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Status Report on Survey of Alternative Heat Pumping Technologies

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ABSTRACT

The Department of Energy is studying alternative heat pumping technologies to identify possible cost effective alternatives to electric driven vapor compression heat pumps, air conditioners, and chillers that could help reduce CO_2 emissions. Over thirty different technologies are being considered including: engine driven systems, fuel cell powered systems, and alternative cycles. Results presented include theoretical efficiencies for all systems as well as measured performance of some commercial, prototype, or experimental systems. Theoretical efficiencies show that the alternative electric-driven technologies would have HSPFs between 4 and 8 Btu/Wh (1.2 to 2.3 W/W) and SEERs between 3 and 9.5 Btu/Wh (0.9 and 2.8 W/W). Gas-fired heat pump technologies have theoretical seasonal heating gCOPs from 1.1 to 1.7 and cooling gCOPs from 0.95 to 1.6 (an SEER 12 Btu/Wh electric air conditioner has a primary energy efficiency of approximately 1.4 W/W).

INTRODUCTION

The Department of Energy is studying alternative heat pumping technologies to identify possible cost effective alternatives to electric driven vapor compression heat pumps, air conditioners, and chillers that could help reach goals to reduce CO_2 emissions nationwide. This review is a status report and includes technologies that have been investigated in the past to determine if recent work could make them viable competitors to vapor compression today or in the future. It also includes refrigeration technologies that have been developed for applications other than space conditioning to see if they could be adapted to this application. Over thirty different technologies are being considered including: engine driven (IC, diesel, Stirling, Brayton, Rankine) systems, fuel cell powered systems, alternative cycles (Rankine, Stirling, Brayton, adsorption, absorption, Vuilleumier), and alternative technologies (thermoacoustic, pulse tube, vortex tube, thermoelectric, magnetic).

Current work has focused on the operating efficiencies of each of these technologies. A future effort will look at estimated life cycle costs of the technologies. Very few of these systems have been developed to the point where efficiencies have been measured at the ARI rating conditions; seasonal performance data are available for even fewer of them. The results presented in this paper are structured in three levels: 1) theoretical cycle COPs at the ARI rating conditions, 2) theoretical system COPs that include power requirements for fans, blowers, pumps, etc. and estimated seasonal performance factors using the calculated system COPs, and 3) measured steady-state and seasonal efficiency data for laboratory prototype, field test, or commercially available systems.

The basis of the cycle and system COPs presented in this paper is explained if the data are computed by the author and references are given for data computed or measured by others. Entries in the tables are left blank if values have not been determined which are considered "comparable" to the efficiencies provided for other technologies. "Comparable" in this sense is taken to mean similar or realistic component efficiencies and operating conditions. Some of the data located in the open literature are either idealized to such a degree or are at operating conditions that it is inappropriate for them to be reported side-by-side with the other data in the tables.

Information is presented separately for electric-driven technologies and gas-fired technologies. Some of the cycles which may be unfamiliar to the reader are described, but it is presumed that the reader is familiar with most of the systems discussed in this paper.

ELECTRIC-DRIVEN HEAT PUMPS

Cycle Efficiencies

Reverse-Rankine Cycle: theoretical steady-state COPs are computed for R-22 and R-290 in a conventional electric heat pump using the refrigerant properties and operating conditions listed as footnotes in Table 1. The corresponding COPs are listed in the first two rows of Table 1 based on a combined compressor and motor efficiency of

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70%. The results for R-290 assume that the mechanical package is located outdoors and that there is a secondary heat transfer loop between the supply air ducts and the condenser. In this case the condensing temperature is assumed to be 5°F (2.8°C) higher in heating mode and the evaporating temperature 5°F (2.8°C) lower in cooling mode so the air-side conditions would be the same as for the non-flammable fluorocarbon heat pump. Measured COPs and rated system effiencies are listed for R-22.

Transcritical CO_2 Cycle: theoretical efficiencies are calculated for a heat pump using carbon dioxide as the refrigerant for a transcritical cycle including an internal heat exchanger transfering heat from the gas cooler discharge line to the compressor suction line. High-side conditions are fixed at the compressor discharge pressure that gives the highest cooling COP. The evaporating temperature is assumed to be 9°F (5°C) below the entering air temperature; the gas cooler exit temperature is calculated using the effectiveness of the indoor (0.90) or outdoor (0.95) heat exchanger for heating and cooling. The compressor efficiency is assumed to be 65%.

