdoor temperature. For this simplified analysis no cycling losses were calculated.

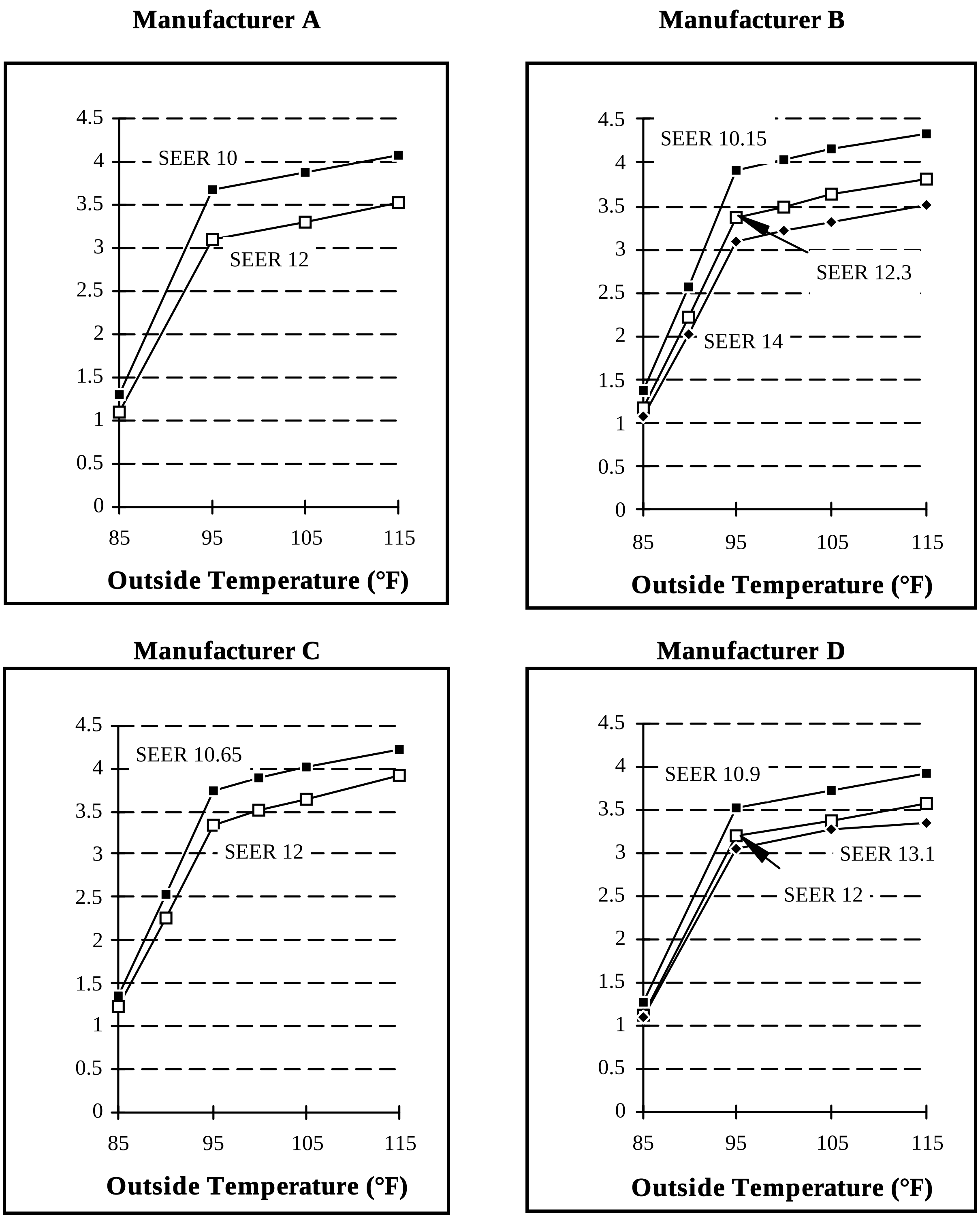

Figure 1. Normalized kW vs. Outdoor Temperature, Single Speed Units

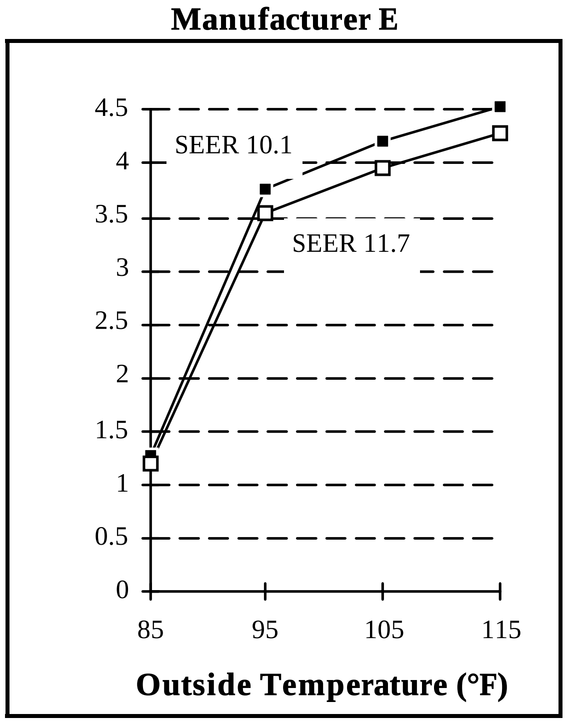

Figure 1 (continued) Normalized kW vs. Outdoor Temperature, Single Speed Units

Observations about Single Speed Units. Single speed units hold few surprises. For a given manufacturer, high SEER units have a higher EER at peak conditions. Each manufacturer has its own preferences in obtaining higher SEER. These methods include:

- increased condenser and evaporator performance due to the increase of the heat exchanger surface area and heat transfer coefficient
- utilizing a scroll compressor
- increased combined fan / motor efficiency
- changes in metering devices
- special control devices such as a time-delay relay on the indoor fan

The higher SEER units usually, but do not necessarily, draw less kW at peak. This potential discrepancy can occur for a number of reasons. First, design changes aimed a reducing cycling losses will not necessarily improve steady state efficiency. Second, changes that increase capacity at high temperatures will increase kW, such as use of a scroll compressor. Third, the actual capacity of nominal 3 ton units can be different. For example, the 3 ton SEER 10 unit of Manufacturer A has 33,800 Btuh design cooling capacity while the SEER 11.7 unit of the Manufacturer E has 37,000 Btuh. In this case the high SEER air conditioner draws 4.35 kW at 115°F and the SEER 10 unit draws 4.07 kW.

Two-Speed Units

Several manufacturers use a two-speed reciprocating compressor in their high efficiency units. Manufacturer A specifies the compressor for 3-ton units with SEERs ranging from 12.0 to 16.6, Manufacturer C for SEERs from 11.85 to 15.5. The compressor operates at the low speed under light and medium loads and at the high speed under heavy cooling loads. The highest SEER is achieved by combining the condensing unit with high efficiency fan coils or furnaces which have variable-speed motors and microprocessor control.

Figure 2 shows the normalized kW draw at various outdoor temperatures and standard ARI indoor conditions. Many of the two speed units are designed with somewhat higher capacity than single speed units, therefore it is important to observe the actual kW draw at high temperatures for these units. This is shown in Figure 3.

