consulting engineers

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GROUND SOURCE HEAT PUMPS

1.0 ABSTRACT

Ground source heat pumps (GSHP's) can be integrated into the design of new and existing building HVAC heating and domestic hot water (DHW) heating systems economically. GSHP's can be packaged heating and cooling water to air heat pumps, which is a conventional use of this equipment. This paper discusses how water to water GSHP's can be integrated into a central heating system. GSHP's can be used as baseloading heat generators in tandem with heating hot water (HW) boilers to reduce primary fuel heating energy consumption by 25 - 30%, site fuel energy consumption by 60 - 70%, CO_2 emissions by 20 - 30% and reduce life-cycle heating energy costs by 35 - 45%.

This paper discusses systems concept criteria, hydraulics and thermal storage, the condenser water loop and heat pump control requirements, optimization of system performance as well as proper economic analysis for the integration of GSHP's into central heating plants.

2.0 INTRODUCTION

Central systems incorporating heat pumps as the heat generator require correct sizing of the compressor to match the load while maintaining trouble free operation and optimal seasonal efficiency.

This paper discusses the physical arrangement as well as some operational requirements of heat pump heating systems. Three areas of design, which are most misunderstood and lead to most failure conditions are:

- Hydraulics and Thermal Storage
- Condenser Water Circuit
- Equipment Sizing and Performance Optimization

The first step, however, in evaluating the energy and dollar saving potential of a baseloading ground source heat pump (GSHP) heating plant is to develop an energy concept. This usually involves gathering together sufficient specific information concerning the project to produce an energy/feasibility study. Section 3.0 discusses general considerations for preparing a meaningful energy study, while the remainder of this paper reviews the equipment, arrangement, control and operation of these energy saving systems.

3.0 ENERGY SYSTEMS CONCEPT CRITERIA

In order to develop the energy concept for a heat pump/heating system several important criteria should be objectively studied: These include energy costs, sizing the baseloading heat generator for economical operation, system operation and control and life-cycle cost.

3.1 ENERGY COSTS

The Department of Energy projected energy price indices for 2005 indicate that energy price escalation rates will vary for various energy sources over the next 30 years.

The Table 3.1 shows a 30 year projected average fuel price escalation indices in 5-year increments for various energy sources. The table projects that electricity, natural gas and distillate fuel prices will rise continuously by over 100% in the next 30 years.

GSHP systems transfer the site energy consumption profile from a primarily fuel dependent system to a primarily electric user. For this reason it is important to run a life-cycle energy cost analysis for each system under consideration to decide which portion of the heating load should be generated electrically and which should be generated by burning other fuels on site. (Refer to Section 3.12, Life-Cycle Cost Analysis).

Table 3.1: DOE Projected Energy Price Indices, excluding Inflation (United States Average)

YEAR[1]	COMMERCIAL			
	Elect.	Nat Gas	Oil	LPG
2005	1.25%	5.48%	-6.97%	-5.13%
2010	1.48%	2.10%	1.06%	1.48%
2015	2.26%	1.80%	2.41%	2.55%
2020	2.53%	2.71%	2.37%	2.89%
2025	2.23%	2.56%	2.47%	2.04%
2030	2.04%	0.73%	2.55%	2.21%
2035	2.07%	2.50%	2.58%	2.22%

[1] DOE 2005 United States Average Energy Price Indices were published in October 2005 and are released annually. These values consider the nominal discount rate, which include the effects of inflation.

3.2 GSHP BASELOADING HEATING PLANT

Planning the integration of heat pump(s) into heating systems should be done during the concept phase of the project to ensure that the system will operate economically, and that initial costs be budgeted properly. Deciding whether a monovalent (hybrid, heat pump only) or bivalent (heat pump and boiler) system should be installed is an important step in the analysis.

3.2.1 MONOVALENT OPERATION

A monovalent system requires that the heat pump provide 100% of the required heating run hours for the application. The heat pump(s) must be sized to handle the design load requirement. In the event of an equipment failure of one heat pump, backup should be available. **(See Figure 1 and 2).**

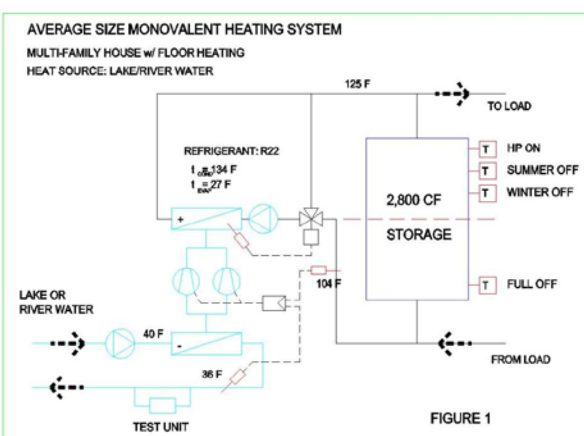

FIGURE 1

Monovalent systems installed in residential applications can operate very efficiently when coupled to radiant floor heating systems, because the condensing temperature is typically lower than with radiator or air side heating systems.

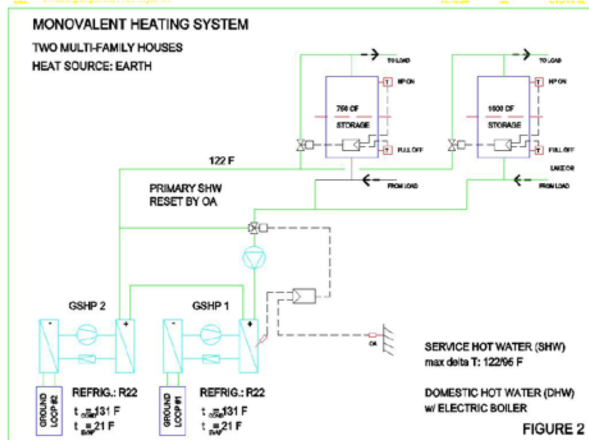

FIGURE 2

3.2.2 BIVALENT OPERATION (HYBRID)

A bivalent system requires that the heat pump provide 100% of the required heating for only a portion of the heating season. During lower outdoor ambient temperatures the additional heating requirement will be met by a boiler or other type of heat generator. **(See Figure 3 and 4).**

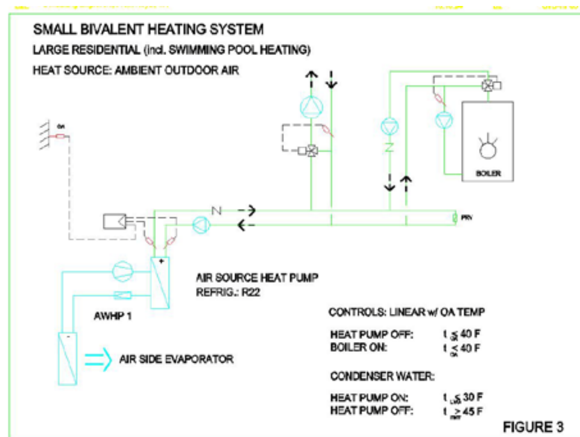

FIGURE 3

There are three types of bivalent heating systems:

3.2.2.1 Bivalent-Alternative Operation

The bivalent-alternative system allows the heat pump and boiler to operate independently of one another. The heat pump meets the heating requirement until the system balance (bivalence) point, and the boiler covers the load during lower outdoor ambient temperatures.

