Evaluation, Measurement & Verification Report for the Residential Ground Source Heat Pump Program

Prepared for the Redding Electric Utility

FINAL

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1. Executive Summary

This report provides Evaluation, Measurement and Verification (EM&V) results for the Residential Ground Source Heat Pump Program implemented by Redding Electric Utility (REU). The program realized energy and peak kW savings by paying incentives to builders for installing high efficiency ground source heat pumps (GSHP) instead of conventional air conditioners and gas furnaces. This EM&V report provides ex post energy and peak savings for the program.

Findings from this study indicate the GSHP units provide advantages for all participants. For the utility, the GSHP reduces peak demand in summer by an average of 2.1 kW per unit and shifts summer cooling loads to winter increasing annual electricity use by 1,355 kWh per year (roughly 10 percent). For the customer, the GSHP reduces annual energy bills for space conditioning by 48 percent saving $639 ± $185 per year. For society, the GSHP mitigates global warming by reducing carbon dioxide emissions for space conditioning by 44 percent, saving 59 million British thermal units (MMBtu) per year of source energy per GSHP.

Ex ante program savings are summarized in **Table 1.1**, and ex post savings are summarized in **Table 1.2**. The ex ante program savings were 455,841 kWh per year and 104 kW. Total net ex post savings for the program are -36,587 ± 22,698 kWh per year, 56.1 ± 0.64 kW, and 14,745 ± 4,368 therms per year. The net-to-gross ratio is assumed to be one since participants wouldn't have purchased the GSHP without incentives due to its high cost relative to conventional AC units.[1] The gross M&V savings and net realization rates are lower than anticipated primarily due to electricity heating usage and lower energy efficiency performance based on field measurements of EER. Nevertheless, the program reduces participating customer space conditioning energy use by 48 percent and carbon dioxide emissions for space conditioning by 44 percent.

Table 1.1 Ex Ante Savings for Residential GSHP Program

Program	Qty.	Ex Ante Full-Year Unit kWh/yr	Ex Ante Unit kW	Ex Ante Net-Unit therm/y	Ex Ante Program Savings kWh/y	Ex Ante Program Savings kW	Ex Ante Program Savings therm/y
GSHP	27	16,883	3.85	n/a	455,841	104	n/a

Table 1.2 Ex Post Savings for Residential GSHP Program

Program	Qty.	M&V Full-Year Unit kWh/y	M&V Unit kW	M&V Full-Year Unit therm/y	Ex Post Program Savings kWh/y	Ex Post Program Savings kW	Ex Post Program Savings therm/y	Net Realization Rate kWh/y	Net Realization Rate kW	Net Realization Rate Therm/y
GSHP	27	-1,355	2.1	546	-36,587	56.1	14,745	-0.08	0.55	n/a

The EM&V study provides average gross savings per unit. The savings are based on in-situ 15-minute true RMS power measurements of 2 air conditioners and 2 GSHP units. Each unit included in the random sample was measured for three months in order to obtain 15-minute

[1] The incremental cost is $14,986 based on average GSHP cost of $23,186 and conventional AC cost of $8,200. The average incentive offered by REU was $12,100. Cost data from Paul Ahern, Key Accounts Manager, REU, and Tim Camacho, General Manager, Palomar Builders, Redding, California.

average kW measurements during the peak period from 2 PM to 6 PM on weekdays. The peak kW for each unit is the maximum kW that occurs during the peak period based on the 15-minute data.

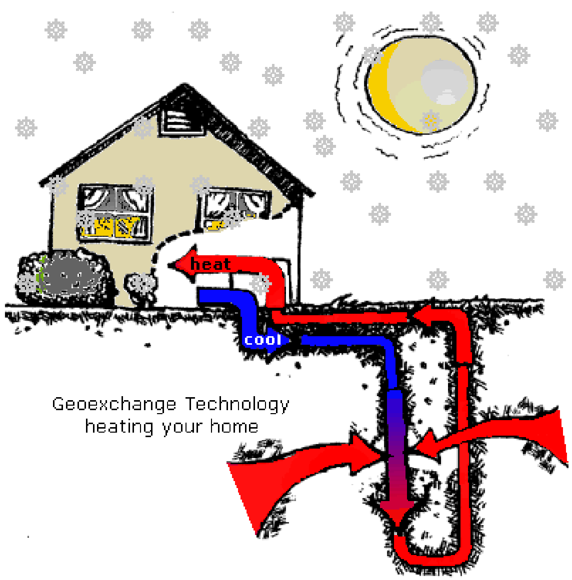
Several studies indicate 50 to 67 percent of new air conditioners have improper refrigerant charge and airflow (RCA), and this reduces efficiency by approximately 10 to 50 percent. This study found improper RCA on two air conditioners with average savings of 1,095 kWh per year and 0.38 kW (see **Section 3.3**). The study also found average duct leakage for all sites of 15.5%. This is 2.6 times higher than the California Energy Commission target value of 6%. The average energy savings found in this study for tight ducts are approximately 337 kWh per year, 0.2 kW, and 38 therms per year. Builders and air conditioning dealers interviewed for this study indicated support for a program involving verification of proper RCA and duct sealing.

Section 2 presents the GSHP measure description. **Section 3** presents the field measurement methodology, findings of the field measurements, energy and peak demand savings for GSHP units, and the impact of proper refrigerant charge and airflow and duct sealing on air conditioner efficiency performance.