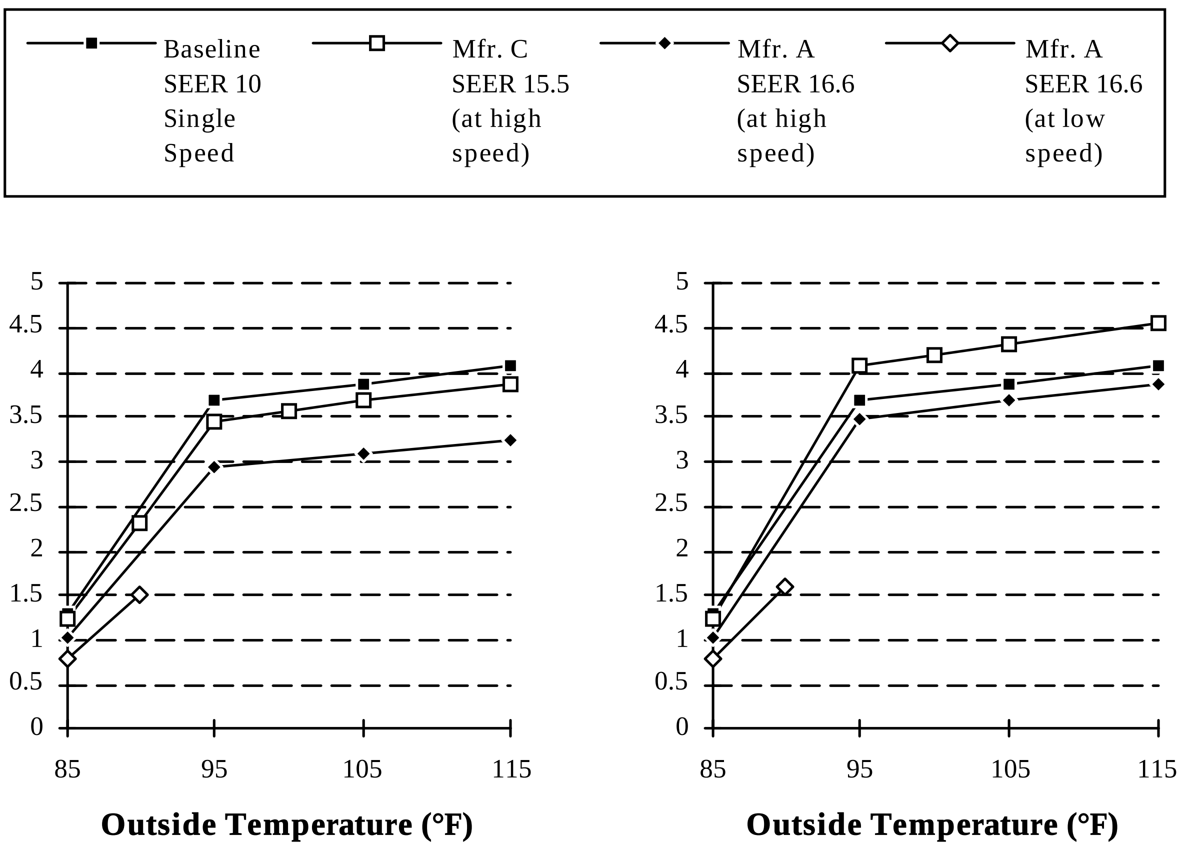

Figure 2. Normalized kW vs. Outdoor Temperature, Two Speed Units

Figure 3. Actual kW vs. Outdoor Temperature, Two Speed Units

Observations about Two Speed Units. At high speed (the most likely operating mode on peak) a two speed unit is not necessarily more efficient than a SEER-10 unit. As shown in Figure 2, the kW draw at 115°F (when normalized to the same capacity) for two of the units analyzed is within .2 kW of a single speed SEER-10 unit.

The improved SEER of these units comes basically from running the unit at low speed as much as possible. When a two speed unit operates at low speed two things happen that improve efficiency. First, the heat exchanger coils are effectively "oversized" for the amount of refrigerant flow. In this condition the evaporating temperature rises and the condensing temperature falls, improving the efficiency of the refrigerant cycle (dehumidification may be reduced). Second, at low speed the capacity is lowered, the unit runs longer, and cycling losses are reduced. The low speed kW line for Manufacturer A shows how effective low speed is in improving efficiency.

Figure 3 illustrates the importance of considering capacity (and proper sizing) when analyzing peak effects. In the case of a straight choice between the baseline unit and Manufacturer C's two speed unit, the baseline unit is a better choice based on continuous running kW. While Manufacturer A's two speed unit used .8 kW less when compared at the same capacity it uses only .2 kW less than the baseline unit at full capacity. While these units are all called 3-ton units, they are vastly different in capacity. Manufacturer A's unit produces a full half ton (6,000 Btu / hr) more capacity at the rating point than the baseline and Manufacturer C's unit produces 9,900 Btu / hr more.

The interaction with the thermal distribution system is more important for two speed units than it is for single speed units. A portion of the energy lost through duct leakage is proportional to:

$$\frac{\text{(Outside Temperature - House Temperature)}}{\text{(House Temperature - Delivery Air Temperature)}}$$

If a two speed unit is run on low speed without a similar reduction in indoor fan speed, the delivery air temperature will approach the house temperature. Since the divisor in the equation gets smaller, the duct losses increase and the seasonal efficiency drops.

Variable-Speed Units

Only one manufacturer presently offers a variable-speed, 3- ton residential split air conditioner. The SEER of this unit varies from 15.45 to 16.2 depending on the evaporator coil and fan used. Like a two speed unit, it runs more efficiently at lower speed. The primary advantage of this unit over the two speed units is its ability to more precisely match the speed to the demand for cooling. While a single speed unit would cycle on and off, and a two speed unit would switch back and forth between its two available speeds, the variable speed unit could closely match the speed with the cooling demand. The compressor and fans are driven by adjustable speed drives and DC motors. The drives are integrated into an electrical system that converts the AC line voltage to stable DC voltage, which is then switched to the motor at variable voltage and frequency.

Among the drawbacks of the technology are high initial cost, low power factor, and high harmonic distortion. The first generation units have passive control for power factor and harmonic distortion. These units have a true power factor up to .79 and a total current harmonic distortion of 54%. A second generation of variable-speed systems is now being prepared for the United States market. The electrical systems in these new air conditioners are expected to use active true power factor control technology to improve the power factor to .98 and total current harmonic distortion to 7% (Sulfstede 1992).

Figure 4 shows the kW draw at various outdoor temperatures and standard ARI indoor conditions on high operating speed. The performance of an SEER 10 unit from the same manufacturer is shown for comparison.

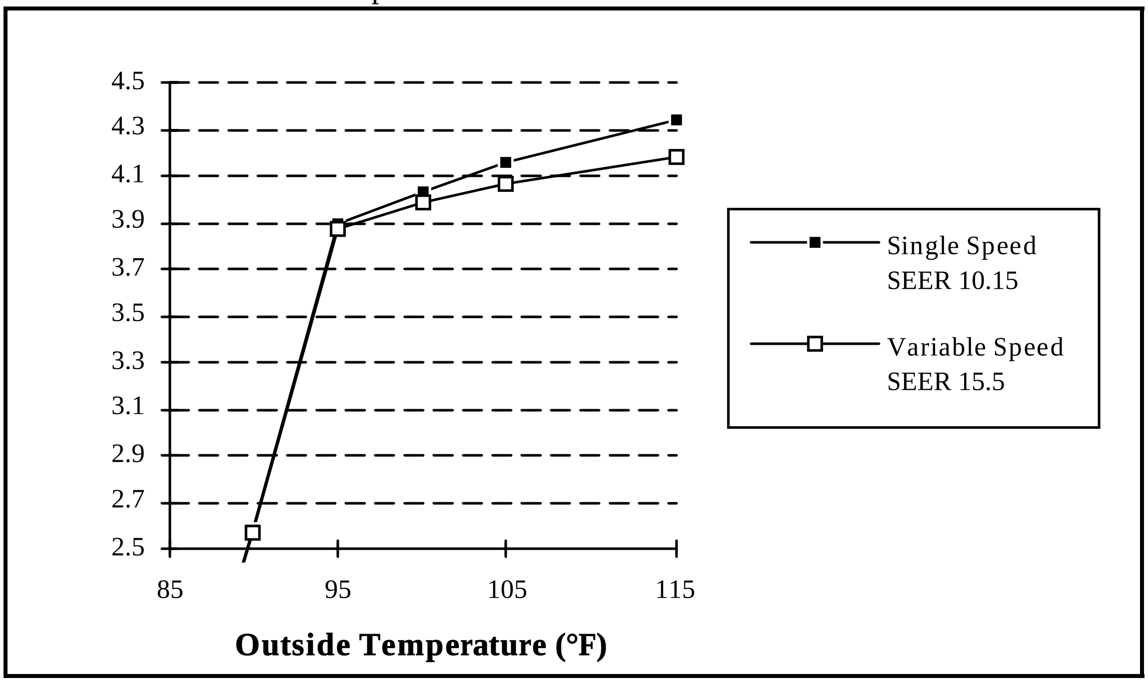

Figure 4. Normalized kW vs. Outdoor Temperature, Variable Speed Unit

Observations about Variable Speed Units. At high speed the kW draw of the variable speed air conditioner is almost indistinguishable from the SEER 10 unit. This means that no significant kW reduction can be expected, in fact with the current design (low true power factor) the kVA will exceed that of the SEER 10 air conditioner by 37%.

As with the two speed unit the installation of a variable speed compressor without a variable speed fan will increase the energy loss in the air distribution system.

DOES SEER RATING PREDICT PEAK KW DRAW?

As noted in the definitions section, SEER for a single speed unit is based on testing at an outside temperature of 82°F (multi speed units use a more complex bin method). Since the tests that establish SEER are run at temperatures substantially below peak temperatures, it is necessary to investigate the relevance of SEER to peak kW. Increased SEER has been a driver of the residential AC market since its introduction. It is one of the most marketable features of an air conditioner and AC manufacturers optimize the unit design for operation at 82°F.

Each year PG&E offers a rebate for air conditioners with upgraded SEER. In 1993, rebates were paid for 6092 high efficiency air conditioners in new construction (78 units with SEER's 15 and over) and 5915 in retrofit situations (57 in the 15 and over SEER category). Addressing PG&E's concern about energy consumption at peak conditions, the appropriateness of SEER for peak reduction prognoses was analyzed.

The actual (not normalized for capacity) performance data were compared with the typical peak kW reduction estimate represented by the ratio:

$$\frac{(SEER_2 - SEER_1)}{SEER_2}$$

Where:

$SEER_1$ = SEER of the base unit

$SEER_2$ = SEER of the proposed unit

Table 2 summarizes the results of the analysis. It shows the change in kW at 115°F outdoor temperature versus the reduction forecast from the unit's SEER.