Heat generator switch-over can occur dependent upon outdoor temperature or by service hot water return temperature.

3.2.2.2 Bivalent-Parallel Operation

The bivalent-parallel system allows the heat pump to provide 100% of the required heating until the system balance (bivalence) point. With outdoor temperatures below the balance point both the heat pump and boiler are operated to meet the heating load. This system is only possible if the service hot water return temperature does not exceed the heat pump condenser discharge temperature.

Heat generator switch over can occur dependent upon outdoor temperature or by service hot water return temperature. This system is applied where low temperature heating hot water distribution will handle the load all year long.

3.2.2.3 Bivalent-Partial Parallel Operation

The bivalent-partial parallel system can allow the heat pump to be downsized compared with the bivalent-alternative system.

The heat pump covers the entire heating requirement until the balance point. With outdoor temperatures below the bivalence point both the heat pump and boiler operate. The heat pump is then switched-off at a predetermined outdoor temperature below the balance point. This allows the heat pump to operate for more heating season run hours.

Heat generator switch-over can occur dependent upon outdoor temperature, by service hot water return temperature, by the availability of the heat source or by the availability of the primary energy source.

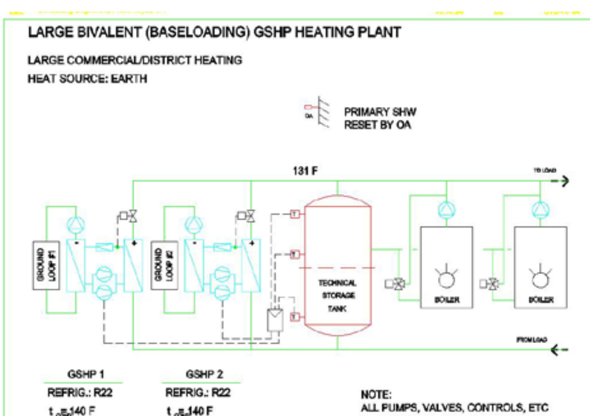

FIGURE 4

3.3 HEATING FREQUENCY

A careful determination of the heating load and a graphical representation of the yearly load over time is required to size the heat pump system.

The heating frequency curve showing the heating load as a function of yearly operating hours is developed using an energy balance diagram and local bin temperature distribution. **(See Figure 5).**

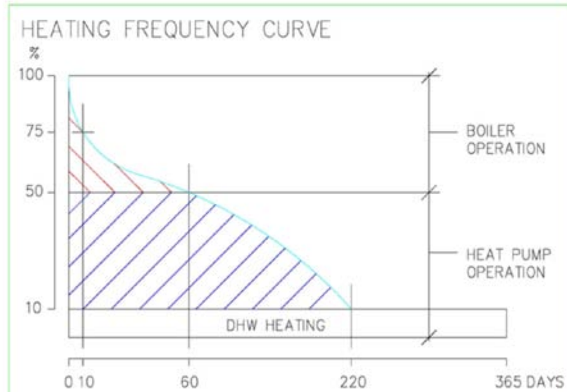

FIGURE 5

3.4 BASELOADING USING GROUND SOURCE HEAT PUMPS

The heating frequency curve can be used to establish the base load for the heat pump. This method is used for multivalent systems. The most expensive and efficient machine should be selected to meet the most frequent point of operation during the year. The less expensive hot water boiler is selected to handle peak heating loads. **(See Figures 6, 7 and 8).**

As mentioned above it is economical to install and control an electric heat pump to handle the base load and a hot water boiler to handle peak system capacity.

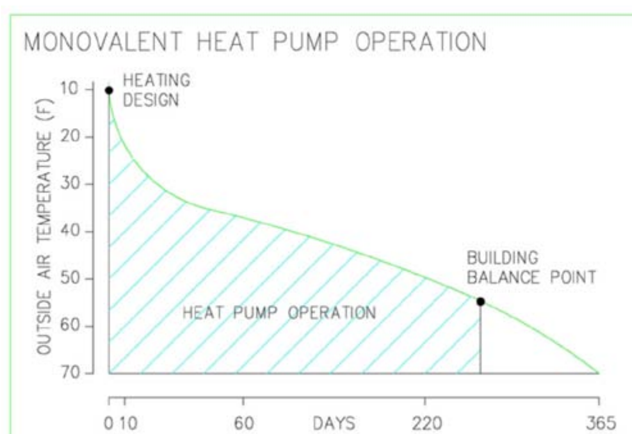

FIGURE 6

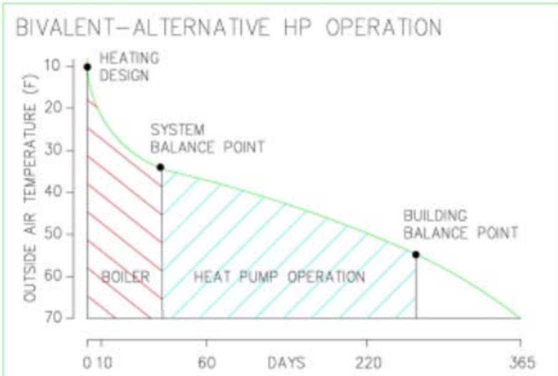

FIGURE 7

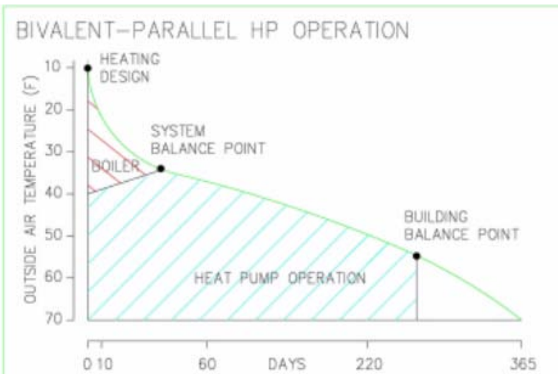

FIGURE 8

3.5 GSHP VERTICAL GROUND LOOP HEAT EXCHANGER

A ground loop heat exchanger (GLHX) can be constructed either vertically or horizontally in the earth. For the purposes of this paper the discussion is limited to vertical systems, because central heating plants involve large equipment capacities and therefore greater heat exchange length. If sufficient land is available, horizontal systems may be considered to reduce investment costs. However, the additional land used will not be available for future building expansion or development other than for parking, athletic fields, etc.

