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oal, for example, can be burned in a power plant roduce steam for conversion into electric power. the resulting ashes and waste heat cannot be coled and burned to produce yet more electricity. The lity of the energy in the ashes and heat is not high ugh for further such use.

Numerous studies have indicated that the United es has enormous reserves of fossil fuels which can vide centuries of energy for an expanding economy, yet few take into account the thermodynamic itations on mining the fuels left. Most cheap and essible fossil fuel deposits have already been exted, and the energy required to fully exploit the may be equal to the energy contained in them. at is significant, and vital to our future, is the *net* rgy of our fuel resources, not the *gross* energy. Net rgy is what is left after the processing, concentrat- and transporting of energy to consumers is subted from the gross energy of the resources in the und.

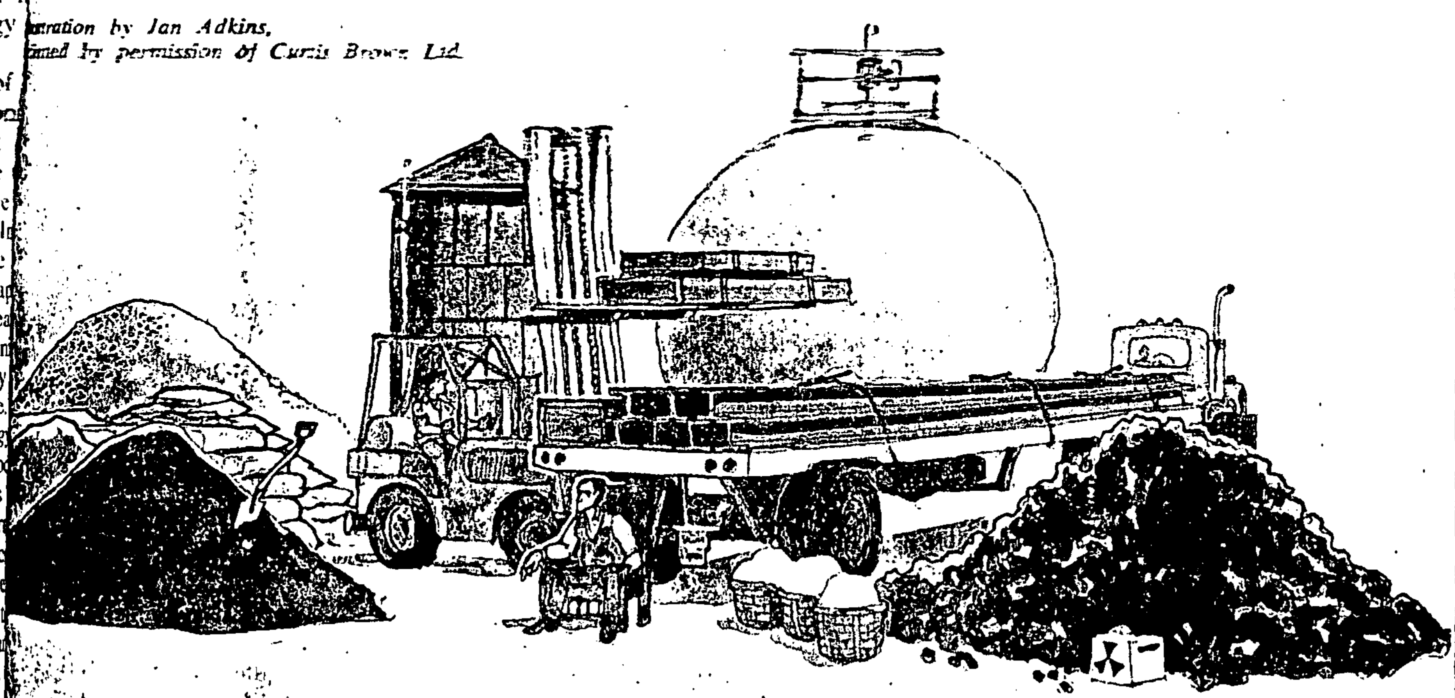
Consider the drilling of oil wells. America's first oil l was drilled in Pennsylvania in 1859. From 1860 1870, the average depth at which oil was found was feet. By 1900, the average find was at 1,000 feet. 1927, it was 3,000 feet; today, it is 6,000 feet. lling deeper and deeper into the earth to find tered oil deposits requires more and more energy.

The elephantine Great Easter, designed for nonstop voyages from England to Australia, required more coal than the ship could carry. Some experts are beginning to wonder if we aren't in the same boat.

Think of the energy costs involved in building the trans-Alaska pipeline (See SMITHSONIAN, October 1974). For natural gas, the story is similar.

Dr. Earl Cook, dean of the College of Geosciences at Texas A & M University, points out that drilling a natural gas well doubles in cost each 3,600 feet. Until 1970, he says, all the natural gas found in Texas was no more than 10,000 feet underground, yet today the gas reserves are found at depths averaging 20,000 feet and deeper. Drilling a typical well less than a decade ago cost $100,000 but now the deeper wells each cost more than $1,000,000 to drill. As oilmen move offshore and across the globe in their search for dwindling deposits of fossil fuels, financial costs increase, as do the basic energy costs of seeking the less concentrated fuel sources.

tration by Jan Adkins.
ted by permission of Curtis Brown Ltd.



maintain the average American at present comfort levels requires 600 lb of nonmetal resources such as sand, gravel, and salt; 0 lb of metal substances; and 18,600 lb of fossil fuel plus a e less than one ounce of uranium each year. That amount of energy, rly twice what the average European uses in a year, is the equivalent each citizen having 300 slaves working 24 hr per day.

Ferdinand Carré
1824–1900

Ammonia Absorption Cycle

e ammonia absorption cycle was e of the first methods employed r the production of refrigeration. e original systems were installed the early 1800's. A more ad-nced ammonia absorption stem was invented by Ferdinand rre, Paris, France in 1850. His ginal invention consisted of a rect fired generator, a condenser, d evaporator, an absorber, and an ua ammonia pump, all of which, h many improvements, remain main components of an am-onia absorption system. The iginal ammonia absorption stem was very inefficient and it s impossible to obtain ammonia uid at above 90% purity. This de the system difficult to erate as excess water collected the evaporator raising the aporation temperature. Further-re, early reciprocating machines ld not approach the compres-on ratios required for low perature applications. Thus, am-ia absorption found its place in refrigeration industry despite disadvantages. As time went on the reciprocating compressor was oved the ammonia absorption em of old passed into obsolescence. e mid-'30s, improved ammonia ab-tion systems were installed which ld operate on waste steam, waste or by direct firing with natural gas her gases. These system employed le column design and spray type ab-ers and provided ammonia at % purity for refrigeration duty. This nce in technology which provided igh purity (commercial grade) am-a overcame the major operating lems of the early systems. The more nt Aqua-Ammonia Absorption ems have several new concepts. g these are an ammonia refrigerant to gas heat exchanger and greater tion of the rectification column of which have further improved the n efficiencies. New and improved ol systems are also available to ize the utilization of valuable heat es.

CONSERVE ENERGY:

REFRIGERATE WITH WASTE HEAT

Refrigerating can be provided by using waste heat with the classic water-ammonia absorption cycle. This cycle was originally employed in the 1800's and has been refined over the years. It lost its economic value in the 1930's as the more efficient centrifugal and reciprocating compressor systems became less expensive. Due to its basic inefficiency, the ammonia absorption system cannot be justified unless low level waste heat is available such as low pressure steam or hot process streams. This paper provides a detailed cycle description including controls, utility requirements and potential heat sources.

GEORGE C. BRILEY
Member ASHRAE

THE cost of power has increased from an average of 0.8¢ per kWh in 1972 to 1.0¢ per kWh in 1974. Predictions are the cost will approach 3.0¢ per kWh during the next three to five years. Fuel costs have also increased at a rate of 2 to 3% each month. The availability of fuel has gone from abundant to scarce or not available with anyone's guess as to the future supply.

All energy-intensive industries have become energy-concious, and many companies have appointed "Energy Chiefs" or "energy committees" charged with the responsibility of reducing cost and consumption. Increasing attention is given to operating efficiency, maintenance and reliability costs. The total net consumption of resources, both human and material has become one of the most important evaluation criteria in equipment selection, particulary in energy intensive processes.

If your plant requires refrigeration you should give consideration to the Aqua Ammonia Absorption System, particularly where waste heat or a high temperature process stream is available (Fig. 1).

G. C. Briley, formerly Executive Vice President, Lewis Refrigeration, Houston Tx, is Vice President of Refrigeration Engineering Co., San Antonio, Tx.

THE SYSTEM

The generator (G-1) is considered to be the heart of the system. It receives strong aqua, (preheated in the aqua heat exhangers), as primary feed and also the returns from the rectification column. These strong aqua and rectification column return are heated by steam or any other type of heat available. This causes a part of the ammonia within the aqua to be vaporized. This vapor flows to the rectification column entering below the bottom plate. The weak aqua is taken from the low end of the generator and heat exchanged with the strong aqua coming from the strong aqua tank.

Rectification column (C-1) utilizes bubble trays to rectify the ammonia (strip it of all of its water vapor). Enough strong aqua is fed into the column at one or two predetermined plates to bring the plates into equilibrium with the strong aqua with the balance directed to the generator. Pure anhydrous ammonia is fed to the top plate from the reflux control receiver (RC-1) which receives condensed liquid ammonia from the condenser (CO-1). This rectification normally provides 99.96% ammonia vapor to the condensor (CO-1).

REFRIGERATE WITH WASTE HEAT

The Ammonia condenser (CO-1) condenses the vapors from the rectification column using primary water. The condensed anhydrous ammonia flows to the reflux control receiver (C-1) providing an accurate temperature reading to the reflux controller. The ammonia flows from the reflux control receiver (C-1) into the ammonia receiver (R-1). The receiver (R-1) is sized to provide storage for ammonia volume variations caused by load changes in the system.

Two reflux pumps (RP-1 and RP-2), (one a 100% spare) can be employed to pump liquid NH3 from the reflux control receiver (RC-1) to the top plate of the rectification column (C-1). If the condenser (CO-1) is high enough in the structure these pumps can be eliminated. Ammonia is expanded in the chiller (CH-1) to the pressure which corresponds approximately to that of the aqua film absorber and vaporizes as it absorbs heat from the cooled liquid or process gas.

The ammonia vapor from the evaporator (EV-1) flows to a refrigerant heat exchanger (HX-1) where it is used as an exhange medium with the liquid ammonia from the receiver thus heating the leaving vapor and cooling the incoming liquid ammonia. The heat ammonia gas from the refrigerant heat exchanger (HX-1) flows directly to the aqua film absorber (AB-1 and AB-2) which is maintained at a slightly lower pressure than the evaproator (EV-1). The ammonia vapor is absorbed into the weak aqua which has been cooled in the aqua heat exchanger (AHX-1). This weak aqua is fed into the spray area of the top aqua film absorbers (AB-1) and in turn flows to the spray area of the aqua film absorber (AB-2) during which time it absorbs the vaporized ammonia from HX-1. The ammonia vapor is directed into the side of AB-1 and AB-2. The unique aqua film absorbers have no static head penalty and respond rapidly to load fluctuations. When weak aqua and anhydrous ammonia gas combine, heat is generated. This heat of absorption must be dissipated. Water is cascaded from the outlet of the ammonia condenser (CO-1) to the aqua film absorbers (AB-1 & AB-2). It is also feasible to use primary water on the condenser (CO-1) and the aqua film absorbers (AB-1 & AB-2) to obtain higher operating efficiencies by reducing the required aqua film absorbers surface. This also reduces the input heat requirement in the generator —(G-1).

The strong aqua from the aqua film absorber flows to a strong aqua tank (SA-1) which is sized to provide for any strong aqua flow variations occuring within the system during load variations. The strong aqua from the strong aqua tank (SA-1) is pumped through the Aqua heat exchangers (AHX-1) by an aqua pump.

Two aqua pumps (AQ-1 & AQ-2) are usually installed (one acting as a 100% spare). The pump drives can be electric, air or steam turbine, or gas engine. These aqua pumps (AQ-1 & AQ-2) can either be of the horizontal split casing type or the vertical type. An aqua pump is the only moving part within the system and requires minimum maintenance. Teh vertical multistage pump si normally recommended due to its low NPSH requirements and excellent efficiency.

As the strong aqua leaves the aqua pump (AQ-1 & AQ-2) it is directed through the aqua heat exchanger (AHX-1) which may either double pipe or shell and tube. This heat exhanger (AHX-1) is critical to the economical operation of the system as it utilizes a large amount of heat that might normally be wasted. The strong aqua from the aqua heat exchanger (AHX-1) is divided proportionately to the rectification column (C-1) and the generator (G-1). The weak aqua from the generator (G-1) is cooled in the aqua heat exchanger (AHX-1) to within 20 to 30 degrees of the temperature of the strong aqua from strong aqua tank (SA-1).

The aqua ammonia absorption system can be made completely automatic with a control system described in Table 1.

Aqua ammonia absorption capacities have been designed in sizes from a minimum of 200 TR (2,400,00 Btuh) at -50 dF and 300 TR (3,600,000 Btuh) at +20 dF to amaximum 2500 TR (30,000,000 Btuh) at -50 dF and 5000 TR (60,000,000 Btuh at +20 dF. Most systems would employ shell and tube condensers and aqua film absorbers however, evaporative cooled absorbers have been used. Air cooled condensers and absorbers could also be used. Various schemes can be used for supplying the water requirements of the condensers and aqua film absorbers other than series flow. Parallel flow can be used to reduce the aqua film absorber size and the heat input to the system. With the parallel flow system a small water temperature rise can be employed and the water used elsewhere in the plant.

The heat source will govern the generator designs. They may be finned surface heat exchangers with aqua pumped through the tubes for vapor heating mediums or double pipe for liquid heating mediums.

ECONOMY

For economic justification an aqua ammonia system *must* be supplied with waste Btu's as the source of heat for the generator. From Table 2 the temperature level of the heat source required for single and two stage systems can be determined. Note that with a heat source of 220 dF -10 dF refrigeration can be obtained in a two stage system. The two stage syftem can be compared directly with a two stage reciprocating refrigeration system. Any source of heat can be utilized including low pressure including low pressure steam, hot oil, hot process streams, stack gas, process gas, etc. Gas fired systems are practical, however the economics are usually not good unless the products of combustion are process components that require cooling.

MAINTENANCE

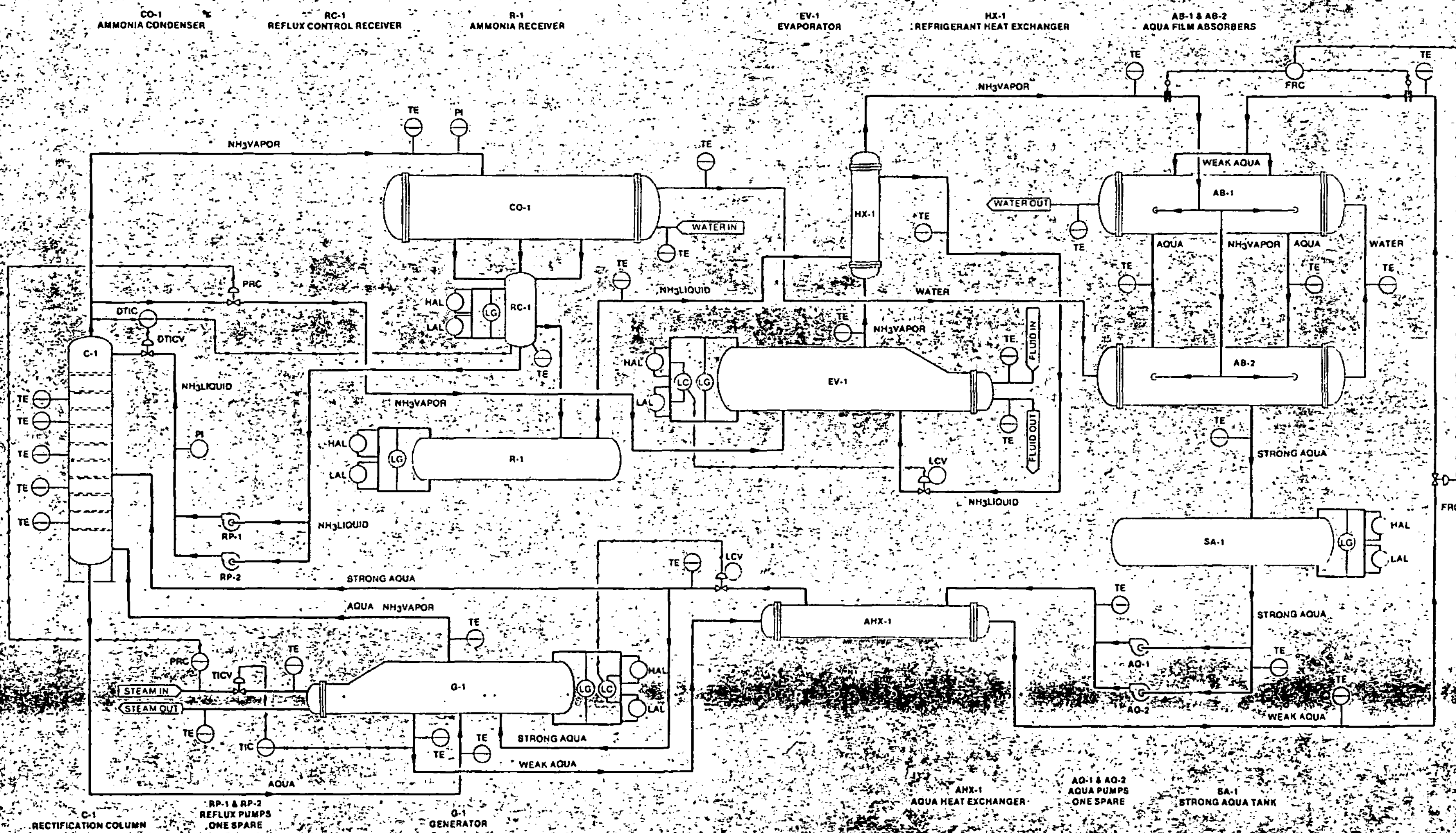
Since the number of moving parts is minimal (an aqua pump and possibly a small reflux pump) total maintenance costs are much less than comparable reciprocating or centrifugal compression systems. Tube cleaning is a normal function of all refrigeration systems however, it is estimated that tube cleaning costs for an absorbtion system employing plain tubes and temperatures less than 150 dF will require much less cleaning than temperatures exceeding 250 dF. Total maintenance cost are expected to be 10% of that of a comparable reciprocating compressor system. Continuous tube cleaning systems are available providing maximum operating efficiency at all times.

RELIABILITY

Aqua ammonia absorbtion systems are normally installed with spare aqua pumps and spare reflux pumps, (when used) offering a comparison to centrifugal and reciprocating systems that have a spare compressor-motor train. Down-time from failure of mechanical item is "zero" due to the 100% spare pumps. Heat exhanger designs are either ASME, TEMA B, C, or R as required by the system.

ADVANTAGES

The system is installed outdoors and requires minimal space. A typical 1500 TR system requires approximately 1600 sq. ft. of ground space. The



CO-1
AMMONIA CONDENSER
RC-1
REFLUX CONTROL RECEIVER
R-1
AMMONIA RECEIVER
EV-1
EVAPORATOR
HX-1
REFRIGERANT HEAT EXCHANGER
AB-1 & AB-2
AQUA FILM ABSORBERS
NH3VAPOR
NH3LIQUID
WATER IN
WATER OUT
WATER
WEAK AQUA
STRONG AQUA
AQUA
FLUID IN
FLUID OUT
STEAM IN
STEAM OUT
PRC
DTIC
DTICV
TICV
TIC
FRC
FRCV
LCV
HAL
LAL
LG
TE
PI
C-1
RP-1
RP-2
G-1
AHX-1
AQ-1
AQ-2
SA-1
C-1
RECTIFICATION COLUMN
RP-1 & RP-2
REFLUX PUMPS
ONE SPARE
G-1
GENERATOR
AHX-1
AQUA HEAT EXCHANGER
AQ-1 & AQ-2
AQUA PUMPS
ONE SPARE
SA-1
STRONG AQUA TANK

Fig. 1

Table 1

Rectification Column (C-1) Reflux Control
This control maintains pure ammonia off the tower for recycle to the evaporators.

A temperature difference recording controller (DTIC) with thermal elements located in the vapor line from the rectificaion column (C-1) and in the reflux control receiver ((RC-1). This controller record temperature difference between these points and controls the reflux flow thru a control valve (DTICV) to maintain the preset difference. By controlling temperature difference the condenser pressure can fluctuate without causing appreciable ammonia purity changes.

Weak Aqua Flow To Aqua Film Absorbers (AB-1 and AB-2)
Calibrated orifice flanges are installed in the vapor line from the evaporator (EV-1) and weak aqua line from the aqua film absorbers (AB-1 and AB-2). DP cells are utilized transmit orifice differentials to a ratio controller (FRC-1). The flow of weak aqua is controlled automatically to maintain a constant pressure in the aqua film absorbers.

Generator Level Control (G-1)
The generator level is maintained by a level control (LC) located on the generator (G-1) balances the flow of strong aqua entering to weak aqua leaving.

Pressure Recorders (Condenser (CO-1)) (Absorbers (Ab-1 & AB-2)
A two pen pressure recorder (PR-1) for condenser (CO-1) pressure and aqua film absorbers, (AB-1 & AB-2) pressure with pressure transmitters.

Generator Temperature Control (G-1)
A temperature recording controller, (TIC) operates a modulating steam or fluid control valve (TICV) to control the flow of heat to the generator. With condensing pressure and generator pressure set by cooling water this temperature determines the concentration of weak aqua produced.
On direct fired generators this temperature is controlled by modulating the fuel to the burners. On waste gas boilers application a modulating damper arrangement on the gas side for by-pass is used, with supplemental firing if required controlled by temperature of weak aqua produced in the generator-pneumatically interlocked with the damper control to prevent over-ride and cycling.
An optional control which would reset the generator (G-1) temperature controller to a lower temperature on large drops in cooling water temperature would permit optimum economical operation. This type control would increase the automatic features but generally could be replaced by simply having the operator reset the control for peak efficiency.

Temperature Indicator
A temperature indicator either manual or automatic for recording or indicating the temperature at the followint points.

Water to condenser (CO-1)
Water to aqua film absorbers (AB-1 and AB-2)
Water from aqua film absorbers (AB-1 and AB-2)
Water between aqua film absorbers (AB-1 and AB-2)
Weak Aqua from generator (G-1)
Weak aqua from aqua heat exchangers (AHX-1)
Strong aqua from aqua heat exchangers (AB-1A)
Strong aqua from aqua heat exchangers (AHX-1)
Strong aqua from aqua pumps (AQ-1 and AQ-2)
Fluid to evaporator (EV-1)
Fluid from evaporator (EV-1)
Ammonia gas from generator (G-1)
Ammonia vapor from rectification column (C-1)
Ammonia vapor to aqua film absorbers (AB-1 and AB-2)
Ammonia liquid in reflux control receiver (RC-1)
Ammonia liquid from reflux pumps (RP-1 and RP-2)
Rectification column (C-1) Five Points

False Load Controller (Optional)
A pressure controller (PRC) sensing inlet steam or heating medium temperature operates a control valve (PRCV) in the ammonia gas line from the rectification column (C-1) to the evaporator (EV-1). This controller overrides the steam control valve to maintain a predetermined steam main pressure by "False loading" the system.

High Level Alarms (HAL) and Low Level Alarms (LAL)
High level alarms (HAL) and low level alarms (LAL) and low level gauges (LG) are installed on teh following vessels:

1. Generator (G-1)
2. Ammonia Receiver (R-1)
3. Strong Aqua Tank (SA-1)
4. Evaporator (EV-1)

Indications of high and low level are registered at the control panel.

Table 2—Utilities—Aqua-Ammonia Absorption System

Evap. Temp., dF	50	40	30	20	10	0	-10	-20	-30	-40	-50
Single Stage											
Steam Pressure, psia	14.1	19.8	24.9	30.9	41.8	53.2	67.0	83.2	103.1	134.6	173.3
Steam Sat. Temp., dF, (or Waste Heat exit temperature)	210	225	240	255	270	285	300	315	330	350	370
Generator Heat Required Btu/min/TR	300	325	347	373	400	430	466	.511	571	645	754
Steam Rate, lb/hr/TR	18.9	20.2	21.8	23.6	25.8	28.1	30.9	34.1	39.1	44.6	53.2
Water rate thru cond. & Absorber, gpm/TR 8C dF On/105 dF Off	3.6	3.7	3.8	4.0	4.3	4.5	5.0	5.5	6.1	7.1	8.8
Two Stage											
Generators Steam Sat. Temp. dF, Exit Temperature	175	180	190	195	205	210	220	230	240	250	265
Steam Pressure psia	6.7	7.5	9.3	10.4	12.3	14.1	17.1	20.7	24.9	29.8	38.1
Generator Heat Required Btu/min/TR	550	557	605	637	670	711	753	799	850	905	970
Steam Rate, lb/hr/TR	33.2	34.9	37.0	39.1	41.3	43.9	46.7	49.9	53.6	57.5	62.3
Water Required thru Cond. & Absorber, gpm/TR 87 dF and 105 dF Off	4.0	4.2	4.3	4.5	4.9	5.3	5.8	6.4	7.2	8.3	10.2

Note: All data approximated. Detailed selections can vary ± 10% depending upon heat source, water available, evaporator constructions etc. A detailed analysis is required for each system. Data courtesy of Lewis Refrigeration Co., Houston, TX.

system is extremely quiet in operation. Noise levels are well below OSHA standards and generally in the 60 to 65 dba area. It is fully automatic and requires only minimal attention. The aqua ammonia absorption system can accept liquid slugs from the refrigeration evaporators and not damage the system in any way. Although the system will be out of balance, it will recover in a few minutes. *"Overloads"* do not damage an absorption system. The system operates automatically from approximately 0% to 100% load with minimal loss of efficiency. As the load requirement is reduced there is a corresponding linear reduction in the consumption of heat. As an additonal advantage, the system can function as a steam condenser or waste gas cooling system while it refrigerates. If the requirement for steam condensing remains at 100% of design while the refrigeration load is reduced a "False Load" control automatically adjust the system to the *dual* demand. This "False Load" control is an additonal plus feature which *may* save additonal steam condensers or waste gas coolers.

DESIGN FLEXIBILITY

Aqua ammonia absorption systems are usually *custom designed* for each specific application. Evaporator temperatures down to -60 dF are practical. Systems can be designed single stage or two stage for one evaporator temperature, an single stage or two stage or multi-stage for several different evaporator temperatures. Systems can be increased in size and evaporator temperatures raised or lowered by the additon of heat exchange surface. Evaporator temperatures are related to heat input temperature. Thus raising the temperature of the heat source lowers the evaporator temperature. The evaporators (chillers) remain 100% efficient at all times as the refrigeration system is oil free. There is no need to add oil fouling factors to the evaporator design, thus saving 5 to 10% in the evaporator cost. The choice of evaporator design has no limitations.

Ammonia is the most efficient refrigerant. Being oil free, it is even more efficient. Its first cost is well below the halocarbons. Ammonia refrigerant piping is normally smaller than that required by the more expensive halocarbons due to its high latent heat and corresponding lower flow rate.