As Table 2 shows, SEER ratings do not predict peak reduction. The most straightforward estimate of kW reductions would be the difference in kW at test conditions similar to peak (modified to reflect diversity). All the manufacturers we contacted made available information sufficient to determine kW and estimate kVA at 115°F. Recalling that kW and kVA are dependent both on the efficiency (EER) and the capacity of the unit, proper sizing of new and replacement air conditioners is critical to achieving peak reduction.

Table 2. Peak Change vs. Change in SEER		
SEER	**Change based on SEER**	**Actual Change in Peak kW** (continuous running)
Manufacturer A	(Base 10 SEER)	
12 16.6	-17% -40%	-5% -5%
Manufacturer B	(Base 10 SEER)	
12.3 14 15.5	-19% -28% -35%	-15% -21% -1%[1]
Manufacturer C	(Base 10.65 SEER)	
12 13.3 15.5	-11% -20% -31%	-4% -1% -8%
Manufacturer D	(Base 10.9 SEER)	
12 13.1	-9% -17%	-3% -8%
Manufacturer E	(Base 10.2 SEER)	
11.7 12	-13% -15%	-2% -2%

1. The Power Factor of this variable speed unit is approximately .7 compared to an average power factor for single and two speed compressors of .96. Installation of this unit instead of Manufacturer B's SEER 10 unit would result in an <u>increased</u> kVA on continuous running of 37%.

III. PHASES TWO AND THREE - POSSIBLE DESIGN CHANGES FOR PEAK REDUCTION

A number of potential design changes were identified based on the analysis in phase one and discussions with industry experts. These design changes were chosen to be within the current capability and state of the industry. This includes the eventual phase out of HCFC's (R-22 is a HCFC)[1]. A similar study of air conditioners and heat pumps was completed at Texas A&M University (O'Neal, Boecker, and Penson 1987). The Texas A&M study calculated the SEER for units of incrementally improved design. The Proctor Engineering Group study concentrated on the kW draw of a number of units at 115°F. The over 300 simulations included calculations of changes in each design parameter. Most of the proposed design changes were modeled with Oak Ridge National Laboratory's MODCON simulation. This simulation program is documented in Rice 1991 and Fischer, Rice, and Jackson 1988.

DESIGN PARAMETERS THAT EFFECT KW DRAW AT PEAK

In this investigation the following parameters were studied:

- Compressor type, efficiency, and size
- Evaporator face area, number of tube rows, fin density, fin type, number of circuits
- Condenser face area, number of tube rows, fin density, fin type, number of circuits
- Fan and motor efficiency
- Cabinet size and design
- Evaporator air flow and temperature
- Condenser air flow and temperature
- Metering device
- Subcooling

Each of these changes was investigated in the literature and many were simulated with MODCON.

Compressor Changes

Potential compressor changes evaluated include reduced size, two speed or variable speed compressors, and scroll compressors.

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[1] According to Title 6 of the Clean Air Act, HCFC production and consumption will be held to baseline levels beginning 2010 and will be eliminated from production and consumption beginning 2020. The application of replacement refrigerants will probably effect selection of lubricant and the flow pattern through the coils. Neither of these should substantially effect this analysis.

Compressor Resizing. When improvements in system efficiency (such as improving the heat transfer at the coils) occur the capacity of the air conditioner will increase. Since it was our goal to reduce peak kW for the same cooling capacity, any such capacity changes had to be adjusted by the use of a smaller compressor.

In the simulation, resizing the compressor sufficient to maintain the same capacity resulted in a normalized peak reduction of 146 watts.

Two Speed and Variable Speed Compressors. Two speed units were of no advantage since at peak conditions properly sized units would run at high speed. For example, the two speed Copeland CTH1-0275-CSV at high speed and its sister single speed unit CRH3-0275-PFV use 3410 watts and 3400 watts respectively under identical conditions.

Presently manufactured variable-speed compressors do not have an efficiency advantage at peak conditions either. For these reasons, only single speed units were analyzed in this portion of the study.

While two speed and variable speed compressors offered no intrinsic value as a source of peak reduction, they could be controlled to run at lower speed (lower capacity and lower kW) on either a temperature or utility controlled signal. This option was not considered since the scope of this study was limited to a 500 watt reduction without a reduction in comfort.

Scroll Compressors. Scroll compressors have many benefits, including higher efficiencies and higher capacities at high temperatures (as shown in Figure 5). At first glance, they seem to be suited to a peak reduction design. However the scroll compressor's increased capacity results in a higher actual kW even with the improved EER. If a scroll compressor AC were designed to the same capacity at 115°F as a reciprocating compressor AC, the scroll unit's capacity at the rating point of 95°F would be substantially lower than the reciprocating unit's.

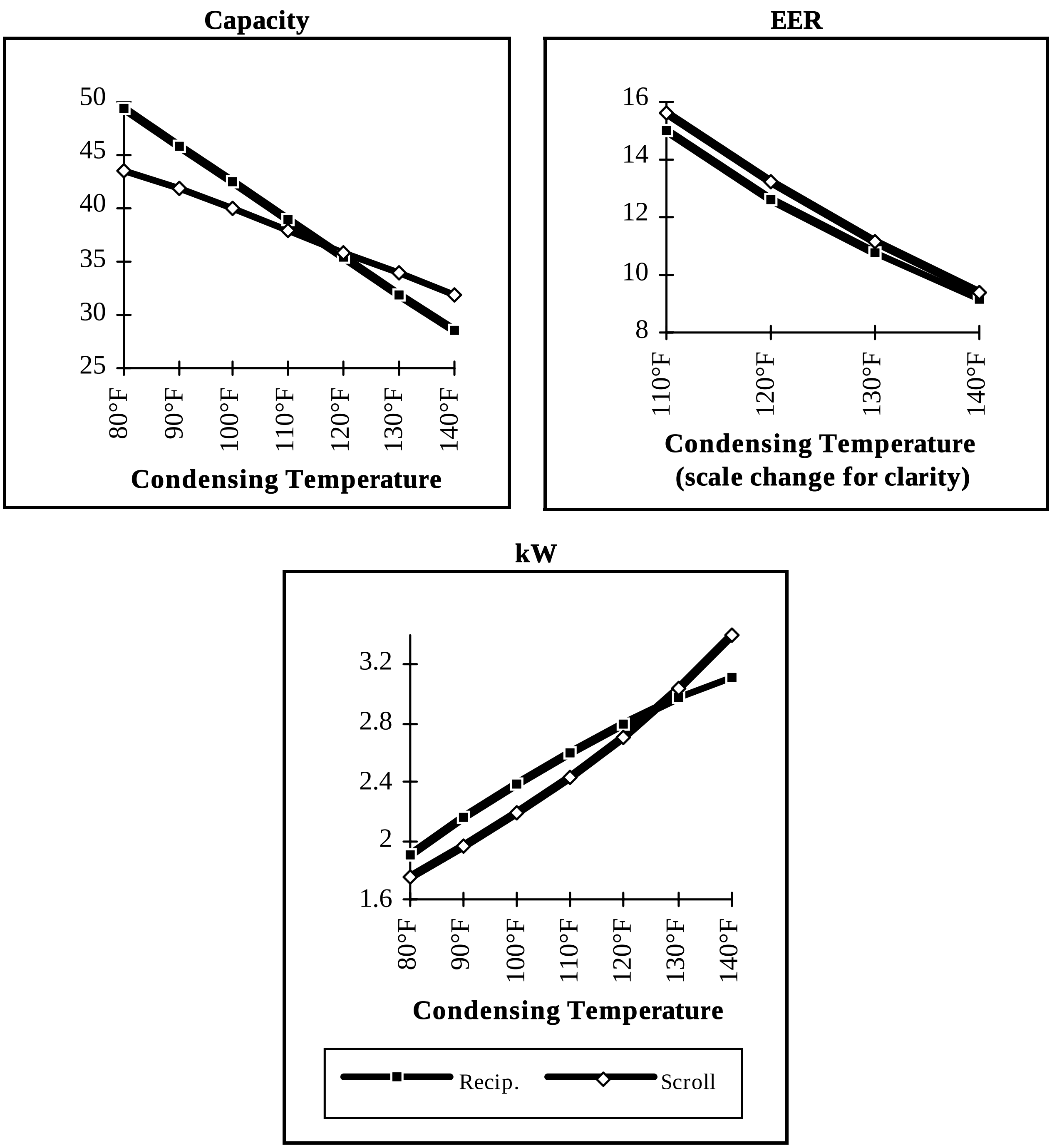

Figure 5. Compressor Performance (Reciprocating vs. Scroll) at 45°F Evaporating Temperature

Both the reciprocating and the scroll compressors have the same capacity near 120°F, which is the approximate condensing temperature for an outdoor air temperature of 95°F (the ARI capacity rating point). Beyond a condensing temperature of 128°F (an outdoor temperature of about 103°F), the scroll compressor draws more than the reciprocating compressor of the same nominal size.

In the simulation the scroll compressor drew slightly less kW than the reciprocating compressor when normalized to the same capacity at 115°F.

Other Potential Compressor Changes. Other compressor changes are possible. It is conceivable that a compressor could be designed with a performance curve that reduces capacity without an efficiency loss at high condensing temperatures. At this time there is no incentive for any manufacturer to develop such a compressor.