Soil Type	Pull-Down Capacity (BTUH/ft)	GSHX Design Length (ft/MBH)
Rock and saturated soil	52 - 57	17 - 19
Heavy damp soil	42 - 47	21 - 24
Dry Soil	max 31	min 32

Vertical ground loop heat exchangers connected with a GSHP baseloading heating plant require special consideration for proper design. Unlike GLHX's connected for heating and cooling duty as with commercial grade factory-built water to air heat pumps, these systems only remove heat from the ground. Therefore, the earth is not mechanically recharged during simultaneous heating/cooling operation or during summer cooling only operation.

Operational measurements made by the "Institute für Geophysik der ETH Zürich" a university in Zürich Switzerland show that by proper system design the mean earth temperature will fall-off gradually leveling out ≈ 1.5 °F below system start-up temperature with sounding spacings of ≈ 15 feet after approximately 3 years. Seasonal heat pump efficiencies for monovalent and bivalent GSHP systems were nonetheless measured with COP's between 2.9 and 3.1.

Important considerations for maintaining seasonal COP ≥ 3 are:

1. Some maximum continuous pull-down capacities of the heat source (Earth) are shown below. When the heat pump capacity at the evaporator exceeds these values, seasonal heat pump efficiency may drop below COP ≈ 3.0.

Relatively inexpensive computer software is available to help the design engineer properly size the ground loop heat exchanger. Test holes (min. 2 soundings) should be drilled on site and a geotechnical survey/report complied to characterize the soil conditions. Using site specific data helps to minimize the first cost of the ground loop heat exchanger.

1. Average annual heat pump run hours should not exceed 1,800 hours in order to maintain seasonal heat pump COP ≥ 3.0. Designs with favorable conditions, as shown below, may operate efficiently up to 2,000 hours per year.

   - Systems designed for smaller pull-down capacity.
   - Good thermal soil characteristics.
   - Favorable geometric arrangement of the vertical soundings (sufficient spacing, min. 15 ft).

2. Average daily heat pump run hours should not exceed 16 hours with an 8 hour relaxation (recharge) period.
3. Turbulent flow, Re ≈ 2500, should be maintained through the ground loop heat exchanger. When variable speed GLHX pumps are used, they should be reset by heating system return water temperature.
4. The heat transfer medium between the ground loop heat exchanger piping and the ground should be continuous. To insure that the installation is properly performed the number of bags of grout

per sounding should be specified. In addition, the pour should be pumped from the bottom to the top of the exchanger circuit. The valves for two pipe spacing thermal conductivities vary widely depending on the grout used. Thermally enhanced grout can be twice as conductive as regular grout. Regular grout has a thermal conductivity of 0.40 BTU/h. ft. °F and thermally enhanced grout has a thermally conductivity of 1.2 BTU/h. ft. °F. This is a three fold increase in thermal conductivity between the earth and the GSHP system.

3.6 DOMESTIC HOT WATER GENERATION

It will be necessary to determine whether the domestic hot water should be generated with the heat pump, with an electric boiler, or with a fuel fired boiler.

A construction cost estimate for the electric heat pump, electric boiler, and fuel fired boiler should be developed. The life cycle cost of these three alternatives should be examined in order to make the final determination.

Operation of the heat pump without considering domestic hot water generation is very economical because the condenser temperature can be adjusted dependent on the local weather pattern to improve system efficiency. This results in better seasonal efficiency.

3.7 STORAGE TANK

The decision of whether to install a technical storage tank (decoupler) or a thermal storage tank is necessary. (See Section 4.1, Thermal Storage).

Electric heat pumps coupled with ground, river or lake water or earth coupled heat wells perform efficiently with thermal storage tanks.

System sizing and rate structures can be discussed with the local electric utility so that 60-70% of the next day heating requirement can be generated and stored at night and at off-peak energy prices or to reduce costly demand charges.

At low outdoor temperatures, if the heat storage is depleted, the remaining load can be handled by the electric heat pump and/or a fossil fuel fired boiler to reduce peak winter time electric demand.

3.8 NUMBER OF COMPRESSORS AND UNLOADING

The number of compressors and the available steps of unloading is important when planning the heat pump system. Unloading compressor systems, whether reciprocating or screw type, affect the energy balance of the operating system. Each system should be evaluated at all operating conditions to ensure proper regulation and control.

Small compressors which cannot be unloaded are simply switched on and off as required.

Larger reciprocating compressors which can be unloaded are furnished with mechanical devices that prevent suction valves in given cylinders from closing. Many compressors of this type are arranged so that all unloaders are effective when the compressors start up. They remain unloaded until the oil pressure reaches a predetermined point. This allows the use of normal starting-torque motors. Hot gas by pass capacity control is not recommended. The compressor often runs hot and efficiencies are typically lower than with cylinder unloaders.

Larger screw compressors can be unloaded linearly and follow fluctuations in the heating system load well. However, efficiencies drop during part load operation.

Both reciprocating as well as screw compressors are suitable for baseloading heat pump duty.

To determine the number of compressors, the heating part load and the influence of the storage tank must be considered.

3.9 ELECTRICAL DEMAND MEASUREMENT AND REGULATION

The electrical demand is often measured and regulated in large commercial systems. This allows peak shaving when the plant electrical demand reaches a predetermined set point. When installing electric heat pumps each unit should be monitored and controlled by the facility management system (FMS) in order to prioritize operation of the facilities energy systems.

Electrical demand supervision helps avoid short electrical peaks which can increase electrical demand charges and increase system life cycle costs.

3.10 HEATING PIPING NETWORK

When bivalent systems are designed it is important to determine whether one or more piping networks should be installed.

Separation of piping networks into high and low temperature circuits is sometimes desirable. These piping systems are used for waste heat recovery at a low temperature source, such as with condensing boiler exhaust gas heat recovery. The reclaimed heat can be used for floor heating, pool water heating, service hot water preheating or DHW boiler feed water preheating.

3.11 ENERGY CONSUMPTION ESTIMATING METHODS

A thorough examination of actual or estimated building/facility HW and DHW design heating loads and monthly and annual energy consumption must be performed on each system under consideration to prepare a life-cycle cost analysis for the project.

3.11.1 EXISTING BUILDINGS

There are several sources for obtaining suitable data required to estimate annual as well as monthly energy consumption.

Site energy consumption can usually be obtained from the local utilities. Monthly electric energy consumption and peak electric demand can be requested from the local electric utility. Fossil fuel consumption can be determined by tabulating monthly energy consumption from the utility bills. Site water consumption for a complete year will also be required to estimate daily domestic hot water consumption.