APPLICATIONS

The aqua ammonia absorption system has applications in three specific areas:

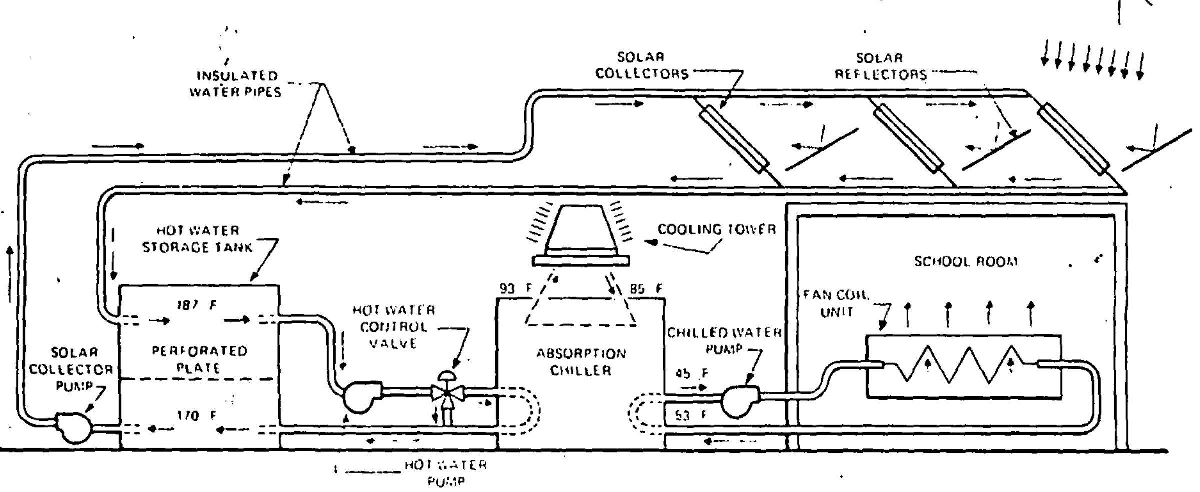
- The production of refrigeration.
- The recovery of NH3 vapors from a mixture of water vapor and ammonia.
- The recovery of NH3 from an aqua-ammonia process.

It can produce refrigeration from waste heat for almost any kind of application in the chemical and petroleum industry. Ammonia absorption systems have been used to chill brine for refrigeration and water for air conditioning while burning *nut hulls* as the heat source. *Wood chips* or *sawdust* are also good sources of wast heat. *Waste steam* has been used as the heat source in many installations in the chemical and petroleum industry providing temperatures from +50 dF to -50 dF. *Process vapor streams* and *hot oil* have also been used as heat sources. *Exhaust gases from gas turbines* would be an excellent source of heat and this heat would normally be capable of providing low temperature refrigeration due to its high temperatures. Refiring could also be added for peak loads. *Stack gases* of many kinds could also be used as a heat source.

The chemical industries have many processes that require heat extraction at a high temperature level in one area and heat extraction at a low temperature (refrigeration) at another. If the extraction requirements are within the limits of the absorption data in Table 2, absorption system could save many dollars in operating cost by reducing the energy requirements. An example, assume that a given process has a liquid or vapor stream at 350 dF that must be cooled to 260 dF or lower and a require ment for refrigeration to cool a process stream to 0 dF with a -10 dF evaporation temperature. From Table 2 note that the heat required in the generator for a -10 dF evaporation temperature is 220 dF thus in dicating a sufficient heating medium exit temperature. The ratio of heat input in the generator (753 Btu/minute) to that extraction in the evaporation (200 Btu/minute) is 3.7. This could probably be reduced by 10 to 20% by additonal refinements in the aqua ammonia absorption system. Variations in the method used to extract the heat from the condenser (temperatures in the 85 to 110 dF range) and the absorbers (temperatures in the 85 to 130 dF range) either by cooling water, air, a combination of both or a process stream add even more possibilities for energy savings.

This sytem is ideally suited to the recovery of NH3 vapors from NH3-water vapor streams. The system can be designed to provide aqua ammonia or pure anhydrous ammonia. The aqua ammonia absorption system is also capable of recovering ammonia in pure anhydrous form from aqua ammonia streams of any concentration. The aqua ammonia systems are the most versatile refrigeration systems available. With the advent of high electrical power cost and *high cost fuel* it is the solution to many refrigeration problems. Its applications are limited only by the imagination and ingenuity for the process engineer. □□

Schematic of Timonium school air conditioning system.



The performance of water cooled lithium bromide absorption units for solar energy applications

Test data such as presented here will assist in capturing the full potential of the solar age

By DONALD K. MILLER,
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Water cooled absorption units are a natural for taking advantage of the abundant energy available from the sun. Many of the currently available lithium bromide units were in fact designed for the use of waste heat or recoverable heat, which in many cases was available at temperatures comparable to those that can be achieved with flat plate solar heat collectors. As technology in the design of solar collectors advances, greater cost effectiveness of the heat source systems will make the widespread use of water cooled lithium bromide units a reality.

From a modest and bold beginning at the Timonium Elementary School in Timonium, Maryland, where a 50 ton lithium bromide absorption unit was installed and is operating today, the industry is progressing to even bolder concepts — limited only by the imagination and the realities of the marketplace. Engineers of all disciplines are actively pursuing the design and application of more efficient and cost effective systems that capture the full potential of the solar age.

The effects of variables external to the water cooled lithium bromide absorption unit, such as the quantities and temperatures of circulated heating, cooling, and chilled fluids, must be known to establish the limits of application.

The effects of internal variables, such as the solution flow rates and the generator and condenser surfaces, must be known by the designer to optimize the effective utilization of the available external variables.

Units commercially available today are capable of furnishing cooling for buildings using the heat source developed from solar energy in the form of circulated fluids. Water is the common fluid used as the heat source, although other fluids can be employed.

With available data on steam coefficients for absorption units, a designer must merely make tradeoffs between the coefficients for steam and the heating fluid used versus the available temperature differences existing in the equipment.

This article explores all of these factors to assist the industry in maximizing the potential of the heat driven equipment available today for widespread application.

Figs. 1 through 12 show a computer analysis of the predicted effects on maximum capacity and the coefficient of performance (COP) for a standard water cooled lithium bromide absorption unit with standard pass arrangements by varying: the steam equivalent heat source temperature (EHST) from 90 to 250 F (32.2 to 121.1 C); the cooling tower water temperature (CTWT) from 45 to 85 F (7.2 to 29.4 C); the temperature of chilled water (CW) from 40 to 50 F (4.4 to 10 C); the cooling tower water flow rate from 64 to 117 percent of nominal (100 percent nominal is 3.6 gpm per ton or 3.88 liters per minute per KW at 85 F cooling tower water temperature); the chilled water flow rate from 50 to 100 percent of nominal (100 percent nominal is 2.4 gpm per ton or 2.59 liters per minute per KW delivered at 44 F); and the lithium bromide flow rate to the generator from 25 percent to 100 percent of the standard absorption unit rate. For a cooling tower water temperature of 75 F (23.9 C), the effects of varying the generator/condenser surface from 80/80 percent to 122/125 percent, respectively, of that for standard absorption units are predicted.

Limits are probed based on restrictions because of crystallization borders (with a standard heat exchanger surface); solution carryover from the generator to the condenser at low condensing temperatures through normally used eliminators; and fluid pressure drop considerations.

Variation of coefficients of heat transfer for steam and other circulated heating fluids with various loadings and fluid velocities is also probed.

Finally, future development possibilities for water cooled lithium bromide absorption units are discussed.

Effects of variables on parameters

The essential parameters that one would generally use to qualify the potential use of water cooled absorption units for solar energy applications would usually be limited to the available equivalent heat source temperature, the cooling tower water temperature, and the expected COP for given capacity requirements at various times of the year for a particular location.

Circulation rates of chilled water, cooling tower water, and heat source fluids can generally be varied to suit the application. The available cooling tower water temperature is generally fixed by the wet bulb temperature; however, the building air conditioning coils and air handling equipment can be varied to some extent depending on the attainable chilled water temperatures.

Therefore, it is useful for the consultant and overall solar energy system designer to know the relative effects on capacity and COP of a water cooled lithium bromide absorption unit for various delivered chilled water temperatures, available equivalent heat source temperatures, and cooling tower water temperatures while easily controlled variables are selected. It is also useful to know that there are actions that a supplier of the absorption equipment can take to enhance the efforts of the total system designer in achieving his goals.

Examination of Fig. 1 shows that for a 100 percent lithium bromide solution rate — based on full nominal tons — and 40 F (4.4 C) delivered chilled water using 100 percent nominal cooling tower water, and chilled water flow rates, and 75 F (23.9 C) cooling tower water temperature, the equivalent heat source temperature is predicted to be 222.8 F (106 C) when producing 100 percent nominal tons at 0.704 COP. COP is defined as cooling capacity produced divided by heat energy input measured in consistent units. A 100 percent nominal capacity for a unit would be that produced using 2.4 gpm per ton (or 2.59 liters per minute per KW) of chilled water delivered at 44 F (6.7 C); 3.6 gpm per ton (or 3.88 liters per minute per KW) of 85 F (29.4C) cooling tower water; and 9 psig (62.06 kilopascal gauge) dry saturated steam or an equivalent supplied to the generator inlet using standard materials and a water side fouling factor of 0.0005 per hr per sq ft per F per Btu (0.000088 per sq meter per C per watt).

One sees in Fig. 1 that a 10 F (5.6 C) reduction in cooling tower water temperature would permit a reduction of the equivalent heat source temperature to 191.5 F (88.6 C) or a reduction of 31.3 F (17.4 C) in the required equivalent heat source temperature, while reducing the maximum capacity to 93 percent of nominal and increasing the COP to 0.745 for a 5.8 percent improvement. Another 10 F cooling tower water temperature reduction is shown to reduce the required equivalent heat source temperature an additional 45 F (25 C) while decreasing the maximum capacity to 76 percent of nominal and increasing the COP to 0.787 for an additional 5.6 percent improvement over the COP achievable with 65 F (18.3 C) cooling tower water.

The reduction in the percent maximum capacity as the cooling tower water temperature is reduced is a natural function of the increasing velocity and specific volume of refrigerant vapor through the eliminators separating the generator and condenser as the condensing temperature is reduced; the reduction of available pressure differences between the generator and absorber sections to move strong lithium bromide solution through the heat exchanger and piping; and the reduction of motive force for the refrigerant condensate from the condenser to the evaporator with a reduction in condensing temperature.

By moving vertically upwards in any column of Fig. 1, one can see that a reduction in the chilled water flow rate to 50 percent of nominal has a negligible effect for a given cooling tower water flow rate. An increase to 117 percent of nominal cooling tower water flow rate reduces the equivalent heat source temperature less than 5 F (2.8 C). A decrease to 64 percent of flow rate increases the equivalent heat source temperature somewhat less than 17 F (9.4 C).

The blanks in Fig. 1 represent situations where the maximum concentration of solution in the generator for the maximum capacity listed to avoid crystallization after the strong solution leaves the heat exchanger may be exceeded. It is possible to achieve higher than 100 percent nominal tons for a given cooling tower water temperature above 75 F (23.9 C) without exceeding the maximum concentration however, for the purposes of this article, the maximum capacity was capped at 100 percent of nominal.

An excessive load may cause carryover of some lithium bromide from the generator to the refrigerant in the condenser. This does not harm the system other than to inhibit optimum evaporator performance. An internal device constantly bleeds some refrigerant liquid from the evaporator section directly to the absorber in order to maintain the refrigerant in an essentially pure state — free of contamination.

Fig. 2 shows what happens to the required equivalent heat source temperature and COP with an increase to 45 F (7.2 C) chilled water off the unit while maintaining 100 percent solution flow rate for the lithium bromide from the absorber to the generator with the same changes in variables previously mentioned.

Fig. 3 shows the effect of raising the chilled water temperature off the unit to 50 F (10 C) with the solution flow rate remaining constant. Notice that for 100 percent nominal chilled water and cooling tower water flow rates, at 85 F (29.4 C) cooling tower water temperature, and 100 percent nominal capacity, the required equivalent heat source temperature is reduced 10.1 F (5.6 C) for a chilled water temperature increase of 5 F (2.8 C), while the COP is increased from 0.690 to 0.713, or an increase of 3.3 percent.

The actual equivalent heat source temperature required, particularly at the lowest cooling tower water temperature of 45 F, may vary somewhat from the computer calculated values. Additional testing in the future would serve as interesting and useful correlations. The values indicated for the lower cooling tower water temperatures should be used in a relative sense rather than as absolute values. Much depends on the degree of refrigerant contamination with lithium bromide and the trimming of the refrigerant and solution charge for a given unit and application. There is a continuous internal redistribution of refrigerant quantity to and from the solution as the load changes and solution concentrations vary in response to the load.

Key: CTW, cooling tower water; CTWT, cooling tower water temperature; EHST, equivalent heat source; COP, coefficient of performance; CAP, capacity.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	115.5	50	.776	158.1	70	.771	207.4	93	.727						
100	107.0	50	.792	146.1	70	.788	190.8	93	.747	221.9	100	.705			
117	104.8	50	.796	143.0	70	.793	186.5	93	.752	217.1	100	.710			
75% CHILLED WATER FLOW															
64	115.6	50	.776	158.3	70	.770	207.8	93	.726						
100	107.1	50	.792	146.4	70	.787	191.3	93	.746	222.5	100	.704			
117	104.9	50	.796	143.2	70	.792	187.0	93	.751	217.5	100	.709			
100% CHILLED WATER FLOW															
64	115.7	50	.776	158.4	70	.770	208.1	93	.726						
100	107.2	50	.792	145.5	70	.787	191.5	93	.745	222.8	100	.704			
117	105.0	50	.796	143.4	70	.792	187.2	93	.751	217.8	100	.709			

1 100 percent solution flow rate with 40 F (4.4 C) chilled water off of water cooled lithium bromide absorption unit. Equivalent heat source temperature, maximum capacity, and COP for various cooling tower water temperatures to unit at various percentages of nominal chilled water and cooling tower water flow rates.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	107.9	50	.799	149.4	70	.794	196.6	93	.754	228.8	100	.711			
100	99.7	50	.815	137.8	70	.812	180.9	93	.773	211.1	100	.731	235.2	100	.692
117	97.5	50	.819	134.7	70	.816	176.8	93	.779	206.4	100	.736	230.4	100	.692
75% CHILLED WATER FLOW															
64	108.1	50	.799	149.7	70	.794	197.4	93	.752	229.3	100	.710			
100	99.8	50	.815	138.0	70	.811	181.3	93	.773	211.5	100	.729	235.8	100	.691
117	97.7	50	.818	135.0	70	.815	177.2	93	.778	206.9	100	.735	231.0	100	.696
100% CHILLED WATER FLOW															
64	108.0	50	.798	149.8	70	.794	197.6	93	.752	229.6	100	.709			
100	99.9	50	.814	138.1	70	.811	181.6	93	.772	211.8	100	.729	236.1	100	.690
117	97.8	50	.818	135.1	70	.815	177.4	93	.777	207.2	100	.734	231.3	100	.695

2 100 percent solution flow rate with 45 F (7.2 C) chilled water off of unit.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	100.8	50	.820	141.4	70	.817	187.8	93	.777	218.7	100	.735	243.4	100	.696
100	92.8	50	.835	130.1	70	.832	172.1	93	.796	201.4	100	.755	225.2	100	.715
117	90.8	50	.840	127.1	70	.837	168.1	93	.801	197.0	100	.760	220.6	100	.720
75% CHILLED WATER FLOW															
64	101.7	50	.820	141.7	70	.816	188.2	93	.776	219.2	100	.734	243.7	100	.695
100	93.0	50	.835	130.3	70	.832	172.6	93	.795	201.9	100	.754	225.7	100	.695
117	90.9	50	.839	127.4	70	.836	168.6	93	.800	197.5	100	.759	221.1	100	.719
100% CHILLED WATER FLOW															
64	101.1	50	.820	141.9	70	.815	188.5	93	.775	219.5	100	.733	244.1	100	.694
100	93.1	50	.834	130.5	70	.831	172.8	93	.795	202.2	100	.753	226.0	100	.713
117	91.0	50	.839	127.5	70	.835	168.9	93	.799	197.8	100	.756	221.4	100	.717

At loads less than the maximums listed for given water flow rates and temperatures, the equivalent heat source temperatures required will be reduced, and the COP may be reduced unless solution flow rates are reduced and the refrigerant is kept free of solution contamination by appropriate means.

The quantity of solution circulated changes the concentrations of the weak and strong solutions throughout the unit and hence the operating efficiency. Generally, the higher the concentration, the better the COP is. A decrease in solution flow rate from the absorber to the generator increases the concentrations and thus the COP. However, this increases the required equivalent heat source temperature. It also places some additional limits on range of variables because of the increased chance of bordering on crystallization tendencies.

Fig. 7 shows that for 64 percent nominal cooling tower water flow at 100 percent nominal chilled water flow, with 65 F (18.3 C) cooling tower water, and 50 percent of the full solution flow rate, the COP increases 4.6 percent compared to that for the 100 percent solution rate represented in Fig. 1. Notice the shaded area on Fig. 7. This is where theoretically some values were attainable except that the strong solution temperature leaving the heat exchanger for the given concentration of solution crossed over the crystallization border limit. With greatly reduced solution flow rates, the weak solution leaving the heat exchanger tends to reach a superheated value — in other words, a value above the boiling point of the solution at the generator pressure level. The generator pressure level is in turn established by the condensing temperature. At low cooling tower water temperatures, the superheated condition of the solution prior to its entering the generator is most apt to occur.

The optimum percent solution flow rate for improved COP is generally somewhat greater than the percent nominal capacity at which the unit must perform. The lowest required equivalent heat source temperatures can be achieved with 100 percent of solution flow.

% NOM.	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
CTW FLOW	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
	50% CHILLED WATER FLOW														
64	116.0	50	.799	159.0	70	.794	209.2	93	.748						
100	107.8	50	.812	147.4	70	.809	193.0	93	.765						
117	105.6	50	.815	144.4	70	.813	188.9	93	.770	219.7	100	.728			
	75% CHILLED WATER FLOW														
64	116.1	50	.799	159.2	70	.793	209.7	93	.747						
100	107.9	50	.812	147.6	70	.808	193.5	93	.764						
117	105.7	50	.815	144.6	70	.812	189.3	93	.768	220.2	100	.727			
	100% CHILLED WATER FLOW														
64	116.2	50	.799	159.4	70	.793	209.9	93	.747						
100	107.9	50	.811	147.7	70	.808	193.7	93	.763						
117	105.8	50	.815	144.7	70	.811	189.5	93	.768	220.5	100	.727			

4 75 percent solution flow rate with 40 F chilled water off of unit.

% NOM.	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23 9°C)			CTWT 85°F (29.4°C		
CTW FLOW	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.
	50% CHILLED WATER FLOW														
64	108.6	50	.818	150.5	70	.816	199.0	93	.773	231.3	100	.731			
100	100.6	50	.830	139.2	70	.829	183.3	93	.789	213.8	100	.748			
117	98.5	50	.833	136.3	70	.833	179.3	93	.794	209.3	100	.752			
	75% CHILLED WATER FLOW														
64	108.7	50	.818	150.8	70	.815	199.4	93	.772	231.9	100	.730			
100	100.7	50	.829	139.5	70	.829	183.8	93	.788	214.3	100	.747			
117	98.6	50	.833	136.5	70	.832	179.7	93	.793	209.8	100	.751			
	100% CHILLED WATER FLOW														
64	108.8	50	.817	150.9	70	.815	199.7	93	.771	232.1	100	.729			
100	100.8	50	.829	139.6	70	.829	184.0	93	.788	214.6	100	.746			
117	98.7	50	.832	136.6	70	.832	179.4	93	.792	210.1	100	.750			

5 75 percent solution flow rate with 45 F chilled water off of unit.

% NOM.	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4		
CTW FLOW	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.(
	50% CHILLED WATER FLOW														
64	101.6	50	.835	142.7	70	.835	189.9	93	.795	221.4	100	.753	245.9	100	.7
100	93.8	50	.846	131.6	70	.848	174.5	93	.811	204.3	100	.770	228.2	100	.7
117	91.8	50	.849	128.7	70	.851	170.6	93	.815	200.0	100	.774	223.7	100	.7
	75% CHILLED WATER FLOW														
64	101.8	50	.834	142.9	70	.835	190.4	93	.794	221.9	100	.752	246.4	100	.7
100	94.0	50	.846	131.9	70	.848	175.0	93	.810	204.8	100	.769	228.7	100	.7
117	92.0	50	.849	129.0	70	.851	171.0	93	.814	200.5	100	.773	224.2	100	.7
	100% CHILLED WATER FLOW														
64	101.9	50	.834	143.1	70	.834	190.6	93	.793	222.2	100	.751	246.8	100	.7
100	94.1	50	.846	132.0	70	.847	175.2	93	.809	205.1	100	.768	229.0	100	.7
117	[illegible]	50	[illegible]	129.1	70	.851	171.3	93	.814	200.8	100	.773	224.5	100	[illegible]

Key: CTW, cooling tower water; CTWT, cooling tower water temperature; EHST, equivalent heat source temperature; COP, coefficient of performance; CAP, capacity.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	118.4	50	.820	163.2	70	.813									
100	110.4	50	.828	151.9	70	.824									
117	108.3	50	.831	148.9	70	.827									
75% CHILLED WATER FLOW															
64	118.5	50	.820	163.5	70	.813									
100	110.5	50	.828	152.1	70	.824									
117	108.4	50	.831	149.1	70	.827	196.9	93	.777						
100% CHILLED WATER FLOW															
64	118.6	50	.820	163.6	70	.812	217.4	93	.759						
100	110.6	50	.828	152.2	70	.824									
117	108.5	50	.831	149.2	70	.827									

7 50 percent solution flow rate with 40 F chilled water off of unit.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	111.1	50	.835	154.9	70	.833	206.6	93	.783						
100	103.3	50	.843	143.8	70	.843	191.0	93	.796						
117	101.2	50	.845	140.9	70	.846	187.0	93	.800						
75% CHILLED WATER FLOW															
64	111.3	50	.835	155.1	70	.832	207.0	93	.781						
100	103.4	50	.843	144.0	70	.843	191.4	93	.795						
117	101.4	50	.845	141.1	70	.845	187.4	93	.798						
100% CHILLED WATER FLOW															
64	111.3	50	.835	155.3	70	.832	207.3	93	.781						
100	103.5	50	.843	144.2	70	.842	191.7	93	.795						
117	101.4	50	.845	141.2	70	.845	187.6	93	.798						

8 50 percent solution flow rate with 45 F chilled water off of unit:

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	114.3	50	.848	147.2	70	.850	197.6	93	.802						
100	96.6	50	.856	136.3	70	.860	182.3	93	.816	213.7	100	.774			
117	94.6	50	.857	133.4	70	.862	178.3	93	.820	209.3	100	.777			
75% CHILLED WATER FLOW															
64	104.3	50	.848	147.4	70	.849	198.1	93	.802						
100	96.7	50	.856	136.5	70	.859	182.7	93	.815	214.2	100	.773			
117	94.7	50	.857	133.7	70	.861	178.8	93	.819	209.8	100	.776			
100% CHILLED WATER FLOW															
64	104.5	50	.848	147.6	70	.849	198.3	93	.801						
100	96.8	50	.855	136.7	70	.859	182.9	93	.814	214.5	100	.773			
117	94.8	50	.857	133.8	70	.861	179.0	93	.818	210.1	100	.776			

9 50 percent solution flow rate with 50 F chilled water off of unit.

Trade-offs can be made with the COP on choosing the optimum solution flow rate to the generator.

Other Figures show the potential effects of changing the variable for various performance parameters at 100 percent nominal surface for the generator and condenser.

With less or more than 100 percent nominal surfaces in the generator and condenser, the required equivalent heat source temperature and COP will change. Table 1 shows these values for surface changes and the same variables explored in Figs. 1 through 12.

Absorption units with 20 percent less generator and condenser surface than that of a standard unit would require a 10.4 F (5.8 C) higher equivalent heat source temperature and have a slightly smaller COP. Those units with 122 percent of a standard generator and 125 percent of the standard condenser surfaces would require a 5.5 F (3.1 C) lower equivalent heat source temperature and achieve a negligible increase in COP with increased costs. There is an optimum trade-off in overall system cost when examining achievable equivalent heat source temperatures.

Trade-offs

Internal steam coefficients, including the effects of scale factor and condensate loading, can vary between 1200 and 3200 Btuh per deg F per sq ft (or 6814 to 18.172 watts per deg Kelvin per square meter).

On a log-log plot of fluid velocity versus typical internal coefficients for various heating fluids in one type generator tube, the data shown in Table 2 plot as straight lines.

A single pass arrangement should have the hottest heating fluid temperature entering the hottest end of the generator (strong solution leaving point). The cooled leaving heating fluid is then at the opposite end where the colder weak solution enters the generator, giving the maximum log mean temperature difference for heat transfer. Velocities approaching 10 fps (3.048 meters per second) for a single pass result in hot water quantities approaching 1.76 gpm per nominal ton (1.897 liters per minute per KW). The hot water range is thus near 20

F (11.1 C), and a minimum size unit can be used for an application while permitting the designer of a total solar energy system to take full advantage of lower overall costs that result from selecting the lower hot water temperature for installations.

Development possibilities

In the lower temperature hot water applications where the maximum solution concentration is relatively low (below 60 percent by weight of lithium bromide), the usual dilution cycle on shutdown of absorption units can be deleted. The inherent lack of susceptibility of these systems to crystallization if there should be an extended power failure assures increased reliability.

Some type of fixed or variable solution flow adjustment to the standard unit would increase the operating COP, particularly at loads considerably below the nominal rating of the unit. The available equivalent heat source temperature will influence the choice of solution flow rate.

For system loads far below the nominal rating of the unit, reoptimization of the heat exchanger, particularly for reduced cooling water temperatures, and use of reduced solution flow rates could be considered.

Improvement in efficiency and increasing the size of eliminators in the generator/condenser area would assist in increasing the attainable capacities where lower cooling tower water temperatures are available and where increased capacity is necessary.

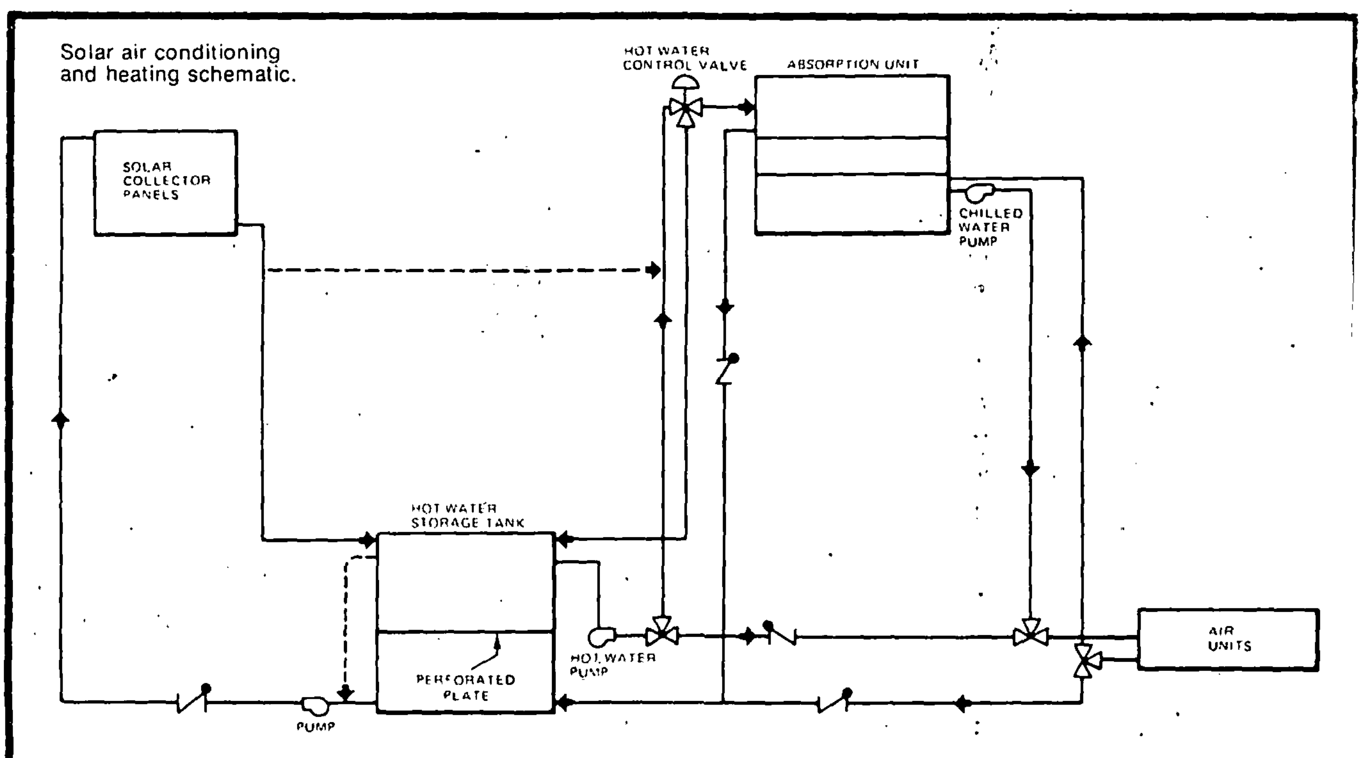
Optimum charge adjustment and optimum size of storage areas for refrigerant and solution where the

Table 1 — Comparison of absorption unit performance with surface variations. Based on 45 F (7.2 C) chilled water off of unit; 100 percent solution flow rates; and 100 percent chilled water and cooling tower water flow rates.