Brayton Cycles: although the Brayton cycle has been examined extensively by others for use in air conditioning applications, little data have been found for either measured or theoretical efficiencies at ARI rating conditions. The results presented in Table 1 are from a model written by Don Gauger (1993) using compressor and expander efficiencies of 85% and regenerator effectiveness of 0.85. Heat exchanger ΔTs are set at 9°F (5°C).

Stirling Cycle: Table 1 also contains cooling COPs for a Stirling cycle air conditioner. Gauger's dissertation, includes results from Kelly's model for cooling mode that can be used to estimate performance at 95°F and 82°F (35°C and 28°C). They assume compression and expansion efficiencies of 85%, a regenerator effectiveness of 0.85, and heat exchanger ΔTs of 18°F (10°C). Heating COPs cannot be backed out of the data presented by Gauger.

Thermoacoustic cooling: thermoacoustic refrigerators employ a resonating chamber in conjunction with an electro-acoustic driver (e.g. loud speaker) and a regenerator or "stack" separating hot and cold heat exchangers. A standing sound wave is used to create a thermal gradient along the stack and the heat exchangers are added to provide heat input and output from the system.While early units had capacities measured in tens of Watts, a three ton system is under development at Penn State.

Very little efficiency data, theoretical or observed, are available on thermoacoustic refrigerators. The systems constructed for cryogenic applications understandably have very low COPs. Recently prototypes have been built and tested claiming COPs of up to 16% of Carnot (Garrett and Hofler, 1992), but the operating conditions were not reported. A thermoacoustic "refrigerator" was built for a Space Shuttle experiment in 1992 that provided cooling at -112°F (-80°C). It later evolved into a multi-purpose cooler to provide for storage of biological samples which is also capable of domestic refrigeration. Minner (1997) performed an analytical optimization of the predecessor to the Space Shuttle cooler to show that the efficiency could be improved two-fold by changes in the geometric parameters (providing 20 Btu/h (6 W) of cooling at cold-side and hot-side wall temperatures of -46°F and 80°F (-43°C and 27°C), respectively).

Theoretically the thermoacoustic and pulse tube systems undergo isentropic compression, isobaric cooling, isentropic expansion, and isobaric heating; the ideal cycle formed by a series of processes undergone by control masses is a reversed Brayton cycle. Gauger developed his model for these idealized cycles using the regenerative Brayton cycle. The results in Table 1 are from his model using 9°F (5°C) ΔTs for both heat exchangers, a pressure ratio of 2.5, compressor and expander efficiencies of 80%, and a regenerator effectiveness of 0.8. These calculations are idealized and unlikely to benefit from the type of optimization demonstrated by Minner; the pressure ratio is optimal for this model and improvements would be needed in the ΔTs and component efficiencies to achieve higher COPs.

Pulse-tube refrigeration: a pulse-tube refrigeration system uses a rotary compressor and control valve or a reciprocating compressor with a high temperature heat exchanger, regenerator, and a multi-component tube. The heat of compression is rejected to the ambient at the high temperature heat exchanger. Additional energy is stored in the regenerator and recovered during the oscillating cycle of the working fluid. The "pulse tube" itself is constructed of high conductivity materials at each end to accept and reject heat with a low conductivity material in the tube wall to minimize conduction along the tube. The wall itself acts as a regenerator as the gas is compressed and expanded in the tube. Prototype pulse tube refrigerators have been built that produce 3.4 to 17 Btu/h (1 to 5 Watts) of capacity at -397°F to -360°F (35 to 55 K). No information has been located on higher temperature applications. The results in Table 1 are from Gauger's model as described for thermoacoustic refrigeration.

Magnetic refrigeration: magnetic cooling is based on entropy changes in a magnetocaloric material such as gadolineum as a magnetic field alternates between 0 and 5 to 7 Tesla. Devices using this concept have been built to achieve temperatures below -450°F (5 K). Superconducting magnets are required to reach the magnetic field strengths necessary to make the magnetic cycle viable. The performance of these systems is heavily dependent on the temperature lift with maximum cycle COPs of 8 for a lift of 14°F (8°C) falling to 1 at a lift of 41°F (23°C) (Zimm 1997). Little information was found on modeling magnetic cycles; Gauger presents a model and results at 95°F (35°C), but cautions that realistically the COP is going to be much lower because the model does not include regenerator losses or energy requirements necessary to achieve and maintain 7 Tesla fields.