Heat Exchanger Changes

Increased evaporator and condenser face area can substantially reduce kW draw (when accompanied by a compressor downsize). Other changes in the coil that improve heat transfer are increased tube rows, increased fin density, revised fin design, and revised tube design. A number of these changes were tested in the simulation program. As single changes, increased evaporator coil face area and increased condenser coil face area were very effective in reducing peak watt draw.

A larger evaporator coil will run at a higher temperature and have somewhat reduced latent capacity. This higher evaporator temperature improves the efficiency of the unit. In hot dry climates such as California's Central Valley it is probable that high latent capacity is not needed, since indoor moisture gains are substantially offset by the infiltration of dry outside air, or positive ventilation. A study of the actual indoor conditions in hot dry locations such as much of California, Nevada, and other states, would be beneficial in guiding design.

In the simulation, increasing the face area of the evaporator coil from 4 sq. ft. to 5.5 sq. ft. showed a peak reduction of 133 watts. When the condenser face area was almost doubled peak watt draw dropped by 363 watts.

Fan and Motor Efficiency Changes

As efficiency improvements are made to the air conditioner compressor, the outdoor evaporator fan and indoor blower fan become more important for power reduction. Fans use about 17 percent of the power for a SEER 7.5 unit, 24 percent for a SEER 12 unit. As the compression cycle is improved, cost-effective improvements are to be found in the fan and motor. Without fan and motor efficiency improvements, SEER improvements will be constrained.

The power required for the fan to move the necessary air volume is determined by three factors.

¥ Fan efficiency

¥ Motor efficiency

- ¥ Air flow resistance

Motor efficiency is the percentage of the input energy which the motor converts to shaft torque. Fan efficiency is the percentage of shaft torque which the fan converts into air movement. The greater the air flow resistance, the more work the fan must perform and the more energy that will be expended.

Outdoor Fan. The outdoor fan draws a large volume of air at very low static pressure, 0.1 to 0.2 inches of water column (WC) through the condenser coil. A typical (SEER-10) 3-ton AC unit would move 2800 cfm of air at a load of 300 watts. Axial fans are well suited to the high volume and low static pressure applications of the condenser fan. The typical outdoor fan / motor combination is a propeller type with a fan efficiency of 20 percent and permanent split capacitor motor with a motor efficiency of 55 percent. The resultant combined efficiency is near 10 percent.

Higher efficiency combinations are available and are used in higher efficiency units. These include larger more efficient propeller fans and high efficiency permanent split capacitor (PSC) motors (combined efficiency 15 percent).

The 15 percent efficiency combination was tested in the simulation and appears cost effective, while reducing the peak draw by 79 watts. A combination of the large propeller fan and a brushless permanent magnet motor was also tested (combined efficiency 19 percent) and is considered too costly under current conditions. These changes are discussed in detail in Appendix B.

Indoor Blower. The efficiency of the indoor blower is far more critical than that of the outdoor blower. When the efficiency of the blower is low not only must more energy be expended in delivering the proper air flow, but also all the energy goes into the air stream heating the air that the air conditioner is trying to cool. The indoor blower draws air from inside the house through the return duct system, passes it over the evaporator coil, and distributes the conditioned through the duct system. The indoor blower moves less air, but at a higher static pressure (.5 external WC), than the condenser fan. A typical (SEER-10) 3-ton AC unit would move 1200 cfm of air at a load of 500 watts. Manufacturers universally choose forward curved centrifugal blowers for the indoor fan with a fan efficiency of 43 to 57 percent (Woods in CMHC 1993). The typical motor is a permanent split capacitor motor with a motor efficiency of 52 percent. The resultant combined efficiency is near 25 percent.

Primary limiting factors of fan efficiency for the indoor blower include blower and cabinet size as well as poor inlet and outlet conditions. Higher fan and motor efficiency combinations (combined efficiency 30 percent) are used in higher efficiency air conditioners. These include improved fans and high efficiency permanent split capacitor motors.

This 30 percent efficient combination was tested in the simulation and appears cost effective. That combination reduced peak draw by 115 watts. A combination including a Brushless Permanent Magnet motor (BPM) was also tested (combined efficiency 39 percent) and was considered borderline under current conditions. These changes are discussed in detail in Appendix B.

Cabinet Size and Design

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 and 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 (CMHC 1993). Attention to plenum design which could improve the inlet and outlet conditions of the blower could also improve overall fan efficiency.

These factors were not modeled in the simulation because too little is known about these effects. They show sufficient promise to warrant further investigation.

Evaporator Air Flow and Temperature

A higher evaporator temperature produces a significantly more efficient refrigeration cycle. Higher evaporator temperatures are accomplished by any method of improving the heat exchange between the inside air and the refrigerant. These methods include changes in heat exchanger efficiency, discussed above, and increased air flow.

Increased air flow across the coil has a number of effects. First, it reduces the latent capacity of the unit. Second, it increases the sensible capacity. Third, it requires more blower energy which brings with it increased heat into the air stream. Fourth, it increases the energy loss due to duct leakage. For these reasons, increased air flow is not a high priority for increased efficiency and reduced peak kW with the present field conditions (low efficiency fans, low efficiency distribution systems, and unresolved questions about latent capacity).

In the simulation increasing evaporator air flow alone resulted in a net increase in peak kW. In the optimization of a design, the air flow is an interactive element along with the coil, blower, and motor design. Increased air flow will have to be accompanied by increased fan / motor efficiency.

Condenser Air Flow

Increasing condenser air flow without any changes in the coil, fan, or blower motor had an adverse effect in the simulation. With increased air flow the compressor watt draw dropped, the capacity increased, but, because of the increased fan power, the total kW draw increased. Increased air flow will need to be accompanied by increased fan / motor efficiency. Some current designs are taking advantage of their higher fan / motor efficiencies by increasing the air flow across the outside coil.

Metering Devices

The most popular expansion devices used in residential air conditioners are: the orifice, capillary tube, and thermostatic expansion valve (TXV). Orifice metering and capillary tube metering is less expensive and can be optimized for one particular design condition. The TXV can be designed to cover a wider range of conditions and is less influenced by off design conditions. Higher SEER air conditioners generally are provided with a TXV.

Studies showed that the SEER of AC with TXV was approximately 3% higher than that of capillary tube systems (Stoecker, Smith, and Emde 1981). Table 3, based on Farzad and O'Neal 1993 (Texas A&M study) shows the tested performance for the two expansion devices under properly charged conditions. The tests were conducted in accordance to the DOE / ARI procedures.

Table 3. Metering Device Comparison - Effect on SEER

Expansion Device	EER_B	C_D	SEER
Capillary Tube	10.7	0.235	9.44
TXV	10.7	0.186	9.7

The lower SEER of the capillary tube unit was caused by cycling losses reflected in the coefficient of degradation (C_D). At high temperatures (highest temperature tested was 100°F) the kW draw was the same with both devices within measurement accuracy.

The Texas A&M study also explored performance at off-design refrigerant charge and found that while at 20% undercharge the SEER dropped by 27% with the capillary tube , it only dropped 3.5% with the TXV.

The simulation did not address changes in metering devices. The laboratory data from Texas A&M indicated that the metering devices are equivalent with respect to peak kW.

Subcooling

Subcooling is the difference between condenser saturation temperature and the liquid line temperature. As subcooling is increased the capacity of the unit increases without an increase in compressor power. The positive effect of subcooling is limited by the availability of a heat sink to cool the liquid. Ambient subcooling is limited to the difference in temperature between ambient air and condenser saturation. Current designs usually take advantage of as much ambient subcooling as is economically practical.

Other Changes

Proctor Engineering Group also examined number of proprietary designs that incorporated alternative heat sinks (instead of or in addition to outside air). Some alternatives are already being utilized (such as geothermal air conditioners). One promising design was only applicable to humid climates and was not pursued. Evaporative and compressor based cooling combinations were not studied.

MODCON SIMULATION BASELINE

For the MODCON analysis a baseline simulation of a 3-ton SEER-10 split air conditioner was created. This model had a simulated capacity and kW draw (at various outdoor temperatures) close to those of most units sold in Northern California. This baseline unit had the following features:

Copeland Compressor	CR-32K6-PFV
Evaporator Face Area / Number of Rows / FPI	4 / 3 / 16
Condenser Face Area / Number of Rows / FPI	10.8 / 1 / 25
Indoor Blower Combined Fan / Motor Efficiency	0.25
Outdoor Fan Combined Fan / Motor Efficiency	0.10

RESULTS OF INDIVIDUAL PARAMETER CHANGES

Simulations of various parameter changes were performed on the basis of the typical SEER 10 unit, described above. Some of the changes were sufficiently interactive that a change in that parameter alone produced higher kW at peak. This is a result of the optimization process involved in air conditioner design. Some of the individual parameter changes that produced positive results are summarized in Table 4. <u>The peak watt draw reductions were calculated by normalizing to the initial unit cooling capacity</u>. The last column in the table shows the cost efficiency (based on kW reduction only and at current contractor pricing) of the changes. These costs reflect not only costs to the manufacturer and distributor, but also margins effected by marketing considerations. The mature market costs may be lower and the cost effectiveness is dependent on a combination of both energy savings and peak reduction.