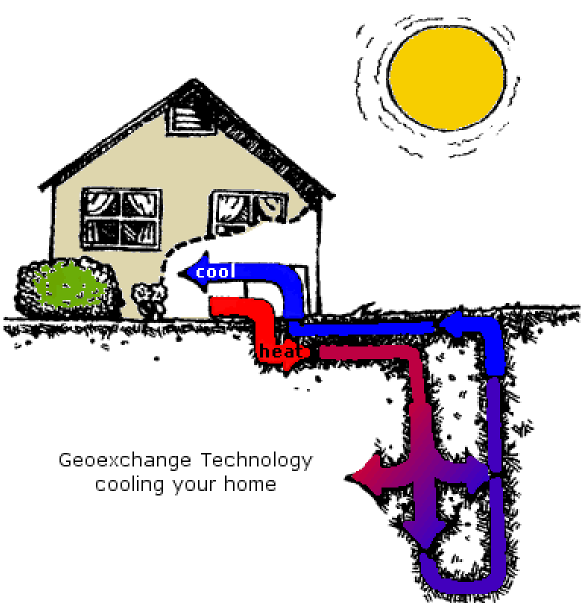
2. GSHP Measure Description

Ground source heat pumps are also referred to as "geoexchange" heat pumps since they exchange heat with the ground instead of the outdoor air. The temperature of the ground remains relatively constant throughout the year, even though the outdoor air temperature may fluctuate greatly with the change of seasons. At a depth of approximately six feet, for example, the temperature of soil in most of the world's regions remains stable between 45 and 70 degrees Fahrenheit (°F). This is why well water drawn from below ground tastes cool even on the hottest summer days. In winter, it is much easier to capture heat from the soil at a moderate 50°F temperature than from the atmosphere when the air temperature is below freezing. This is also why GSHP systems can provide warm air through a home's ventilation system, even when the outdoor air temperature is extremely cold. Conversely, in summer, the relatively cool ground can absorb the home's waste heat more readily than the hot outdoor air.

The GSHP system circulates water through polyethylene pipes buried in the ground (ground loop), using a small circulating pump. The soil heats the water as it flows through the buried pipes. The warmed water is then passed through the GSHP located in the building, where heat is taken out of the water by the refrigerant system in the heat pump. The refrigerant system concentrates the heat to produce refrigerant at a high temperature. The high temperature refrigerant is then passed through a coil (similar to a car radiator) and a blower directs the building's air through the coil to produce hot air which heats the building.



To cool a building, the heat pump reverses the flow of the refrigerant system and cold refrigerant is passed through the coil as warm building air is blown across it. This process absorbs heat out of the building air and heats the refrigerant. This heat is then rejected out of the refrigerant system and into water in the ground loop system where the water is circulated through pipes buried in the ground. While water is circulating through the buried pipes it passes heat back to the earth, and cooler water is carried back to the heat pump in the building to absorb more heat.



3. Field Measurement Results for Residential HVAC

The measurement and verification approach for the study was based on the *International Performance Measurement & Verification Protocols* (IPMVP).[2] Ex post energy savings were determined using IPMVP Option B (i.e., retrofit isolation), Option C (i.e., whole facility billing analysis), and Option D (calibrated simulations). Peak demand savings were determined using IPMVP Option B (i.e., retrofit isolation). Field measurements of kW, kWh, and energy efficiency ratios (EER) were used to estimate peak kW savings. The study examined proper refrigerant charge and airflow (RCA) for new GSHP units and air conditioners and how improper RCA and other factors influence efficiency.

3.1 Field Measurement Methodology

Field measurements of the Energy Efficiency Ratio (EER) were made to evaluate energy and peak demand savings and determine in-situ efficiency before and after correcting refrigerant charge and airflow (RCA) on a sample of two air conditioners with thermostatic expansion valves (TXVs) and two GSHP units.[3] Field measurements, measurement equipment, and measurement tolerances are provided in **Table 3.1**.

Table 3.1. Field Measurements, Measurement Equipment, and Tolerances

Field Measurement	Measurement Equipment	Measurement Tolerances
Temperature in degrees Fahrenheit (°F) of return and supply wetbulb and drybulb and outdoor condenser entering air	4-channel temperature data loggers with 10K thermisters. Calibration of wetbulb and drybulb temperatures were checked using sling psychrometers	Data logger: ± 0.1°F Thermisters: ± 0.2°F Sling psychrometer: ± 0.2°F (wetbulb and drybulb)
Pressure in pounds per square inch (psi) of vapor and suction line	Compound pressure gauge for R22 and R410a	Refrigerant pressure: ± 2 % for R22 and ± 3 percent for R410a
Temperature (°F) of vapor and suction lines	Digital thermometer with clamp-on insulated type K thermocouples	Digital thermometer: ± 0.1°F Type K thermocouple: ± 0.1% °F
Temperature (°F) of actual and required superheat and subcooling	Digital thermometer with clamp-on insulated type K thermocouples	Digital thermometer: ± 0.1°F Type K thermocouple: ± 0.1% °F
Airflow in cubic feet per minute (cfm) across air conditioner evaporator coil	Digital pressure gauge and fan-powered flow hood, flow meter pitot tube array, and electronic balometer	Fan-powered flowhood: ± 3% Flow meter pitot tube array: ± 7% Electronic balometer: ± 4%
Ounces (oz.) of refrigerant charge added or removed	Digital electronic charging scales	Electronic scale: ± 0.5 ounces or ± 0.1% whichever is greater
Total power in kilowatts (kW) of air conditioner compressor and fans	True RMS 4-channel power data loggers and 4-channel power analyzer	Data loggers, CTs, PTs: ± 1% Power analyzer: ± 1%
Duct Leakage in cfm at 25 Pascal (Pa)	Digital pressure gauge, controller, fan, extension duct, and flow conditioner.	Fan flow: ± 3%
Building envelope leakage in cfm at 50 Pa and Effective Leakage Area (ELA) in square inches.	Digital pressure gauge, controller, fan, and blower door.	Air leakage and ELA: ± 3%

[2] See *International Performance Measurement & Verification Protocols*, DOE/GO-102000-1132, October 2000.