		Cooling tower water temperature 75 F (23.9 C)			
Percent generator surface	Percent condenser surface	Equivalent heat source temperature F	Equivalent heat source temperature C	Percent capacity	COP
80	80	222.2	105.7	100	0.725
100	100	211.8	99.9	100	0.729
122	125	206.3	96.8	100	0.730

Table 2 — The following data plot as straight lines on a log-log plot of fluid velocity versus typical internal coefficients for various heating fluids in one type generator tube.

Velocity fps	Velocity m per sec	Average bulk temperature F	Average bulk temperature C	Pure water Btuh per sq ft per F	Pure water watts per Kelvin per sq meter	Organic fluids Btuh per sq ft per F	Organic fluids watts per Kelvin per sq meter
3	0.9144	160	71.1	1,200	6,814	173	982
		200	93.3	1,380	7,837	196	1,113
		240	115.5	1,530	8,688	215	1,221
10	3.048	160	71.1	3,070	17,433	450	2,555
		200	93.3	3,600	20,443	515	2,925
		240	115.5	4,000	22,714	570	3,237



Key: CTW, cooling tower water; CTWT, cooling tower water temperature; EHST, equivalent heat source temperature; COP, coefficient of performance; CAP, capacity.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	130.9	50	.826												
100	122.9	50	.832												
117	120.8	50	.834												
75% CHILLED WATER FLOW															
64	131.1	50	.826												
100	123.0	50	.832												
117	120.9	50	.834												
100% CHILLED WATER FLOW															
64	131.1	50	.826												
100	123.1	50	.832												
117	121.0	50	.834												

10 25 percent solution flow rate with 40 F chilled water off of unit.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	123.6	50	.840												
100	115.7	50	.845												
117	113.6	50	.846												
75% CHILLED WATER FLOW															
64	123.7	50	.840												
100	115.8	50	.845												
117	113.7	50	.846												
100% CHILLED WATER FLOW															
64	123.8	50	.840												
100	115.9	50	.845	166.4	70	.827									
117	113.8	50	.846												

11 25 percent solution flow rate with 45 F chilled water off of unit.

% NOM. CTW FLOW	CTWT 45° F (7.2°C)			CTWT 55°F(12.8°C)			CTWT 65°F (18.3°C)			CTWT 75°F (23.9°C)			CTWT 85°F (29.4°C)		
	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.	EHST °F	% CAP.	C.O.P.
50% CHILLED WATER FLOW															
64	116.6	50	.851												
100	108.8	50	.856												
117	106.7	50	.857	155.5	70	.846									
75% CHILLED WATER FLOW															
64	116.7	50	.851												
100	108.9	50	.856												
117	106.9	50	.857	155.7	70	.846									
100% CHILLED WATER FLOW															
64	116.8	50	.851												
100	109.0	50	.856	158.8	70	.843									
117	106.9	50	.857	155.9	70	.845									

12 25 percent solution flow rate with 50 F chilled water off of unit.

lower operating solution concentrations would be used could be considered.

There is always room for optimization. However, the standard units available today are an immediate and effective answer to the use of solar energy for cooling buildings. An ongoing joint effort of system and absorption unit designers with testing correlations and optimizations will further enhance the possibility of utilizing this natural source of abundant energy.

Conclusions

This article includes a comprehensive review of the variables that can influence the proper choice of all the solar energy system components for the cooling of buildings.

Optimization of absorption unit COPs and the required temperature levels of the heat source directly affect the choice of systems external to the unit. Improved COPs influence the effectiveness of hot water storage systems and the cyclic demand on solar energy collectors.

The ability of absorption units to essentially respond to matching seasonal changes in available cooling tower water and equivalent heat source temperatures as well as the attendant load changes makes absorption units the ideal choice of equipment for the cooling buildings. The consultant can match the building load profile and cooling capabilities of absorption to the available solar energy source, once guidelines are available on water cooled absorption unit capabilities when utilizing heated fluids such as water.

The state of the art for solar collectors and systems is advancing. The further development and testing of absorption units will assist in capturing the full potential of the solar age within our lifetime.

This article is based on a talk delivered by Mr. Miller at the University of California, Los Angeles, during a workshop on the use of solar energy for cooling buildings.

Kinetics Corp. Starts Mass Production of Rankine Cycle Power Systems

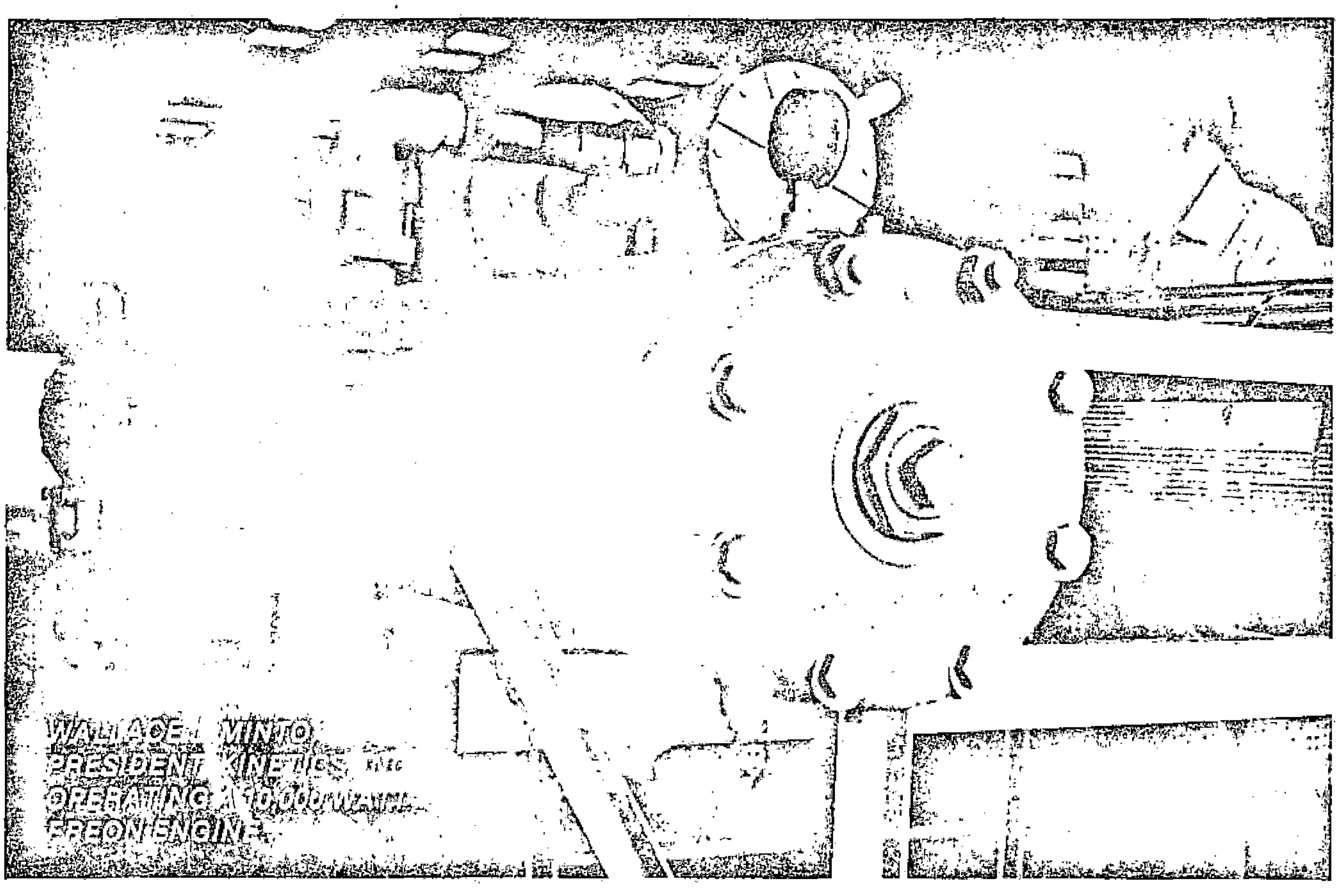
Sarasota, Florida, March 1, 1977 — Kinetics Corporation, a Sarasota research and development firm, today announced that packaged freon Rankine Cycle power generating units will be commercially available for the first time on a mass production basis. Each unit will produce 10,000 watts of power when supplied with hot water at 185°F.

Wallace L. Minto, president of Kinetics, said the firm is designing and building as large as two thousand kilowatt systems using any heat source of 150°F or higher. Industrial waste heat, geothermal sources, solar collectors, or any natural heat source can be employed to supply the necessary low-grade inlet temperature. Kinetics keeps a demonstration unit operative at all times and the system has been successfully endurance tested for a period of 20,000 hours.

Prior to the decision to mass produce these systems, at a rate of 100 systems per month, Kinetics offered only custom-built designs. The move was made on the basis of the system's demonstrated efficiency and cost effectiveness. An important consideration was Kinetic's determination that they could be offered for sale at costs low enough to enable buyers to recover capital costs within two years due to the present value of power.

Over the past year, eight pilot units have been demonstrated and sold worldwide, two of which have been sold recently to the United Nations. These engines are destined for installation in Africa and Asia and will be shipped to Senegal and Sri Lanka for production of electricity using solar-heated water. Both systems are part of a United Nations Environment Programme, (UNEP), an effort to develop an integrated solar, wind, and biogas energy center which can produce electricity for remote villages.

A major American automobile manufacturer has also purchased two systems from Kinetics for evaluation and has plans to buy additional larger systems which will produce electricity and compressed air to power automobile assembly tools. Among other applications now being considered is a 1600 Kilowatt system for use by a



WALLACE L. MINTO, PRESIDENT KINETICS, INC OPERATING A 10,000 WATT FREON ENGINE major American steel producer. This system will utilize blast furnace flue gas as its heat source. The systems will also be furnished as prime movers for air and gas compressors, pumps, or for any other application requiring reliable, continuous shaft horsepower.

The Kinetics engine system efficiently converts low temperature, low level heat energy into high intensity mechanical shaft output. This output may be used to drive an electric generator, hydraulic pump, air compressor, or any other device performing useful work. The low level heat energy may be supplied by industrial process waste streams, such as waste steam, cooling water, quench oil or flue gases. In many cases, a polluting nuisance is abated and converted into a valuable energy source.

According to 1976 figures the total annual energy consumption in the United States is approximately 80 quads, (1 quad = 1 quadrillion BTU = 172 million barrels of oil = 41.7 million tons of coal = 970 trillion cubic feet of natural gas) of which about 60 quads are now supplied by petroleum and natural gas. Domestic reserves of these are dwindling and sharp additional price rises are anticipated within the next twelve months.

Minto considers the use of power systems operating on industrial waste heat as a significant step forward in solving the world-wide energy crisis. By adapting this proven technology to the energy crisis in what Minto describes as Kinetics' first "commercial landing," he foresees an alleviation of energy shortages. Substantial economic benefits will result from using plentiful present waste heat sources to produce shaft power or electricity. Expected increases in industrial electricity rates also will serve to stimulate the demand for waste heat power recovery systems in the United States and abroad.

As an inventor, Minto previously developed a non-polluting, freon-powered engine under contract with Nissan Motor Co. Ltd., for possible use in Datsun automobiles, and the "Minto Wheel" . . a gravity engine driven by a small temperature differential. The Minto Wheel converts very low level geothermal, solar or waste heat into small amounts of shaft horsepower, and the wheel is already being applied in the developing areas of the world.

Kinetics Corporation
1121 Lewis Avenue
Sarasota, Florida
Phone: (813) 366-3050

GEOTHERMAL EFFLUENTS WORKSHOP

FEBRUARY 15-17, 1977

An audience of 110 government and industry representatives heard the speakers during a comprehensive seminar on geothermal effluent problems.

The second conference on effluents by EPA-EMSL was held in the luxurious Marina Hotel in Las Vegas. It was sponsored by the EPA and Dr. Tsvi Meidav and his staff from Geonomics Inc., aided in an efficiently-organized and informative 3-day meeting.

Abstracts from the meeting will be printed from time to time in the GEM, but those who wish to receive the complete proceedings should contact either EPA-EMSL (Attn. Dr. Morgan) at Box 15029, Las Vegas, NV. 89114 or Dr. S. Sanyal, Geonomics, 3165 Adeline Street, Berkeley, California.

A particularly interesting presentation was the showing of slides of eruptionf in Nicaragua.

The geothermal community benefits from such well-planned and interesting seminars. Thanks should go to the EPA personnel, and to Geonomics for their worthy efforts.

ERDA PRESENTS McCABE DISTINGUISHED CAREER AWARD

The Energy Research and Development Administration (ERDA) has presented its **First Distinguished Career Award**, for Outstanding contributions to the founding and continued success of the Geothermal Industry in America to **B.C. McCabe**, President of **Magma Power Co.**, Los Angeles, California. "Mac" McCabe, who recently observed his 80th birthday, was among the first to recognize the energy-generating potential of The Geysers steam field in northern California.

McCabe's citation reads: "For his pioneer work in founding the Geothermal Industry in America and his outstanding contributions toward its continued success. In 1925, he first visited The Geysers in California and sensed its potential. In 1954 he drilled the first steam well there which is still being productively used. In 1958 he attracted a utility company to utilize the steam. Through joint efforts, the first Geothermal Power Plant began operations in 1960 in a field which now supplies more than half the power requirements of San Francisco . . ."

Gas-Fired Heat Pumps: An Emerging Technology

L. A. Sarkes
J. A. Nicholls
M. S. Menzer

Up until the late 1960s when natural gas was in abundant supply, the technological goals of the gas industry were aimed at improvements in safety, reliability, economy and efficiency of operations. But, whereas the industry was even then conscious of the need to use gas wisely (economy and efficiency of operations are synonymous with conservation), the consumer had no real reason to be frugal in his use of this plentiful and cheap energy form. Nor could he be convinced of the need to replace old appliances with new, more efficient equipment that used less gas when the incremental savings thus obtained were small compared to the higher first cost penalties of the new appliances.

THE declining supply of gas and the increasing costs of all energy forms has changed public attitudes over the past few years. The need for energy conservation is now better understood by an increasing number of cost-conscious consumers and the gas industry, for its part, is aggressively pursuing all avenues of technology development that offer the potential for realizing significant savings in gas energy utilization. Development of an ultra-high efficient central heating furnace, improved commercial cooking equipment, design of domestic appliances to use gas at elevated pressures, gas-fired fuel cells, heat pumps for domestic and commercial establishments, and solar-assisted gas heating and cooling systems are but a few examples of the industry's involvement. The market potential and development status of three gas-fired heat pump systems is the subject of this paper.

MARKET ANALYSIS

Heat pump technology offers a potential for realizing significant savings in energy. During 1975, 46% of the net marketed natural gas retailed in the U. S. was sold to the industrial sector, 34% to the residential consumer, and 20% to the commercial and other segments. Two-thirds of residential gas usage and over half of commercial consumption were used for space conditioning (heating and cooling) purposes. In total, more than 30% of the natural gas retailed was sensitive to heating/cooling systems technology. Any improvement in this technology would have a significant national impact on the consumption of gas and could be more than twice as great as technological advances in any other market sector.

The seasonal efficiency of a properly sized, properly installed conventional gas furnace is about 65% and the COP of electric air conditioners is about 0.6. Using the contemporary gas furnace-electric air conditioner system as a base, it is estimated that seasonal improvements in overall gas utilization of 2.0 to 2.5 can be achieved by using gas fuel heat pumps.

Business market analyses conducted to date indicate a cumulative total available heat pump market of 10,278,000 units during the period of 1981 to 1990. Based on low, nominal and high percentage market potential, the cumulative total of gas-fired heat pumps would be 208,000 (low), 419,000 (nominal), and 879,000 (high) for the projected 10-year period. These data are shown in Table 1.

L. A. Sarkes is Director of Research; J. A. Nicholls is with R & D Planning; and M. S. Menzer is with R & D Environmental Systems—all with the American Gas Association, Arlington, VA.

The breakdown of the 1981 gas-fired heat pump market by size is shown in Table 2. Operating costs in the study were calculated for 4 building types in 9 regions of the U.S. to obtain heating/cooling requirements, weather, building/cooling characteristics, and energy cost data. Gas and electrical energy costs from known data were used to estimate the price of energy as a function of regions of the country. These cost comparisons, shown in Table 3, were projected for 1980 based upon constant 1975 dollars.

To measure the overall energy savings impact of the gas heat pump, the energy savings related to the product concept was combined with the market unit shipment figures expected for the product. The projected energy savings was made up of savings in natural gas and electricity consumption. The data shown in Table 4 reflect the range of savings expected and correspond to the range projected for unit shipment in both residential and commercial market segments.

The commercial market represents approximately 20% of the total energy savings. This provides a significant incentive for an accelerated commercial development, in that by 1990, 20% of the projected new commercial gas customers could be satisfied by the savings generated by units in the field.

While it is recognized that the gas heat pump is not the total answer to conservation in the use of gas energy, it does have the potential for reducing the consumption level of gas for heating and cooling purposes in a reasonable period of time. To better appreciate the magnitude of these savings, the data generated to date can be expressed in some meaningful ways. For example, the average 1975 natural gas consumption per household was 124.8 mcf. In 1990, with reduced consumption through the use of heat pumps, 238,000-528,000 additional households could be satisfied with the gas savings alone, without increasing supply requirements.

The incentive is clear: The greater the number and the sooner gas-fired heat pump products can be introduced to the marketplace, the sooner benefits will be realized. However, there are still developmental and product-line complexities to be overcome.

HEAT PUMP R&D

Recognizing the energy conservation potential and advantages of an on-site heat-activated heat pump, the American Gas Association (AGA) initiated a gas heat pump research program long before the energy crisis became a pressing public concern. Within the overall objective of developing gas-fired heat pumps for the residential and commercial markets, this program has four specific goals:

- To have gas-fired heat pumps available for the residential market by 1981.
- To have gas-fired heat pumps available for the commercial market by 1983.
- To demonstrate gas-fired heat pumps with a reliability equivalent to that of a conventional gas-fired air furnace.
- To achieve a first cost on initial gas-fired heat pumps within 15% of a combination forced air furnace-electric air conditioner system with a payback period of less than 4 yr.

Currently, the program has matured to the point where AGA has three concepts in various stages of hardware development:

AGA On-Site Heat Activated Heat Pumps
Development Projects

Cycle	Application	Status
Stirling/Rankine	Residential/Commercial	Breadboard
Brayton/Rankine	Commercial Only	Breadboard
Organic Absorption	Residential/Commercial	Prototype

Table 1

Year	Potential Market (Thousands)	Market Penetration % lo	nom	hi	Industry Shipments (Thousands) lo	nom	hi
1981	938	1.2	1.4	1.6	11	13	15
1982	986	1.3	1.7	2.2	13	17	22
1983	956	1.5	2.1	3.0	14	20	29
1984	985	1.6	2.5	3.9	16	25	38
1985	1041	1.8	3.0	5.3	19	31	55
1986	1053	2.0	3.8	6.8	21	40	72
1987	1063	2.2	4.6	9.2	23	49	98
1988	1073	2.5	5.6	12.3	27	60	132
1989	1085	2.8	6.8	16.3	30	74	177
1990	1098	3.1	8.2	21.9	34	90	241
Cum. Total	10,278				208	419	879

Table 2

Total Market Gas-Fired Heat/Cooling Systems
Year: *1981;* All Regions

Units ($\times 10^{3}$)

Unit Size	Residential New	Replace	Total	Commercial New	Replace	Total
2 Tons	139.7	85.3	225.0	—	—	—
2-1/2	118.8	69.9	118.7	—	—	—
3	143.4	62.6	211.0	22.8	3.3	26.1
3-1/2	32.3	15.0	47.3	13.0	1.9	14.9
4	33.1	13.4	46.4	24.9	3.3	28.2
5	26.5	10.5	37.0	30.2	4.4	34.6
7-1/2	—	—	—	27.5	3.9	31.2
10	—	—	—	21.9	3.1	25.0
15	—	—	—	11.5	1.6	13.1
20	—	—	—	4.1	.6	4.7
25	—	—	—	2.8	.4	3.2
30	—	—	—	1.4	.2	1.6
Total	498.8	257.2	756.0	160.0	22.7	182.7

Table 3

Fuel Price Projection—$/MMBtu (1980, 1975 $)

Region	Residential Gas Heating	Residential Electric Non-Heating	Commercial Gas Heating	Commercial Electric Non-Heating
New England-Boston	3.96	18.18	3.89	17.08
Mid-Atlantic-Philadelphia	3.02	18.94	2.82	18.57
E.N. Central-Madison	2.11	15.53	1.95	15.23
W.N. Central-Bismarck	2.09	15.72	1.70	14.85
S. Atlantic-Charleston	2.76	13.26	2.40	13.37
E.S. Central-Nashville	1.99	9.66	1.73	11.51
W.S. Central-Ft. Worth	1.82	13.45	1.33	11.89
Mountain-Denver	1.82	13.82	1.53	12.25
Pacific-Seattle	2.15	11.37	1.95	11.32

Conversions:
$/MMBtu × 1 = $/MCF $/MMBtu × 0.341 = ¢/kWh

Table 4

GHP Potential Energy Savings

Year	Annual Gas Savings ($CF \times 109^9$)		Annual Electric Savings ($kWH \times 10^6$)		Total Annual Energy Savings ($Btu \times 10^{12}$)	
	Residential	Commercial	Residential	Commercial	Residential	Commercial
1980	0.64- 0.74	0.09-0.10	5.31- 6.44	12.15- 14.05	0.66- 0.76	0.13- 0.15
1981	1.46- 1.80	0.20-0.24	11.81- 14.89	27.64- 33.97	1.51- 1.85	0.29- 0.36
1982	2.43- 3.19	0.34-0.44	19.26- 23.75	47.21- 61.80	2.50- 3.27	0.50- 0.66
1983	3.62- 3.19	0.51-0.73	28.42- 38.00	71.54- 96.32	3.72- 5.17	0.76- 1.07
1984	5.15- 7.74	0.72-1.10	40.17- 58.77	101.79- 150.00	5.29- 7.94	1.07- 1.63
1985	7.11-11.20	0.99-1.60	55.04- 84.94	140.95- 220.16	7.30-11.52	1.47- 2.37
1986	9.50-15.98	1.33-2.29	72.80-120.51	190.65- 319.47	9.75-16.42	1.98- 3.40
1987	12.43-22.43	1.77-3.24	94.46-168.07	252.12- 454.19	12.76-23.03	2.62- 4.81
1988	16.02-31.02	2.31-4.54	120.81-231.40	328.43- 637.55	16.44-31.84	3.42- 6.74
1989	20.39-42.70	2.97-6.24	152.84-316.96	422.20- 887.39	20.92-43.81	4.40- 9.29
1990	25.77-57.31	3.80-8.51	191.49-425.30	539.33-1209.92	26.43-58.79	5.63-12.66

The Stirling/Rankine system uses a Free Piston Stirling Engine and a vapor compression Rankine refrigeration loop. This project is envisioned for use in residential and light commercial buildings requiring up to 10-tons capacity. The hardware for this project is now in the laboratory breadboard stage with a 3-ton unit. The Brayton/Rankine system uses a subatmospheric gas turbine with a regenerator and a vapor compression Rankine refrigeration cycle. This project is envisioned for use in commercial applications requiring in excess of 10-tons capacity. The hardware for this project is in the laboratory breadboard stage with a 10-ton unit. The absorption system is based on a proprietary organic refrigerant and organic absorber pair developed by Allied Chemical. This project is envisioned for use in residential applications and is now in the prototype stage with a 3-ton unit.

The most promising of these three technologies, for meeting the industry goals defined earlier, appears to be the Stirling/Rankine system developed by the gas industry over the past 5 yr.

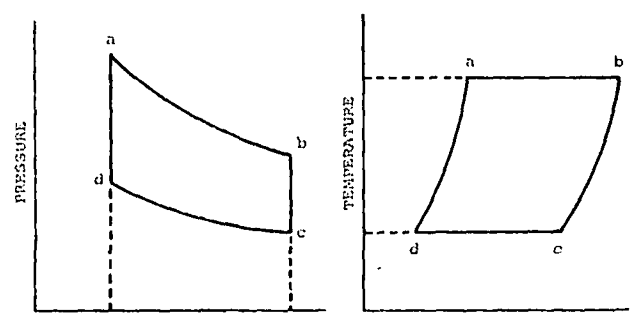
STIRLING/RANKINE

The ideal Stirling cycle, shown in Fig. 1, has received considerable attention over the years because of its potential for high efficiency. The constant volume-constant temperature cycle has the same efficiency as the Carnot cycle, but unlike Carnot, a practical Stirling engine can and has been built. In cost-effectiveness terms, however, the Stirling engine lost out to other engine cycles, particularly the Rankine cycle steam engine, but modern technology combined with other benefits of the cycle have injected new interest into the Stirling engine. Recent developments have tended to focus upon automotive applications because of the inherent low pollution levels of the external combustion Stirling engine. These automotive engines have rotary shaft power output and require high pressure differential, low leakage and dynamic seals. However, the engine does not necessarily have to have rotary shaft power output.

The original application for a Stirling engine in the early 19th century was to operate a piston pump to drain deep mines. In most refrigeration and heat pump devices, a single-piston positive displacement pump is used to compress the vapor. This type of pump needs linear power. Fortunately, a Free Piston Stirling Engine which is a linear engine that could be matched with a reciprocating inertia compressor in a hermetically sealed assembly has been developed. Additionally, the engine is self-starting, does not require oil lubrication, has no gears, and has low loadings on the piston rings. Recognizing all these advantages, AGA negotiated a position on patent rights and, in 1972, initiated a project to further develop the engine for gas industry applications.

As shown in Fig. 2, the Stirling cycle consists of a constant temperature expansion, a constant volume temperature and pressure reduction, a constant temperature compression, and a constant volume heating. To accomplish this cycle, two pistons are required. The displacer piston, usually called the displacer, moves the working gas through the heating and cooling steps of the cycle. The power piston, usually called the piston, provides the working gas compression and takes the power out during the expansion. Obviously, the phase relationship between the displacer and piston is important if the cycle is to work.

In the rotary Stirling engines, the displacer and piston are placed in separate cylinders and connected with a rather elaborate set of linkages. In the Free Piston Stirling Engine, the displacer and piston are placed in a single cylinder. This engine depends upon the mass differences between a light displacer and a heavy piston and a gas spring formed by the pressure variations in the bounce space or what would typically be called the engine's crankcase.



$$\text{CYCLE EFFICIENCY } \eta = \frac{T_a - T_c}{T_a}$$

Fig. 1 The Stirling cycle

Starting at State I (Fig. 2), the working gas is basically in the hot space and at a pressure higher than the bounce space. The working gas expands driving the displacer and piston to the left. As the working gas expands, its pressure drops below that of bounce space. The higher bounce space pressure brings the displacer and piston to a stop.

At State II, the bounce space pressure is higher than the working space pressure and begins to expand and drive the displacer and piston to the right. Since the displacer is light compared to the piston, it accelerates faster and displaces the working gas from the hot space to the cold space.