Table 4. Simulation Results for Individual Changes					
Design Parameter	**Parameter Change**		**Norm.[1] Peak Reduction (watts)**	**Added Cost ($)**	**Cost of Reduction ($/kW)**
	from	to			
Evaporator Face Area	4 sq ft	5.5 sq ft	133	53	400
Condenser Face Area	10.8 sq ft	19 sq ft	363	124	342
Condenser Tube Rows	1	2	57	50	877
Compressor	Reciprocating	Scroll	52	50	962
Compressor Size	Copeland CR32K6-PFV	Copeland CR28K6-PFV	146	-30	-205
Compressor Size	Copeland CR32K6-PFV	Copeland, Scroll ZR28K1-PFV	133	0	NA
Combined Indoor Fan/Motor Efficiency	0.25	0.3	115	18	157
Combined Indoor Fan/Motor Efficiency	0.25	0.39	232	192	828
Combined Outdoor Fan/Motor Efficiency	0.1	0.15	79	27	342
Combined Outdoor Fan/Motor Efficiency	0.1	0.19	113	186	1646

1. Normalized to same capacity at 115°F.

AIR CONDITIONER DESIGNS THAT ACHIEVE TARGETED PEAK KW REDUCTION

The individual component change simulations provide information to prioritize the design process. Based on this and information gathered from design experts, a number of new designs were developed from the baseline 3-ton SEER-10 unit. Four of the units met the initial design criteria.

Reduced Peak kW Designs

The reduced peak kW designs incorporate the design features shown in Table 5

Table 5. Major Parameters of Reduced Peak kW Designs

Unit	**Compressor** (Copeland)	**Evaporator** (face area / number of rows / FPI)	**Condenser** (face area / number of rows / FPI)	**Indoor blower** (combined fan / motor efficiency)	**Outdoor fan** (combined fan / motor efficiency)
Baseline	CR-32K6-PFV	4 / 3 / 16	10.8 / 1 / 25	0.25	0.1
A	CR-28K6-PFV	4 / 3 / 16	19 / 1 / 25	0.3	0.15
B	CR-28K6-PFV	5.5 / 3 / 16	19 / 1 / 25	0.3	0.15
C	CR-28K6-PFV	4 / 3 / 16	23 / 1 / 25	0.3	0.15
D	CR-28K6-PFV	6.5 / 3 / 16	23 / 1 / 25	0.3	0.15

Table 6 shows the simulated units' performance at 115°F. Achieved peak draw reduction is in total (not normalized) watts compared to the baseline unit.

Table 6. Proposed Designs vs. Baseline Unit at 115°F

Unit	**Capacity at 115°F** (Btuh)	**Watts at 115°F**	**Actual Peak Reduction** (W)	**Added Cost** ($ per unit)
Baseline	31163	4046	NA	NA
Unit A	30235	3331	715	139
Unit B	31050	3304	742	192
Unit C	30276	3310	736	199
Unit D	31435	3273	773	287

All simulated units (A, B, C, and D) meet the goal of at least 500 watts reduction at 115°F. At the same time, the units have different capacities and EERs at this temperature. Units B and D have similar capacity to the baseline unit at 115°F. Both of these units utilize a larger evaporator coil which would necessitate some cabinet redesign. Such a redesign could also improve the efficiency of the indoor fan, an improvement which was not taken into account in the simulation.

The larger coils will also increase the amount of refrigerant charge (by as much as 30% or more depending on passage size) which may become more important if replacement refrigerants are more expensive than R-22.

Since the compressor size of the simulated units was decreased, it is necessary to ensure that the units can still provide 3-ton cooling capacity and operate efficiently at standard ARI conditions. Results of the computer simulation at 95°F and 82°F outdoor temperatures are shown in Table 7.

Table 7. Proposed Designs vs. Baseline Unit at 95°F and 82°F							
Unit	**Capacity at 95°F (Btuh)**	**EER "A"**	**Watts at 95°F**	**Capacity at 82°F (Btuh)**	**EER "B"**	**Watts at 82°F**	**SEER (C_D=0.2)**
Baseline	35767	9.7	3697	38364	11.1	3453	10
Unit A	34906	11.7	2974	37717	13.9	2715	12.5
Unit B	35889	12.2	2942	38774	14.5	2679	13.0
Unit C	34964	11.9	2946	37786	14.1	2682	12.7
Unit D	36372	12.5	2902	39342	15.9	2631	14.3

All four new designs have a cooling capacity at 95°F, within about three percent of nominal three tons (36,000 Btuh). This is well within the accuracy limits of engineering cooling load calculations. The simulated units A, B, C, and D have higher EERs than the baseline unit at 95°F and 82°F and run at substantially reduced watt draws. MODCON does not simulate transient tests needed to determine SEER. However, based on a C_D of 0.2 these units would also have SEER's in the 12 to 14 range. This is a reasonable assumption, however a number of parameters have changed in these units including the total refrigerant charge and the true SEER will have to be determined.

These units are expected to perform better than existing SEER 12 to 14 units at peak, and they also save the customer cooling kWh compared to a SEER 10 unit.

Reduced Peak kW Designs Compared to Current Designs

The peak performance of the simulated units was compared with the performance of the most popular SEER 10 air conditioners as well as the presently available high efficiency units with SEERs ranging from 11 to 14. All units were nominal 3-ton, single-speed air conditioners with cooling capacity ranging from 33,800 to 37,600 Btuh at 95°F outdoor temperature. The units were compared at 80°F dry bulb / 67°F wet bulb, 1200 cfm evaporator entering air, and 115°F, 2400 cfm condenser entering air. The reductions in watt draw at 115°F of the simulated units below those of the manufactured units are summarized in Table 8.

Table 8. Continuous Operation Watt Draw Reduction (115°F) Reduced Peak kW Designs vs. Current Designs					
Current Design		**Watt Draw Reduction, $W_{\text{Reduced Peak kW Design}} - W_{\text{Current Design}}$**			
Manufacturer	**SEER Rating**	**Simulated Unit A**	**Simulated Unit B**	**Simulated Unit C**	**Simulated Unit D**
Manufacturer A	10	739	766	760	797
	12	519	546	540	577
Manufacturer B	10	1131	1158	1152	1189
	12.3	459	486	480	517
	14	183	210	204	241
Manufacturer C	10.65	809	836	830	867
	12	653	680	674	711
	13.2	663	690	684	721
Manufacturer D	10.9	609	636	630	667
	12	509	536	530	567
	13.1	289	316	310	347
Manufacturer E	10.2	1129	1156	1150	1187
	11.7	1019	1046	1040	1077
	12	1019	1046	1040	1077

As Table 8 shows, the continuous operation watt draw of the new designs at 115°F is less than that of the current designs. It should be noted that the units have slightly different cooling capacities as detailed in Appendix C. It is also important to note that the highest efficiency air conditioners are often specified with special indoor air handlers in order to boost SEER. If the same condensing unit is specified with another indoor section, the total unit SEER may be substantially lower. For example, Manufacturer B's SEER 14 unit ranges from 11.65 to 14.75 for various combinations of the indoor section.

DIVERSIFIED LOAD IMPACT AT PEAK CONDITIONS, MODEL P

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. This factor is the diversified kVA at peak. To determine the effect of these simulated air conditioner designs on a diversified (population wide) basis a 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 data. Model P takes the following variables 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

Four customer classes have been identified and subclasses established. With this model peak reduction is estimated based on the customer / home class, subclass, and proposed change. Model P was applied to the simulated designs and a resultant diversified load projected (see Table 9). For these units and the California Central Valley population, the multiplier for diversity at system peak was .46 and for local peak was .65. Since the diversity is so much higher at local peak, benefits increase and the cost per kW drops.

Table 9. Diversified Peak Load Reduction and Cost per kW for Simulated Designs

Unit	**System Peak Draw Reduction** (W)	**System Peak Cost per kW** ($)	**Local Area Peak Draw Reduction** (W)	**Local Area Peak Cost per kW** ($)
Unit A	329	422	465	299
Unit B	341	563	482	398
Unit C	339	587	477	417
Unit D	355	808	502	571

If the replacement of air conditioners is targeted at customers with high peak watt draw the diversified peak reduction can be increased. Each class of customer needs to be examined for potential and for motivation (a seldom used air conditioner is a poor marketing target since there is very little kWh savings available to the customer).

IV. PHASE FOUR - DISCUSSION OF RESULTS WITH MANUFACTURERS

The five air conditioner manufacturers were asked their opinion of the design improvement simulations. Each contact received a letter emphasizing the utility's concern about the peak load impact of residential air conditioners and describing the simulation techniques. The letter enumerated parameters of the simulated units and gave peak watt draws in comparison with those of the units produced by the particular manufacturer. The estimated added cost of the design changes was also discussed. The manufacturers were asked the following questions:

- Is the MODCON simulation a reasonable model for AC performance?
- Are the estimated contractor costs for the air conditioner parameter changes reasonable?
- Can any presently manufactured 3-ton, single-speed split unit (condensing unit plus coil) provide a watt reduction at 115°F similar to that of the simulated units?
    - What is the cost difference at the contractor level between that unit and typical SEER 10 and SEER 12 units (and higher, depending on the manufacturer)?
- If they were to manufacture 3-ton, single-speed split units to match the watt draw (at 115°F) of the simulated units, what parameters (components) would that unit have?
    - What would be the added cost in comparison with SEER 10 and SEER 12 units (and higher, depending on the manufacturer)?
- Would it be feasible to manufacture a 3-ton, single-speed split unit with the parameters of the simulated units? How closely would the watt reduction at 115°F match the simulated numbers?
    - What would be the added contractor cost?
- What techniques would you recommend for watt draw reduction at peak load conditions?