[3] EER is the cooling capacity in thousand British Thermal Units per hour (kBtuh) divided by total air conditioner electric power (kW) including indoor fan, outdoor condensing fan, compressor, and controls. The Btu is the energy required to raise one pound of water one degree Fahrenheit. EER values are typically measured under laboratory conditions of 95°F condenser entering air and 80°F drybulb and 67°F wetbulb evaporator entering air.

Return and supply temperatures were measured inside the return and supply plenums, and entering water temperatures (EWT) were measured for the GSHP units. Temperature and power were measured at one minute intervals. Airflow cubic feet per minute (cfm) was measured before and after making any changes to the supply/return ducts, opening vents, or installing new air filters that would affect airflow. Return and supply enthalpies were derived from the temperature measurements using standard psychrometric algorithms.[4] EER was derived from the combination of enthalpy, airflow, and power measurements. Duct leakage in cfm was measured at 25 Pascals pressure between the ducts and house with all air vents sealed. Building envelope leakage in cfm was measured at 50 Pascals house pressure with respect to outdoor ambient pressure and duct air vents sealed to exclude duct leakage. Measurements were made to evaluate the relative change in efficiency not the absolute efficiency, and all measurements of air conditioner performance were made within minutes of any efficiency improvements, but at least 15 minutes after any refrigerant charge adjustments. Measurement tolerances are less important than the relative performance change. The rated Seasonal Energy Efficiency Ratio (SEER) was evaluated based on field measurements of EER.[5] Billing data was collected for a seven month period from May 2004 through November 2004. These data were used to develop annual energy savings.

3.2 Findings of the Field Measurements

Field measurements of participant and non-participant air conditioners were made to determine duct leakage, infiltration, and the in-situ energy efficiency ratio (EER) before and after making any service adjustments to the units. Data loggers were installed at 4 sites with identical floor plans in Redding, California, to measure peak demand and energy use for air conditioners and GSHP units. Two homes had 12 SEER/10 EER conventional air conditioners with nominal capacities of 5 tons, and two homes had 16.5 EER GSHP units with nominal capacities of 4 tons. The following field measurements were made at each house (see **Table 3.2**): airflow in cubic feet per minute (cfm), duct leakage (% of total airflow), infiltration (cfm at 50 Pascal house pressure), capacity in thousand British thermal units per hour (MBtuh), and service adjustments (i.e., proper refrigerant charge and airflow and purging air from the GSHP ground loop). The AC and GSHP units were found to have comparable cooling capacities even though the AC units were rated at 5 tons and the GSHP units were rated at 4 tons. Average duct leakage for all sites in the study was 15.5%. This is 2.6 times higher than the California Energy Commission target value of 6% (for more information, please refer to last section of the report).

The air conditioners at sites #1 and #2 did not perform at their rated efficiency. Refrigerant charge and airflow (RCA) were checked, and found to be outside the manufacturer's specifications. All vents were opened and fan speed was checked on the units to improve airflow. Refrigerant charge was added to both units as per the manufacturer's specifications. The relative efficiency gain due to proper RCA for the two TXV-equipped air conditioners was 42 percent (see **Table 3.2**). Even with correct refrigerant charge the TXV-equipped units did not perform at their rated efficiency due to being installed outside the evaporator coil box in hot attics and TXV

[4] Kelsey, J. 2004. Get Psyched™ Psychrometric Software for MS Excel, Available online: www.kw-engineering.com. Oakland, Calif. kW Engineering.

[5] SEER is an adjusted rating based on steady-state EER measured at standard conditions of 82°F outdoor and 80°F drybulb/67°F wetbulb indoor temperature multiplied by the Part Load Factor with a default of 0.875 (ARI 2003).

sensing bulbs installed on the vapor line two feet from the vapor manifold. The installer and manufacturers were contacted to evaluate additional service improvements. The condensing coil manufacturer indicated no guarantee regarding ARI (Air-Conditioning and Refrigeration Institute) SEER/EER ratings since the evaporator coil was manufactured by an independent coil manufacturer and was not listed in the ARI directory as a proper match for the condensing coil.[6]

The GSHP units at sites #3 and #4 did not perform at their rated efficiency. Refrigerant charge and airflow were checked on the GSHP units, and found to be within the manufacturer's specifications. The GSHP installer and manufacturer were contacted to evaluate additional service improvements. The GSHP installer purged air from the ground loop at site #3, and this improved the efficiency from 9 to 9.5 EER. The entering water temperatures (EWT) from the ground loop were monitored at each GSHP site. The peak EWT values were 94.9 ± 0.07 F for site #3 and 98.2 ± 0.11 F for site #4. The peak capacity for these GSHP units is approximately 42.5 MBtuh based on manufacturer's literature and the EWT measurements, water loop flow rates (gpm), and airflow (cfm) measurements. Using this upper-limit capacity and peak power input of 3.7 kW for site #4, the upper limit on GSHP efficiency would be 11.4 EER. This is 7 percent higher than the 10.7 EER measurement found at site #4 and compares favorably to the field measured EER values.