In State III, the working gas pressure is much lower than the bounce space pressure and accelerates the motion of the piston. The piston now compresses the cold working gas to State IV. During this compression the working gas pressure exceeds the bounce space pressure and slows the piston.

In State IV, the working gas pressure is higher than the bounce space pressure and is forcing the displacer and piston to the left. Since the displacer is lighter, it again moves first and displaces the working gas from the cold space to the hot space and brings us back to State I.

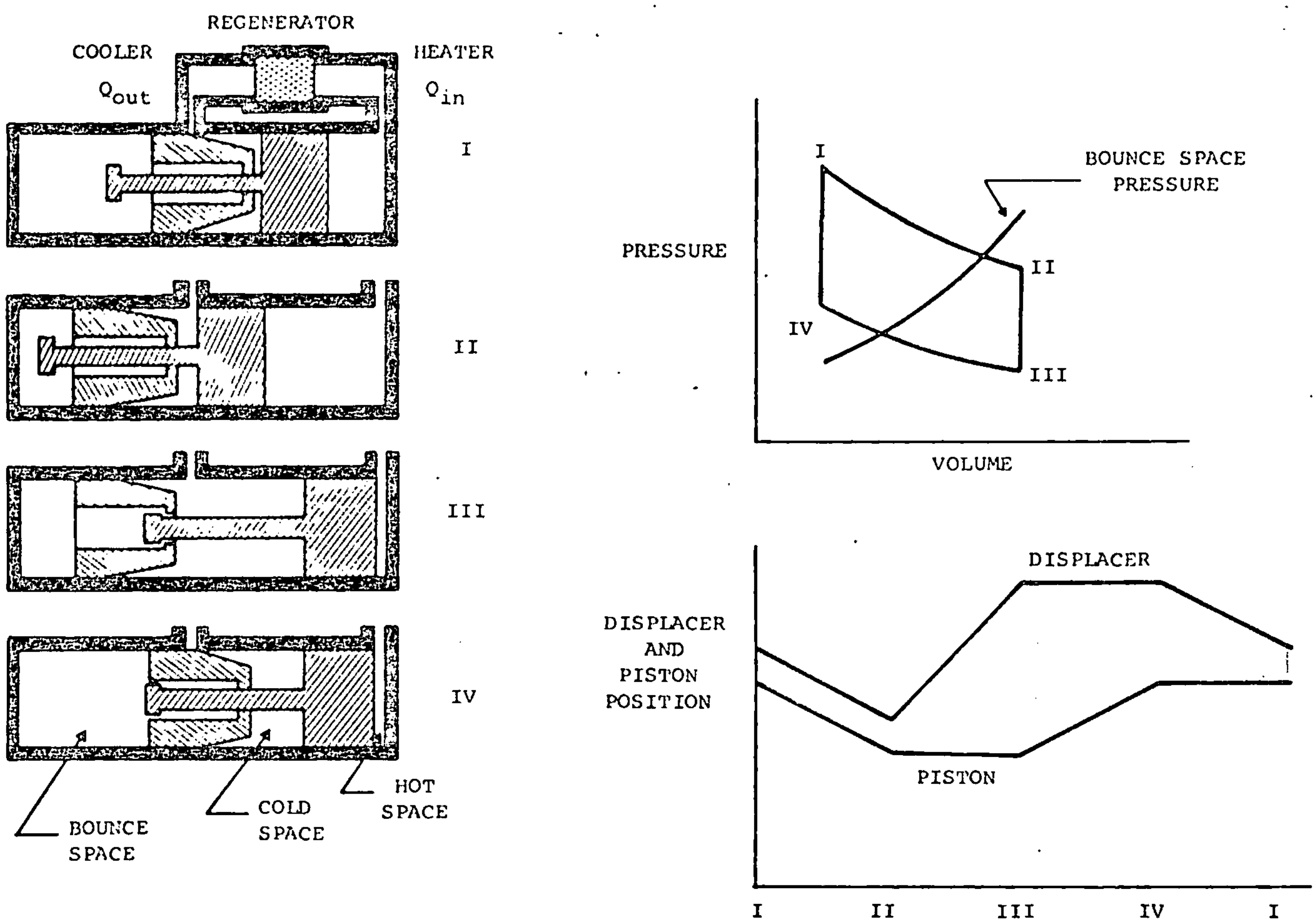


Fig. 2 Free piston Stirling engine cycle

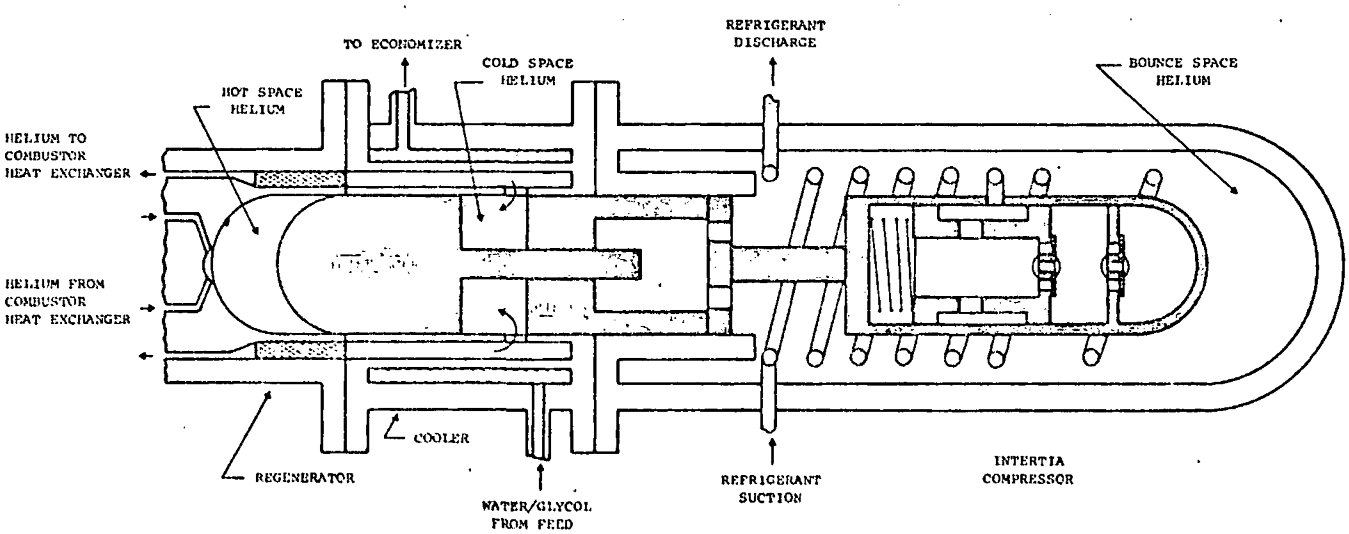


Fig. 3 Free piston Stirling engine with inertia compressor

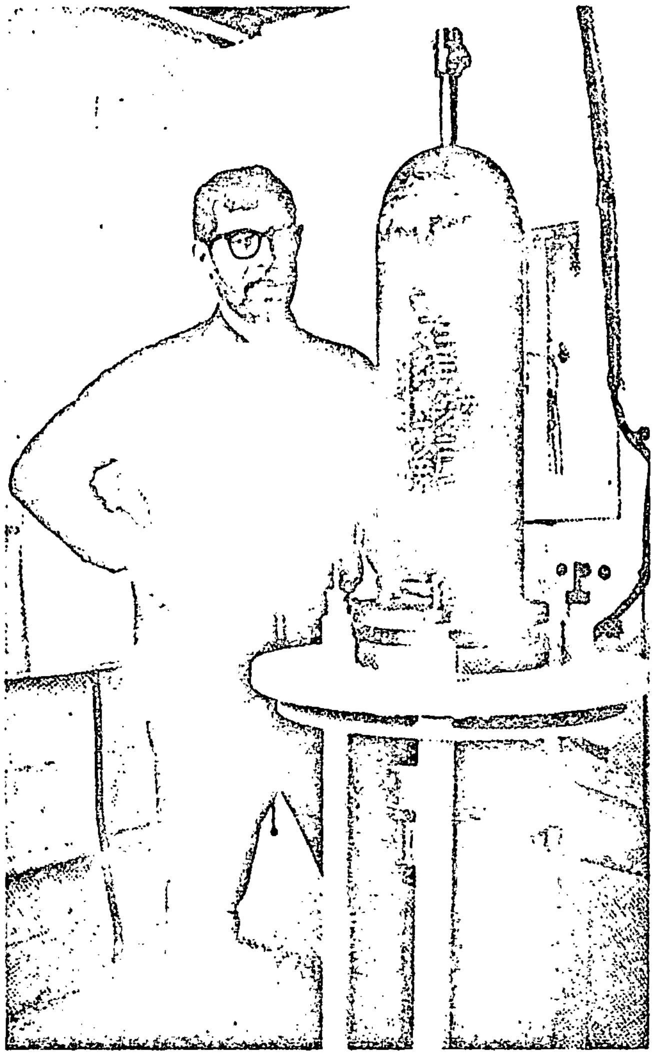


Fig. 4 Three-ton gas-fired free piston Stirling engine with inertia compressor assembly

In order to take the work out of the engine, a shaft could be brought through the engine housing and connected to the piston of a reciprocating compressor. However, this would require a sliding seal that would be effective with the 1000 psi helium in the bounce space. An alternative technique is to place an inertia compressor within the engine's crankcase and bring the refrigerant through the engine casing. This would permit the engine to be hermetically sealed. Fig. 3 illustrates how this is accomplished. The engine's piston is coupled to the compressor cylinder housing of the inertia compressor. The mass of the compressor aids in keeping the proper phase relationship between the engine piston and displacer. As the engine piston-compressor cylinder oscillate, the compressor piston tends to oscillate out of phase with the cylinder providing a pumping action. The compressor assembly is hermetically sealed in the bounce space of the engine. The refrigerant is transmitted to the compressor through a steel tube spring.

This Free Piston Stirling Engine design was invented by Dr. William Beale, and the development of the 3-ton gas-fired engine and compressor assembly shown in Fig. 4 was carried out under AGA funding by Dr. Beale's company, Sunpower, Inc.

The Free Piston Stirling Engine and inertia compressor can be used in a heat pump system as shown in Fig. 5. The refrigerant loop operates in the standard heat pump cycle. An additional heat transfer medium is used to recover the engine's waste heat for supplemental heat. In the heating mode, the water-glycol loop will transfer heat to the indoor coil; in the cooling mode, to the outside coil.

BRAYTON/RANKINE

Another gas-fired heat pump concept being considered by AGA is the Brayton power cycle (gas turbine) driving a vapor compressor in a Rankine cycle. The objective of the program is to develop a subatmospheric gas-fired Brayton cycle engine driving a centrifugal R-12 compressor via a hermetically sealed magnetic coupling. The development is tailored to a 10-ton rooftop type vapor cycle air conditioner with the engine and compressor replacing the electric motor and positive displacement compressor.

During heat pump operation, the exhaust heat from the engine will be utilized in addition to the vapor cycle unit operating in the heat pump mode, thus providing a heating COP as high as 1.5 and a cooling COP of 1.10 as an air conditioner. When developed, this concept can provide an all gas year-round space conditioning unit which can reduce the gas consumer's energy expense, reduce seasonal peak load demands, and conserve source energy.

The program is now well into the fourth phase of development. Phase I of the program was cycle optimization and development of a combustion heater and heat exchanger; Phase II was development and test of the R-12 compressor and magnetic coupling; and Phase III (which was completed in 1975) was to conduct a breadboard system test, demonstrating the feasibility and performance of the system. The breadboard system, after compensating for laboratory losses, showed a corrected COP of 1.18, indicative of what a production configuration can achieve. The breadboard system included the R-12 Freon compressor developed in Phase II, a breadboard gas engine driving the Freon compressor and a Dunham-Bush 15-ton refrigeration system.

Phase IV of the program includes development of the prototype rotating group to match with the R-12 Freon compressor. The air side rotating group will be a 90,000-rpm machine incorporating a magnetic coupling to the Freon compressor, air bearings and designed for 1500F air inlet temperature. The machine will provide approximately 13 hp to drive the Rankine cycle system. Testing of the rotating group in the laboratory test setup will be accomplished during this phase of the program. Other activities included in this phase of the program are a marketing survey and analysis for a production system in the size range from 3-ton to 100-ton in the 1980 time frame. This effort includes a first iteration cost analysis of the system and comparison to existing equipment to aid in determination of the potential market. Cycle optimization and initial design-to-cost system studies are simultaneously being conducted.

ORGANIC ABSORPTION

Units operating on the absorption cycle to provide space cooling and refrigeration have been commercially available for many years. The most common systems have working fluids of lithium bromide/water and ammonia/water. These systems are well known and characterized. In the lithium bromide/water systems, water is the refrigerant, thus restricting cycle operation to temperatures above the freezing point of water. This limitation makes the application of lithium bromide/water systems as heat pumps, impractical.

However, ammonia/water absorption refrigeration systems can be converted to operate as heat pumps, just as mechanical refrigeration systems can be converted. Studies using ammonia/water working fluids have predicted the performance for a gas absorption heat pump at nominal heating and cooling conditions. These studies show potential advantages for gas absorption heat pump systems over conventional systems, particularly for heating.

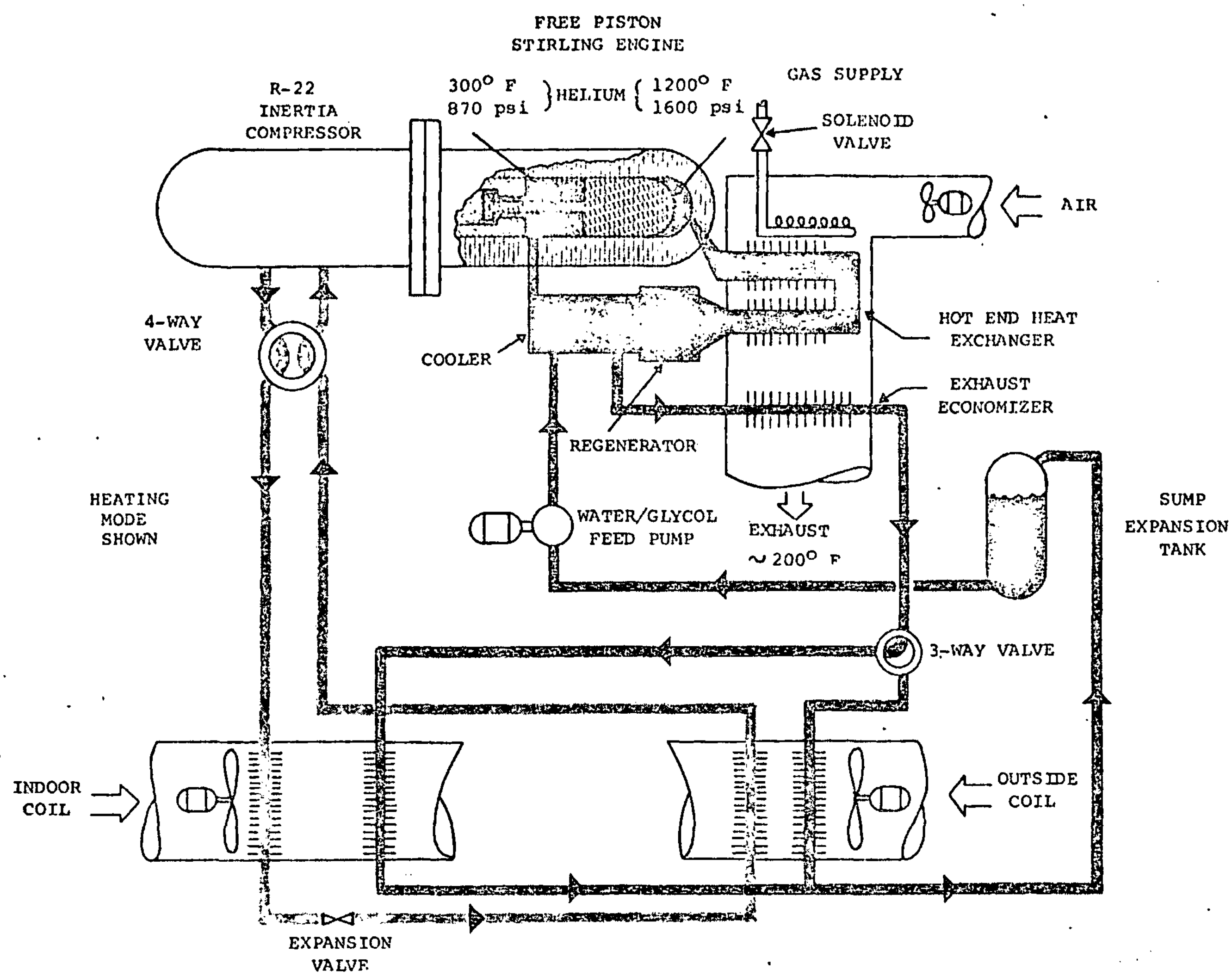


Fig. 5 Stirling/Rankine heat pump schematic

Three gas absorption heat pump concepts have been investigated in a study conducted for AGA by General Electric. An ammonia/water system, like today's ammonia/water absorption chiller, was taken as the baseline system. The Whirlpool Corp. has designed and built a prototype 3-ton ammonia/water heat pump which demonstrated the performance indicated by the studies mentioned above. Because this system requires relatively little modification of currently marketed hardware, it represents a potentially near-term gas-fired heat pump and was therefore selected as the base-line gas absorption heat pump concept.

Two advanced concepts considered to be representative of performance improvements achievable by 1980 were evaluated. The first of these is an upgraded ammonia/water system which operates at boiler temperatures on the order of 350-400F and employs a novel generator-absorber heat exchanger to reduce the gas heat input requirements, thereby increasing the cycle COP. Refrigeration COP improvements of 75% over today's ammonia/water systems appear achievable. The second advanced system consists of a cycle employing a new absorption working fluid combination. The advanced fluid is used in a system having the same basic components as the ammonia/water system.

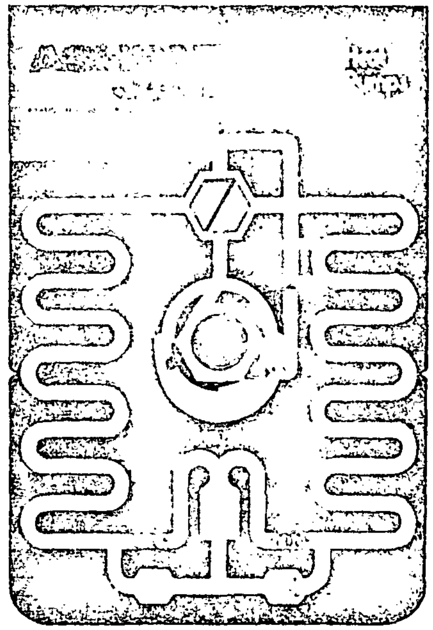
SUMMARY

Of three gas-fired heat pump concepts being investigated, the Stirling/Rankine system is, by far, the best performing system. The COP of the gas absorption system does outperform the Stirling/Rankine heat pump at ambient temperatures below 0F, but this is a result of an assumption that the Stirling/Rankine system is operated at constant speed. On the cooling mode side, the Stirling/Rankine is the only gas-fired heat pump system that can compete with an electrically-driven vapor compression system.

In system capacity relative to the design point cooling capacity relative to the design point cooling capacity, the Stirling Engine efficiency is so high that the engine waste heat is not as significant in contributing to the overall heating capacity as the Brayton/Rankine system. As a consequence, the heating-to-cooling ratio of the Stirling/Rankine gas-fired heat pump may require considerably more make-up energy in the cooler climates, unless multi-load operation proves feasible. □□

This report was prepared by the Research and Engineering Division Staff of the American Gas Association from data provided by the General Electric Corporation under A.G.A. Contract HC-115-1 and from the paper "On-Site Heat Activated Heat Pumps" by D. D. Colosimo.

THE UNITARY HEAT PUMP INDUSTRY—25 YEARS OF PROGRESS



The heat pump industry came up the hard way. It had no government funding or subsidies. The industry did the work and took the risks, made errors and corrected them. After 25 years, the industry is scarred, but strong, with a demonstrated record of responsibility and accomplishment. This paper was presented before the 39th Annual Meeting of the American Power Conference last April, sponsored by Illinois Institute of Technology.

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Unitary Air Conditioners & Heat Pumps

QUIETLY and without fanfare, the unitary heat pump industry is celebrating its 25th anniversary this year. It all started in 1952. For about the first ten years the industry experienced excellent growth. Then, as a result of some severe equipment reliability problems, the industry found itself operating in a survival mode. Maintaining momentum was made more difficult by the continued decline in electrical energy rates which brought increasing competition from electric furnaces. This caused the industry to operate on a plateau for about ten years. Then came the early 1970's. The industry had equipment reliability under control, but still suffered from the poor image established in its early years. At that time we entered the era of the energy crisis—soaring fuel costs, allocations, curtailments, and rapidly rising electric energy costs. In many cases, due to the lack of a reasonable alternative, many were forced to try a heat pump. They did—and found them not only completely satisfactory, but preferred. As a result, the heat pump industry is now experiencing a sales boom.

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PRIOR TO 1952

The basic principle of the heat pump was first proposed by Nicholas Carnot in 1824. This theory was advanced 30 years later in the early 1850's when Lord Kelvin proposed that refrigerating equipment could be used for heating. The heat pump remained a researcher's curiosity for several decades. Many scientists and engineers pursued investigations in an attempt to develop systems and hardware which were feasible for comfort heating. These investigations continued for 80 years.

In the mid-1930's several manufacturers became interested in the possibility of developing cost-effective products based on the heat pump principle. Customized systems were designed using the heat pump principle for comfort heating and demonstration installations were made. There were a limited number of these demonstration projects as they were all privately financed.

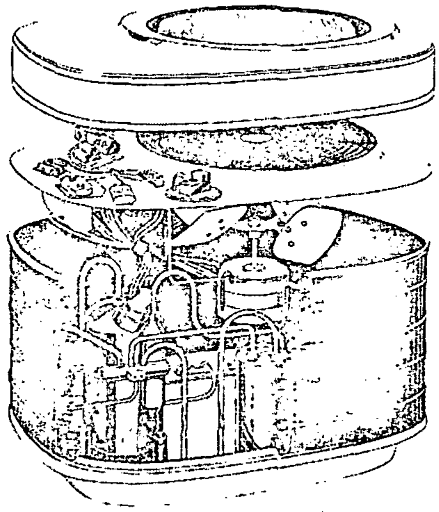
These projects confirmed the applicability the heat pump principle for comfort heating which had been predicted by Kelvin. They also provided performance data which enabled more accurate prediction of operating efficiencies. This activity was interrupted by World War II which diverted the technical skills of industry to more urgent matters. Interest resurned after the war and there were many more demonstration projects installed in the late 1940's. It became evident at that time that if there was to be broad acceptance of heat pumps for comfort heating, products based on the unitary concept would have to be developed. By unitary, is meant a refrigeration system which is factory-engineered and factory-built, then shipped to the field in one or two assemblies. So, in the late 1940's and into the early 1950's development work continued on unitary heat pumps for residential and small commercial installations. Products of this type were offered for sale in quantity for the first time in 1952.

1952 TO 1963

In the first full year of production of unitary heat pumps, approximately 1000 units were shipped by the industry. The early growth was slow—2000 units in 1954, 10,000 units by 1957—but by 1963 shipments has increased to 76,000 units per year (Fig. 1).

It was found in the early days that two essential conditions had to exist to generate consumer interest in a heat pump: A need for cooling as well as heating to justify the equipment cost, and relatively low electrical rates so that operating costs would be competitive with fossil fuel furnaces. For these reasons most of the early sales were in the southeastern states in areas where electric utilities encouraged usage through favorable energy rates for electric heating. Even though most of the installations were in the South, a significant number were also installed in northern climates—and equipment problems began to surface.

Early heat pumps used unitary cooling equipment hardware. Reversing valves and appropriate control hardware were added to cooling equipment. When operating at mild winter temperatures, the stress of these components is about the same as found when operating in the cooling mode. As outdoor winter temperatures drop, the stresses increase on these refrigerating components and the early heat pumps which lacked sufficient durability under these conditions had a high mortality. So, along with the growth through this period, the product was also achieving a poor reliability reputation. Something had to be done.



1964 TO 1971

The experiences of the 1950's and the early 1960's almost destroyed the heat pump industry. A product with such great promise had its reputation tarnished by low product reliability and high service costs. Many manufacturers dropped heat pumps from their product lines. Some curtailed sales by region and would only make installations in the more southern climates. Also, during this period the federal government which had purchased many heat pumps in the early days for military housing, imposed a ban on the further installation of heat pumps due to the poor reliability records.

Then a new threat to the industry emerged. During the 1960's, electric rates continued to decline (Fig. 2) and the most formidable competitors to heat pumps were electric furnaces rather than fossil fuel furnaces. Many potential heat pumps purchasers were switching to electric furnaces. The low first cost and higher reliability of the electric furnace were extremely attractive, and even though energy usage was higher, the lower energy cost per kWh kept current monthly utility bills in the competitive range. The continuing declining energy rates suggested even lower monthly utility costs in the future. For an eight-year period, industry shipments remained essentially flat. But the industry perservered.

During this period, improved designs were developed. These designs used refined components which were designed to withstand more severe heat pump stresses. During this period, recognizing their responsibility to their customers, many electrical utilities set up programs to assure good heat pump installations. These programs consisted of major training efforts for installers, certification of qualified installers, and the collection and dissemination of service and reliability information. This information was fed back to manufacturers to help them determine areas of design weakness.

Also, to add integrity to manufacturer's performance claims, the industry trade association, the Air-Conditioning and Refrigeration Institute, initiated in 1964 a program which certified the basic performance characteristics and the cooling and heating capacity of the product.

The industry had done its homework—it had corrected its faults—it was ready—but the consumers were not.

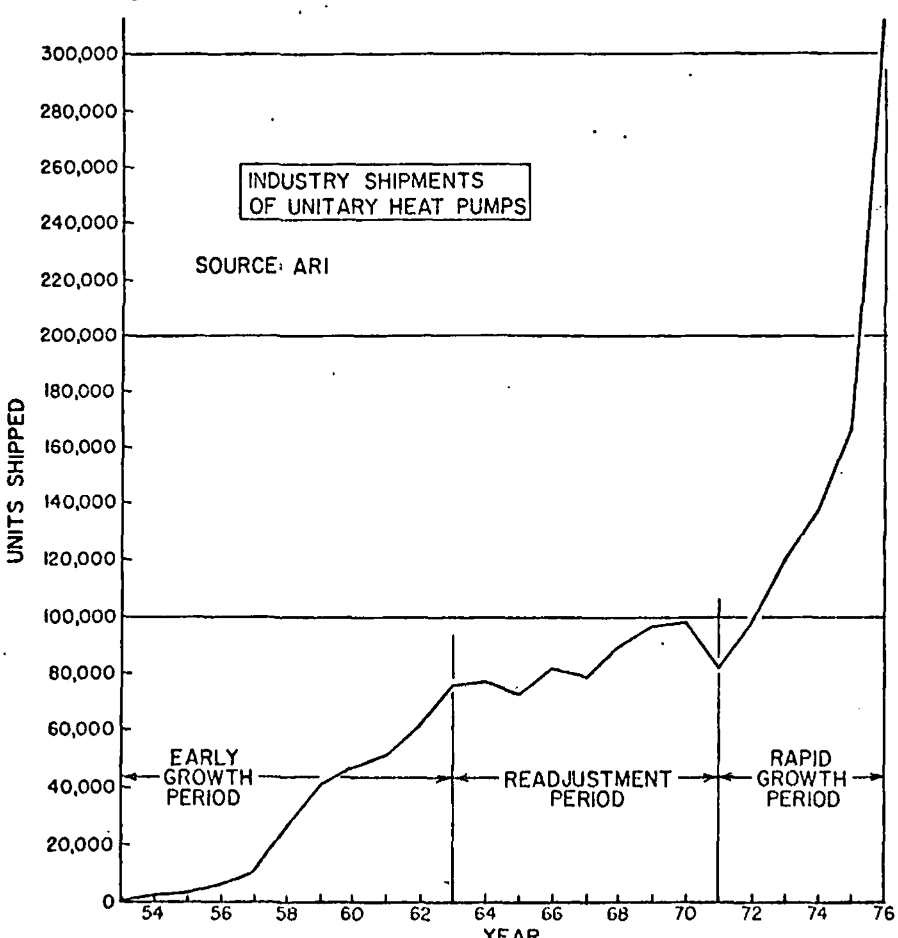


Fig. 1

1972 TO 1977

In the early 1970's, it began to be widely recognized that the energy resources of our planet were finite. We also entered an era of energy shortages, either due to lack of availability of fossil fuels or due to the limited capacity of electric generating plants and the related distribution system. This energy situation which has been referred to as the "energy crisis," also brought about a reversal in the downward trend of electric energy rates and they started to rise sharply. In many areas of the country, curtailments were placed on the use of some fossil fuels for heating and the only alternatives were oil furnaces, electric furnaces and heat pumps.