Once these values are tabulated monthly a baseline energy model can be constructed. Using commercially available software, the building/facility can be modeled to approximate monthly and annual site energy consumption to within +/- 5% of the actual metered data. At this point the baseline energy model is complete and the building load computer files can be reused to estimate monthly and annual energy consumption for various HVAC systems and plants.

Baseloading GSHP Heating Plants are not modeled by commercially available energy simulation programs. Therefore, the base case and alternative systems selected must be modeled using a boiler(s) to cover 100% of the heating requirement of the building/facility, and the output must be adjusted by hand to approximate the actual GSHP Baseloading Heating Plant monthly and annual energy consumption. When the heat source on the evaporator side of the heat pump is relatively constant, as is the case with vertical ground loop heat exchanger systems, this is relatively easy. However, if the heat source fluctuates substantially throughout the year, the calculations must be performed for each operating and evaporator temperature condition throughout the year.

3.11.2 NEW BUILDINGS

The procedure for estimating monthly and annual site energy consumption for new buildings is easier than for existing buildings. This is primarily due to the fact that baseline energy consumption need not be matched to an existing system.

The requirements for estimating the base case and alternative heating plant energy consumption, however, are similar to those discussed in Section 3.11.1.

3.11.3 SPECIFIC ENERGY CONSUMPTION

It is sometimes easier to think of annual energy consumption in an energy value per square foot area or per cubic foot volume of the net conditioned space in the building/facility. This can be a good indicator of the energy performance of a certain building type. It can not, however, indicate the performance of specific equipment, or systems in the building. This is because HVAC and DHW systems are often custom built as a building/facility package. Some systems may incorporate heat recovery systems, thermal storage systems, DDC energy management systems etc. which affect the specific energy consumption of the heating plant. Climate and hours of operation are also an influential factor. The specific energy consumption value (kWh/sf. yr or KBTU/sf. yr.) is a good indicator of the energy saving potential of an existing facility. For instance, if elementary schools consume 25 kWh/sf annually, but one school is consuming 35 kWh/sf annually it is in need of a energy audit to determine why it is not operating efficiently.

3.12 LIFE-CYCLE COST ANALYSIS

A life-cycle cost analysis should be performed for each heating system under consideration for a particular application. For the purposes of this paper we will discuss GSHP Baseloading Heating Plants however, the methodology is valid for all systems. In order to determine the life-cycle net savings of a GSHP Baseloading Heating Plant over another type of plant, a base case heating plant should be identified.

The base case heating plant should have the least first cost, but highest life-cycle cost of the systems studied. Alternative systems will have correspondingly higher investment costs, but provide lower life-cycle costs. This may be true for a number of reasons. The annual operation, maintenance and repair costs may be lower, the non-annual repair and replacement costs may be less frequent and lower, the Annual energy costs may be lower and/or the residual (salvage) value of the alternatives may be higher than the base case heating plant.

In addition, monetary one-time incentives may be provided by the local electric utility for integration of GSHP Baseloading Heating Plants. These incentives must be deducted from the initial investment cost of the system.

Some electric utilities offer incentives to large customers for integration of innovative heat pump technology into central heating plants, because GSHP's save energy while transferring consumption to electric utilities rather than fossil fuel distributors.

Once the life-cycle costs for the base case and alternative systems are determined, then life-cycle net savings, savings to investment ratio, discounted payback period and adjusted internal rate of return values can be calculated. For more comprehensive information concerning proper building life-cycle analysis methods, we recommend publications and software available from the U.S. Department of Energy, Federal Energy Management Program, Washington DC.

3.13 CO_2 EMISSIONS

CO_2 emissions can be reduced from between 20 and 30% with GSHP baseloading heating plants. This is due to the fact that heat pumps operate with good seasonal efficiencies, COP ≈ 3.0.

The following energy flow diagrams show approximate flows of energy from the source (primary heating energy) to the site (demand heating energy). **(See Figures 9, 10 and 11).**

The table below shows CO_2 combustion emissions per energy unit of fuel chosen.

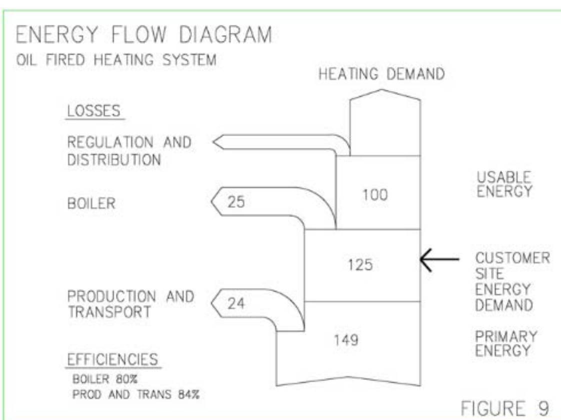

FIGURE 9

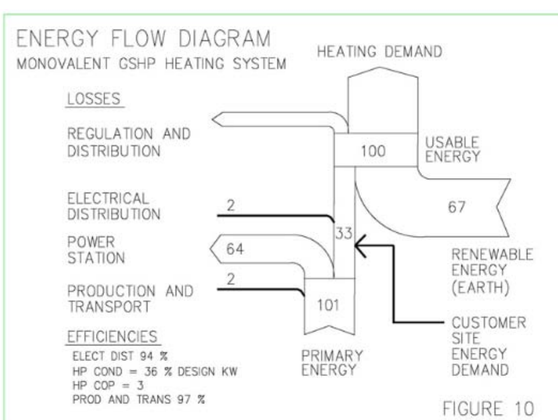

FIGURE 10

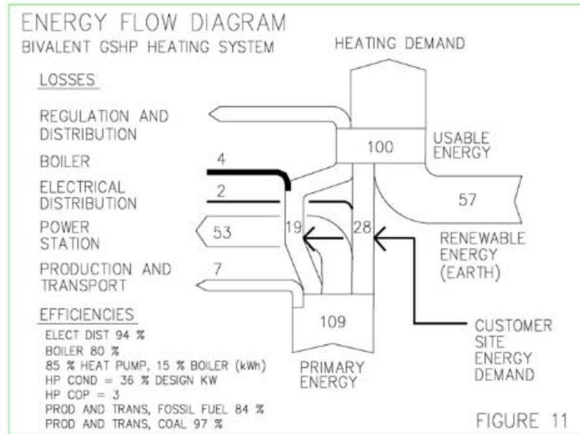

FIGURE 11

TABLE 3.13.1 APPROXIMATE CO_2 COMBUSTION EMISSIONS PER ENERGY UNIT AND PER KWH

Waste Gas	NatGas	LPG	# 2 Fuel Oil	Coal
CO_2	15.1 (lb/Therm)	16.0 (lb/gal)	22.0 (lb/gal)	3.0 (lb/lb)
CO_2	0.52 (lb/kWh)	0.59 (lb/kWh)	0.54 (lb/kWh)	0.73 (lb/kWh)