Table 3.2 Field Measurements for Conventional Air Conditioners and GSHP Units

Site	HVAC System	EER	Rated Capacity MBtuh	Measured Cooling Capacity Post MBtuh	Average Outdoor, Indoor Dry/Wet Bulb °F	Airflow (cfm)	Duct Leak cfm @ 25 Pa	Infil. cfm @ 50 Pa	EER Pre	EER Post	Service Adjust Oz.	Percent Charge Adjust per Factory Charge
#1	AC	10	51	38.5	105/81/65	1631	19%	1830	3.9	6.5	+98.2	+49.4%
#2	AC	10	51	41.6	105/80/64	1734	12%	1537	5.5	6.5	+12.5	+6.3%
#3	GSHP	16.5	47	39.7	105/75/63	1470	16%	1447	9.0	9.5	Purged	
#4	GSHP	16.5	47	39.9	103/74/62	1443	15%	1928	10.7	10.7	n/a	

Note: Rated EER values are based on manufacturers' data. The GSHP 16.5 EER is for 77°F EWT, 1500 cfm, and 12 gpm.

Energy and peak demand savings are based on three months of 15 minute kW measurements and eQuest/DOE-2.2 simulations calibrated to utility billing data consistent with IPMVP Option D. The eQuest/DOE-2.2 model is shown in **Figure 3.1**, and modeling inputs are based on data collected during site audits. All homes in the sample have the same plan with 2,200 ft^2 of conditioned floor area, R30 ceiling insulation, R19 wall insulation, slab foundation, and low-E windows. The homes were built in 2004 and occupied during late spring or summer. Space cooling and heating UEC values were normalized using Typical Meteorological Year (TMY) weather data for CEC climate zone 11.[7]

[6] The Air-Conditioning and Refrigeration Institute's On-Line Directories of Certified Equipment are available online at http://www.ariprimenet.org.

[7] *California Thermal Climate Zones*, California Energy Commission, 1516 9th St., Sacramento, CA 95814, 1992.

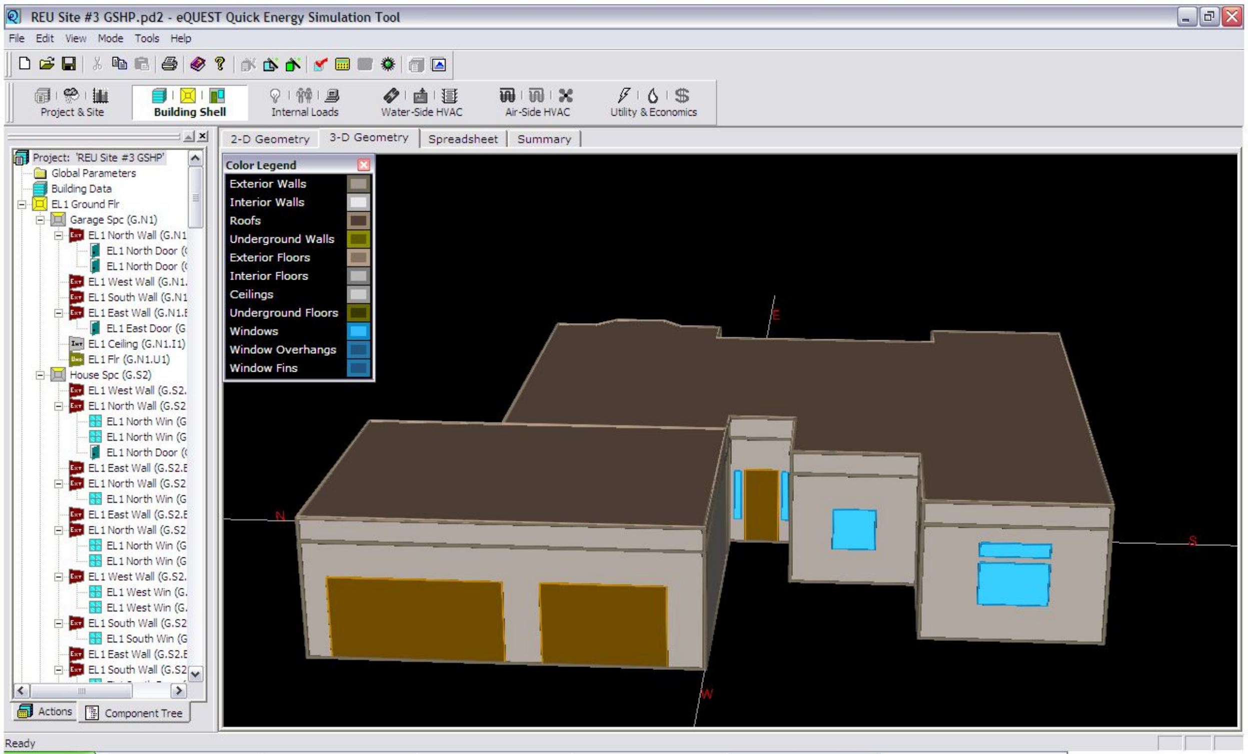


Figure 3.1 eQuest/DOE-2.2 Model

The peak demand savings are based on three months of data logger measurements for the four sites according to IPMVP Option B. Average measured kW savings are 2.1 ± 0.02 kW based on 15-minute data shown in **Figures 3.2** and **3.3**.[8] The GSHP peak demand occurs later in the day and later in the year than the conventional AC (i.e., late August or early September compared to July for the air conditioner). Space cooling and heating usage and savings for each site are summarized in **Table 3.3**. The average cooling use for the GSHP is 3,869 ± 1,189 kWh per year and peak demand is 4.02 ± 0.04 kW (including fans). The average cooling use for conventional air conditioners is 5,145 ± 1,220 kWh per year and peak demand is 6.10 ± 0.06 kW. The confidence intervals for the peak demand values are smaller due to the large sample size of 960 15-minute readings.