The price of fuel oil also advanced rapidly during this period and with the rising costs of electric energy for many customers the only reasonable alternative was the heat pump. Also, as we entered this period, the readjustment of the 1960's had been effective. The reliability of improved designs had been established, and while questions about the reputation earned in the 1950's still remained, the evidence indicated that the heat pump industry was ready. The combination of these events had led to a period of explosive growth and the industry moved from a level of 82,000 units shipped in 1971 to over 300,000 units shipped in 1976.

In a short five year, shipments have increased by a factor of 4. The industry which stumbled through its early years—the industry which corrected its early errors—the industry which had persevered had come of age.

In the first 25 years of the heat pump industry, there have been over 1,800,000 unitary heat pumps installed (Fig. 3). Some of these have been installed as replacements, so if we consider an average life of 15 years, there are approximately 1,600,000 unitary heat pumps in use today. These products are not experimental, they are proven devices which were selected by consumers in a highly competitive heating industry.

THE FUTURE

What does the future hold for the heat pump industry? For the near term, all projections show great growth. The desired energy scenario for this country indicates a decline in the use of scarce fuels such as natural gas and oil for comfort heating. Since the heat pump is the most cost effective electric heating system available, it appears to be the most reasonable alternative. Some of the energy usage aspects of various electric input heating systems are shown in Fig. 4. An *electric furnace* is 100% efficient so the energy delivered is equal to the energy consumed. A heat pump has an efficiency of about 200, so that for every 100 units of energy delivered it consumes only 50. The energy saved by using a heat pump instead of an electric furnace is 50 units. In a life-cycle costing exercise, this savings in energy usage is used to offset the higher first cost of the heat pump.

As we progress into the future, heat pumps will become more efficient, moving toward the 300% level. Note that the savings with a 300% heat pump over a 200% heat pump is only 17 units. To maintain cost-effectiveness, the incremental first costs to achieve the higher efficiency must not be excessive. Note that as we move to increments of higher efficiency, the energy saved becomes less for each increment.

Solar heating systems have seasonal efficiencies which range in the 400 to 600% area. The energy consumed in these systems includes the electric power required for pumps, fans and blowers and may include back-up heating for extended cloudy periods. Even though these efficiencies are high, note that the pay-back funds available from decreased energy usage is relatively small. Solar heating will find wide usage only if its incremental first cost over a heat

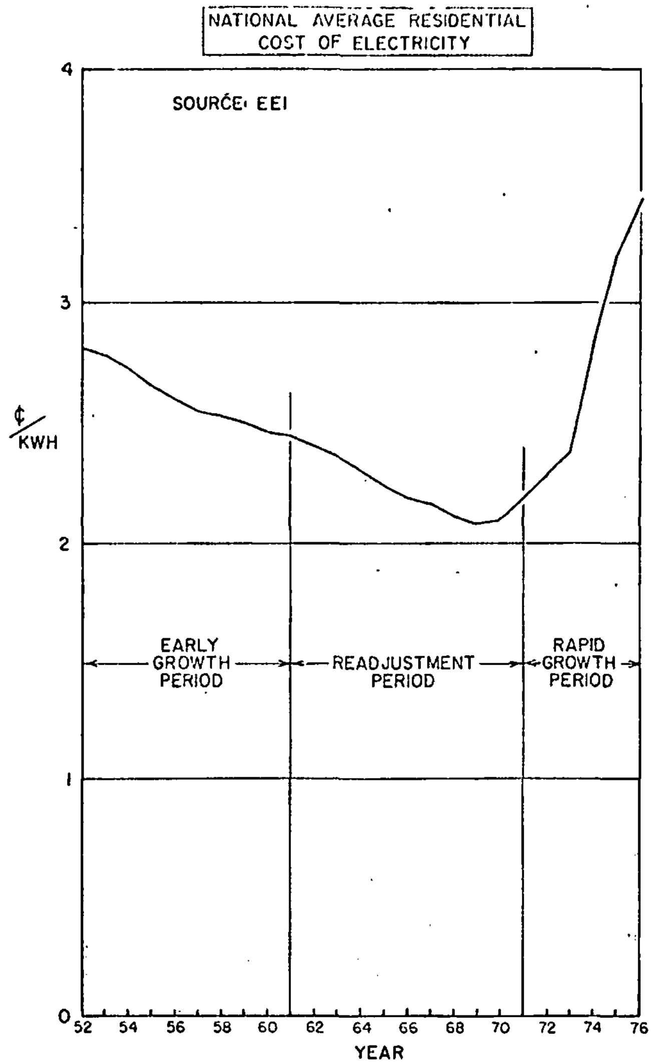


Fig. 2

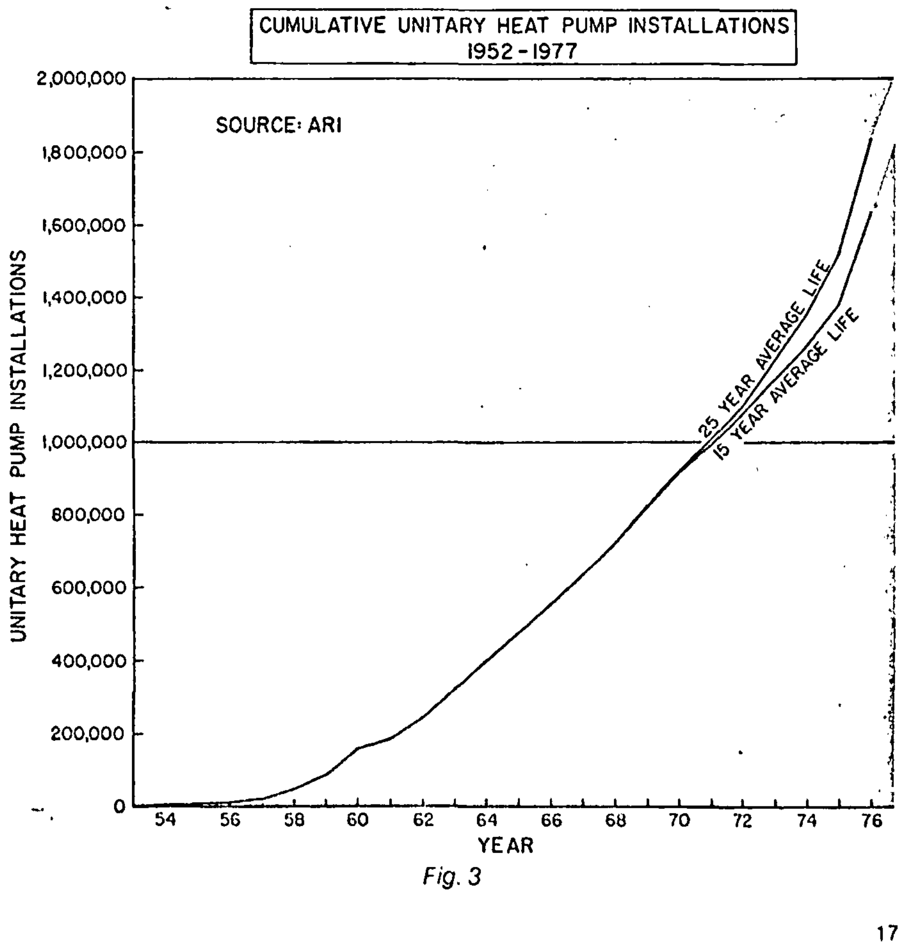


Fig. 3

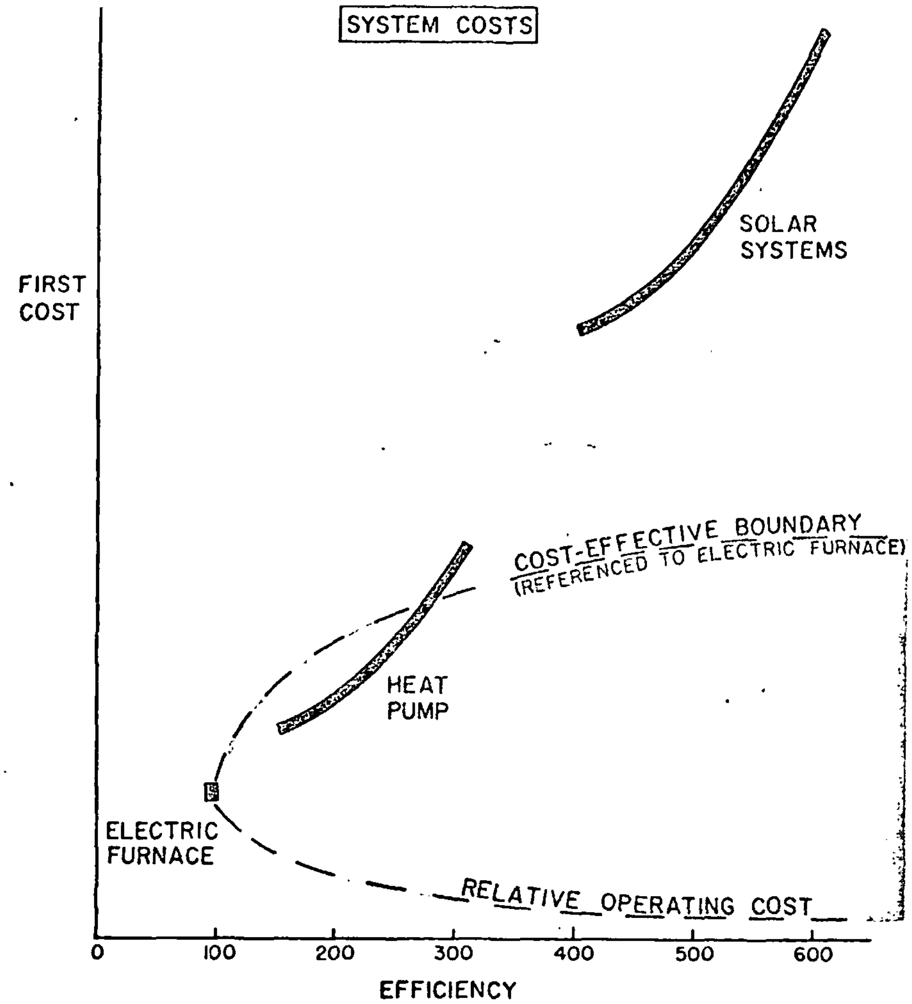


Fig. 4

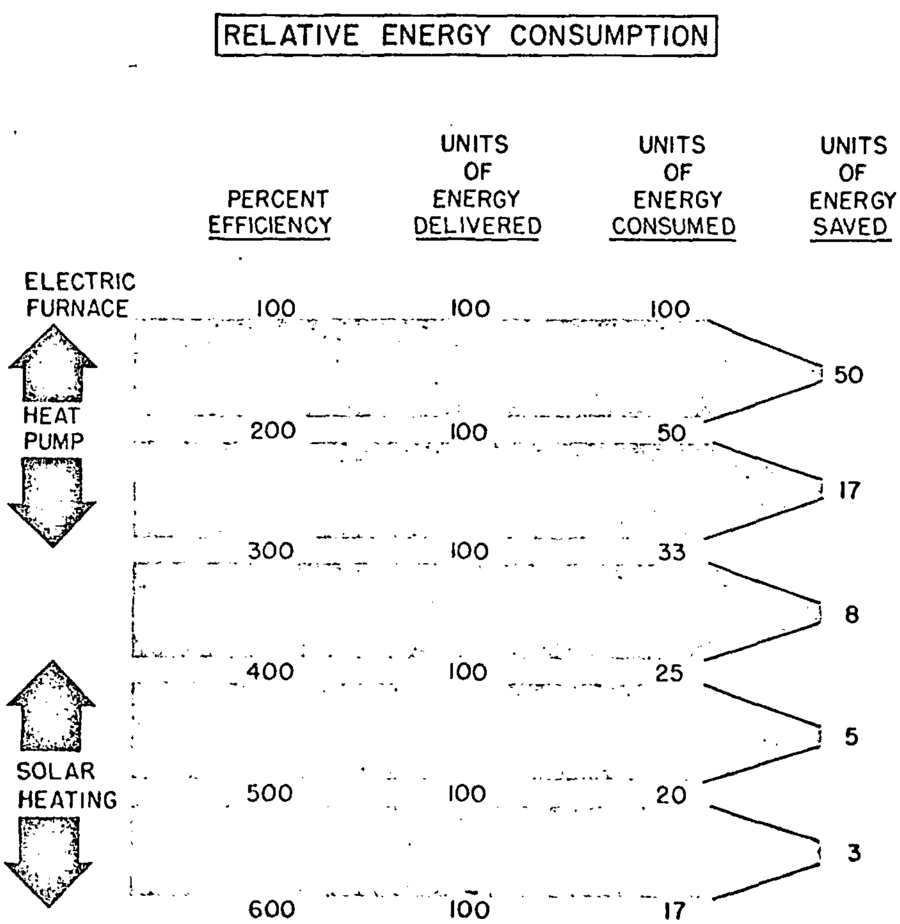


Fig. 5

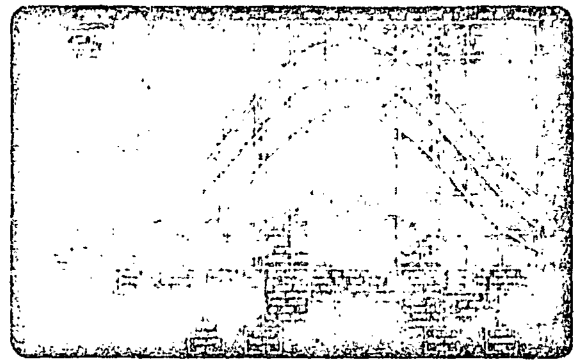
pump is reduced significantly from today's levels. This is demonstrated in Fig. 5 which shows the relative first cost positions of electric furnaces, heat pumps and solar systems. Note that for heat pumps costs increase with improved efficiency. Solar systems also have varying costs depending on efficiency levels. As efficiency improves, the relative operating costs decline. For a certain set of conditions; that is, energy costs, interest rates, and pay-back period, a cost-effectiveness boundary can be drawn. This line is proportional to a mirror image of the first derivative of the relative operating cost line. For simplicity, we have shown only one such boundary. The boundary would shift as energy costs, interest rates and pay-back periods are varied. A first cost above this boundary is considered not to be cost effective. A first cost below the boundary is considered to be cost effective. Note that heat pumps are cost effective at most levels of efficiency; however, the higher efficiency heat pumps, due to higher first costs, may cross the cost-effective boundary and be unattractive in the marketplace. The crossover point could shift out in time as heat pump designs improve in terms of cost-performance relationships, as energy costs rise, or as acceptable pay-back periods lengthen. Today solar systems are well above the cost effective line. It is questionable whether they will ever be cost effective. Does that mean that we will be limited to the efficiencies that can be achieved by heat pumps alone? Not necessarily. The best solution for the future may be a hybrid system which integrates the cost effectiveness of the heat pump with improved performance levels of solar systems.*

CONCLUSION

Lord Kelvin expressed the concept for a heat pump in the early 1850's. It was 80 years before this principle was reduced to practice for use in customized installations and another 20 years before unitary products became commercially available. The unitary heat pumps industry took 25 years to achieve maturity. It had a troubled adolescence; it went through a period of readjustment; and is now established on a firm, sound base. All indications are that in the next 25 years the heat pump will become the dominant comfort heating product. The unitary heat pump has such outstanding credentials, we doubt if there will be any contenders for this position. □□

***EDITOR'S NOTE: See article by Richard A. Biehl in this issue.**

The Annual Cycle Energy System: A HYBRID HEAT PUMP CYCLE



Suppose there was a hybrid heat pump system that could supply all of a building's heating demands, at any temperature whatsoever, without the use of additional heat: No electric resistance, no oil, no gas, no coal. The Annual Cycle Energy System (ACES) does this and much, much more.

RICHARD A. BIEHL
Member, ASHRAE

ASK someone outside our profession to identify the best system to supply a building's heating and cooling requirements while consuming the least amount of energy. The answer you will probably receive is, "The heat pump." To those of us who exist in the world of COP's, the heat pump is realized to be more efficient than most conventional systems. In spite of its significant advantages, however, there is still one major facet of the heat pump's operation that gives us pause for concern. For example, an air-source heat pump with an outdoor evaporator heats very well down to ambient temperatures of about 4C. Below this temperature, frosting on the coil or merely depressed evaporator temperatures make the use of supplementary heat necessary. Even water-source heat pumps often require supplementary heat to maintain the source temperature. The addition of this heat causes COP's to fall off—tarnishing the image of an otherwise efficient system. Until the Annual Cycle Energy System (ACES) came along, that is.

THE CONCEPT

Very simply, ACES operates as an air-source heat pump down to the 4C temperature. Below that, the system becomes a sort of water-source heat pump which uses the latent heat of fusion given up by water in its change of phase to ice (3.35×10^5 J/kg.

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Astute readers will remark that for a heat pump cycle to produce this phase change, depressed evaporator temperatures with their correspondingly lower efficiencies are still required. Do not forget, however, that, as the name implies, this is an annual system. The ice that is produced while supplying the winter heating requirements of a building is stored. This ice is melted during the summer months to satisfy the cooling requirements in the building. Feasibility studies, adapting ACES to a variety of structures, have shown that the system will provide anywhere from 35 to 70% of the building's annual cooling requirement from the stored ice. The actual amount depends upon the occupancy and geographical location of the building. This is done at the expense of running only a small circulator to move water over the ice. In fact, the Energy Research and Development Administration (ERDA) at Oak Ridge, TN, did a computerized simulation of a system now under construction at the Veterans Administration Center in Wilmington, DE. It predicted that, with no attempt to control the ice "inventory," the ice would last until mid-July. Expressed differently, this is six weeks of cooling at a COP of 26! It certainly doesn't take a full-blown life cycle analysis to see that six weeks cooling is a fair trade against reduced heating efficiencies below 4C.

Greater understanding of the full potential of the Annual Cycle Energy System can be achieved from studying an actual design. One of the better examples is the only commercial scale application of ACES at this time. It is the previously mentioned system now under construction in Wilmington. The system is housed in a separate 12.19m × 15.24m building which the Veteran's Administration has appropriately nicknamed the "Energy Bank." It supplies the annual heating and cooling requirements for a new 60 bed Nursing Home Care Unit which is also under construction at the same site. Keep in mind, that although the system has numerous methods or "modes" of operation, there are three overriding objectives that ACES seeks to attain:

- Energy conservation;
- Load management;
- Use of renewable resources.

There are seven (7) basic modes of operation by the Energy Bank. Four of these modes are used to produce heat for one use or another. The remaining three are used to produce cooling. The first modes to be investigated are the four heating modes. Two of these heating modes supply heat to the Nursing Home by way of a dual temperature water system. They are identified as Mode A and Mode B.

In Mode A heat is removed from outdoor air and imparted to the dual temperature water. To accomplish this, the system performs as a typical air source heat pump. Air passing over the coil of the outdoor unit (acting as an evaporator) imparts its heat to refrigerant within the coil causing it to vaporize. This refrigerant is then compressed to a hot gas and transferred to the double-bundle condenser. Here, the hot gas condenses on the tubes giving up its heat to the dual temperature water. The refrigerant drops to the receiver on its way back to the outdoor unit to complete the cycle.

A micro-processor within the Energy Bank will periodically accumulate data from a multitude of sensing locations and execute COP calculations. This is to determine if some other mode of operation could be employed which would result in the use of less energy. If, for instance, the building is still calling for heat, and if low evaporator (outdoor) temperatures cause the system COP to drop, the system will revert to Mode B. In Mode B, heat is removed from the ice tank and imparted to the dual temperature water. The ice tank functions as a heat source for the evaporator. A solution of 30% methanol and water is circulated inside coils of the ice tank. The brine then circulates to the shell of the brine cooler giving up its heat content to vaporize refrigerant within the tubes of the cooler. This refrigerant is then compressed, and the hot gas transferred to the double-bundle condenser. Heat from the gas is given up to the dual temperature water as it condenses. The liquid refrigerant returns to the brine cooler completing the cycle.

Mode B is also employed when the outdoor temperature is right at the 4C temperature range. At this temperature, moisture in the air will usually form frost on the outdoor unit coil, virtually preventing heat transfer. When this occurs the micro-processor will automatically switch the system from Mode A to Mode B for half an hour. During this time, the outdoor unit fan is energized at high speed for the purpose of defrosting the coil. This gives rise to a new method of operation which, while it is not totally distinct from Mode B, is identified as Mode B2. Mode B2, therefore, is identical to Mode B except that the outdoor unit fan is also run.

The third heating mode is identified as Mode H. This mode is used for "inventory" control within the ice tank only. The ice within the tank is not permitted to exceed 75% of the volume of the tank. To keep from exceeding this level, solar heat is used to melt the ice when it is available. Specifically, solar radiation is transferred to the methanol brine through a low-cost solar collector/nocturnal radiator. This heated brine is then circulated through the coils in the tank, melting the ice. Naturally, the micro-processor will be equipped with seasonal operation parameters to prevent melting too much ice at the beginning of the cooling season.

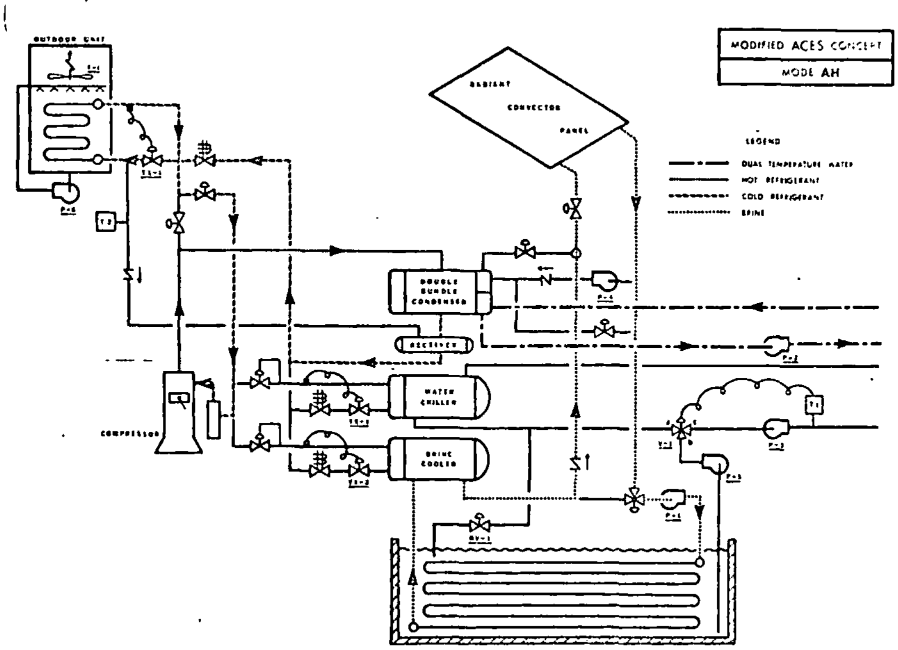


Fig. 1 Mode A—Heating the building using outside air as a heat source

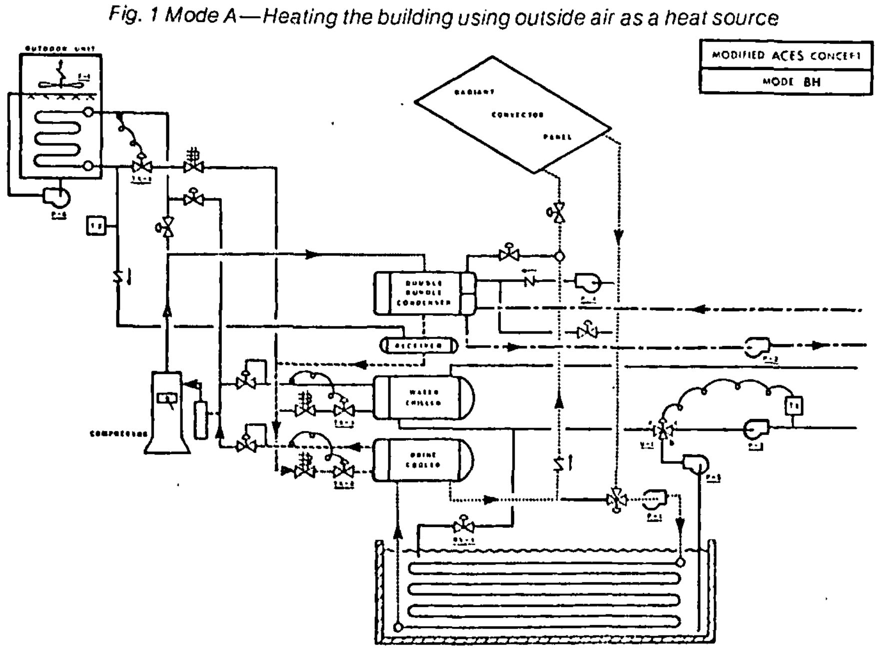


Fig. 2 Mode B—Heating the building using the ice tank as a heat source

HYBRIDS

At times, the micro-processor may deem it advantageous to use the collector/radiator at the the same time that another heating mode is being employed. These modes are hybrids of the four basic heating modes and are identified as: Mode AH, Mode BH, and Mode B2H. In Mode AH, the dual temperature water heats the building, and both the outdoor unit and the collector/radiator are used as heat sources. Mode BH heats the building using both the ice tank and the collector/radiator as heat sources. Model B2H is, of course, identical to Mode BH with the exception that the outdoor unit fan is energized in an effort to defrost the coil.

The fourth and last basic heating mode is identified as Mode I which provides heat both to the building and to the tank through the heat pump cycle using the outdoor unit as a heat source. This mode would be used during periods of severely cold weather when the ice tank tends to approach its maximum capacity. If a break in the weather occurs, and the temperatures moderate above the 4C level, the Energy Bank is run to produce heat at its maximum capacity. The Nursing Home will not need all the heat output to satisfy its demands in this period of temperature weather. The excess heating capacity is used to melt ice in the tank by rejecting it through the summer bundle of the double-bundle condenser. Mode I uses the moderate outdoor temperatures to produce heat at a relatively high COP for use on another day when temperatures are lower. Additionally, it provides an alternate method vke inventory control for long sieges when solar heat is unavailable.