Using the energy flow diagrams shown above to approximate primary energy demand for the HW boiler (conventional), GSHP monovalent and GSHP baseloading (bivalent) heating plants, we can approximate the CO_2 emissions in lb per kWh or MBH. These values are shown in the table below:

TABLE 3.13.2 APPROXIMATE CO_2 EMISSIONS PER KWH AND MBH OF HEATING PLANT PRIMARY (DEMAND) ENERGY CONSUMPTION

CO_2 Emissions				
Heating Plant	**Input (kWh)**	**Output (kWh)**	**lb/kWh**	**lb/MBH**
NatGas[1] (Conventional)	1.49	1.00	0.77	0.23
LPG[2] (Conventional)	1.49	1.00	0.88	0.26
# 2 Fuel Oil[3] (Conventional)	1.49	1.00	0.80	0.24
Electricity[4] (Monovalent GSHP)	1.01	1.00	0.63	0.18
Electricity[4] NatGas[1] (Bivalent GSHP)	0.93 0.16	1.00	0.58 0.08 ------ = 0.66	0.17 0.02 ------- = 0.19

Notes:

1. Assumes 80% efficient HW NatGas boiler.
2. Assumes 80% efficient HW LPG boiler.
3. Assumes 80% efficient HW # 2 oil boiler.
4. Assumes electricity generated by 85% coal fired boiler/steam turbine with efficiency 33%, and 15% nuclear generation.

4.0 HYDRAULICS AND THERMAL STORAGE

4.1 THERMAL STORAGE

The term "Thermal Storage" can be misleading when applied to baseloading heat pump systems. In some cases a tank is installed to decouple (buffer) the heat pumps (primary heat generator) from the heating load and provide sufficient thermal storage capacity to control compressor cycling. In other cases tanks are installed to store service heating or domestic hot water to reduce system operating costs as well as to decouple and control heat pump operation.

Storage tanks are separated into two general classifications:

- Technical Storage Tank – The storage tank acts as a decoupler to maintain a duty cycling frequency between two and three times per hour during design winter conditions.
- Thermal Storage Tank - The storage tank is designed to be loaded over a longer time period for regulated discharge at a specified condition. This type of storage tank should be designed to limit compressor(s) from cycling more than twice per hour during the loading cycle.

4.1.1 TASK OF THE TECHNICAL STORAGE TANK

- Hydraulic decoupling
- Reduce duty cycling frequency to increase the operating life of the compressor(s)
- Regulation of the heat generator, on/off heat pump cycling, boiler or other heat generators.

4.1.1.1 Technical Storage Tank Volume

The minimum tank volume required for a specified maximum compressor cycle frequency, **n** is given below:

$$V_{ST,\,MIN}\ [\text{gal}] = 0.03 \left(\frac{Q_c\ [\text{BTU}]}{n \times \Delta T\ (°\text{F})} \right)$$

The temperature difference should be calculated for 50% design heating conditions this will increase the size of the tank providing an additional safety factor.

The following equations are provided for calculation of the technical tank temperature differences:

Mixing loading

$$\Delta T_{MIX} = \left(\frac{(T_{C,MAX} + T_{OFF})}{2} \right) - T_{R,MAX}\ [°\text{F}]$$

Stratifying loading

$$\Delta T_{STRAT} = T_{STORAGE} - T_{R,MAX}\ [°\text{F}]$$

These equations have been separated into mixing loading and stratifying, loading control strategies, which will be discussed in subsequent sections of this paper.

4.1.2 TASK OF THE THERMAL STORAGE TANK

(In addition to the functions described for the technical storage tank)

- Shift electric energy consumption to off peak periods.
- Increase number of operating hours of the heat pump.
- Increase the number of operating hours for waste heat utilization (especially when the heat source fluctuates).

4.1.2.1 Thermal Storage Tank Volume

The minimum tank volume required for a maximum compressor cycle frequency,
$n < 2$ is given below:

$$V_{ST,\,MIN}\ [\text{gal}] = 0.12\ \frac{Q_c\ [\text{BTU}]}{\Delta T\ [°\text{F}]}$$

The temperature difference should be calculated at the design loading conditions. The following equations provide a good rule of thumb:

Mixing loading

$$\Delta T_{STEP} = \frac{(T_{C,MAX} + T_{OFF}) \quad - T_R}{2}\ [°\text{F}]$$

Stratifying loading

$$\Delta T_{STRAT} = T_{STORAGE} - T_R\ [°\text{F}]$$

The volume of the tank, V_{ST}, is calculated as useable volume. The actual useable volume is the volume of the tank between the heat pump on and off temperature sensors minus mixing zone volume.

4.1.3 MIXING OR STRATIFYING STORAGE TANK

The loading control strategy for both technical and thermal storage tanks can be of the mixing or stratifying type. **(See Figure 12).**

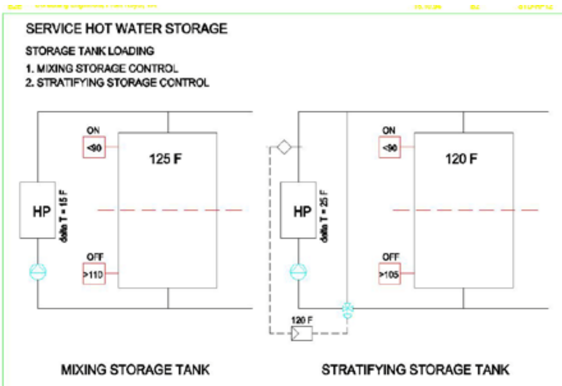

FIGURE 12

- Mixing Storage Tank - Loading occurs stepwise by circulating the mixed water in the tank through the heat pump condenser with progressively higher condenser leaving water temperature. This type of control strategy is only recommended for smaller systems, systems with only one heating zone or for technical storage tanks.
- Stratifying Storage Tank - Loading occurs evenly with the hot water level moving downward in the tank. It occurs in one pass of the heat pump leaving condenser water through the tank at a constant loading temperature.

4.1.3.1 Mixing Tank

The mixing storage tank does not provide supply temperature regulation at the condenser. It offers fewer hours at full load operation. In addition, the storage tank control sensor could be satisfied from a high load side return water temperature. The temperature difference between supply and return water to the heat pump fluctuates according to the capacity of the machine. The water flow must be regulated to keep the working pressure of the refrigerant within operating limits. Frequent cycling is required if a leaving condenser water set point is desired.

Advantages:

- Better coefficient of performance (because of lower average condenser return water temperature)
- Lower first cost (no loading controls)

Disadvantages:

- Fluctuations in hot water supply temperature
- Smaller useful storage capacity of the tank because of mixing
- Larger condenser water pump(s)

4.1.3.2 Stratifying Storage Tank

Larger storage capacity, longer run times for the heat pump, and simple control possibilities as well as lower loading temperatures.