Vortex tubes: vortex tubes are a relatively recent discovery (1930) that have been developed into commercial products for industrial spot cooling applications. Compressed air (e.g. 100 psi, 70°F; 690 kPa, 21°C) is forced into a cylindrical generator where it is separated into hot and cold streams. The air is forced to rotate at speeds up to 1,000,000 rpm along the outside of the tube. The flow is split, with part of the air exiting through a needle valve at high temperature (e.g. 212°F; 100°C) while the remainder flows back down a helical coil through the center of the tube exiting at low temperature at the opposite end (e.g. -40°F; -40°C). COPs of vortex tube coolers at 100 psi (690 kPa) are approximately 0.08 for an SEER of 0.3 Btu/Wh (0.08 W/W) (ITW Vortec 1996, ACP 1998).

Thermoelectric cooling: the direct conversion of electricity to useful heating or cooling based on the Peltier effect is well known and has been developed commercially ranging in capacity from space conditioning for an office building in the 1950s to low capacity portable coolers and circuit boards heat sinks in the 1990s. Marketable consumer products have been produced, but they have low COPs at the current state of the art. Breakthroughs are needed in semiconductor pairs in order to achieve efficiencies that could be competitive with conventional compression systems. The efficiency of the thermoelectric pair is measured by its figure of merit, Z; presently the highest value of Z obtained in a laboratory thermoelectric couple is approximately 0.0018/°F (0.00325 K^{-1}), the COPs shown in Table 1 are computed for a commercially available semiconductor pair with 0.0015/°F (Z=0.00274 K^{-1}) with heat exchanger ΔTs of 9°F (5°C).

Malone cycle: very little development work has been done on the Malone cycle. This cycle employs the little known property of the compressibility of a liquid near its critical point. Malone cycle machines have been built around both Stirling and Brayton cycle cycle configurations. A Malone/Stirling refrigerator was built at a national laboratory using liquid propylene as the working fluid. This system was designed for ease of instrumentation, rather than high efficiency, and reportedly had a COP about half that of a conventional refrigerator using CFCs. Subsequently a 6800 Btu/h (2 kW) free-piston Stirling cycle Malone system using liquid CO_2 as the refrigerant but performance data are not available. A 11.4 ton (40 kW) liquid CO_2 Brayton cycle Malone system was under development in Annapolis (Swift 1993); the current status of this project is unknown.

Compressor driven metal hydride adsorption: conventional adsorption cycles use combustion of natural gas or propane to provide the energy input necessary to regenerate the adsorption beds. A metal hydride heat pump is being developed that uses an electrically-driven compressor to create a pressure differential to drive the desorption process. Measurements on a water-to-refrigerant prototype heat pump confirmed analytical predictions, although the overall system efficiency was very low because of a low compressor efficiency (31%). Efficiencies are expected to be much higher with a compressor optimized for use with hydrogen, although additional losses will be incurred using air-to-refrigerant, or adding an air-to-water, heat exchanger.

Calculated System Seasonal Efficiencies

The steady-state cycle efficiencies are modified using Equ. 1 to estimate system efficiencies for each system. Parasitic energy use is included for the outdoor fan and blower powers (140 and 70 W/ton; 40 W/kW and 20 W/kW, respectively) plus energy use for a secondary fluid pump in the case of the hydrocarbon refrigerant (100 W/ton; 28 W/kW). Seasonal performance

$$COP_{system} = \frac{1}{\dfrac{1}{COP_{cycle}} + \dfrac{P_{blower} + P_{fan} + P_{pump}}{3516}} \quad (1)$$

factors, HSPF and SEER, are then computed according to DOE rating algorithms using a computer program by Domanski and Pannock. The cycling parameter Cd is set to 0.25 in each case and system capacities are assumed to be equal to those of a commercially available SEER 12 Btu/Wh (3.5 W/W) electric heat pump. These results are also listed in Table 1.