[8] Peak kW savings are based on three months of 15-minute logger data evaluated during the peak period from 2 PM to 6PM weekdays within the months of May through October.

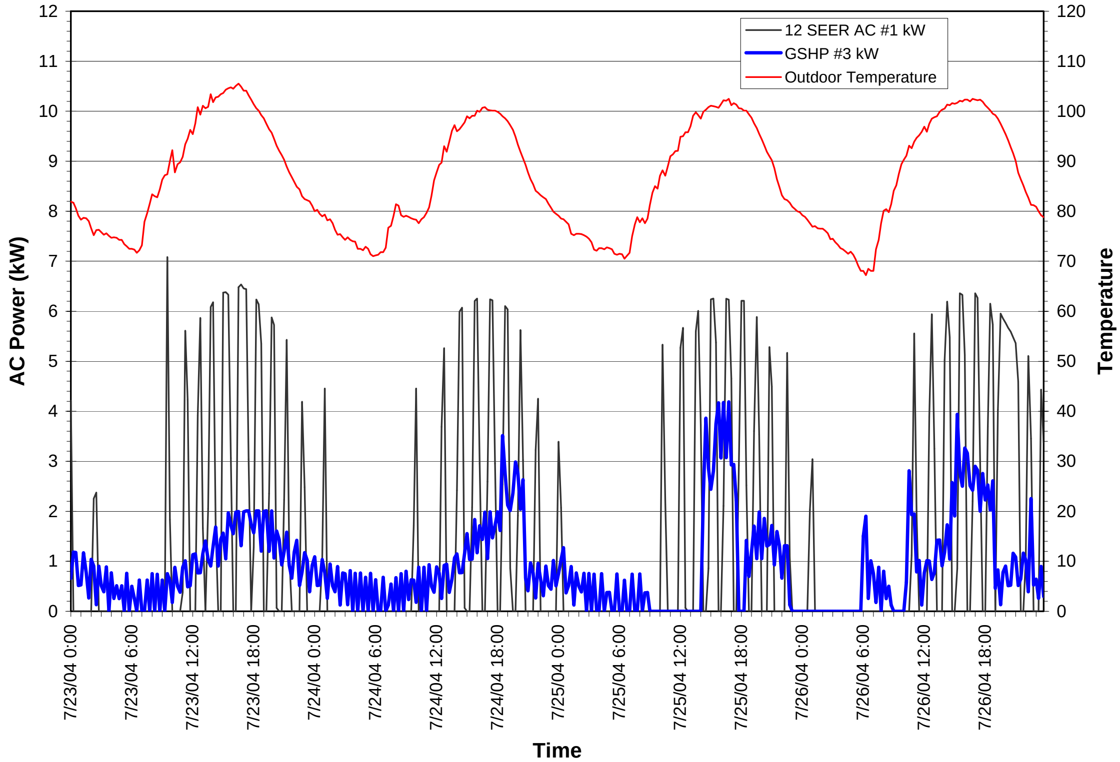


Figure 3.2 Measurements of Standard AC #1 and GSHP #3 Peak Demand (kW)

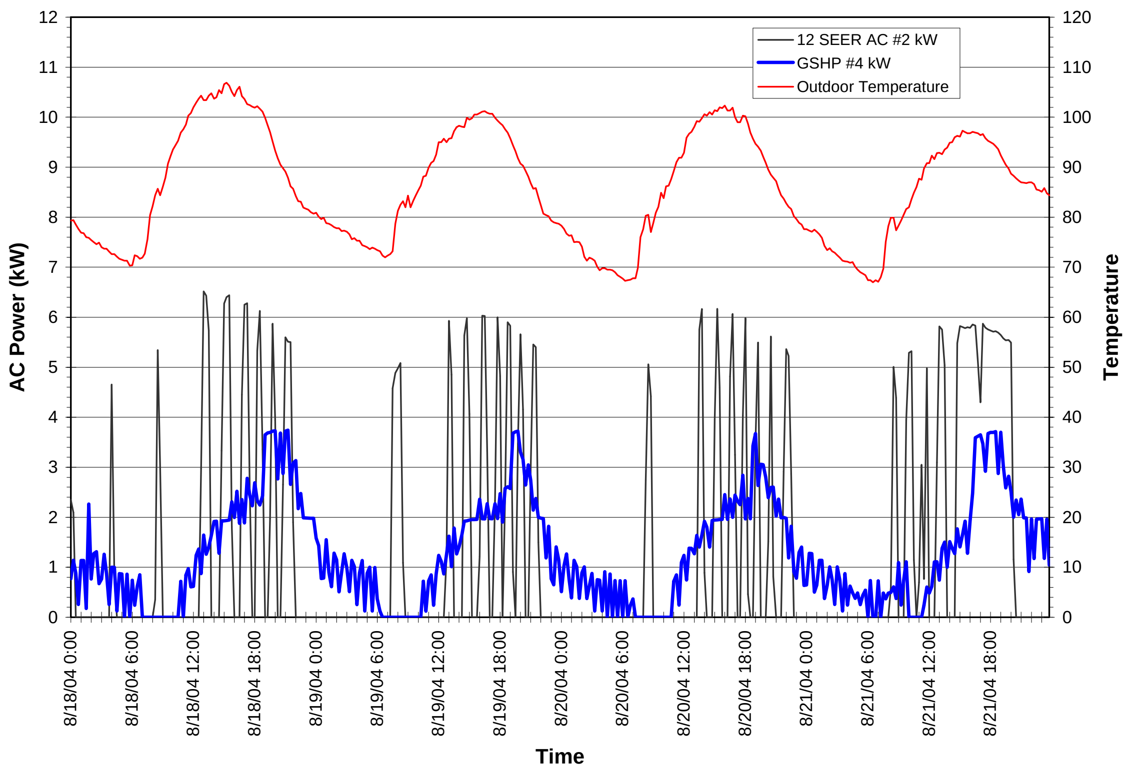


Figure 3.3 Measurements of Standard AC #2 and GSHP #4 Peak Demand (kW)