COOLING

The remaining three of the seven basic modes of operation are cooling modes. These are designated as Mode E, Mode F, and Mode G. Mode E is the first cooling mode to be considered. This method of operation demonstrates the true value of an annual cycle, for it is here that the Nursing Home's cooling demands are totally satisfied at the expense of a few

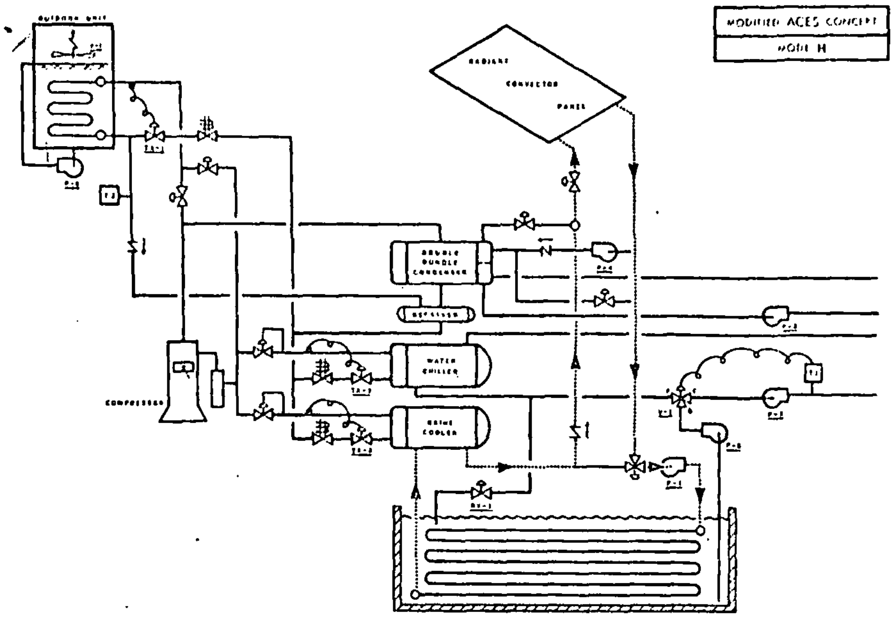


Fig. 3 Mode H—Heating the tank using solar heat

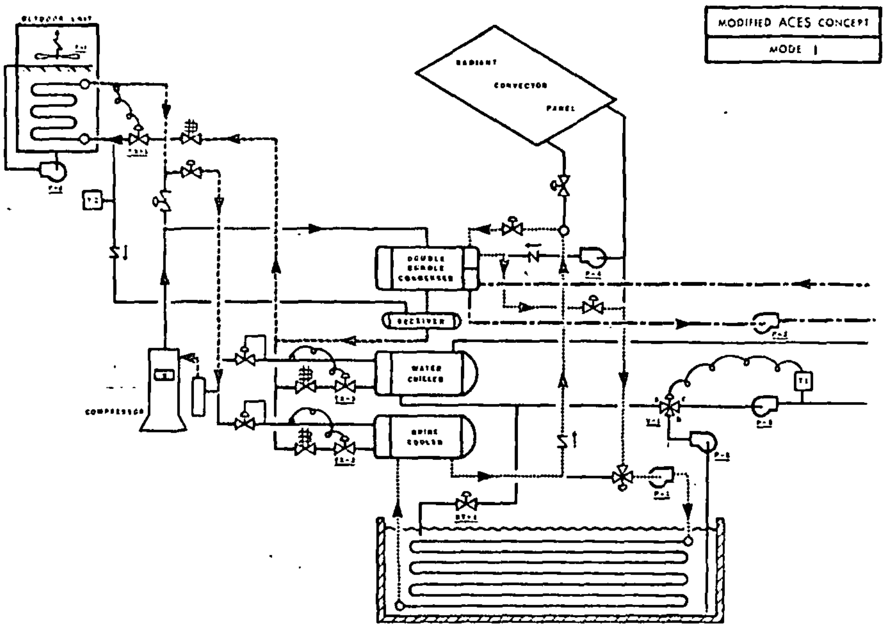


Fig. 4 Mode I—Heating the ice tank and the building using outside air as a heat source

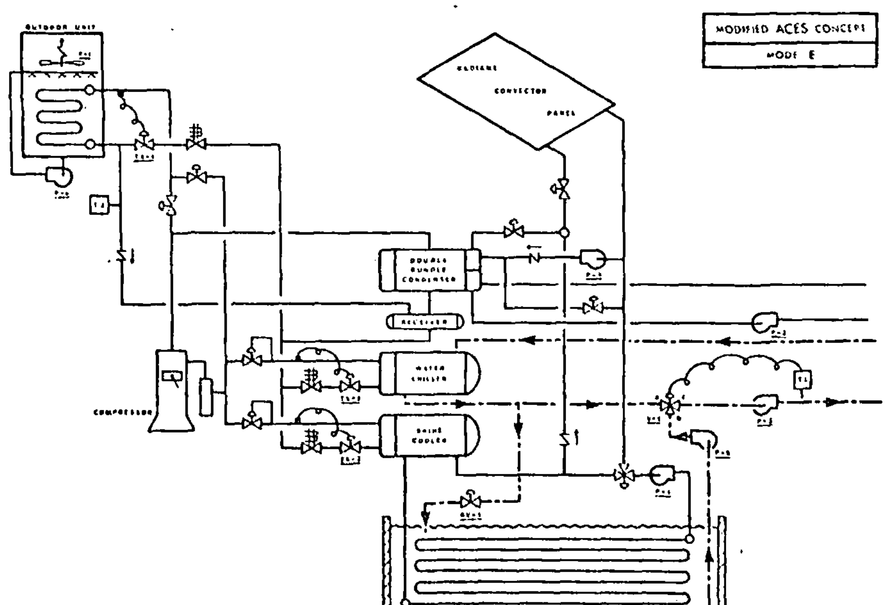


Fig. 5 Mode E—Cooling the building while rejecting heat to the tank

watts input to a circulator. All the ice which was created as a by-product of heating in the winter has water circulated over it. This 1.7C water is then mixed with the 12.8C return water to provide 7.2C to the dual temperature system of the building for cooling. This mode can be used to great advantage even after the ice inventory is depleted. In the latter part of the summer, the cooling for the Nursing Home will be done during the "off-peak" hours and stored in the tank as ice until the next day. At that time, the microprocessor will opt for Mode E and only a circulator will run. This mode is especially beneficial on that sweltkg August 21st at 3 p.m. because the time will eventually come when the power companies are no longer able to satisfy the high refrigeration loads of such a day. The normal demands of commerce and industry will be too great, and excess capacity for comfort cooling will not be available. The residents of the Nursing Home needn't be concerned. They can have their comfort cooling without adding to the power company's burden.

The second of the cooling modes is Mode F. In Mode F, the dual temperature water is cooled and the heat rejected through the outdoor unit. This is a conventional refrigeration cycle. The collector/radiator may also be used in conjunction with this mode, just as it was used with the heating modes. A hybrid mode, which is identified as Mode FH, results. In Mode FH, the collector radiator serves as a nocturnal radiator. The Energy Bank uses this mode on a clear night when on-line cooling is still required in the Nursing Home. If the temperature in the collector/radiator is lower than the temperature of the refrigerant leaving the outdoor unit, a small circulator is energized and brine is circulated from the collector/radiator through the summer bundle of the douuble-bundle condenser. The cooler brine causes the double-bundle condenser to become an area of lower pressure than the outdoor unit. Refrigerant will tend to migrate toward this shell to condense, substantially reducing the load on the outdoor unit.

The third and last of the basic cooling modes is the one that provides the ice for use at later periods in Mode E. In Mode G, the ice tank is cooled and heat is rejected through the outdoor unit. Heat of fusion is removed from the ice tank by freezing water to ice on the coils of the tank. This heat is carried by the methanol brine to the brine cooler. Here the vaporizes the refrigerant in the tubes of the brine cooler. The refrigerant then goes to the compressor where it is compressed to a hot gas and sent to the outdoor unit, which, functioning as an *evaporative condenser*, condenses the hot gas causing it to give up its heat. The liquid refrigerant returns to the brine cooler completing the cycle. As with Mode F, the collector/radiator may be used with Mode G. Once again, sensors in the collector/radiator and the refrigerant leaving the outdoor unit are used in comparing the two temperatures. If the temperature of the stagnant brine is lower, a circulator is energized to circulate the cool brine through the summer bundle of the double-bundle condenser. The two condensers—the outdoor unit, and the double-bundle condenser—then operate in parallel.

Although the Energy Bank has seven basic modes, this is only because the Nursing Home has a two-pipe fan coil system. If the distribution system were a four pipe rather than a dual temperature system, there would be another method of operation possible. Chilled water would be distributed to core of the building to remove the sensible heat of lights and equipment there. This heat would be used in a water chiller to vaporize refrigerant. The refrigerant would then be compressed to a hot gas and sent to the double-bundle condenser where it would give up its heat to the hot water system. The heat is returned to the perimeter of the building by way of the hot water. This mode has often been employed in the past as an energy conserving measure and is commonly referred to as "bootstrapping."

ADDITIONAL FEATURES

An additional operational feature has been incorporated into an ACES Demonstration House located on the campus of the University of Tennessee. The house is a cooperative venture of the Energy Research and Development Administration represented by the Oak Ridge National Laboratory, the University of Tennessee, and the Tennessee Valley Authority. The system in operation there is equipped with a de-super heater. The heat removed by the de-super heater is used for the domestic hot water. There is another difference between the VA's Energy Bank and ERDA's Demonstration house: The house does not have an outdoor unit. Its system relies entirely upon the collector/radiator and the ice tank for a heat source or for any heat rejection, depending on the season. The Energy Bank because of its size would require a huge tank to hold all the ice produced in the winter. Since the tank is the single most costly item of the system, some of the ice storage was sacrificed in the interests of the lower first cost. Consequently, every effort was made

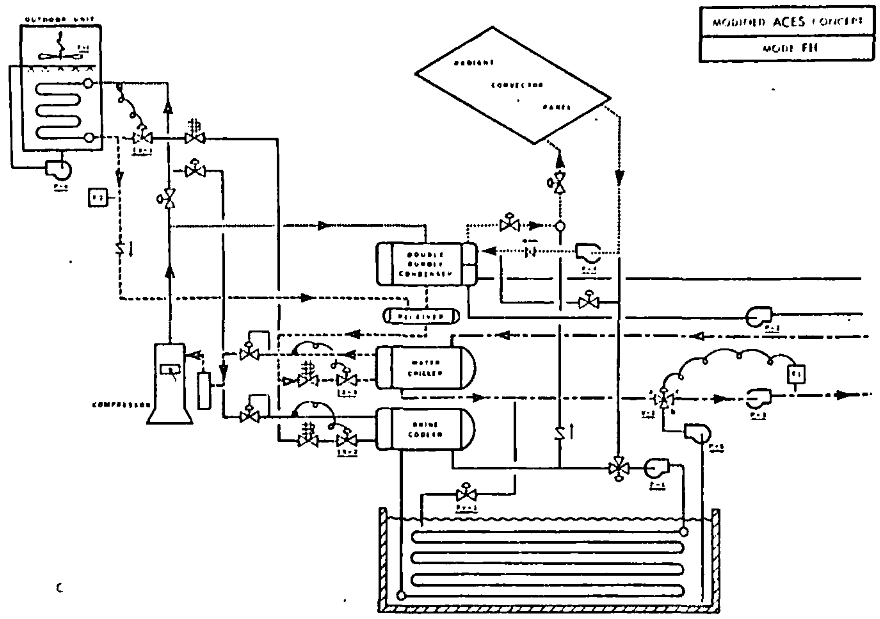


Fig. 6 Mode F—Cooling the building while rejecting heat to the outside air

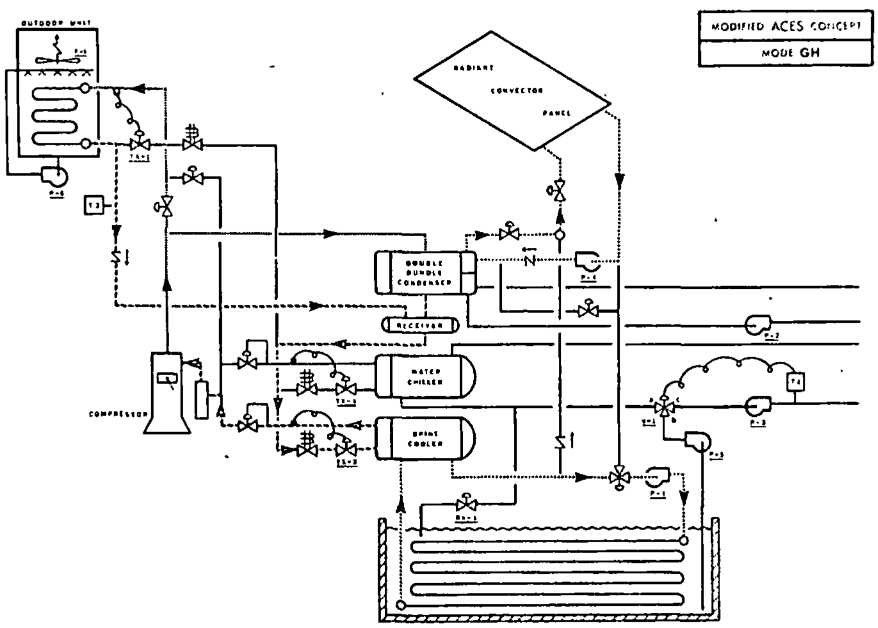


Fig. 7 Mode G—Cooling the ice tank while rejecting heat to the outside air

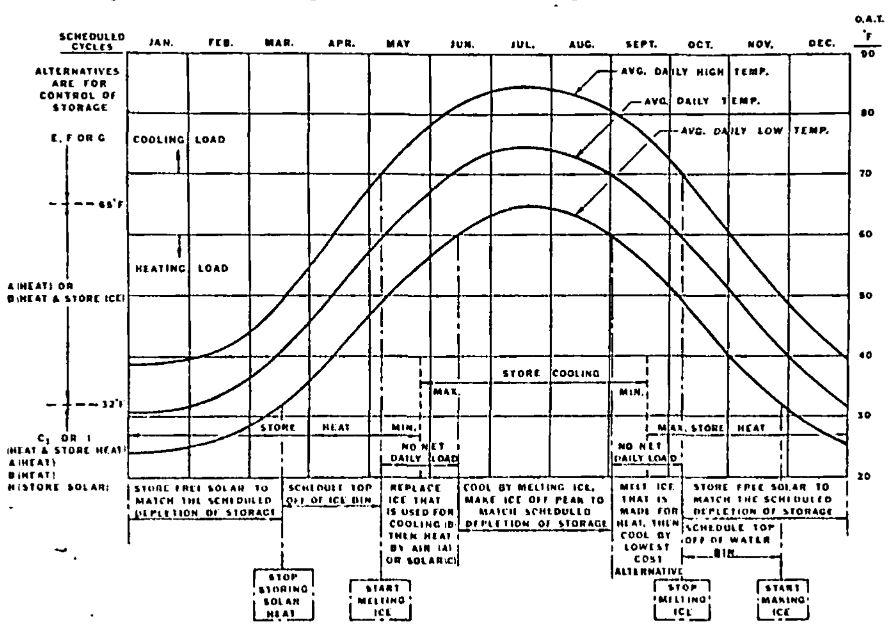


Fig. 8 Energy use strategy

to have as an efficient an outdoor unit as possible. An evaporative-condenser/evaporator was selected over a dry-type unit to lower the summer condensing temperatures. Also the unit was equipped with a two-speed, two-winding fan so that motor horsepower can be conserved when less air is required.

The solar collector/nocturnal radiators of both the Demonstration house and the Energy Bank are very similar. The Energy Bank's consists of 96 aluminum "fins;" each is 6.63m long. They are arranged in two rows of 48 fins each. The most optimum design for this project is a 5.08×10^{-3}m thick by 0.20m wide aluminum extrusion having a 1.52×10^{-2}m I.D. tube located on its central axis. The fin selected for the house has the same shape but different dimensions. This fin is 5.08×10^{-4}m thick by 7.62×10^{-2}m wide aluminum extrusion with a 1.27×10^{-2}m O.D. tube on its central axis. The fin on the house has essentially the same heat transfer properties as the one on the Energy Bank but must be supported more often.

The ice tank for the Energy Bank is constructed of reinforced concrete. It is 15.24m long by 12.19m wide by 3.35m deep and serves as a sort of basement for the Energy Bank. The inside of the tank has been cleaned and given a coating of waterproofing. There are 70 coils within the tank—35 in each half of the tank. They are separated by a distance of 0.33m center-to-center. The coils are fabricated of 1-1/4 inch, Schedule 40, black steel pipe. They are eight pipes high with hairpin bends on 0.33m centers. There are, effectively, two nests of pipes, each eight pipes high by 35 pipes wide by 6.71m long.

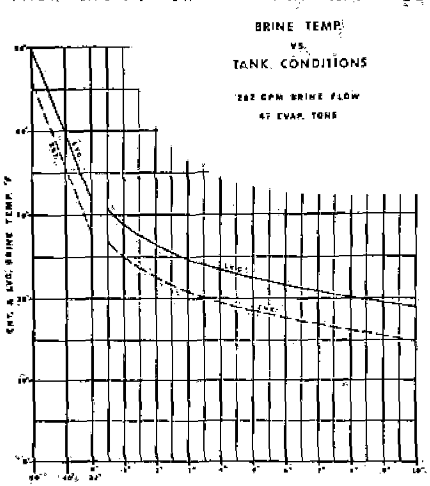


Fig. 9 Plot of ice thickness vs. Brine temperatures

The ice tank for the demonstration house is fabricated from formed blocks of expanded polystyrene held together with expanded metal ties. Waterproofing is accomplished by a vinyl liner like the ones used in backyard swimming pools. It is 5.33m wide by 5.87m long by about 3.05m deep. ERDA elected to use the same finlike extrusion for their ice coils that they used for their solar collector/nocturnal radiator. The fins are also arranged on 0.33m centers with 16 parallel circuits.

Both projects are equipped with micro-processors. Their responsibility is to control the system so that not only the immediate heating and cooling demands of the building are satisfied but also that the storage of heating and cooling in the ice tank is at its optimum potential. The overall philosophy of the ice storage principle is to enter the cooling season with the maximum storage of ice and use it all before the onset of the heating season. The processors have been equipped with seasonal parameters. Based on data regarding the ice inventory and this seasonal strategy, the micro-processor will override the COP calculations if another mode proves to be the most expedient over the long run. The processors also serve to acquire data from various sensing devices throughout the system. Information regarding heating output, outdoor air temperature, solar insolation, brine temperatures, ice tank temperature, percent of ice in tank, refrigerant temperature and power consumption of the compressors, pumps and outdoor unit is recorded and totalized at each change of mode. This information will eventually be compiled and used in writing guidelines for future ACES designers.

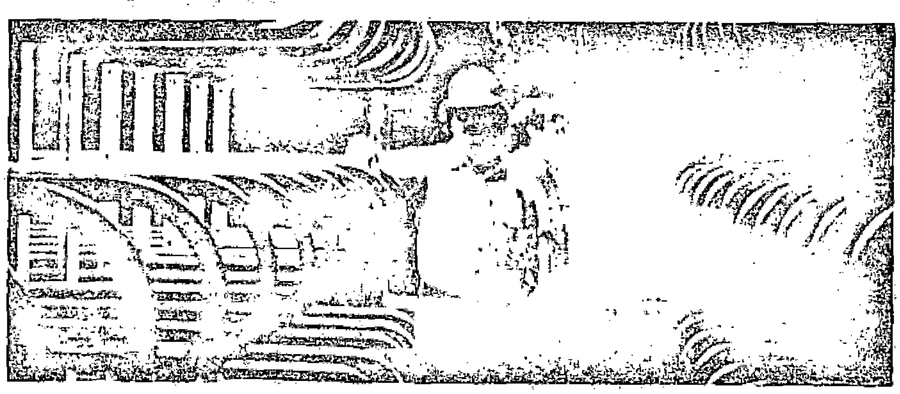


Fig. 10 View from inside ice tank of Energy Bank

The Energy Bank is now under construction in Wilmington. Most of the equipment is ready for installation and the system should be completely "shaken-down" and prepared to go on line by January 1, 1978. At that time, the Energy Bank will begin supplying the heating and cooling needs of the Nursing Home while using an estimated 40% of the energy consumed by a conventional system. The annual COP is expected to be in the range of 4.5 to 5.0. In spite of this impressive performance, the estimated payback period for the Energy Bank is in the vicinity of 11 years (allowing for cost of money and escalating fuel costs.) The reason for this somewhat lengthy payback is the high first cost of the equipment selected for the system. Our primary concern was that the system perform as intended, thereby providing useful data for "fine tuning" future designs. Time tested components were, therefore, selected over others less costly but of uncertain reliability. Ultimately, great savings should result from advances in the design of the ice tank which was the single most costly item of the project. Also, the collector/radiator, while less expensive than most other solar collectors, can be expected to drop in cost as the scale of production increases. With the unusually high energy savings realized by ACES, a technological development of any consequence at all should put the payback in the commercially acceptable six to eight year range.

The Demonstration house at the University of Tennessee has been in operation for nearly a year. The valuable information, obtained in this short time, has given the scientists and consultants of Oak Ridge a course to follow in their system adjustment. Pumps with high efficiencies are being investigated. The same is true with the fan-coil unit which distributes conditioned air throughout the house. Various methods of storing and using the ice are also being explored. The automatic controls which were initially a major expense are being replaced with inexpensive, readily available valves; and a solid-state replacement for the processor is well on its way to becoming a reality.

This past winter brought us another energy crisis in the form of a gas shortage. Temperatures in homes were drastically reduced and laborers were kept from their jobs for want of natural gas. In these times of energy shortage, the day is almost upon us when comfort cooling will be viewed as a luxury we cannot afford. The Annual Cycle Energy System, which heats efficiently while simultaneously providing the means for cooling, offers a solution to this inevitable dilemma. □□

HEAT PUMPS WORLDWIDE

AUSTRALIA

For the more populous three Eastern States of Australia, general consensus is that application of heat pumps is at present confined to room air conditioning units and to small packaged units.

After some large heat pumps systems were tried between 1950 and 1965, they are now avoided in large or complicated systems because of their history of unreliability, and in spite of their undoubted potential for energy saving.

To be more specific on the probable causes of unreliability the following can be mentioned:

- Excessive pressure loss through refrigerant changeover valves.
- Insufficient pressure loss to actuate self-acting refrigerant changeover valves.
- Distortion and consequent jamming of refrigerant changeover valves through the relative expansion of the multiple pipe connections.
- Refrigerant leakage through stems of auxiliary powered changeover valves.
- Refrigerant leakage through non-return valves
- Compressor overheating due to excessive suction super-heat from leaking changeover valves.
- Return of liquid slugs to compressor after mode changeover.
- Refrigerant liquid holdup due to improper drainage of condensed refrigerant in one mode of operation.
- Oil accumulation in the evaporator in one mode of operation.
- The unbalance between plant capacity and requirements in both winter and summer makes the plant more vulnerable to an unstable or unreliable control system.

GERMAN FEDERAL REPUBLIC

Principle and theory of the heat pump are well-known to refrigeration and air conditioning experts in the Federal Republic of Germany. Refrigeration and air conditioning engineers are familiar with all essential problems of computation, projection and installation of heat pump equipment. Questions relating to policies of power distribution—as far as heat pumps and their effects on the environment are concerned—are likewise well-known. Today, a number of large public news media and many politicians regard the heat pump as an important heat source with a great future. It must be said, however, that the importance of the heat pump and its poeential applications are now frequently overestimated.

Today, single pump units—combined into systems by joint water connections—are widely used. A large number of buildings have been equipped with such installations and it is estimated that approximately 5,000 heat pumps are in regular use.

A main obstacle for a wider use of the heat pump is the rate of electricity, and the general power supply situation. It is obvious that the power supply companies are, not interested in a promotion of the heat pump because the capacity of their power lines is limited. As a result, extremely high kW/h rates and exceedingly high connecting fees per kW/h are demanded. In many cases, secondary calculations have shown that the use of heat pumps is far from economical. The situation is likely to remain unchanged unless the high connecting fees are dropped and the rates of electric current, now much higher than the energy cost for gas and oil installations, are considerably reduced.

GREAT BRITAIN

The history of heat pumps in the U.K. has not been very successful due to the availability of cheap energy and the inability of engineers to obtain the instantaneous relationship of the heating and cooling demands of air conditioning loads. The heat pump has been successfully used in a number of chemical installations in the evaporating and condensing of liquids as well as for fluid concentration plants.

With the advent of higher energy costs and the capability of analyzing the energy systems of a building in depth with computer modelling techniques, energy costs are now beginning to be predicted in the design stage of a project. These techniques have led to analyzing various energy systems and, where appropriate, the comparison of a convectional air conditioning and boiler plant to part heat recovery and heat pump systems can be easily made. In many cases, this analysis shows the owning and operating cost of heat pump systems to be below that of conventional oil fired and vapor compression plant.

It is felt that the future building energy systems will rely more on electrical power with the greater use of nuclear stations. Since the electrically powered heat pump is a means to obtain heat energy at comparative costs to heavy fuel oil installations more installations must therefore be built. A further advantage of the heat pump is that localized pollution aspects of oil fired heating plants can be removed from city centers.

JAPAN

Heat pump applications in Japan go back to 1924 when a technical paper relating to their use first appeared. The first experimental operation of a heat pump was conducted in 1930. In 1937 a large building was equipped with a heat pump air-conditioning system using turbo refrigerators and drawing well water. Because of its performance and existing economic conditions after the War, use of this system increased. However, rapid progress of the Japanese economy in the 1950's caused the exhaustion of well water. Soon a law restricting use of well water as a heat source was enacted in 1962.

Heat pump air-conditioning systems using air as a source were already in use in 1957. The growth of this system however remained slow, and its application rather limited. Further progress brought air pollution to urban areas, forcing enactment of the 1968 Air Pollution Control Law, restricting heavy oil combustion and calling for mandatory use of a system that neigher polluted, nor wasted energy. Thus, the air source heat pump system and the waste heat recovery heat pump systems have rapidly increased in number in the past few years.

Packaged air conditioners using air source heat pumps were first marketed in 1960. By 1974, 17% of all packaged air conditioners were based on this sytem.

SWEDEN

The use of heat pumps now is rather common in Sweden and this market is growing rapidly. Most, mainly in department stores and supermarkets, are air-to-air systems with summer cooling cycle and winter heating cycle. The largest units have a capacity of about 12,000 cfm and 300,000 Btu.

Also operating are about twenty plants with air-to-water systems. During heating cycle, the outdoor air is the heat source at night and weekends, and exhaust air from the building during working hours.

Heated or summertime cooled water is distributed to the inlet air coils. The largest plant, in an office-building in Stockholm, has two units with screw compressors and a cooling capacity of 2,300,000 Btuh each. The heating capacity is the same with exhaust air as heat source. This plant was completed in July 1974 and the first one of this kind in 1971. Most have run very successfully with minor maintenance requirements.

In March 1975, installation of an air-to-water heat pump was completed for a multi-family house of 84 apartments. The heated water from the condensers of the twelve separate refrigerant systems, each with a 9 hp hermetic compressor, is distributed to radiators in the apartments. There is also in each system a heat exchanger for cooling hot refrigerant gas from the compressor with water for domestic use. Thus it is possible to heat domestic water up to as much as 160F wintertime with the heat pump. The total heating capacity from the heat pump is 630,000 Btuh by 25F outdoor air, heat source, temperature. It is installed on the roof of the building as an outdoor unit.

Interest in heat pumps in one-family houses is great though the number of installations is rather small, perhaps some one hundred, and mostly for test use. Hitherto air-to-air systems are most common. Air-to-water heat pumps with considerably higher COP than normal heat pumps and also with domestic hot water heating are now ready for production in Sweden. □□

COLD WATER POTENTIAL FOR HEAT SOURCE EXPLORED

BY BILL CLARK
OREGON INSTITUTE OF TECHNOLOGY

In the city of Klamath Falls, Oregon, the use of hot geothermal waters to heat homes and industry is not a new thing. Several hundred homes, schools and industries in the region tap geothermal hot waters of 140 to 235 degrees Fahrenheit to heat their facilities. Only persons living in an area called the "Hot Springs" have been fortunate enough to tap the resource thus far, while other residents look on wishfully.

Now, however, comes word of the potential use of relatively COOL waters to accomplish both heating and air conditioning. Current installations could mean good news to the millions of Americans who live in areas where underground water sources yield water of relatively cold or mild temperatures (50 to 80 degrees Fahrenheit). Recently, the Barnhisel-Ganong Realty in Klamath Falls installed two heat pump systems which tap into the Klamath Falls underground reservoir, whose temperatures average approximately 68 degrees Fahrenheit.