GAS-FIRED HEAT PUMPS

Engine-Driven Heat Pumps

Cycle efficiencies are estimated for IC-engine driven heat pumps, Stirling engine, Brayton engine, and Rankine-engine driven heat pumps. Steady-state COPs are calculated in the same manner as they are for the electric-driven heat pump COPs in Table 1. Gas COPs are computed by Equ. 2 where $\alpha = 1$ for heating and 0 for cooling. The waste heat recovery efficiency, $\eta_{recovvery}$, is set at 0.50. Similar results are shown in Table 2 for Stirling and Brayton engine driven reverse-Rankine cycle heat pumps, a fuel cell powered heat pump, and a gas furnace/electric air conditioner combination with primary energy COPs. Data on engine efficiencies are from Miyairi (1989), Kazuta (1989), Volvo (1998), Nowakowski (1995), Monahan (1987), Brodrick (1987), Garrett/AiResearch (1982), and MTI (1986).

Significant development efforts have gone into engine driven heat pumps with prototypes being built and tested. Table 2 also contains results for comparison with the theoretical values. Some of the theoretical values are in close agreement with what was achieved in prototype equipment, some are not.

$$gCOP_{system} = \eta_{engine} \cdot COP_{cycle} + \alpha \cdot \eta_{recovery} \cdot \left(1 - \eta_{engine}\right) \qquad \textbf{(2)}$$

The seasonal efficiencies of engine-driven heat pumps listed in Table 2 are computed using the steady-state gCOPs using the same procedure employed for estimating electric heat pump HSPFs and SEERs. In this instance, the cycling parameter Cd is set to 0.10 to account for modulation of the engine speed to follow the building load. This comparison also shows some good and some poor agreement between the theoretical and observed performance data.

Vuilleumier cycle: a Vuilleumier machine resembles a duplex Stirling machine; the principal difference between these two systems is the use of pistons by the duplex Stirling system to maintain large pressure differences in the cylinders while the Vuilleumier machine uses displacers in the cylinders for small or 0 ΔPs. A gas burner is used at the high temperature heat exchanger in the Vuilleumier cycle to raise the pressure in the cylinders by increasing the temperature of the working fluid. A small amount of mechanical energy is applied using a motor and crank mechanism to move the displacers shifting the working fluid from one chamber to another. Heat is absorbed at the low temperature heat exchanger, providing useful cooling in air conditioning mode; heat is rejected at the intermediate heat exchangers, to the ambient for air conditioning or into the conditioned space for heating. Variations of the basic Vuilleumier cycle include free-piston configurations and constant volume machines that employ pistons and mechanical- rather than thermo-compression. Vuilleumeir cycle heat pumps were built and tested in a field demonstration in Japan. Data from Gauger's model and the Japanese field test are listed in Table 2.

Absorption and adsorption cycles: both absorption and adsorption systems are also under development and much data on theoretical and measured efficiency appear in the open literature; operating conditions are not always given. The Generator/Absorber Exchanger (GAX) absorption cycle has been developed and is commercially available as a chiller for cooling applications and is under development as a heat pump for both heating and cooling. Double-effect absorption chillers are successful commercial products and triple-effect chillers are being developed. Data from the literature are listed in Table 2 for single-, double-, triple-, and GAX absorption cycles. Independent theoretical calculations of the absorption systems has not been performed by the author. Different adsorbent/-adsorbate combinations are being investigated for adsorption cycles by different individuals and organizations to take advantage of the regeneration temperature, cost, and size advantages of each. A discussion of the relative merits of the different pairs is beyond the scope of this paper; the efficiency data listed in Table 2 is from the literature.