Average cooling savings per GSHP are 1,276 ± 172 kWh/yr and 2.1 ± 0.02 kW. The GSHP uses 3,116 ± 1,044 kWh per year for heating. It saves 480 ± 105 therms per year on heating and 65 ± 18 therms per year on domestic hot water use. Total annual savings for the GSHP are -1,355 ± 841 kWh per year, 2.1 ± 0.02 kW, and 545 ± 161 therms per year. The average estimated simple payback is 4.7 ± 1.6 years.[9]

Table 3.3 Cooling and Heating Use for Air Conditioners and GSHP Units

Site	AC Cooling kWh/yr	AC Cooling kW	Heating (therm/yr)	Heating (kWh/yr)	DHW (therm/yr)	Energy Cost $/yr	Simple Payback (years)
AC #1	3,560	6.44	494	640	282	$1,218	
GSHP #1	2,489	3.56		1,071	237	$565	4.4
AC #2	6,655	6.54	230	555	282	$1,180	
GSHP #2	5,478	4.56		1,177	192	$778	7.2
AC #3	5,462	5.79	661	838	282	$1,581	
GSHP #3	4,056	3.69		1,406	215	$702	3.3
AC #4	4,902	5.62	534	889	282	$1,397	
GSHP #4	3,453	4.27		1,449	223	$664	3.9
Savings	**1,276**	**2.1**	**480**	**-1,900**	**65**	**$639**	**4.7**

Note: Results for AC #1, AC #2, GSHP #3, and GSHP #4 are based on calibrated simulations of installed equipment at the sites. The results for GSHP #1, GSHP #2, AC #3, and AC #4 are based on simulating the substitution of either an AC or GSHP unit for the actual installed equipment.

Ex ante program savings are summarized in **Table 3.4**, and ex post savings are summarized in **Table 3.5**. The ex ante program savings were 455,841 kWh per year and 104 kW. Total net ex post savings for the program are -36,587 ± 22,698 kWh per year, 56.1 ± 0.64 kW, and 14,745 ± 4,368 therms per year. The net-to-gross ratio is assumed to be one since participants wouldn't have purchased the GSHP without incentives due to its high cost relative to conventional AC units.[10] The gross M&V savings and net realization rates are lower than anticipated primarily due to electricity heating usage and lower energy efficiency performance based on field measurements of EER.

Table 3.4 Ex Ante Savings for Residential GSHP Program

Program	Qty.	Ex Ante Full-Year Unit kWh/yr	Ex Ante Unit kW	Ex Ante Net-Unit therm/y	Ex Ante Program Savings kWh/y	Ex Ante Program Savings kW	Ex Ante Program Savings therm/y
GSHP	27	16,883	3.85	n/a	455,841	104	n/a

Table 3.5 Ex Post Savings for Residential GSHP Program

Program	Qty.	M&V Full-Year Unit kWh/y	M&V Unit kW	M&V Full-Year Unit therm/y	Ex Post Program Savings kWh/y	Ex Post Program Savings kW	Ex Post Program Savings therm/y	Net Realization Rate kWh/y	Net Realization Rate kW	Net Realization Rate Therm/y
GSHP	27	-1,355	2.1	546	-36,587	56.1	14,745	-0.08	0.55	n/a

[9] The simple payback is based on REU electricity rates of $0.0848/kWh and PG&E natural gas rates of $1.11/therm (base) and $1.34/therm over base).

[10] The incremental cost is $14,986 based on average GSHP cost of $23,186 and conventional AC cost of $8,200. The average incentive offered by REU was $12,100. Cost data from Paul Ahern, Key Accounts Manager, REU, and Tim Camacho, General Manager, Palomar Builders, Redding, California.

Findings from this study indicate the GSHP units provide advantages for all participants. For the utility, the GSHP reduces peak demand in summer by an average of 2.1 ± 0.02 kW per unit and shifts summer cooling loads to winter increasing annual electricity use by 1,355 ± 841 kWh per year (roughly 10 percent). For the customer, the GSHP reduces annual energy bills for space conditioning by 48 percent saving $639 ± $185 per year. For society, the GSHP mitigates global warming by reducing carbon dioxide emissions for space conditioning by 44 ± 9 percent saving 59 ± 17 MMBtu per year of source energy per GSHP.[11]

3.3 Proper Refrigerant Charge and Airflow and Duct Leakage

Several studies indicate approximately 50 to 67 percent of new air conditioners have improper refrigerant charge and airflow (RCA), and this reduces efficiency by approximately 10 to 50 percent.[12] These studies also report considerable savings from reducing duct leakage. Three studies have shown that improper RCA can be mitigated by installing a TXV device.[13] The studies found TXV systems only had a clear advantage when the system is undercharged, and found no difference in performance at the rating condition between TXV and non-TXV (i.e., fixed orifice) when systems were properly installed. Unfortunately, TXVs can have their own performance problems associated with incorrect installation leading to a phenomenon known as "valve hunting." This can occur when the evaporator coil experiences reduced heat loads caused by many problems including low airflow, dirty or icy coils, and low refrigerant charge.[14] Under these circumstances the TXV can lose control and successively overfeed and then underfeed refrigerant to the evaporator while attempting to stabilize control causing reduced capacity and efficiency. Overfeeding liquid to the evaporator can also damage the compressor. The tendency for hunting can be reduced by correcting RCA, by relocating the TXV sensing bulb to a better location inside the evaporator coil box, and by insulating the sensing bulb.