The heat pumps extract BTU's from the water and transport that heat throughout the office facility, for both heating and air conditioning. Once BTU's have been removed from the water, reduced to about 43 degrees, it is then dumped into the city drain. Water returned to the main reservoir in a chilled condition could reduce reservoir temperature. However, only about eight gallons per minute are emptied into drains at the occasional times when the pump motor is running.

The principle of a heat pump involves the use of heat as a free natural resource. Heat exists in abundance in air, in water and in the soil. A heat pump does not create heat, but rather transfers it from one medium to another. It may help the reader to consider that heat is not limited to high temperatures such as 150 or even 90 degrees Fahrenheit.

All substances contain heat until they reach absolute zero. Heat can be removed from 35 degree air or water and even minus 35 degree air. Heat pumps may serve a dual purpose transferring heat into our homes in winter and out of our homes in the summer. During the seasons when weather tends to vary much, a heat pump will warm a home or office on cool mornings and then automatically reverse itself to cool indoor air as the temperature rises in the afternoon. This briefly is an explanation of what happens in the Barnhisel-Ganong facility.

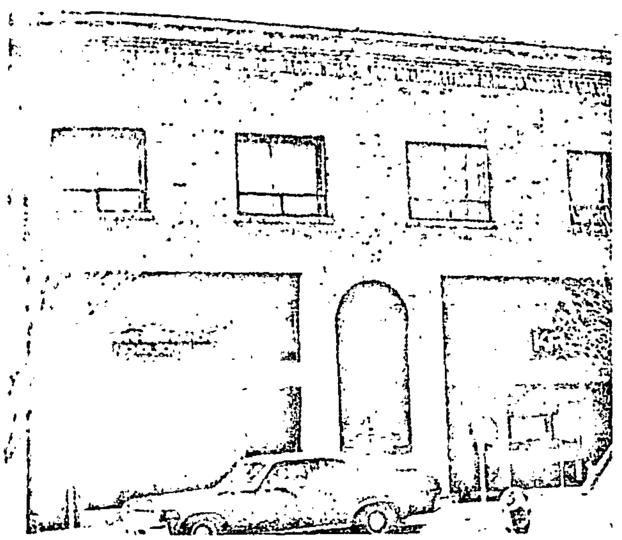
A heat pump works like a refrigerator with a m[otor] driven compressor that pumps a refrigerant (... gas) through two separated coils of metal tubing ... of the coils is inside the box around the ice ... trays, the second coil is outside the box. When ... compressor runs, making refrigerant flow, the ... inside the box becomes cold and begins absor[bing] heat. The heat (brought in by food or seepin[g] through the door and walls) is absorbed by the i[nside] refrigerator coil outside the refrigerator.

The second coil, as it heats, gives off heat to ... surrounding air. Similarly, a heat pump coll[ects] nature's free outside heat and releases it into ... home or building and in the process pumps twic[e as] much or even more heat than the energy it use[s to] operate its own motor. In the Barnhisel-Gan[ong] building, you will recall heat is extracted from ... cool waters fed through a condensing unit and a ... then blows across the heat exchange unit for[cing] heated air out into the room.

Is the cold water unit at Barnhisel-Ganong eff[ec]tive? Is it economical? Yes. Frank Ganong of ... realty says their heat pumps serve a seven thous[and] square foot area with two five-ton units, which w[ere] more than adequate and about half the size of ai[r] air requirements. The 65 degree Fahrenheit w[ater] permits the heat pump to put out 120 degree air ... into a two story building. The realty office elec[tric] bills average from $128 to $190 per month du[ring] coldest periods.

After deducting power consumption for 91 4-... 160 watt fluorescent lights, office equipment, ... water heaters, etc., it is estimated the heat and ... conditioning cost is approximately $65 a mon[th]. (Many local homeowners would gladly exchange t[heir] utility bills for the realty office two story build[ing] fare). Bear in mind that the heat pump provides b[oth] heating and air conditioning. Initial cost of the ins[tal]lation was approximately $7000, not including w[ell] drilling.

Installation of the heat pump unit at Barnhis[el-] Ganong was done by Ray Green, manager of Ba... Heating and Air Conditioning in Klamath Falls. Gre[en] says even normal drinking water temperatures of 45 ... 50 degree Fahrenheit range can be used with the h[eat] pump principle, but efficiency is quite low. ... recommends temperatures ranging from 60 to ... degrees Fahrenheit as ideal. Temperatures above ...



degree Fahrenheit are not too efficient where air conditioning is also desired.

Green says commercial systems incorporate a water-cooled air conditioner and add a reversing valve to make a heat pump. Air conditioning manufacturers don't build many water supplied heat pumps because there is not sufficient demand, according to Green. He says the public needs to be educated to the potential use of this type of heat pump. The commercially produced heat pump does not have much efficiency, according to Green, to produce the most benefit from the waters availablė. It is for this reason that Green builds his own units which he says are approximately one-third more efficient.

The reason for lack of efficiency on commercial units, Green says, is because the condenser in use has been designed for an air conditioner. Green currently converts existing chillers as condensors. He also is considering a tie-in with some commercial manufacturer to mass produce units of the type desired. Additionally, a "closed system" for commercial use is still in the design stages. (A "closed system" permits recirculating the water, thereby using and wasting less water.)

In addition to the Barnhisel-Ganong use of low temperature waters, a second well has been drilled for motel unit two blocks away. The motel plans to use 2 water-cooled heat pumps for individual heater/air conditioner units costing in the vicinity of $650 each. (Well drilling, ducting and installation costs not included). At least two other homeowners in the Klamath Falls region also use warm waters of 53 to 80 degree Fahrenheit temperature range for heating and air conditioning, with others in the negotiating stages.

An area of new tract homes in the Moyina Heights area have access to warm waters. If a "closed system" could be designed to get rid of excess water, Moyina residents might also use warm waters for heating and air conditioning. It is acknowledged that initial installation of this type of unit will cost about 15 to 25 per cent more than the traditional air to air unit, however, the cost of the operation is reduced on a monthly basis by at least one half, according to Green.

What does the heat pump principle mean to the average person residing in Klamath Falls? First of all, it means many who have felt they were not lucky enough to reside in the "Hot Springs" area may take heart. For millions of other Americans, a rosy future may also be in store for home heating. Each month, new geothermal discoveries throughout the United States reveal water sources of sufficient temperatures to make use of the heat pump principle. Vast areas of the West, including California, Oregon, Idaho, Nevada, Arizona, New Mexico and Wyoming lie above relatively warm geothermal water reservoirs. Some geothermal experts predict that for every geothermal well, which produces steam for power generation, there will be five times as many warm water reservoirs discovered.

Who knows how many of these warm or "cold" geothermal water sources await the well-driller's bit to tap their energy for heating AND air conditioning?

BILL GREEN

UNITED STATES
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
WASHINGTON, D.C. 20545

September 20, 1977

Dr. Phillip M. Wright
University of Utah Research Institute
Earth Science Laboratory
Research Park
580 Arapean Drive
Salt Lake City, Utah 84108

Dear Phil:

Enclosed for your information is a copy of the Regulations pursuant to Section 432(d) of the Energy Policy and Conservation Act (P.L. 94-385). During your visit last month, I mentioned that heat pumps were considered "renewable resource measures" and that ground-water heat pumps to replace non-renewable resources in existing dwellings were applicable for government Loan Guaranties under the Act. The list of renewable resource measures that qualify under the provisions of the Act are noted in the enclosed regulations.

You might also be interested in several recent articles on ground water energy that were published in the April 1977 issue of the Journal of the National Well Association. For your information copies of the articles are also enclosed.

Sorry I missed you on a recent trip to Salt Lake City, but I expect to return there September 28-30; if you are available I would like to visit and learn of your programs on geothermal resource assessment in Utah.

Regards,

Burton B. Barnes
Hydrothermal Mission Team Leader
Division of Geothermal Energy

2 Enclosures:
1. Reprint from Federal
   Register dtd Jul 25, 1977.
2. Ground Water Energy Article.

Reprint from the Federal Register

of Monday, July 25, 1977

Title 10—Energy

CHAPTER II—FEDERAL ENERGY ADMINISTRATION

PART 450—ENERGY MEASURES AND ENERGY AUDITS

Energy Measures List

AGENCY: Federal Energy Administration.

ACTION: Final rule.

SUMMARY: The Federal Energy Administration, by final rule, establishes a list of energy conservation measures and renewable-resource energy measures it has developed after consultation with the Secretary of Housing and Urban Development. This document provides a subpart to, and thereby combines the list of energy measures with, the energy audit rulemaking. Depletion of the Nation's domestic resources of fossil fuels has focused attention on the need to identify energy conservation and renewable-resource energy measures which can be implemented in existing residential or commercial buildings and industrial plants.

DATE: Effective date: July 15, 1977.

FOR FURTHER INFORMATION CONTACT:

Allen Jaisle, Office of Program Development, Office of Conservation, Federal Energy Administration, Washington, D.C. 20461 (202-566-7856).

John Pulice, Office of Synfuels, Solar and Geothermal Energy, Office of Energy Resource Development, Federal Energy Administration, Washington, D.C. 20461 (202-566-6192).

SUPPLEMENTARY INFORMATION: 1. Introduction. 2. Section 450.3 Definitions. 3. Section 450.31 Energy Conservation Measures, and Section 450.32 Requirements and Limitations For Energy Conservation Measures. 4. Section 450.33 Renewable-Resource Energy Measures, and Section 450.34 Requirements for Renewable-Resource Energy Measures.

1. Introduction

On June 10, 1977, the Federal Energy Administration (FEA) published a notice of proposed rulemaking, 42 FR 29906 et seq., to amend Part 450, Chapter II of Title 10, Code of Federal Regulations (CFR), to establish a list of energy conservation and renewable-resource energy measures (energy measures) developed by FEA, after consultation with the Department of Housing and Urban Development (HUD), under Section 432 (d), 42 U.S.C. 6325 (e)(1), of the Energy Conservation and Production Act (Act); Pub. L. 94-385, 90 Stat. 1125 et seq. Section 432 amends Part C of Title III of the

Energy Policy and Conservation Act (EPCA), 42 U.S.C. 6321-6326.

FEA published proposed procedures for energy audits on April 15, 1977, at 42 FR 20012, and published a final rule for energy audits on June 29, 1977, at 42 FR 33158. This rulemaking for energy measures amends the energy audit rulemaking by adding a new Subpart D to combine energy audits and energy measures in one consolidated part and by providing additional necessary definitions.

FEA solicited comments for its consideration from interested persons on the proposed energy measures list if submitted by June 30, 1977, and FEA held a public hearing in Washington, D.C. on June 29, 1977.

In response to the notice of proposed rulemaking, six persons presented oral testimony and 19 written comments were received by FEA. This final rule contains revisions to the proposed rule which reflect FEA's consideration of the comments as well as other information available to FEA.

Some comments mistook an energy measure for a regulatory requirement applicable to construction, alteration or retrofit of a building or industrial plant. No one is required to use or install any energy measures in a residential or commercial building or an industrial plant. Anyone may use an energy measure, with or without regard to the requirements prescribed in this rulemaking.

The energy measures list does not establish performance or prescriptive standards but instead, as explained in the proposed rulemaking, has only three identifiable uses.

First, the energy measures list will be used by States participating in the program for supplemental State energy conservation plans. To be eligible for financial assistance, section 432 of the Act (42 U.S.C. 6326), provides that a supplemental plan shall include procedures for carrying out a continuing public education effort to increase significantly public awareness of energy and cost savings which are likely to result from implementation of energy measures. Moreover, each supplemental plan must also contain procedures to encourage energy audits. An energy audit is a process which identifies energy and cost savings likely to be realized through the purchase and installation of energy measures such as those identified by the proposed list. Final guidelines for supplemental plans were published by FEA on May 24, 1977, at 42 FR 26413, and Congress has appropriated $12 million dollars for this program.

Secondly, an energy measure on the list is eligible, on a national or regional basis, for financial assistance pursuant to section 509 of the Housing and Urban Development Act of 1970 or thirdly, for obligation guarantees under section 451 of the Act. No funds have been appropriated to provide financial assistance for implementation of either program. Therefore, for all practical purposes, this list is to be used at the present time only in connection with the program for supplemental State energy conservation plans. Whether Congress may at some future time appropriate funds for the other two programs or enact a new Federal program to provide financial assistance for or otherwise provide for the use of an energy measure included in this list is purely speculative and beyond the scope of this rulemaking.

Some comments questioned whether and, if so, how FEA coordinated the rulemaking process for the energy measures list and energy audits. Both rulemakings have been consolidated in Part 450 because they are interdependent. Both the list of energy measures and the energy audit procedures identify a modification as an energy measure so that both must employ identical definitions, administrative findings and interpretations.

The energy measures list and energy audits use the qualifying conditions for an energy measure prescribed in § 450.4 as the basic criteria for assessing a modification and are predicated on the fuel price projections contained in Appendix A to Part 450. FEA has made certain that the consolidation of the list of energy measures with energy audit procedures is internally consistent.

Numerous comments concerned energy price projections contained in Appendix A of Part 450 which FEA employed to evaluate cost recovery for a modification. These prices were criticized as being too low, being average "rolled-in" prices, and being representative of past policies. Similar questions were raised regarding the proposed energy audit rulemaking which published these prices, and these comments were considered by FEA prior to issuance of the final rule for energy audits which only slightly changed the proposed price projections. Reconsideration of these prices by FEA would be untimely and unnecessary since the final energy audit rule was only published on June 29, 1977.

At that time, FEA stated:

> FEA intends to revise its projections when appropriate and may hold public hearings where significant changes are made to these projections. FEA will endeavor to provide up-to-date projections but not with such frequency as to cause undue burdens, uncertainty, and confusion for energy auditors. 42 FR 33159.

One comment urged FEA to be less rigorous in its application of cost recovery criteria in assessing energy measures because of the degree of imprecision and because so many significant factors are excluded from consideration. The comment also pointed out that the approach of the energy measures list is much more detailed than the lists of energy saving investments in the National Energy Act as proposed by the Administration.

FEA has fully considered the restrictions on precision necessary to develop a broadly inclusive energy measures list. For this reason, FEA has not included modifications unless FEA obtained information and data which FEA determined provided a reasonable level of precision and reliability. The explicitness of the cost recovery provisions of the Act leaves little room for establishing a less restrictive evaluation which could provide a more inclusive list with fewer requirements and limitations. FEA is aware of the different approach proposed in the National Energy Act. However, this rulemaking is confined to the authorities established by the Act and cannot at this time reflect or otherwise anticipate future changes in legislation which may be enacted by Congress.

One comment suggested a 30-day extension of the comment period. FEA does not believe this was necessary because no similar request was received from any other person, and the comment presented a full and carefully considered discussion of the commenter's concern. Accordingly, FEA has decided to adhere to the original comment period because an adequate discussion of the issues has been made possible by this comment period.

Some comments urged that FEA undertake an active program to educate consumers and businessmen about energy measures, to encourage their use and to identify effective conditions for installation of an energy measure. While these comments do not directly address this rulemaking, FEA appreciates these comments on educational and informational efforts to complement rulemaking in this area. FEA is undertaking such an effort in conjunction with the States under the program for supplemental State energy conservation plans.

2. Section 450.3 Definitions

One comment suggested that the energy measures list be broadened to include modifications other than "hardware". Presumably this means design or operational adjustments should be considered modifications eligible for inclusion in the energy measures list. Because of the definitions contained in the Act for "energy conservation measure" and "renewable resource-energy measure," an energy measure may only include a modification to a building or industrial plant (the latter term including an industrial process) which has been purchased and installed. For this reason, FEA has interpreted the Act to require that a "modification" cannot include a design or operational change which cannot be purchased and installed.

One comment drew attention to the proper definition for the abbreviation of ASTM as the American Society for Testing and Materials. The definition has been so changed.

A few comments indicated that "heating degree days" and "cooling degree days" should use a different base temperature than 65° F. The data is provided by the National Oceanic and Atmospheric Administration, Department of Commerce. FEA has retained this definition because it represents the best currently available standardized collection of climatic information which can be readily adapted to the needs of FEA's conservation program.

Some comments indicated that the "R-Value" definition should include specification of the mean temperature at which the R-Value is measured because thermal conductivity varies with temperature. The definition in § 450.3 is modified so that the R-Value is to be measured at the mean temperature of the insulation under design conditions. FEA, however, decided it would be unnecessarily complex to specify the mean temperatures for different types of uses.

FEA has also provided definitions for "ANSI Standard" and "IEEE Standard" to clarify the source of referenced commercial standards.

3. SECTION 450.31 ENERGY CONSERVATION MEASURES AND SECTION 450.32 REQUIREMENTS AND LIMITATIONS FOR ENERGY CONSERVATION MEASURES

Proposed § 450.31 identified and described energy conservation measures included in the energy measures list. Proposed § 450.32 stated the requirements and limitations for each energy conservation measure identified in proposed § 450.31. Comments on both these sections are combined for clarity and presented below. Comments proposing the addition of an energy conservation measure to the proposed list are addressed either in the discussion of closely related energy conservation measures or separately.

A. CEILING INSULATION

Some comments suggested that ceiling insulation not be limited to installation between the heated top level living area and the unheated attic space but also include installation of insulation on the surface of the ceiling. FEA agrees and has changed the definition accordingly.

One comment suggested the R-Value level be broadened to include up to R-38 for residential ceiling insulation in all climate zones when the residence is both heated and air conditioned. FEA does not believe the evidence of energy saving in the cooling season is sufficiently clear to justify this increase in the thickness of insulation.

B. WALL INSULATION

The use of insulation on the building interior surface of a wall with an exterior exposure was recommended by some comments. Upon review, FEA agrees and has changed the provision in § 450.31(b) to include this installation of interior insulation. FEA has also revised the requirement so that only a wall with an exterior exposure is to receive wall insulation. FEA has found that insulation of partition walls is generally not worth the cost and the regulation is not formulated to take into account exceptional conditions.

Some comments suggested that § 450.32(b) restricting installation of wall insulation to a 4 inch wall cavity was too restrictive. FEA agrees and has deleted reference to a maximum thickness for the wall cavity. For the climate zones specified, it is unlikely that variations in wall cavity size will result in applications with insufficient cost recovery. Further it seems unlikely that too much interior insulation would be applied since there is a natural inclination to maximize interior space.

One comment suggested insulated aluminum siding as an energy conservation measure. If this modification were eligible, it could be included within the definition of wall insulation. The insulating value of such siding is typically quite low in relation to its cost. FEA does not believe that a reasonable judgment can be made that insulated aluminum siding will have sufficient cost recovery from energy savings so that this modification meets neither the primary purpose nor the cost recovery criteria.

C. FLOOR INSULATION

No comments were received and no changes have been made.

D. PIPE INSULATION

Some comments pointed out that a level of R-19 may be too broad where there are space limitations or where pipe sizes are very large. FEA, however, only specified R-9 in proposed § 450.32(d) as a maximum which may not be exceeded. A lower R-Value could be installed. Upon further consideration however, FEA concludes that pipe insulation has such a high cost recovery in nearly all applications that no R-Value limitations are considered necessary for this energy measure. Moreover, FEA is reluctant to be overly restrictive concerning pipe insulation because cost recovery is expected in any reasonable application. Any excessive use of pipe insulation application would most likely occur in commercial and industrial installations where purchasers have both the knowledge and incentive to avoid such a misallocation of resources. Accordingly, § 450.32(d) does not contain an R-Value restriction.

Some comments suggested that FEA Conservation Paper No. 46, entitled "Economic Thickness of Industrial Insulation," should be used to establish mandatory requirements for pipe and other insulation applications. FEA has not followed this suggestion because the degree of control is not warranted for the reasons discussed but FEA recommends its use as a source of information on sound business practice.

E. CAULKING AND WEATHERSTRIPPING

No comments were received and no changes have been made.

F. ROOF INSULATION

Some comments suggested proposed § 450.31 include insulation installed on the surface of a roof facing a building interior. FEA agrees and has changed the definition to include this installation.

G. CLOCK THERMOSTAT

One comment suggested that an additional energy conservation measure be included for a very advanced form of time and temperature controller. FEA believes that the definition in proposed § 450.31(g) is sufficiently broad to include this device.

H. HOT WATER HEATER INSULATION

Some comments indicated that R-19 may be too large a thickness for hot water heater insulation because of space limitations. Proposed § 450.32(h) set R-19 as a maximum which may not be exceeded. Where space limitations preclude use of the maximum amount of insulation, FEA finds it unnecessary and redundant to provide that a lower level of insulation may be used. FEA has retained the proposed maximum requirement.

I. DUCT INSULATION

Comments on space limitations similar to those noted above have been made concerning duct insulation. FEA has retained the proposed maximum R-Value for reasons similar to those discussed above.

J. STORM WINDOWS

After review of its cost recovery calculations, FEA has discovered an error resulting from the exclusion of the savings multiplier. The corrected calculation justifies extension of his energy conservation measure to heating zone 2, and § 450.32(j) has been so changed.

One comment suggested inclusion of thermalized and thermal break aluminum windows. Another comment suggested various additional types of replacement windows. One comment suggested that a modification of double glazed to triple glazed windows be considered an energy conservation measure for areas which have not less than 8000 heating degree days.

The suggested modifications are essentially intended to achieve energy savings by preventing infiltration and temperature transmission through penetration similar to those savings achieved by storm windows combined with caulking and weatherstripping. In FEA's judgment, the suggested modifications have insufficient cost recovery for inclusion in the list of energy conservation measures. Essentially the modifications provide similar energy savings to storm windows combined with weatherstripping but are substantially more expensive because of the increased cost of purchase and installation of replacement windows. Accordingly, FEA has not changed the storm window definition or otherwise included the suggested modifications on the list.

K. LIGHTING

One comment suggested that proposed § 450.31(k) be expanded to include a change from mercury vapor to high pressure sodium lighting systems. The comment further suggested that proposed § 450.32(k) delete the word "incandescent" so that a doubling of useful light output is required regardless of the lighting system used. FEA believes the suggested changes will usefully broaden this energy conservation measure while achieving cost recovery; therefore, the suggested changes have been made

One comment suggested specifications regulating the amount of illumination allowable for different uses. FEA has not adopted this recommendation because it is extrinsic to the energy and cost savings likely to be derived from the purchase and installation of an energy measure.

Regulation of how much energy may be consumed for a particular use is beyond the scope of this rulemaking and the provisions of the Act.

L. MIXING VALVE

No comments were received and no changes have been made.

M. FLOW RESTRICTOR

No comments were received and no changes have been made.

N. RESIDENTIAL OIL BURNER

One comment objected to the requirement in proposed § 450.32(n) that residential oil burners be eligible only in heating systems which cannot be adjusted to achieve a minimum efficiency level specified for each heating zone. FEA has retained this requirement because residential users should get the immediate energy conservation benefits and cost savings of furnace adjustment, the cost of which is low relative to installation of a replacement oil burner. Moreover, it is likely that cost recovery for a residential oil burner will not be attained where indicated efficiency levels can be achieved by furnace adjustment. As for its impact on businessmen, FEA finds retention of this requirement will not be burdensome or overly complex for oil dealers and others who install this equipment because these businessmen are presumed knowledgeable concerning furnace adjustment and their operations are generally restricted to one heating zone.

FEA's judgment that 82 percent efficiency is a reasonable expectation from new burners was confirmed by comments. However, FEA recognizes that this performance is predicated on proper matching of the new burner with other components of the heating system.

O. INDIVIDUAL METERS

One comment suggested that this energy conservation measure be limited to changes from master to individual metering without changing energy sources. FEA believes this requirement was self evident in the proposed language of § 450.32(o) and has therefore not made any change.

P. BURNERS AND CONTROLS FOR COMMERCIAL BUILDINGS AND INDUSTRIAL PLANTS

One comment noted that the requirements for this energy conservation measure are very stringent. The proposed requirements are retained in the final rule because cost recovery is only likely where there is a large gain in efficiency. Other comments concurred with the requirements of this energy conservation measure. Again, FEA recognizes that expected performance is predicated on properly matched heating system components.

Q. LIGHTING CONTROLS

No comments were received, and no changes have been made.

R. HVAC CONTROL SYSTEM

One comment suggested that automated computer, microprocessor and logic controller devices be included in this energy conservation measure. FEA agrees and has added § 450.32(r)(6) to include these modifications.

S. HIGH EFFICIENCY ELECTRIC MOTORS

One comment noted that the specifications in the proposed rule were based on totally enclosed fan cooled motors but that other categories of motors can achieve similar efficiencies at even lower costs. Upon further consideration, FEA concludes that the energy measure should be both broadened and simplified. Therefore, FEA has changed the requirement so that § 450.32(s) requires only that the efficiency improvement be as specified and that the service duty be substantially continuous, which means 5,000 hours minimum annual use.

Broadening of this energy conservation measure to include motor controls as well as motors to achieve high efficiency electric motor operation was suggested by another comment. Motor controls capable of achieving the stated efficiency increase are judged by FEA to be likely to cost less than a high efficiency motor itself. Accordingly, § 450.32(s) includes such controls.

One comment noted a discrepancy between the technical support document and the proposed rulemaking. The technical support document stated that the attributed life was 10 years, and proposed § 450.32(s) was calculated upon the basis of an attributed life of 7.5 years. FEA finds the attributed life to be 10 years as stated in the technical support document, and the revised subparagraph is consistent with this conclusion.

T. VENTILATION

Some comments suggested that ventilation devices other than the whole house ventilation fan be included in the list of energy conservation measures. Test results submitted have been reviewed, and FEA remains unconvinced that other ventilation devices can be included at this time. After joint industry-government tests which have been planned are conducted, FEA may obtain more reliable information which could serve as a basis to reconsider inclusion of other ventilation devices.

U. ADDITIONS TO THE LIST

Several comments suggested the addition of modifications to the list which were not previously proposed by FEA. FEA is reluctant to add such modifications to the list without first providing an opportunity for public review and comment. FEA believes that some of the modifications recommended appear unlikely to meet the criteria for an energy measure. An example is a respiratory personnel protection system, which appears to be primarily for occupational health and safety purposes.

FEA will modify the list when a modification can be properly assessed and when a revision of the energy measures list is appropriate.

4. SECTION 450.33 RENEWABLE-RESOURCE ENERGY MEASURES AND SECTION 450.34 REQUIREMENTS FOR RENEWABLE-RESOURCE ENERGY MEASURES

Proposed § 450.34 has been revised because many comments apparently misunderstood the requirement for a verification audit. FEA has determined that, with two exceptions, the likelihood of cost recovery cannot be ascertained, at this time, without a verification audit. Therefore § 450.34(a) requires use of an energy audit except for the two applications set forth in § 450.34 (b) and (c) respectively, each of which requires the use of a simple formula for establishing cost recovery. However, a person may choose to use a verification audit in lieu of the formulas. A person may select this option, for instance, to obtain more precise information about energy and cost savings or where information required for the formula cannot be obtained.

Three comments expressed concern that FEA was favoring electricity over more efficient fuel systems by mentioning only measures which use or generate electricity. FEA has selected energy measures which it believes most effectively meet the requirements of the Act. Some of these may require the use of electricity; others, such as a wood-fired boiler or an urban waste pyrolysis system, may not. The purpose of designating renewable-resource energy measures is to encourage the use of nondepletable resources.

The Act requires an increase in efficiency for an energy conservation measure but not for a renewable-resource energy measure. The reason for the distinction is that increased efficiency need not be demonstrated for a renewable-resource energy measure so long as the use of a renewable-resource results in the decreased use of a depletable energy source. Accordingly, although the list is not intended to favor the use of electricity, the renewable-resource energy measures are intended to diminish the use of oil and gas which are depletable sources of energy by substitution of nondepletable sources. Furthermore, FEA has developed the list of renewable-resource energy measures so that there will be a net decrease in the consumption of a depletable energy source resulting from a change of energy systems required by the installation of a modification.

A. SOLAR WATER HEATER

No comments were received and no changes have been made.