The general response to these questions was:

- The simulation and costs were reasonable.
- They could manufacture a unit that would meet the design criteria (each had a different spin on how they would approach the task).
- They want to manufacture units that have the largest possible market.
- There is no financial incentive to manufacture low peak kW air conditioners.

- Some manufacturers were concerned with latent capacity and cabinet size for the large evaporator designs.

V. CONCLUSIONS AND RECOMMENDATIONS

This study investigated possible changes to residential air conditioners to reduce kVA 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

- Based on manufacturer supplied data on current high market penetration air conditioners, the SEER rating is not an accurate predictor of peak kW or kVA.
- Oversized units are likely to produce a higher kW and kVA than properly sized units (under a number of peak conditions this increase will be substantial).
- A number of techniques currently employed by the AC manufacturers to increase SEER are also effective at reducing peak draw. These include:
  - Increased condenser area and efficiency
  - Increased evaporator area and efficiency
  - Improved condenser fan / motor efficiency (sometimes with increased air flow)
  - Improved evaporator fan / motor efficiency
- A number of techniques currently employed by the AC manufacturers to increase SEER are ineffective or detrimental to reducing peak draw. These include:
  - Two speed compressor
  - Variable speed compressor (Current variable speed units have a low power factor and high current harmonic distortion.)
  - Scroll compressor
  - Evaporator fan time delay
- It is possible to build a 3-ton residential air conditioner with existing technology that will have a diversified local peak of .4 to .5 kW less than existing SEER 10 units. The reduced peak kW (RPK) unit would have a peak kW draw less than current SEER 12 designs. The incremental cost increase should price the RPK unit at or below the cost of current SEER 12 units. These designs would have a SEER between 12 and 14.

The incremental cost (at the contractor) of the RPK design would be:

- between $300 and $600 per local area peak kW (diversified)
- between $400 and $800 per system peak kW (diversified)

RECOMMENDATIONS

Based on this investigation, Proctor Engineering Group makes the following recommendations:

- PG&E enter into discussions with ARI on rating methods to assure peak reduction. For the utility, a direct rating of steady state kVA at a high temperature would be most effective. The cost effectiveness of one air conditioner over another is determined by both demand and energy considerations.
- Properly size new and replacement units to reduce peak kVA.
- Monitor the actual indoor conditions (temperature and humidity) of a sample of air conditioned residences in PG&E's service territory. This would substantially add to the current information and assist development of a low peak kVA unit. Little is known about the latent capacity needs in hot dry climates. Some effective design changes to reduce peak may result in reduced latent capacity. Whether a reduction in latent capacity would effect comfort or energy use in hot dry climates has not been tested.
- Through an alliance with the manufacturers, build and lab test the most promising RPK designs. Based on the results undertake a limited field test of the designs.
- Lab test alternative cabinet designs matched to more efficient blowers. This could result in substantial efficiency gains at a low cost.

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APPENDIX B-FAN AND MOTOR EFFICIENCY

The design of an air conditioner is a very interactive process, changes in one design feature effect the functioning of other design features. This is quite clear in the fan and motor area. This appendix explores fan and motor efficiency.

FAN EFFICIENCY

Fans convert shaft torque into flow and increased pressure in the air stream. There are two basic type of fans, axial and centrifugal. Axial fans direct the airflow parallel to the axis of fan rotation; the common house fan is an axial fan. Centrifugal fans move the air by centrifugal force and rotate perpendicular to the air stream; the squirrel cage blower in a furnace is a centrifugal fan. There are several subgroups within the axial and centrifugal categories.

Each type of fan has its strengths and weaknesses. The choice of fan for each application involves tradeoffs. Differences important to fan choice include:

¥ Fan efficiency at the operating conditions

¥ Static pressure at design conditions

¥ Operating characteristics at off-design conditions

¥ Noise

¥ Space requirements

¥ Cost

Outdoor Fan

The outdoor fan draws a large volume of air at very low static pressure (0.1 to 0.2 WC) through the condenser coil. Typical electrical load and air flow for a 3-ton AC unit are 300 watts and 2800 cfm. Considerations for design and manufacture are cost, durability in outdoor conditions, efficiency, and noise. Manufacturers universally choose axial propeller fans for this application.

The axial fans are well suited to the high volume and low static pressure applications of the condenser fan. However, a number of inefficiencies are typically present in the fan and fan mounting design. These include :

¥ Unoptimized fan blade, cross section, and shape

¥ Excessive clearance between fan blade and shroud

- ¥ Recirculation at the fan hub
- ¥ Restricted fan inlet
- ¥ Obtrusive fan mounting hardware
- ¥ Lack of a diffuser or stator vanes

The fan blades are typically stamped out of thick sheet metal. This creates a blade with sharp edges and a constant cross section Optimum blade design has smooth edges and an air foil cross section. Some stamped blades even have ribs for extra rigidity which further disturb air flow. Often the fan outlet is covered by a sheet metal cover to protect the fan from debris and for safety. This cover can cause a large static pressure drop.

Table 10 presents the efficiency options and approximate contractor costs for several condenser fan designs.

Table 10. Condenser Fans

Fan Design and Application (2800 cfm, 0.13")		**Efficiency**	**Cost**	**Savings (Watts) Cost ($/kW) 55% Motor Eff..**	**75%**
Regular	SEER 10	20%	$18	---	---
Premium	SEER 12-15	25%	$26	78 W $103 / kW	57 W $140 / kW
Future technical potential		60%	$140	259 W $471 / kW	190 W $643 / kW

* Assumptions: 2800 CFM, 0.13" wc

Typical propeller fan efficiencies are about 20 percent. Larger propeller fans with efficiencies of about 25 percent are currently employed in higher SEER air conditioning units. This change alone can cause a peak reduction of 78 watts over a standard unit. The incremental cost of the larger propeller fans is quite small.

A vaneaxial, rather than propeller, fan can provide an efficiency of 60 percent and stable operating characteristics at only a slightly higher price (Wright, 1983). This change would result in a peak reduction of over 200 watts for a typical unit. At this time, this option is not commercially available.

Indoor Blower

The indoor blower draws air from inside the house through the return duct system, passes it over the evaporator coil, and distributes the conditioned air through the duct system. The indoor blower moves less air, but at a higher static pressure, than the condenser fan. Typical values for a 3-ton (SEER-10) unit are 500 watts, 1200 cfm, and 0.5" WC of external static pressure, 1" total static at the fan[2]. Considerations for design and manufacture are cost, efficiency, and noise. Manufacturers universally choose forward curved centrifugal blowers for the indoor fan.

One of the most important design considerations is that the external static of the duct work is unknowable until the final installation. Additionally, the static pressure changes as the filter gets dirty. In order to deliver proper air flow under all of the possible installation and operating conditions, various compromises to efficiency are necessary. In particular, the fan cannot be set to achieve maximum efficiency at design conditions. If it were, then at higher static pressure, the increased pressure and efficiency drop-off would cause excessive motor torque requirements and a rapid loss of air flow. Instead, the fan is set to operate below maximum efficiency. As the static pressure increases, the fan efficiency also increases, creating a more level air flow and power requirement (see Figure 6). To adapt even further to various conditions, a multi speed motor is used.

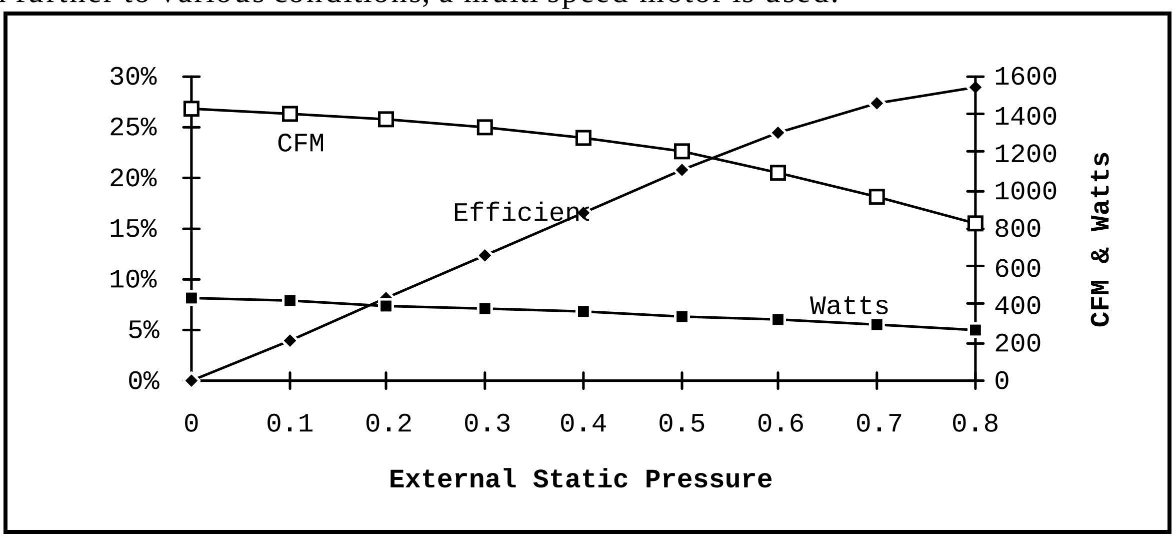

Figure 6. Indoor Blower Operating Characteristics

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[2]Total static includes the pressure drop from the cabinet, evaporator coil, and filter. External static measures only flow resistance from the return and supply ducting system

Table 11 shows potential costs and savings for indoor blower options.