DISCUSSION

Significant R&D efforts have been directed toward commercializing alternative electric-driven space conditioning technologies in the past, but the Rankine cycle system is the only one that has achieved significant success (in air-to-air, water-source, and geothermal). Rankine cycle heat pumps using either hydrocarbons or CO_2 as the refrigerant have lower efficiencies than comparable equipment using conventional non-flammable refrigerants. Vortex tubes have been developed into successful commercial products for spot cooling applications, but they are not viable for large loads. Capacities are constrained by the technology itself to below about 6,000 Btu/h and the efficiencies are very low. Brayton cycle equipment has been investigated for space conditioning, commercial water heating, automobile air conditioning, air conditioning on passenger trains, and transport refrigeration. It has the advantage of light weight for transport applications but suffers from low efficiency. The Stirling cycle was revisited repeatedly by R&D projects attempting to apply it to consumer products for space conditioning or refrigeration. This cycle has advantages over the Rankine cycle for cryogenic applications, and projects targeting household refrigeration have concluded that it can be as good as the Rankine cycle, but the electrically-driven Stirling cycle does not compare well at the lower lifts for comfort heating and air conditioning. More effort has been directed toward the gas-fired duplex Stirling configuration for the space conditioning application. The low efficiencies or very early stages of development of the remaining electrically-driven technologies makes it difficult for them to compete effectively with Rankine cycle heat pumps in the foreseeable future.

Extensive development efforts have also gone into gas-fired heat pumping technologies. Successful commercial consumer products have been developed for engine-driven chillers and refrigeration systems and absorption chillers. Engine-driven unitary systems are also marketed for commercial and residential applications; they are gaining market share in Japan but having difficulty competing with lower priced electric-driven equipment in the U.S. Stirling, Rankine, and Brayton engines have not succeeded in supplanting IC engines (Otto and Diesel) despite potential advantages in emission reductions and potentially longer engine life. GAX cycle absorption machines are also being developed for this application and will be competing with both electric- and engine-driven machines.

CONCLUSION.

A literature search and computer simulations have been used to tabulate theoretical and measured efficiencies for electrically-driven and gas-fired heat pumps. These data will eventually be used to estimate life-cycle costs for the alternative technologies represented and compared with the life cycle costs for conventional compression systems. There are conspicuous gaps in the tabulated results which dictate the methods that can be used for the economic analysis. The most consistent sets of information are the theoretical values for steady-state and seasonal performance. These should be compared wherever possible with measured efficiencies for commercial products or laboratory prototypes to ensure that the modeled results are accurate.

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Table 1. Theoretical Cycle, System, and Seasonal Efficiencies and Measured Efficiencies for Electric-Driven Heat Pumps.

Technology	Theoretical: Cycle Only				Theoretical: System**						Measured: Prototype or Commercial Unit					
	47°F 8.3°C	17°F -8.3°C	95°F 35°C	82°F 28°C	47°F 8.3°C	17°F -8.3°C	95°F 35°C	82°F 28°C	HSPF Btu/Wh	SEER Btu/Wh	47°F 8.3°C	17°F -8.3°C	95°F 35°C	82°F 28°C	HSPF Btu/Wh	SEER Btu/Wh
Rankine Cycle HP†	5.09	4.74	3.98	5.30	3.90	3.69	3.20	4.02	9.0	12.2	3.84	2.60	2.66		8.3	12
Rankine Cycle Hydrocarbon HP††	4.46	4.18	3.28	4.29	3.19	3.04	2.54	3.10	8.2	9.2						
Thermoacoustic HP	1.45	1.46	0.45	0.48	1.28	1.29	0.43	0.46	3.8	1.4						
Pulse Tube HP	1.45	1.46	0.45	0.46	1.28	1.29	0.43	0.46	3.8	1.4						
Thermoelectric HP	2.41	1.53	1.99	3.53	2.09	1.39	1.76	2.88	4.6	8.8						
Brayton Cycle	1.94	1.83	0.98	1.10	1.70	1.62	0.91	1.02	4.5	3.1						
Transcritical CO_2 Rankine Cycle HP	4.48	2.65	1.91	2.77	3.48	2.26	1.70	2.35	7.8	6.9						
Stirling Cycle HP			1.59**	2.17**			1.44**	1.90**		5.8***						
Vortex Tube													0.08	0.08		0.3
Malone Cycle HP																
Magnetic HP																
Compressor Driven Metal Hydride			3.0													

** auxiliary power requirements based on indoor blower power of 0.140 kW/ton (40 W/kW) and outdoor fan power of 0.07 kW/ton (20 W/kW) scaled by the relative heat rejection requirement (baseline cycle COP 4.45), secondary fluid pump power of 0.100 kW/ton (28 W/kW), HSPF and SEER estimated using spreadsheet based on algorithms in Domaniski computer codes.

*** COPs for Stirling cycle extrapolated / interpolated from Kelly's data in Gauger's dissertation; SEER computed using Domanski algorithms.