B. AIR SOURCE HEAT PUMP

Four comments recommended the exclusion of heat pumps because such equipment does not utilize a renewable-resource. FEA has determined that an air source heat pump extracts a portion of the heat contained in the ambient air which has been heated by the sun outside of a building and transfers it into the building, thereby using solar energy as a resource to heat the building. The electric energy necessary to run the heat pump is not assumed to be derived from a renewable resource, so only a part of the fuel requirements of a building are changed to a renewable energy source by installing a heat pump. However, the

Act specifically defines a renewable-resource energy measure to include a change "in whole or part" from a depletable to a non-depletable source of energy, and FEA has determined that a heat pump satisfies this requirement.

Three comments recommended that an air-source heat pump be removed from the list because its use tends to encourage purchase by the consumer of a reverse-cycle heat pump which also provides air conditioning. They argued that the resulting use of air conditioning could cause an increase in the consumption of a depletable energy source in excess of the system it replaced. FEA recognized this in the proposed rulemaking and, for this reason, requires an energy audit for the installation of a reverse-cycle heat pump except when the installation replaces both electric resistence heating and air conditioning. Where electric air conditioning already is in operation, FEA concluded that it is highly likely there would be a net saving in energy costs attributable to the heating cycle performance of the heat pump.

Six comments recommended that the heat pump be excluded from the list because a heat pump uses electricity and stated that electricity is a very inefficient energy system by comparison with heating systems which use oil or gas.

FEA does not deny that oil and gas home heating systems may, in many cases, be more efficient in their use of depletable resources than a heat pump system. It is for precisely this reason that FEA structured the list so that the procedure specified in § 450.34(c) only be used when a heat pump is to be installed in place of an electric resistance heating system. Several comments indicated that a heat pump will significantly reduce fuel consumption when installed to replace an electric resistance heating system, and FEA has also reached this conclusion. The installation of a heat pump requires an energy audit to ensure that there is a change to a nondepletable source of energy pursuant to § 450.34(a) except where—(1) a heat pump with only heating capability replaces electric resistance space heating; or (2) a reverse cycle heat pump replaces electric resistance space heating and air-conditioning.

One comment indicated that insufficient information is available to a consumer considering the purchase of a heat pump to enable him or her to carry out the calculation provided in proposed § 450.34(a). FEA, however, believes that in most instances a consumer can obtain the needed information. Where this is impracticable, a verification audit is required.

One comment reflected a concern that prices for electricity used to develop the values in the Table of Climatic Factors for heat pumps in proposed § 450.34(c) do not accurately reflect electric heating costs the homeowner will actually pay. The homeowner uses his or her actual previous year's heating cost to compensate for variations between actual heating costs and regional electric prices. The purpose of the Table is to determine the estimated cost savings attributable to the installation of a heat pump taking into consideration regional use projections. The result, however, is adjusted for the homeowner by the use of his or her actual heating costs.

Three comments questioned FEA's decision not to use a heat pump coefficient of performance (COP) in its formulation of the Table of Climatic Factors. FEA used a heat pump seasonal performance factor (SPF). The use of SPF includes the heat pump COP and additional factors. SPF takes into account heat pump capacity and degree hours at various temperature ranges and includes consideration of the supplemental resistance heat required at low temperatures. SPF, but not COP, allows for a evaluation of the performance of heat pump in changing temperatures and FEA considers it to be a better overall indicator of annual performance.

FEA used data provided by manufacturers in its analysis of a heat pump. One comment mentioned a heat pump study conducted by the National Bureau of Standards which reported discrepancies between the data found by testing and data published by one manufacturer. FEA does not conclude that this study provides a reasonable basis for an inference that data supplied by a manufacturer is likely to be inaccurate.

FEA believes that its calculations are based upon the performance of a representative heat pump and that it has used accurate data. However, additional research is being conducted in both the government and the private sector, and FEA welcomes the submission of new information. FEA will review its Table of Climatic Factors in the light of new information and data as may be appropriate.

Two comments recommended the exclusion of a heat pump because it is an "appliance". FEA finds that an air-source heat pump is an integral part of the central heating system of a structure, and thus is not an appliance. For purposes of Part 450, the definition of "appliance" in § 450.3 makes this clear. A modification may constitute a "consumer product" under Part B of Title V of the Energy Policy and Conservation Act, 42 U.S.C. 6201. However, this does not necessarily indicate that such a consumer product is an "appliance" as that term is used by FEA for an energy measure under the Act. Part B does not use or define the term "appliance".

One comment addressed the reliability and repair cost of heat pumps, suggesting that heat pump failure rates and associated repair costs were excessive, and requested that heat pumps be eliminated from the list for that reason. FEA has determined that maintenance and repair costs which can reasonably be anticipated are likely to be more than repaid by the energy savings resulting from replacing an electric resistance heating system with a heat pump. Two comments expressed concern over FEA's determination of a 15 year heat pump life. FEA held discussions with the National Bureau of Standards and several major heat pump manufacturers to collect information to determine a valid system life for residential heat pumps. On the basis of these discussions, FEA has chosen a system life of 15 years as the most valid assumption for and has thus retained the 15 year life figure in §450.34(c).

One comment expressed concern that FEA does not provide a method for calculating annual electric space heating costs in proposed § 450.34(c)(3)(ii). FEA does not believe that one method should be prescribed by regulation. Several methods can be used, each of which provides satisfactory results. A method acceptable to FEA for determining the annual space heating costs is to choose a month when neither heating nor air-conditioning was used. The electricity cost for this month is the appliance base load. Subtracting this amount from each month's electric bill when heating was required will give the heating cost for each month. The previous year's total heating cost is the sum of the monthly heating costs.

C. WATER SOURCE HEAT PUMP

One comment reflected concern that FEA did not establish requirements for the installation of a water-source heat pump. This results from a misunderstanding. A verification audit is required for this energy measure in accordance with § 450.34(a), regardless of the system it is replacing. The energy audit will indicate whether the installation of a water-source heat pump will result in a decrease or an increase in the use of depletable energy resources.

D. ADDITIONAL COMMENTS

One comment reflected concern over the inclusion of a water-powered generator proposed in § 450.33(q). The renewable-resource used by a water-powered generator is the potential and kinetic energy in moving water, and FEA believes that the energy audit requirement will assure that its use will not result in an increased use of depletable resources.

Another comment reflected a concern that the use of a skylight may not necessarily result in a reduction in energy consumption. FEA finds that the requirement that this measure undergo an energy audit ensures that any such installation will meet the requirements of the law, and a skylight is listed as an energy measure in § 450.33(g).

One comment recommended a wording change with regard to proposed §§ 450.33 (k) and (n) to allow the inclusion of the word "system" to describe these measures. The final rule has been changed by referring to a "system" with regard to an energy measure to make clear that all components necessary for the proper operation of an energy measure are included as part of the measure, and proposed § 450.33 (b), (c), (f), (i), (j), (k), (n), (o), and (q) have been revised accordingly.

One comment recommended the inclusion of "hydraulic ram pumps" on the list. An hydraulic ram pump is a device which uses the energy in moving water to provide pressurized running water for a building. FEA finds that it has insufficient information to determine whether such a modification is an energy measure. However, FEA welcomes the submission of additional information concerning this technology.

One comment urged the addition of ocean thermal energy conversion to the list of measures. While ocean thermal gradients constitute a potentially renewable source of energy, FEA has determined that the technology does not currently meet the standards for selection of energy measures.

E. SECTION 450.35 CLIMATE ZONES

A comment suggested that Puerto Rico and the American Virgin Islands be included in cooling zone 1. FEA agrees and has changed the climate zones to include these jurisdictions in cooling zone 1 and heating zone 0.

The proposed regulation was reviewed in accordance with Executive Order 11821 and OMB Circular A-107 issued November 7, 1976, by Executive Order 11949 and has been determined not to be a major proposal requiring an evaluation of its inflationary impact.

(Part B of Title IV of the Energy Conservation and Production Act, Pub. L. 94-385, 90 Stat. 1125 et seq.; also issued under Part C of Title III of the Energy Policy and Conservation Act, Pub. L. 94-163, 89 Stat. 871 et seq., 42 U.S.C. 6321 et seq.; Federal Energy Administration Act of 1974, Pub. L. 93-275, as amended, 15 U.S.C. 761 et seq.; Executive Order 11790, 39 FR 23185).

In consideration of the foregoing Part 450, Subchapter D, Chapter II of Title 10 Code of Federal Regulations is amended as set forth below, effective immediately.

Issued in Washington, D.C., July 15, 1977.

ERIC J. FYGI,
Acting General Counsel,
Federal Energy Administration.

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1. In 10 CFR Part 450, the title of the part is revised to read as set forth above.

2. Section 450.1 is revised to read as follows:

§ 450.1 Purpose and scope.

This part designates energy measures and the types of, and requirements for, energy audits as required by the Federal Energy Administration, pursuant to section 432(d) of the Energy Conservation and Production Act, Pub. L. 94-385, 90 Stat. 1125 et seq., which adds section 365(e) (1) and (2), 42 U.S.C. 6325(e) (1) and (2), to the Energy Policy and Conservation Act, 42 U.S.C. 6201 et seq. This part also contains the projections of future energy prices which shall be used in calculating the changes in energy costs which will result from installation of a particular modification in a building or industrial plant, and includes the criteria for determining whether the installation of a particular modification meets certain requirements of the Act for designation as an energy measure.

3. 10 CFR 450.3 is amended to add, in appropriate alphabetical sequence, definitions of ASTM Standard, ANSI Standard, climatic zone, compressor hours, cooling degree days, cooling zone, Federal Region, heating degree days, heating zone, HVAC, IEEE Standard, and R-Value.

§ 450.3 Definitions.

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"ANSI Standard" means a standard prescribed by the American National Standards Institute.

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"ASTM Standard" means a standard prescribed by the American Society for Testing and Materials.

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"Climatic zone" means a geographical area of the United States designated by FEA.

"Compressor hours" means the average number of hours which an air conditioning compressor must operate to provide the cooling needed for space conditioning for a cooling zone.

"Cooling degree days" means the annual arithmetic sum of the negative differences of the average daily outside temperature, in degrees Fahrenheit, subtracted from 65 degrees Fahrenheit.

"Cooling zone" means a climatic zone based on cooling degree days or compressor hours.

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"Federal Region" means one of the 10 standard regions as described in OMB Circular A-105, Standard Federal Regions.

"Heating degree days" means the annual arithmetic sum of the positive differences of the average daily outside temperature, in degrees Fahrenheit, subtracted from 65 degrees Fahrenheit.

"Heating zone" means a climatic zone based on heating degree days.

"HVAC" means heating, ventilating and air conditioning.

* * * * *

"IEEE Standard" means a standard prescribed by the Institute of Electrical and Electronic Engineers.

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"R-Value" means a measurement of the ability of insulation to resist the flow of heat, expressed in English units at the mean temperature of the insulation under design conditions.

4. Part 450 is amended by establishing Subpart D as follows:

Subpart D—Energy Measures

AUTHORITY: Part B of Title IV of the Energy Conservation and Production Act, Pub. L. 94-385, 90 Stat. 1125 et seq.; also issued under Part C, Title III, of the Energy Policy and Conservation Act, Pub. L. 94-163, 89 Stat. 871 et seq. (42 U.S.C. 6321 et seq.); Federal Energy Administration Act of 1974, as amended, Pub. L. 93-225 (15 U.S.C. 761 et seq.); E.O. 11790, 39 FR 23185.

§ 450.30 Purpose and scope.

This part establishes a list of energy conservation and renewable-resource energy measures developed by FEA after consultation with the Secretary of Housing and Urban Development Depletion of the Nation's domestic resources of fossil fuels has created a need to identify energy measures which can be carried out in residential and commercial buildings and industrial plants.

§ 450.31 Energy conservation measures.

Subject to the requirements and limitations set forth in § 450.32, an energy conservation measure shall be—

(a) Ceiling insulation in a residential or commercial building, which is a material which is installed on the surface of the ceiling facing the building interior or between the heated top level living area and the unheated attic space and which resists heat flow through the ceiling;

(b) Wall insulation in a residential or commercial building or industrial plant, which is a material which is installed on the surface facing the building interior, or in the cavity, of an exterior wall and which functions to resist heat flow through the wall;

(c) Floor insulation in a residential or commercial building, which is a material which resists heat flow through the floor between the first level heated space and the unheated space beneath it, including a basement or crawl space;

(d) Insulation for hot bare pipes in a residential or commercial building or industrial plant, which is a material which resists heat flow from the pipes to the surrounding space;

(e) (1) Caulks and sealants in a residential or commercial building or industrial plant, which are nonrigid materials placed in joints of buildings to prevent the passage of heat, air and moisture;

(2) Weatherstripping in a residential or commercial building or industrial plant, which consists of narrow strips of flexible material placed over or in movable joints of windows and doors to reduce the passage of air and moisture;

(f) Roof insulation in a commercial building or industrial plant, which is insulation placed on the surface of the roof facing the building interior or between a roof deck and its water repellent roof surface;

(g) Clock thermostat in a residential building, which is a temperature control device for interior spaces incorporating more than one temperature control point and a clock for switching from one control point to another;

(h) Exterior insulation for a hot water heater in a residential or commercial building or industrial plant, which is a material placed around the tank which resists the heat flow from the hot water heater to its surrounding space;

(i) Insulation for forced air ducts in a residential or commercial building or industrial plant, which is a material which resists heat flow from the duct to its surrounding space;

(j) Storm window in a residential or commercial building, which is an extra window, normally installed to the exterior, but which may be installed to the interior, of the primary or ordinary window, to increase resistance to heat flow and to decrease air infiltration;

(k) Efficient lighting fixture or lamp in a residential or commercial building or industrial plant, which is one which—

(1) Replaces an incandescent fixture or lamp with a type of lighting system including fluorescent, mercury vapor, metal halide, and high pressure sodium or ellipsoidal reflector lamps; or

(2) Replaces a mercury vapor fixture or lamp with a high pressure sodium lighting system.

(l) Mixing valve for a hot water supply line in a residential or commercial building or industrial plant, which is a type of valve mounted in the hot water supply line, close to the water heater, which mixes cold water with hot, reducing the temperature of the water in the hot water distribution system;

(m) Flow restrictor for hot water lines in a residential or commercial building or industrial plant, which is a device that limits the rate of flow of hot water from shower heads and faucets;

(n) Burner for oil fired heating equipment in a residential building, which is a device which atomizes the fuel oil, mixes it with air and ignites the fuel-air mixture, and is an integral part of an oil fired furnace or boiler, including the combustion chamber;

(o) Individual meters to replace a master meter for gas, electricity and hot water in a commercial building, which are meters that measure the consumption of gas, electricity or centrally distributed hot water for individual users, instead of the total consumption which is measured by a master meter;

(p) (1) New oil burner in a commercial building or industrial plant, which is a device that meters, atomizes, ignites and mixes the oil with air for the combustion process of a boiler; or

(2) New boiler controls in a commercial building or industrial plant, which are devices that sense the need for reducing or increasing the firing rate and change the combustion air and oil flow rate accordingly;

(q) Controls for lighting in a residential or commercial building or industrial plant, which are manual or automatic cut off switches for lighting systems that allow cut off of all lighting or a portion of the lighting systems when lighting is not required;

(r) Automatic HVAC control system in a commercial building or industrial plant, which is a device which adjusts the supply of heating or cooling to meet space conditioning requirements;

(s) High efficiency electric motor or motor controls in a commercial building or industrial plant, which replace an existing motor or motor controls, resulting in not less than a specified increase in efficiency at a specified level of use, as determined by FEA; and

(t) Whole house ventilation fan in a residential building, which is a fan which removes air from the inside of a residential building to the outside.

§ 450.32 Requirements and limitations for energy conservation measures.

(a) Ceiling insulation shall be that amount which is required to raise the total ceiling insulation in a heating zone as measured by an R-Value, to levels not greater than—

Heating zone:	*R-Value*
0	26
1	26
2	26
3	30
4	33
5	38

(b) Wall insulation shall be eligible in heating zones 2, 3, 4, and 5.

(c) Floor insulation shall be that amount which is required to raise the total floor insulation in a heating zone, as measured by an R-Value, to levels not greater than—

Heating zone:	*R-Value*
0	0
1	0
2	13
3	19
4	22
5	22

(d) Insulation for hot bare pipes shall be eligible in all climate zones.

(e) Caulks, sealants, and weatherstripping shall be eligible in all climate zones.

(f) Roof insulation shall be no greater than that amount which is required to raise the total insulation to the level of R-Value 20 in heating zone 5. In all other heating zones, this amount of roof insulation is eligible where the structure is air conditioned by an absorption chiller.

(g) Clock thermostats shall automatically change a temperature setting to match heating and cooling demands, and complete not less than one cycle of adjustment in a 24 hour period, in all climate zones.

(h) Exterior insulation for hot water heaters shall be no greater than that amount which is required to raise the total exterior insulation to the level of R-Value 19, in all climate zones.

(i) Insulation for forced air ducts shall be no greater than that amount which is required to raise the total insulation to the level of R-Value 19, in all climate zones.

(j) Storm windows shall be eligible in heating zones 2, 3, 4, and 5, provided that existing windows are single glazed.

(k) Efficient lighting fixtures and lamps shall produce more than twice the useful light per watt of the lighting system they replace, in all climate zones.

(l) Mixing valves for an hot water supply line shall be capable of manual adjustment of water temperature, without water shut off or disconnection, in all climate zones.

(m) Flow restrictors for hot water lines shall be eligible for all shower heads and faucets in all climate zones.

(n) Residential burners for oil fired heating equipment shall—

(1) Cost less than $340 installed;

(2) Be certified by the manufacturer to be capable of yielding an efficiency rating of 82 percent or higher in a new furnace as measured by a standard steady state efficiency test measuring CO_2 and stack temperature; and

(3) Replace inefficient burners, which shall be burners that cannot be adjusted using the procedures and tests prescribed in E.P.A. publication 600/2-75-069A, entitled Guidelines for Residential Oil Burner Adjustments, Oct., 1975, to perform at an efficiency not less than—

Zone:	*Efficiency after adjustment*
0	0.57
1	.65
2	.69
3	.73
4	.74
5	.75

(o) Individual meters to replace master meters for gas, electricity and hot water shall be permitted in all climate zones.

(p) New commercial or industrial oil burners and controls shall—

(1) Replace oil burners that cannot maintain 10 percent CO_2 at ¼ firing rate, 11 percent CO_2 at ½ firing rate and 14 percent CO_2 at full firing rate, while producing less than number two smoke spot number for No. 2 oil or less than number three smoke spot number for No. 6 oil, using the test prescribed in ASTM Standard D2156-65(70); and

(2) Be certified by the manufacturer to maintain 11 percent CO_2 at ¼ firing rate, 12.5 percent CO_2 at ½ firing rate, and 14.5 percent CO_2 at full firing rate, while producing less than number two smoke spot number for No. 2 oil or less than number three smoke spot number for No. 6 oil using the test prescribed.

(q) Controls for lighting shall be installed on a circuit having a wattage of more than 1,500 watts for automatic controls and 400 watts for manual branch circuit switches, in all climate zones.

(r) HVAC controls, in all climate zones, shall be—

(1) Automatic, turn down, time actuated thermostats;

(2) Steam controls, valves, thermostats, timers, or external temperature sensors to limit space temperatures;

(3) Economizer controls and systems to utilize outside air in lieu of conditioned air when outside air temperatures will assist;

(4) Controls to reduce air distribution volume to meet demand;

(5) Controls to reduce heating or air conditioning systems output to minimum levels during unoccupied periods; or

(6) Automated computer, microprocessor and logic controller associated with HVAC control.

(s) High efficiency motors or motor controls shall have substantially continuous annual use, 5,000 hours minimum, and shall increase efficiency of operation not less than—

Motor horsepower:	Efficiency increase percentage points[1]
1.5	5.7
2	5.4
5	3.9
7.5	3.9
10	3.4
15	3.0
20	2.4
25	2.3

[1] Efficiencies shall be determined by IEEE standard 112A method B, under ANSI standard C.50.2. Percentage points efficiency increase is the arithmetic sum of the efficiency of the new motor operation less the efficiency of the existing motor operation.

(t) Whole house ventilation fans shall have the capacity to provide one complete exchange of air in less than two minutes, provided that the residential buildings are air conditioned and located in cooling zones 1, 2, or 3.

§ 450.33 Renewable-resource energy measures.

Subject to the requirements set forth in § 450.34, a renewable-resource energy measure shall be a—

(a) Solar water heater, which is a system which captures energy radiated by the sun, and uses it to heat water;

(b) Air source heat pump, which is a system which is part of the central heating system and which has the capability of extracting heat from a body of air and transferring this heat to a body of liquid or to another body of air for space conditioning purposes;

(c) Water source heat pump, which is a system which is part of the central heating system and which has the the capability of extracting heat from a body of water and transferring this heat to another body of liquid or to a body of air for space conditioning purposes;

(d) Solar space heating or cooling system, which is a system which captures energy radiated by the sun, and uses it for space conditioning purposes;

(e) Solar process heating system, which is a system which captures energy radiated by the sun for use in industrial or agricultural processes;

(f) Solar powered pump, which is a system which captures energy radiated by the sun, and uses this energy to power a pump;

(g) Skylight, which is a device which is installed to replace small portions of a roof for the purpose of supplying a portion of the lighting requirements of a building;

(h) Solar electric dispersed photovoltaic system, which is a system which involves the use of small arrays of cells which convert solar radiation into electric power for on-site use;

(i) Wind powered generator, which is a system which captures and stores the energy transmitted by the wind and transforms this energy into electric power;

(j) Wind powered water pump, which is a system which captures the energy transmitted by the wind and uses this energy to extract water from a reservoir;

(k) Urban waste-fired boiler, which is a system which is partially or entirely fueled by refuse or a refuse derived fuel;

(l) Urban waste pyrolysis system, which is a system which uses urban wastes as a fuel and processes the wastes into a liquid or gaseous fuel;

(m) Agricultural waste-fired boiler, which is a system which is partially or completely fueled by agricultural residues;

(n) Wood-fired stove, which is a stove fueled by wood and which is installed primarily for space conditioning purposes;

(o) Wood-fired boiler, which is a system which is partially or completely fueled by wood or wood residues;

(p) Geothermal space heating or cooling system, which is a system that uses heat extracted from the earth for either electrical generation or space conditioning purposes; and

(q) Water powered generator, which is a system which captures and stores the energy contained in moving water and transforms this energy into electricity.

§ 450.34 Requirements for renewable-resource energy measures.

(a) Except as provided in paragraphs (b) or (c) of this section, a renewable-resource energy measure listed in § 450.33 must be evaluated by a verification audit in accordance with the procedures in Subpart C of this part to determine whether cost savings in a specific application are sufficient to recover the costs of purchase and installation within the attributed life of the energy measure.

(b) A solar water heater shall not be required to be evaluated by a verification audit if it is a system which—

(1) Is installed in a residential building; and

(2) Replaces an electric resistance water heater; and

(3) Is purchased, installed, and maintained at a total cost which shall not exceed the maximum allowable cost which shall be computed by—

(i) Selecting the correct system life factor in the Federal Region in which the system will be installed for the attributed life of the solar heater specified by the manufacturer in the following table—

Table of system life factors

Federal region	Years			
	10	15	20	25
1	280	340	380	410
2	270	330	370	390
3	280	350	390	420
4	300	370	410	440
5	290	360	400	430
6	330	410	460	490
7	290	360	400	430
8	280	350	390	410
9	300	370	420	445
10	320	400	460	490

(ii) Multiplying the system life factor by the current year's electricity rate for water heating in effect for the user of the system to be installed expressed in cents per kilowatt hour; and

(iii) Multiplying the product by the percent of the total hot water demand the system will supply.

(c) An air source heat pump shall not be required to be evaluated by a verification audit if it is a system which is—

(1) Installed in a residential building; and

(2) (i) A heat pump with only heating capability which replaces electric resistance space heating; or

(ii) A heat pump with both heating and cooling capability which replaces electric resistance space heating and air conditioning; and

(3) Purchased, installed, and maintained at a total cost which shall not exceed the maximum allowable cost which shall be computed by—

(i) Selecting the correct climate zone factor for the Federal Region and heating zone in which the system will be installed using the following table—

Table of climatic factors

Federal region	Heating zones					
	0	1	2	3	4	5
1	—	—	—	3.9	2.8	—
2	—	—	3.9	3.8	2.7	—
4	5.1	4.8	4.3	—	—	—
5	—	—	4.2	4.1	2.9	2.7
6	5.6	5.4	4.8	4.7	3.4	—
7	—	—	4.3	4.2	3.0	—
8	—	—	—	4.0	2.8	2.6
9	—	4.9	4.4	4.3	3.1	—
10	—	—	4.7	4.6	3.3	—

NOTE.—Dashes in table correspond to points where a heating zone and a Federal region do not coincide.

(ii) Multiplying this factor by the previous year's heating cost for the user of the system to be installed.

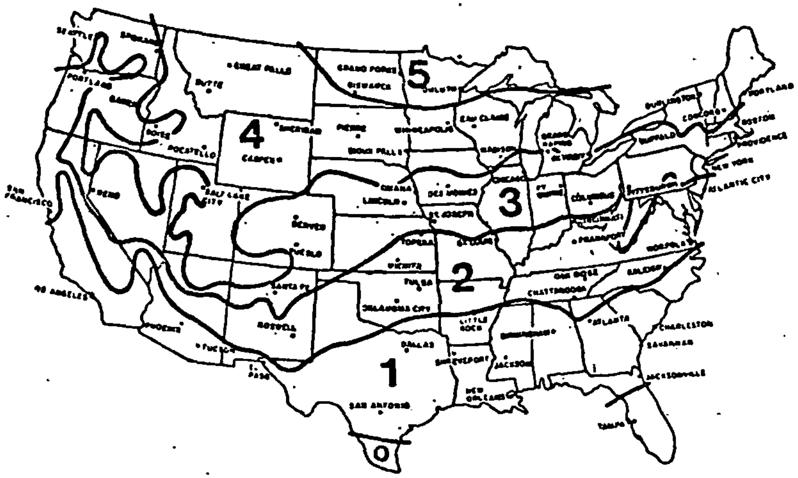
§ 450.35 Climate zones.

(a) FEA shall determine whether to restrict an energy measure to a climate zone.

(b) FEA shall designate climate zones, consisting of heating zones as shown in Appendix B or cooling zones as shown in Appendix C.

Appendix B

HEATING ZONES FOR ENERGY MEASURES

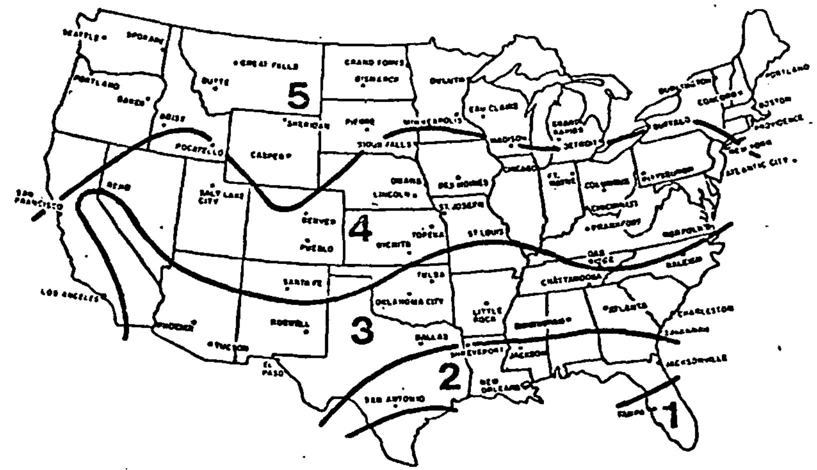


Notes:

1. Alaska is included in Heating Zone 5.
2. Hawaii, Puerto Rico and the Virgin Islands are included in Heating Zone 0.

Appendix C

COOLING ZONES FOR ENERGY MEASURES



Notes:

1. Alaska is included in Cooling Zone 5.
2. Hawaii, Puerto Rico and the Virgin Islands are included in Cooling Zone 1.

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