Advantages:

- Constant supply temperature
- Larger useable volume (with smaller stratification losses)
- Smaller condenser water pump(s)

Disadvantages:

- Lower coefficient of performance (because of higher average condenser water return temperature)
- Higher first cost (loading controls)

4.1.3.3 Differential Stratifying Storage Tank

To achieve optimal stratification it is recommended to only allow the differential in flow rate between the heat generator and load through the tank. **(See Figure 13).**

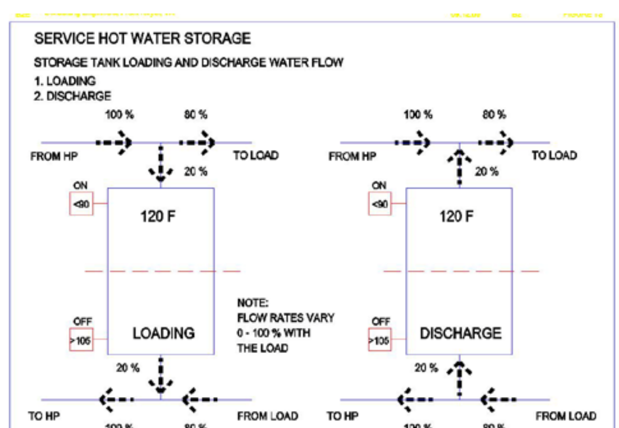

FIGURE 13

4.1.4 STORAGE STRATIFICATION

Loading the storage tank requires careful planning. Each initial flow of water into the tank generates a wave impulse and transfers energy to the water volume in the tank. This sets the water volume in motion. The supplied energy, especially with over sized circulation pumps, is significantly greater than the driving force creating the thermo-cline, because of the relatively small temperature differences in heat pump systems.

This situation is complicated because loading of the storage tank occurs from the top and unloading produces an impulse at the bottom. This means that the tank must be specially designed for stratification to be achieved.

4.1.5 SPECIAL TANK DESIGN FOR STRATIFICATION

Factory-fabricated steel tanks with integral loading distribution piping is very important. This system provides adequate stratification when the loading SHW circulating pumps are properly balanced. **(See Figure 14).**

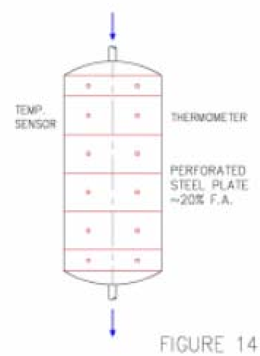

FIGURE 14

Various stratification tanks can be found on the market today. Each tank under consideration should be carefully selected by the design engineer because all tanks are not equal.

4.2 HYDRAULICS OF THE PRIMARY /SECONDARY DISTRIBUTOR

Heat pumps are sensitive to elevated return water temperatures from the heating system.

A temperature difference between supply and return water of 27°F (15°C) will tolerate almost no deviation of return temperature. A deviation of plus 9°F (5°C) requires a flow rate 33% higher than design across the condenser.

In order to achieve proper return temperatures a hot water mixing control must be installed for each heating zone. This means that a variable water flow will be necessary on the primary hot water loop. **(See Figure 15).**

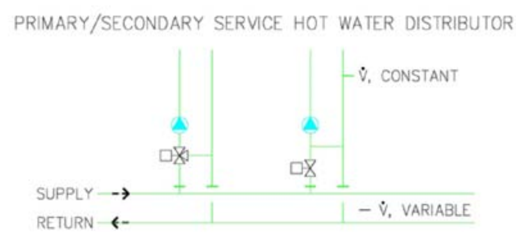

FIGURE 15

4.3 SYSTEM HYDRAULICS

In order to synchronize the capacity and water flow rate of the heat generator (heat pump) and the service hot water consumption it is necessary to incorporate a thermal storage tank into the system.

A load dependent variable water flow control and constant temperatures are required for proper function of the system. The water temperature can be reset (mixed) by the zone temperature controller as the load changes. **(See Figure 16)**.

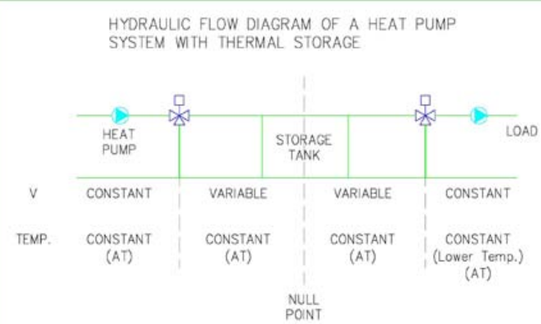

FIGURE 16

5.0 CONDENSER WATER LOOP

Only very small systems should be directly (hard) connected with the service hot water piping network. These systems are typically unregulated. This means that severe temperature fluctuations can occur which may cause excessive system cycling and premature compressor failure.

For example:

Supply under 95°F (35°C) = Heat Pump "on"
Return over 122°F (50°C) = Heating "off"

Larger systems require the following examination:

Figure 17 shows a heat pump system with the condenser water loop as well as stepped control, regulated from temperature sensors in the heating return water piping and the evaporator leaving water temperature.

5.1 REGULATION OF THE HEAT PUMP

5.1.1 ON/OFF CONTROL

The on and off command for the heat pump is regulated from the storage tank. The system on command switches the circulation pump across the evaporator first, then the circulation pump across the condenser is switched on. The heat pump is switched on automatically and maintains set point upon detecting flow across both exchangers. When set point is reached the heat pump is automatically switched off. The circulation pumps operate on a delayed shut-off mode to prevent under cooling of the evaporator and overheating of the condenser.

It is possible, for example, that after the heat pump is switched off the remaining heat in the condenser can activate the high-pressure cut-out control. If this occurred the heat pump controller would need to be reset by hand.

5.1.2 STEP CONTROL

After the heat pump is activated the step control for parallel compressors regulates operation of the heat pumps to maintain set point. The sensor is located in the condenser entering water piping which is the heating system return.

For example, three compressors are in operation in a 122/95°F (50/35°C) heating system when the return water temperature is 95°F (35°C). When the return water temperature climbs to 104°F (40°C) one compressor is switched off, leaving two compressors in operation. When the return water temperature climbs to 113°F (45°C) another compressor is switched off, leaving one compressor in operation. It is important that by proper selection of the compressors, the secured compressor(s) do not counter flow while the parallel system is operated at reduced capacity.

5.1.3 SUPPLY WATER CONTROL

The supply water temperature (heating water leaving the condenser) climbs very quickly after the heat pump is switched on. A normally closed heating valve with a 30 second reaction time, regulated by the leaving water temperature, is seldom able to avoid overheating at the condenser.