Table 11. Indoor Blowers

Fan Design and Application		**Efficiency**	**Cost**	**Savings (Watts) Cost ($/kW) 57% Motor Eff..**	**75%**
Regular	SEER 10	48%	$53	---	---
Premium	SEER 12-15	52%	$61	36 W $224 / kW	27 W $295 / kW
Future technical potential		70%	$125	146 W $494 / kW	111 W $651 / kW

The potential efficiency[3] of the forward curved centrifugal blower is 70 to 75 percent. However, in actual practice, an efficiency of only about 45 to 50 percent is achieved due to the constraints and losses associated with cabinet and air flow design. Little innovation is present yet in the indoor blower design. The blower wheels are highly standardized. One fan manufacturer states that improvements in fan blade design are not warranted since so little of any improvement would end up as increased unit efficiency. The problem in the cabinet and airflow design must be improved first.

Increasing the chamber and blower size and reducing the fan speed will improve efficiency. The backward curved centrifugal blower has a higher efficiency than the forward curved. Performance issues with the backward curved design include performance at lower static pressures, operating speed, and noise. Higher efficiency backward curved blowers that operate well at low static pressures are possible, but are not yet commercially available.

MOTOR EFFICIENCY

Three types of motors are in current use in air conditioner applications:

¥ Shaded pole motors

¥ Permanent split-capacitor (PSC) motors

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[3] The fan efficiency is measured at the fan itself. It uses the total pressure at the fan. The air handler efficiency measures the total pressure gain across the air handler; this discounts the internal static resistance and the work done to overcome it.

- ¥ Brushless permanent magnet motors (BPMs), [General Electric refers to them as Electronically Commutated Motors (ECM™)

Two additional types of motors are of potential interest:

- ¥ Switched reluctance motors
- ¥ Three-phase motors

Shaded pole motors are the least expensive small motors, but have the lowest efficiency and lowest power factor. The more efficient PSC motors are now generally used rather than shaded pole motors. PSC motors are available in regular and high efficiency models with peak efficiencies of 63 and 68 percent, respectively.

Even higher efficiencies, up to 82 percent, are possible with brushless permanent magnet motors (BPM). The BPM is a DC motor with a permanent magnet for the rotor. The brushes and commutator are replaced by an integrated circuit which electronically switches the stator winding polarities. The reversal rate is directly controlled at the motor, making the BPM motor inherently variable-speed. The best known of the BPM's is the General Electric ECM™.

Traditionally, multi-speed motors have used such technologies as alternate taps on the windings to produce multiple speeds. The lower speeds are much less efficient than full speed. With BPMs, the high efficiency is maintained at the lower speeds. This type of motor is used by manufacturers on their highest efficiency product lines. Contractor cost is about $200 additional per unit.

The BPM has several other advantages. The integrated circuitry which controls the motor can be programmable. This allows such innovations as soft start, in which the motor starts slowly and gradually reaches running speed. Different speeds (at similar efficiency) can be set for the motor in heating and cooling modes. Current manufacturer development efforts are aimed at cost reduction and the incorporation of additional features. General Electric estimates a 30 percent price decline for the motor by 1997 (GE, 6/29/1993).

Future Developments

The switched reluctance motor is an alternative AC motor that does not require electrical excitation of the rotor, thus reducing losses. Reluctance motors have traditionally been used in small, constant-speed devices such as timers and turntables. The switched reluctance motor provides variable speed operation, low losses, and high power factors. Development of this motor is ongoing. Efficiencies greater than those of the BPM have not been achieved and this motor is not yet used in AC equipment.

Three-phase motors are inherently more efficient and reliable, and potentially cheaper than single-phase motors. They are typically found in most commercial and industrial applications. If a cost-effective way to synthesize three-phase power from single-phase residential service can be developed, the use of this type of motor may become practical.

Condenser Fan Motors

The operating environment of the condenser fan motor can be fairly accurately predicted. The air flow and static resistance of the condenser are known. Therefore, the motor can be optimized to the load. A single speed is customary, with two-speed operation on higher SEER units.

Table 12 lists typical efficiency and cost values for condenser motors. Improvements of the condenser motor and fan efficiency are interrelated. Efficiency improvements of the motor are less valuable with a more efficient fan, and vice versa. Current power draws for the condenser range from 180 to 400 watts. The technical potential appears to be as low as 60 watts. The cost-effectiveness of the motor efficiency improvements ranges from $734 to $2,148 per kW for current condenser fans, to $1,761 to $5,156 per kW for a future highly efficient fan.

Table 12. Condenser Motor Efficiencies

Motor (2800 cfm, 0.13")	Applica-tion	Effic-iency	Cost	Savings (Watts) Cost ($/kW) 25% Fan Eff	60%
Permanent Split Capacitor (PSC)	Regular SEER 10	55%	$47	------ ------	------ ------
High efficiency PSC	Higher SEER 12	60%	$66	26 W $734 / W	11 W $1761 / W
Brushless Permanent Magnet (BPM)	Premium SEER 15+	75%	$225	83 W $2148 / W	35 W $5156 / W
Future BPM	Future	82%	$150	102 W $1007 / W	44 W $2416 / W

Indoor Blower Motors

Motor efficiency is doubly important for the indoor blower. All of the energy used by the motor ends up as heat gain in the conditioned space. As with the condenser fan, two motor types are in common use, PSC motors and BPMs (ECM™).

The indoor blower motor is larger than the condenser motor. While it moves less air than the condenser, the static pressure is much higher. As discussed earlier, the external static pressure of the final installation is not known in advance and changes as the filter gets dirty. In order to allow for higher than design external static pressure, the motor must be over-sized for design conditions. Typically, a three-speed PSC motor is set to operate on medium speed at the design external static of 0.5". Motor efficiency drops off rapidly at the slower speeds.

BPMs are particularly advantageous compared to PSC motors for the indoor blower. BPMs, which are inherently variable-speed, maintain higher efficiencies at slower speeds, allowing the installation to be optimized. Also, being more efficient, BPMs cause less heat gain. BPMs are now used in the highest SEER units of several manufacturers. Table 13 lists estimated costs and savings. As the incremental cost of the BPM is reduced, its use should become increasingly common.

Table 13. Indoor Blower Motor Efficiencies

Motor (1200 cfm, 0.9")	Applica-tion	Effic-iency	Cost	Savings (Watts) Cost ($/kW) 48% Fan Eff	75%
Permanent Split Capacitor (PSC)	Regular SEER 10	52%	$76	------ ------	------ ------
High efficiency PSC	Higher SEER 12	57%	$86	45 W $224 / kW	29 W $351 / kW
Brushless (BPM) Permanent Magnet	Premium SEER 15+	75%	$250	156 W $1117 / kW	100 W $1746 / kW
Future BPM	Future	82%	$175	186 W $533 / kW	119 W $833 / kW

AIR FLOW RESISTANCE

Condenser Coil Pressure Drop

The static pressure caused by the condenser coil is already low and its reduction by using a larger face area is fairly expensive. Increased compressor efficiency from a larger condenser coil is the prime consideration for condenser sizing.

The top outlet of the condenser fan is covered by a fan guard. The guard is necessary to protect the fan from the elements and for safety, but some guards are restrictive and cause high back-pressure. Redesigning the guard to reduce this back-pressure is assumed to have no incremental cost and is therefore highly cost-effective.

Indoor Blower Unit Pressure Drop

The indoor blower unit has three basic sources of pressure drop -- the evaporator coil, the filter, and the cabinet design.

A major source of static pressure drop in the unit is the evaporator coil itself. The coil commonly causes 0.35" of static drop. This pressure drop can be reduced by increasing the coil face area and reducing the fins per inch (see Figure 7). Increased compressor efficiency from a larger evaporator coil is the prime consideration for evaporator sizing.