Field and factory-installed TXV sensing bulbs are often installed without insulation (see **Figure 3.4**) or without adequate linear contact, and at incorrect orientations (see **Figure 3.5**). This practice is not recommended by manufacturers who instruct technicians to insulate the sensing

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[11] Source energy is valued at 0.010239 MMBtu/kWh and 0.1 MMBtu/therm.

[12] Downey, T., Proctor, J. 2002. "What Can 13,000 Air Conditioners Tell Us?" In the *Proceedings of the 2002 ACEEE Summer Study on Energy Efficiency in Buildings.* 1:53-67. Washington, D.C.: American Council for an Energy-Efficient Economy. Palani, M., O'Neal, D., and Haberl, J. 1992. *The Effect of Reduced Evaporator Air Flow on the Performance of a Residential Central Air Conditioner*, The Eighth Symposium on Improving Building Systems in Hot and Humid Climates. Parker, D. 1997. *Impact of Evaporator Coil Air Flow in Residential Air Conditioning Systems,* FSEC-PF-321-97. Cocoa, Fla.: Florida Solar Energy Center. Rodriguez, A. 1995. *The Effect of Refrigerant Charge, Duct Leakage, and Evaporator Air Flow on the High Temperature Performance of Air Conditioners and Heat Pumps*, Palo Alto, Calif.: Electric Power Research Institute.

[13] Farzad, M., O'Neal, D. 1993. "Influence of the Expansion Device on Air Conditioner System Performance Characteristics Under a Range of Charging Conditions." Paper 3622. *ASHRAE Transactions*. Atlanta, Ga.: American Society of Heating Refrigerating and Air-Conditioning Engineers. Davis, R. 2001a. *Influence of the Expansion Device on Performance of a Residential Split-System Air Conditioner.* Report No.: 491-01.4. San Francisco, Calif. Pacific Gas and Electric. Davis, R. 2001b. *Influence of Expansion Device and Refrigerant Charge on the Performance of a Residential Split-System Air Conditioner using R-410a Refrigerant.* Report No.: 491-01.7. San Francisco, Calif.: Pacific Gas and Electric.

[14] Tomczyk, J. 1995. *Troubleshooting and Servicing Modern Air Conditioning and Refrigeration Systems*. ESCO Press. Mt. Prospect, Ill.: Educational Standards Corporation.

bulb to prevent ambient air from causing false readings, and to tightly clamp the bulb to the vapor line with good linear thermal contact at the recommended orientation (i.e., 4 or 8 'o'clock when viewed in cross section) to guard against false readings due to air or liquid in the suction line.[15] Unfortunately, most installers do not do this. Field inspections at the sites in this study found sensing bulbs installed with insulation, but they were installed outside the evaporator coil box in hot attics on the vapor line two feet from the vapor manifold.

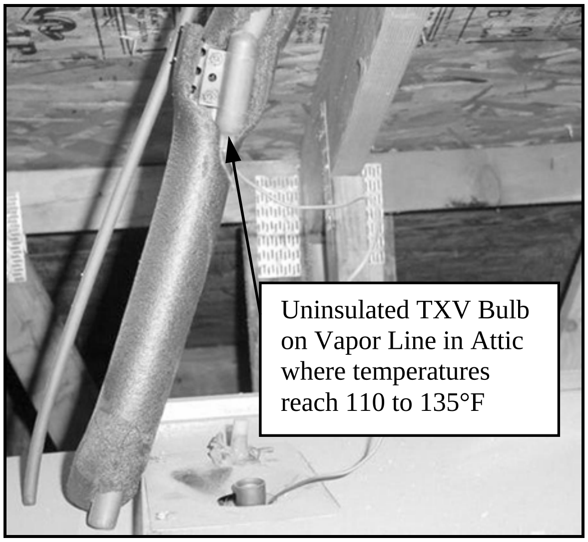

Figure 3.4 Uninsulated TXV Bulb in Attic

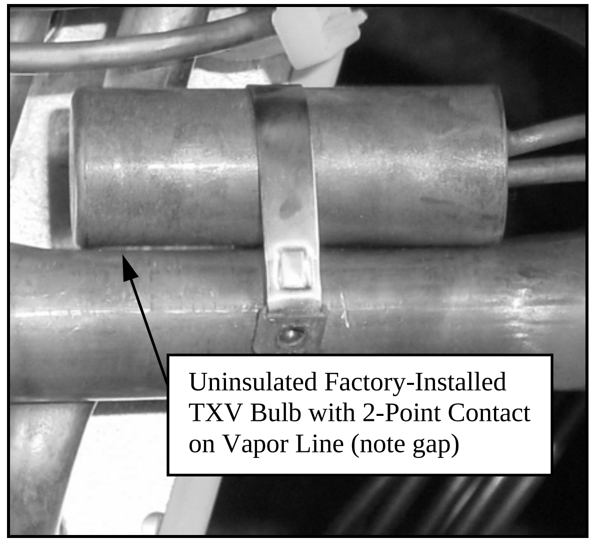

Figure 3.5 Uninsulated Factory TXV Bulb

Factory-installed TXVs with uninsulated sensing bulbs inside the evaporator coil box will be influenced by the mixed supply-air temperatures which are typically 10-20°F higher than vapor line temperatures. Field-installed TXVs with uninsulated sensing bulbs located in attics or garages will be influenced by attic or garage temperatures which are 50 to 80°F higher than vapor line temperatures (e.g., attic temperatures range from 110 to 130°F compared to vapor line temperatures of 35 to 50°F). The three laboratory studies (mentioned above) measured TXV-equipped air conditioners with the evaporator coil box, TXV, and well-insulated sensing bulb located in conditioned space and this is not typical of field conditions. Furthermore, none of these three studies recommended TXVs as a substitute for proper RCA.