In order to eliminate this problem it is necessary to delay switching on the compressor. One method opens the valve, allowing water circulation, then the compressor is switched on. Another method used is supply water flow control dependent on the condenser head pressure.

When the valve open delay is shut off the temperature sensor will control the water flow and heat transfer.

The condenser head pressure climbs parallel with supply water temperature. With stepwise compressor control a temperature sensor and fast acting flow control valve can be installed. The control valve and sensor should be located near the heat pump to reduce reaction times due to greater water volumes in longer runs of pipe.

5.1.4 EVAPORATOR SAFETY

It is usually necessary to place an additional temperature sensor in the evaporator leaving water piping. This sensor is a low water temperature (low pressure) cut out safety. In addition, evaporators using water for heat transfer are susceptible to freezing if the heat source has been pumped down or if the heat source temperature varies seasonally. For this reason a freeze control temperature sensor is usually placed in the evaporator entering water piping to switch off the compressor below set point. When water/methanol solutions are used this problem can be eliminated.

5.1.5 SUMMARY

- ON/OFF control from a temperature sensor in the storage tank
- Step control from a temperature sensor in the condenser return water loop
- Supply temperature regulation dependent on condenser head pressure

5.2 STEP CONTROL AND THE CONDENSER WATER FLOW RATE

The condenser water control valve automatically regulates the water flow rate as compressors are switched on and off. Therefore a variable water flow rate is circulated with constant leaving water temperature to the primary hot water piping network. The internal water flow rate across the condenser remains constant. **(See Figure 17).**

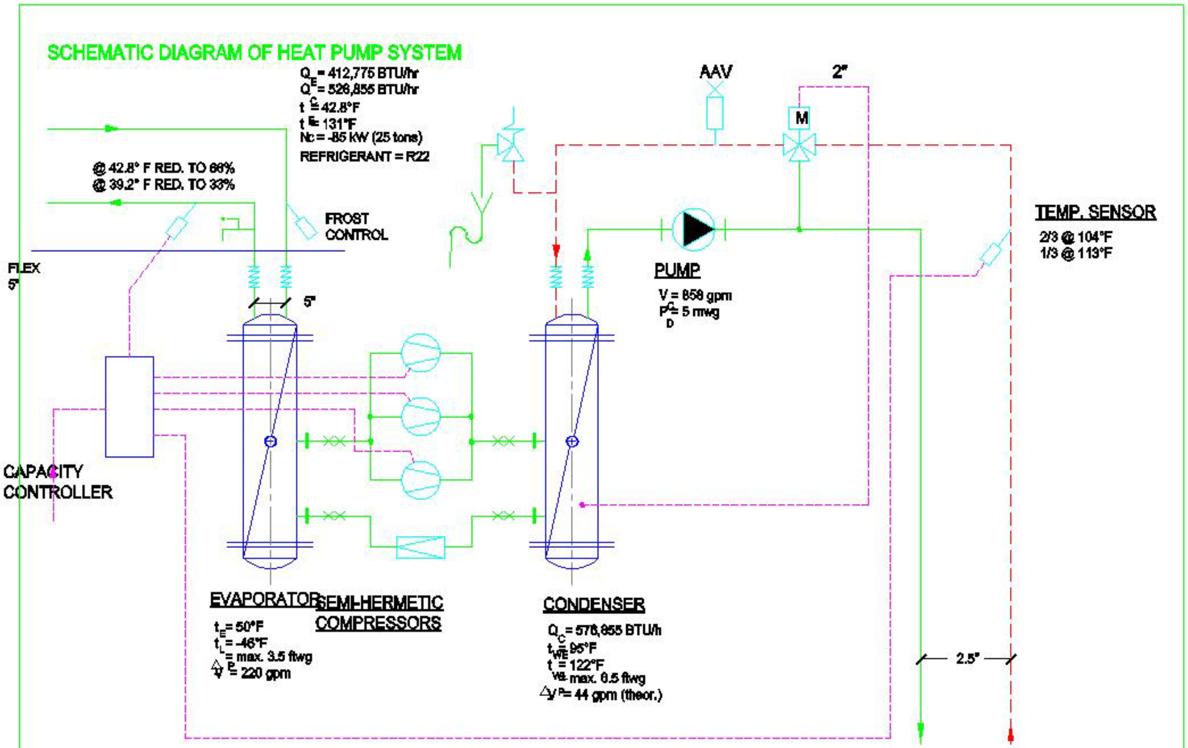

FIGURE 17

5.3 RETURN WATER FLOW CONTROL

As previously discussed a return water temperature sensor influences the parallel compressor step control. It is therefore also possible to provide a bypass in the supply water piping dependent on return water temperature. When the return water temperature is too high, compressors switch off automatically and the return water is mixed into the supply water preventing the supply water temperature from falling. This type of control is usually not required.

High return water temperatures are usually the result of problems in the design or sizing of the load side heat exchanger surface area. The installation of this type of return water control is used in exceptional cases and can be avoided with the development of a proper energy balance and flow diagram.

5.4 OPTIMIZATION OF THE WATER FLOW RATE ACROSS THE CONDENSER

Changes in the capacity of the condenser require adjustment of the required water flow rate across the condenser. The resulting balanced water flow rate is influenced by the following factors:

- Higher evaporator temperatures.
- Lower condensing temperature (due to seasonal changes).
- Differences in the evaporator or condenser heat exchange surface area from the theoretical calculation.
- Differences in the system head pressure.

5.4.1 ADJUSTING THE CONDENSER WATER FLOW RATE, V_C

The following equation can be used to calculate the water flow rate across the heat pump condenser:

$$V_c\ [\text{gal/min}] = \frac{Q_c\ [\text{BTUH}]}{500 \cdot \Delta T\ [°\text{F}]}$$

5.4.1.1 Mixing Storage and Condenser Water Flow Rate

A compromise must be reached considering the following factors:

- Greater flow rate, whereby the condensing temperature is held low, storage temperature fluctuations are small and usable storage volume is large.
- Lower flow rate, whereby the required condenser water pumping brake horsepower is small.

The following adjustments can be made to the condenser water temperature difference, ΔT_c:

Heat source nearly constant

Condensing Temperature	=	131°F
Balance Pt. Temp Diff		122/95°F
Heating Capacity	Q =	358 MBH
Water Flow Rate Across Condenser	V =	26 gpm

- monovalent operation
  $\Delta T_c = 0.5 \cdot \Delta T_{DESIGN}$
- bivalent operation
  $\Delta T_c = 0.7 \cdot \Delta T_{BALANCE\ PT.}$

Heat source varies greatly

- monovalent operation
  $\Delta T_c = 0.5 \cdot \Delta T_{DESIGN}$
- bivalent operation
  $\Delta T_c = 0.5 \cdot \Delta T_{BALANCE\ PT.}$

5.4.1.2 Stratifying Storage and Condenser Water Flow Rate

In general the following is valid:

- With a nearly constant heat source the condenser water flow rate can be minimized.
- With a variable heat source the condenser water flow rate must be raised.