† 47°F (8.3°C) 100°F (38°C) condensing, 29°F evaporating(-1.7°C), 17°F (-8.3°C), 85°F (28°C) condensing, 10°F (-12°C) evaporating, 95°F, (35°C) 120°F (49°C) condensing, 49°F (9°C) evaporating, 82°F (28°C), 105°F (40°C) condensing, 49°F (9°C) evaporating; 70% compressor efficiency; superheat 18.4°F (10°C) heating, 16 °F (9°C) cooling; subcooling 11°F (6°C) heating, 0°F (°C) cooling

†† 47°F (8.3°C) 105°F (40°C) condensing, 24°F (-4°C) evaporating, 17°F (-8.3°C), 90°F (32°C) condensing, 5°F (-15°C) evaporating, 95°F (35°C), 125°F (52°C) condensing, 44°F (6.7°C) evaporating, 82°F (28°C), 110°F (43°C) condensing, 44°F (6.7°C) evaporating; 70% compressor efficiency; superheat 18.4°F (10°C) heating, 16°F (9°C) cooling; subcooling 11°F (6°C) heating, 0°F (0°C) cooling

Table 2. Theoretical Cycle and System and Observed Efficiencies for Gas-Fired Heat Pumps.

	Theoretical: Cycle Only					Theoretical: System		Electrical Auxiliaries (W/ton)	Measured: Prototype or Commercial Unit					
Technology	$\epsilon_{conversion}$	47°F 8.3°C	17°F -8.3°C	95°F 35°C	82°F 28°C	Heating Btu/Wh	Cooling Btu/Wh		47°F 8.3°C	17°F -8.3°C	95°F 35°C	82°F 28°C	HSPF Btu/Wh	SEER Btu/Wh
Gas Furnace / Electric A/C	0.80			1.18††	1.57††	0.80	1.36				0.79‡		80%	1.04‡
IC Engine Rankine Cycle HP	0.30	1.44	1.36	0.97	1.27	1.26	1.19	100-133*	1.44		0.90		1.27	1.05
Stirling Engine Rankine Cycle HP	0.28†	1.25	1.19	0.77	1.01	1.12	0.95		1.09**	1.08**	0.42**	1.0**		
Brayton Engine Rankine Cycle HP	0.27	1.35	1.28	0.88	1.14	1.19	1.07		1.30		1.00			
Vuilleumier Heat Pump	0.80	1.13	1.12	0.19	0.23	1.0	0.2		1.34		0.63			
Absorption: Single Effect	0.80				0.62		0.62	12‡‡			0.60			
Adsorption: Water / Zeolite						1.8	1.2				1.2			
Diesel Engine Rankine Cycle HP	0.35	1.60	1.51	1.14	1.48	1.37	1.39						1.3	0.9
Rankine Engine Rankine Cycle HP	0.24	1.25	1.19	0.78	1.01	1.13	0.95							
Fuel Cell / Rankine Cycle HP	0.40	1.75	1.65	1.30	1.69	1.47	1.59							
Adsorption: Ammoniated Carbon		1.9	1.5	1.0	1.3	1.30	1.00	185						
Absorption: GAX	0.80	1.2			0.80	1.2	0.80							
Adsorption: Complex Compounds				1.0		1.7	1.0							
Absorption: Open Cycle	1.0	1.4			0.50	1.4	0.5							
Adsorption: Open Cycle														
Duplex Stirling HP														
Ejector		1.3		0.3										
Absorption: Double Effect	0.80				0.96		1.07	25-35‡‡			1.0			
Absorption: Triple Effect	0.80			1.20			1.20	100‡‡						

* measured electrical auxiliary power for engine-driven heat pump (source: Nowakowski 1995, p.1385).
** Stirling engine driven unitary heat pumps (Source: Brodrick 1987).
† excludes assumed 85% transmission efficiency
†† cycle COPs converted to gCOPs based on 11,500 Btu/h (3.37 kW) power plant input to generate 1 kW electricity delivered
‡ results for electric a/c converted to gCOPs using 11,500 Btu/h (3.37 kW) power plant input to generate 1 kW electricity delivered and SEER converted to SCOP
‡‡ solution pumping power only, does not include air handling requirements; single-effect per DeVault, double-effect per manufacturer literature