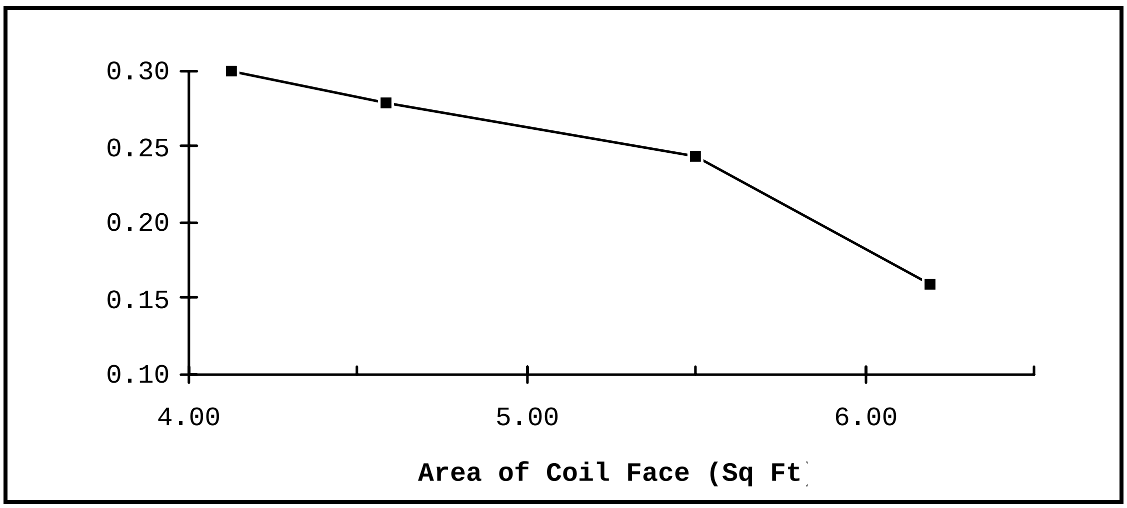

Figure 7. Coil Face Area vs. Pressure Drop (1200 cfm)

The more roomy the cabinet and the straighter the air flow within it, the less the pressure drop. Ideally, the pressure drop from the cabinet itself would be nearly zero. However, manufacturers are often more concerned with size constraints than pressure drop. Many blower wheels are two sided; even if one side is unrestricted the other side will be restricted. Air flow must often pass by the opening between the blower housing and cabinet side and then rotate 90 degrees to get into the blower wheel. Cabinet air flow restriction often significantly increases static pressure drop. Less constricted cabinet air flow could reduce or eliminate this. Carrier does manufacture a high efficiency model with a larger cabinet and blower.

A second possibility to reduce indoor blower unit pressure drop is to use a vaneaxial fan rather than a centrifugal blower. The vaneaxial fan may allow a straighter, less restricted flow path and overall higher efficiency.

In conclusion, a moderate amount of power reduction is possible through improved fan and motor efficiency and reduced air flow resistance of components. A number of manufacturers, trade organizations, government bodies, and researchers are working on improvements. PG&E could contribute to and accelerate this process if it chose to. The following areas might most benefit from such an effort:

¥ Cabinet redesign to reduce air flow resistance

¥ Use of backward curved centrifugal fans for indoor blowers

¥ Use of vaneaxial fans for indoor blowers

¥ Use of low static resistance evaporator coils

¥ Modification of intake and discharge locations for indoor blowers

¥ Study of performance of BPMs (ECM™) in the field

¥ Reduction of condenser static resistance

APPENDIX C-EXISTING AND SIMULATED UNIT PERFORMANCE AT 115°F

Table 14 Existing and Simulated Unit Performance at 115°F

Manufacturer Designation	Unit SEER rating	Capacity at 95°F, Btuh	Capacity at 115°F, Btuh	Watts at 115°F
Manufacturer A	10	33800	29200	4070
	12	35000	31900	3850
Manufacturer B	10	36000	31800	4462
	12.3	36200	31700	3790
	14	37100	32600	3514
Manufacturer C	10.65	36250	31750	4140
	12	35420	32816	3984
	13.2	35919	33219	3994
Manufacturer D	10.9	35000	29600	3940
	12	35400	31700	3840
	13.1	35600	32000	3620
Manufacturer E	10.2	35400	29700	4460
	11.7	37000	31100	4350
	12	37600	31600	4350
Simulation A	12.5	34906	30235	3331
Simulation B	13	35889	31050	3304
Simulation C	12.7	34964	30276	3310
Simulation D	14.3	36372	31435	3273

APPENDIX D-RETROFIT METHODS OF REDUCING PEAK KW

There are a number of methods of reducing the peak load contribution of residential air conditioners. One of the most common attempts at peak reduction is to reduce heat gain or to improve delivered air conditioner efficiency. Unfortunately, this provides a less than certain peak reduction. In order to guarantee the peak reduction, a method of “locking in” the peak reduction that should come from reduced heat gain or increased delivered efficiency is necessary. The most certain peak reduction occurs when the connected load of the air conditioner is reduced.

This appendix discusses connected load reduction that can be accomplished after efficiency improvements have been made. These include:

- Refrigerant changeover
- Compressor speed reduction
- Duty cycling

REFRIGERANT CHANGEOVER

Almost all residential air conditioners use R-22 as the refrigerant. R-500 can be used as a drop-in replacement. Its use will result in a 25 percent power reduction, 28 percent capacity reduction, and 3 percent efficiency reduction[4]. The problem with R-500 is that it is a chlorinated refrigerant. R-500 is an azeotropic mixture of 73.8 percent R-12 and 26.2 percent R-152a. Because of ozone depletion problems and greenhouse warming effects, R-500 is not acceptable as an alternative refrigerant.

Non-chlorocarbon-based refrigerants are being developed as replacements for R-12 in automotive applications. The newer refrigerants require different lubricants. The refrigerants and lubricants have different chemical reactance with the internal system components such as motor winding insulation and metal surfaces, and therefore may necessitate reworking system design. They are not drop-in replacements. Refrigerant alternatives to R-22 are being developed, however finding an alternative refrigerant that will lower capacity is not even on the priority list. Alternate refrigerant development is not yet advanced enough to provide an immediate solution.

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[4] Springer, James, North Carolina Alternative Energy Corporation, personal communication.

COMPRESSOR SPEED REDUCTION - "THE BLACK BOX"

The speed of the compressor can be changed by changing the frequency of the electric current. Many AC compressors are originally manufactured to operate both on U.S. current (230 volt, 60 Hz) and reduced voltage European current (200 volt 50 Hz). Operation at 50 Hz rather than 60 Hz reduces the speed, capacity, and watt draw by approximately 16 percent. The efficiency remains the same.

When U.S. compressors are shipped overseas for European use, a simple step-down transformer is used to reduce the 50 Hz current from 220 volts to 200 volts. The frequency does not change. Changing the frequency is much more difficult than changing the voltage. There are two major ways to accomplish this: using a motor generator set or solid state electronics.

With a motor generator set, the original power can be used to operate a motor which then drives a generator. The equipment for this is bulky and expensive for a residential application, weighing several hundred pounds and costing several thousand dollars. This option is not feasible, since it would be more cost-effective to replace the existing unit.

Using solid state electronics, the original current can be converted to DC and then reformed into the desired frequency AC. The major problems with this approach are inrush current, harmonic distortion, availability, and cost. When the compressor first turns on, there is a large initial surge of current, the inrush current. The electric power converter must be designed to handle or avoid this initial current. It is most practical to avoid the current by ramping up the frequency from 0-50 Hz slowly enough to keep the current draw below the run load amps. The start capacitor must also be isolated from the converter's electronics. The solid state electronics produce about 30 percent harmonic distortion. Additional expense would be needed to correct this.

The solid state power converters that are commercially available are for three-phase motors and produce a pulse output. The pulse output causes harmonics in the motor which produce harmonic heating, about 10 to 15 percent extra heat. Therefore, unless the converter produces a clean sine wave form, the harmonic heating would reduce or eliminate the gain from frequency reduction.

The approximate cost of a commercially available, uncorrected current converter for a residential AC load of 3,000 to 4,000 watts is $800, when purchased in quantity. The estimated installation cost is $200, yielding a total cost of $1,000. Converter improvements necessary to avoid feeding harmonics into the power grid, to produce a pure sine wave, and to isolate it from the compressor motor operation would cost significantly more. In contrast, a new smaller compressor would cost only about $300 for materials, $500 installed.

While it is technically possible to achieve a 16 percent power reduction by running the compressor at 50 Hz rather than 60 Hz, it is less cost-effective than compressor replacement.

DUTY CYCLE

With duty cycling, the capacity is reduced by forcing the compressor off periodically. There are several advantages to this technique. First, none of the operating conditions of the system have to be changed (this still doesn't ensure that the warranty will stay in effect). The necessary equipment is relatively inexpensive and commercially available. Installation of the necessary timer is fairly straightforward and simple. The major drawback is that it can be fairly easily disconnected, although in most cases only by a knowledgeable serviceperson.

Solid state timers to accomplish this are commercially available for $13 to $26 each, depending on the quantity ordered. They are completely solid state, encapsulated, and suitable for outdoor installation. Installation is estimated at $60 to $75 per unit, for a total cost below $100 per unit.

CONCLUSION

The most feasible method to "hard wire" peak reduction for an AC unit made oversize by some efficiency improvement is by connected load reduction. Three methods were evaluated, compressor speed reduction, refrigerant changeover, and duty cycling. Compressor speed reduction is not cost-effective because of technical considerations involved with frequency change. Refrigerant changeover to R-500 would work but involves a chlorocarbon and is therefore unacceptable. Duty cycling alone is a simple inexpensive method. However, duty cycling has been controversial throughout the years. It is probably not likely to find wide acceptance among air conditioning professionals.