The following adjustments can be made to the condenser water temperature difference, ΔT_c:

Heat source nearly constant

- monovalent operation
  $\Delta T_c = \Delta T_{DESIGN}$
- bivalent operation
  $\Delta T_c = \Delta T_{BALANCE\ PT.}$

Heat source varies greatly

- monovalent operation
  $\Delta T_c = 0.5 \cdot \Delta T_{DESIGN}$
- bivalent operation
  $\Delta T_c = 0.7 \cdot \Delta T_{BALANCE\ PT.}$

Considering the usual temperature fluctuations in heating systems, the following parameters which are shown in the example below should be given and are recommended to size the heat pump at the chosen operating conditions.

Example: Bivalent System with Heat Pump and Boiler

The temperatures shown are the theoretical operating temperatures. Actually from practical experience return water temperature fluctuations cannot be avoided. Based on this the condenser should be sized according to the following conditions:

Adjusted System Sizing Parameters:

Selecting the condenser for greater water flow (at the same water side pressure drop) increases the available condenser heat transfer surface area. In this case it results in an increased water flow of 30%.

This method considers entering condenser water temperature fluctuations and results in easier system balancing and fewer disturbances in system operation.

In each case the condenser heat transfer surface area should be calculated for each of the design conditions under which the system will operate. This will allow for optimal condenser selection.

5.5 HYDRAULICS WITH REDUNDANT SYSTEM ARRANGEMENT

Careful design of the switching concept for heat pumps arranged in series or in parallel is necessary.

Recommended system switching control is different for both cases:

- Switching the condensers in series with one temperature sensor and controller and one circulation pump.
- Systems with a parallel arrangement require two temperature sensors and controllers and two circulation pumps.

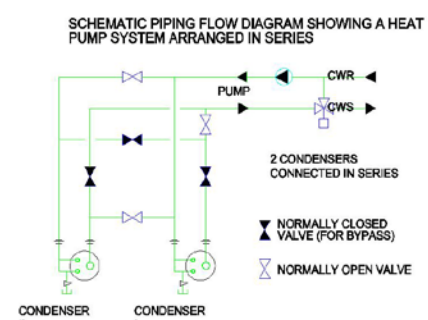

FIGURE 18

5.6 SYSTEM CONCEPT FOR TWO OR FOUR COMPRESSOR SYSTEMS

5.6.1 REDUNDANT SYSTEM WITH SEPARATE REFRIGERANT CIRCUIT

This concept includes two separate heat pump systems each with a condenser, compressor and refrigerant circuit.

Advantage:

- 100 % redundancy

Disadvantage:

- No improvement of the coefficient of performance in part load operation

5.6.2 ONE SYSTEM WITH TWO COMPRESSORS

A refrigerant circuit with one evaporator and one condenser.

Advantage:

- Good coefficient of performance in part load operation.

Disadvantage:

- 50 % redundancy

When planning redundant systems the aforementioned solutions must be weighed against one another.

The dual compressor solution represented in section 5.6.1 is suitable when 100% redundancy is absolutely necessary.

The staged compressor solution represented in section 5.6.2 is suitable when longer part load operation can be expected.

6.0 OPTIMIZING SYSTEM PERFORMANCE

Optimization means that all load and temperature conditions must be calculated and that all system components must be designed and sized, including the refrigerant piping.

- System operation at lower condensing and storage water temperatures allows the system to operate with a higher capacity. This can be a problem if the required heat energy entering the evaporator is not available.
- The condenser and evaporator heat exchange surface area provide a significantly higher capacity at part load operation than the available work of the compressor.

In addition it is a good idea to electrically submeter the GSHP baseloading heating plant as well as provide additional energy measurement devices which will be helpful in optimizing operation of the plant.

Installation of BTU meters on the secondary distributor will provide accurate measurement of heating zone (demand) energy consumption. kWh energy meters should be provided at each heat pump and fuel consumption should be measured at each boiler. This will allow the consulting engineer to calculate the efficiency of the baseloading GSHP system, and provide needed information for making operational adjustments to the system to improve operating efficiency.

7.0 FINAL REMARKS

GSHP baseloading heating plants operating with seasonal heat pump efficiencies of COP $\approx$ 3 can save between 25 - 30% of the primary heating energy and 60 - 70% of site heating energy consumption required by conventional HW boiler systems.

GSHP systems transfer heating energy consumption from fossil fuel burning HW boilers to electric heat pumps. This lowers life-cycle energy costs, because DOE projected electricity rates and fuel rates will escalate similarly over the next 30 years, and currently the fuel rates are higher than the electricity rates per input BTU.

GSHP monovalent and bivalent central heating systems can save between 20 - 30% of the CO_2 emissions created by conventional HW boiler systems.

Careful planning of the heat pump system chosen is a challenging task for the engineering design professional. As the need grows to develop new innovative technologies to save energy and money, the building owner will be faced with increased initial investment costs to realize reduction in life-cycle costs. The mechanical systems will become increasingly more sophisticated with applicable equipment and controls. The heat pump offers a excellent opportunity to save energy over the life-cycle of the HVAC system. Incorporation of bivalent (hybrid) systems offer numerous advantages over monovalent (heat pump only) systems.

Electric utilities are beginning to offer incentives for commercial building owners who are interested in commissioning professional engineers to perform life cycle cost analysis studies and implement engineering designs using innovative energy saving technology. The benefit to the electric utility is leveling annual electric power demand which assists in improving power plant operating efficiency and helps to reduce utility costs.

The benefit to the client is development of a customized system which meets the individual needs of the client and minimizes the life cycle cost of operating the system, while maintaining a specified minimum rate of return on the long term investment.

In addition to the possible economic benefits which can be realized by incorporating heat pumps into the heating system, something must be said about the ecological benefits using bivalent (hybrid) heat pump systems. These systems can reduce energy costs, reduce primary energy consumption, and reduce CO_2 emissions.

Proper design and formulation of the specification with possible system alternates is desirable. It is also a good idea to involve the consulting engineer to carefully review contractor prices and product substitutions. System component costs should be broken out. This will allow the engineer to develop a cost control spreadsheet which simplifies the contractor selection and cost control process.

The ground-coupled, closed-loop, water-source heat pump system (GSHP) can be configured to operate in a conventional arrangement with packaged water to air heat pumps, or it can be integrated into the central heating plant. The engineering principals and calculations are similar for both types of systems. These systems have proven to reduce the life-cycle cost over conventional 4-pipe HVAC systems by reducing site fuel consumption. At a moderate fuel escalation rate of 3.0% per year the cost of fuel will approximately double in 24 years. As the cost of fuel increases the GSHP systems will prove to be a cost effective HVAC system.

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