Other studies found average load impacts for proper RCA of 266 ± 60 kWh for TXV and 388 ± 133 kWh for non-TXV air conditioners or 15.2 ± 6.8 percent for TXV and 9.2 ± 2.5 percent for non-TXV units.[16] The relative efficiency gains versus electricity savings for both systems are

[15] Advanced Distributor Products (ADP). 2003. *TXV Installation Instructions.* 0991710-01 Rev 1, October 03. Stone Mountain, Ga.: Advanced Distributor Products, Available online: www.adpnow.com. AllStyle Coil Company, L.P. (Allstyle). 2001. *Evaporator Coil Installation Instructions.* Brittmore, Texas: AllStyle Coil Company, L.P. Carrier Corporation (Carrier). 2002. *Installation Instructions: Thermostatic Expansion Valve Kit.* KAATX, KHATX (R22), KSATX----PUR (R410a). Syracuse, N.Y.: Carrier Corporation. Emerson Climate Technologies, Inc. 1998. *Installation Instructions Expansion Valve Kits TXV153 & TXV355.* Lewisburg, Tenn.: Emerson Climate Technologies, Inc.

[16] *Field Measurements of Air Conditioners with and without TXVs,* Mowris, R., Blankenship, A., Jones, E., 2004 ACEEE Summer Study on Energy Efficiency in Buildings, August 2004.

shown in **Figure 3.6**. Average load impacts for proper refrigerant charge and airflow are approximately 316 kWh per year and 0.31 kW. This study found average load impacts for proper RCA of 1,095 kWh per year and 0.38 kW. These savings are greater than the average values due to one unit being undercharged by 49.4%. The study also found average duct leakage for all sites of 15.5%. This is 2.6 times higher than the California Energy Commission target value of 6%. The average energy savings found in this study for tight ducts are approximately 337 kWh per year, 0.2 kW, and 38 therms per year. Builders and air conditioning dealers interviewed for this study indicated support for a program involving verification of proper RCA and duct sealing.

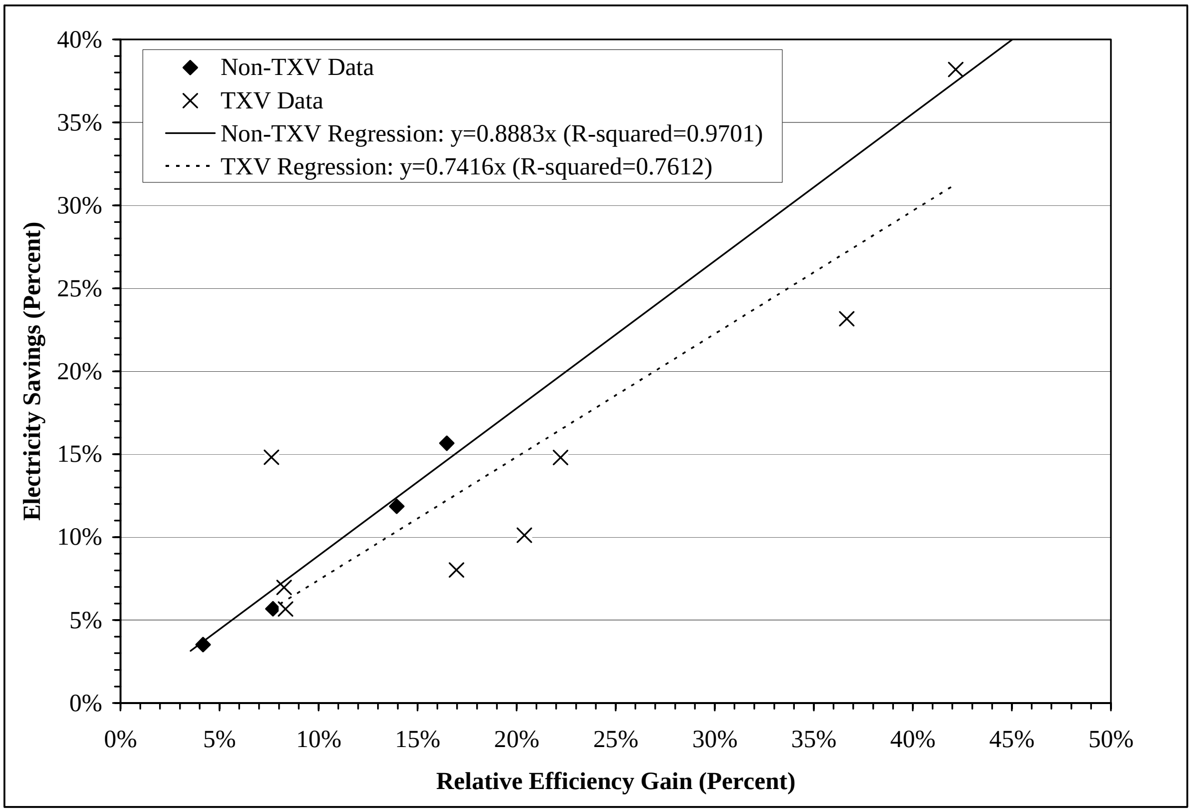

Figure 3.6 Relative Efficiency Gain Vs. kWh Savings for TXV and non-TXV Systems