#5954

BA-92-9-2

AN ENERGY-EFFICIENT HVAC SYSTEM AT A HIGH SCHOOL

R.B. Stotz, P.E.
Fellow/Life Member ASHRAE

R.L. Hanson

ABSTRACT

This paper examines a method of using heat recovery and aquifer wells to provide an HVAC system that is economically and ecologically friendly as well as being energy efficient. It is applicable for any new or existing building with a high internal load and where aquifer wells can produce water in the 50°F to 77°F (10°C to 25°C) range. It specifically examines a large school building with an HVAC system designed for use with high-temperature water from natural gas boilers. It compares results obtained by incorporation of an energy management system with those obtained by modification of system design. The paper demonstrates the important role geothermal energy can play in the future.

INTRODUCTION

The objective of this study was to design an energy-efficient HVAC system for a high school to replace one that had been designed with the classroom portion being essentially a two-pipe system. All heating for these rooms was accomplished by the ventilation air system, which used high-temperature hot water from a gas-fired boiler also required to power the cooling system.

The problem was further complicated by climatic conditions that might require heating of all classrooms in the morning and cooling of some or all rooms in the afternoon. Variations in student load also had to be considered if even temperatures were to be maintained.

The high school is a windowless complex that was put into initial operation by the Bay County, Florida, school board in Panama City in August 1973. The overall complex includes a single floor of classrooms, shops, band room, choral room, library, and cafeteria and a two-story gymnasium area. The complex has an overall floor area of approximately 210,000 square feet, of which 174,743 square feet is controlled, conditioned space.

The student population is more than 2,000 and activities include those normal to a high school of this size. Bay County fields an excellent swimming team, which trains in an outdoor Olympic-size pool located on the campus.

Cooling and heating for the original HVAC system was supplied by two high-temperature hot water, two-stage, absorption chillers and two gas-fired, natural gas boilers. All heating for classrooms was accomplished by ventilation air units since classroom units had no heating coils. Large assembly areas, such as the gymnasium, library, cafeteria, and band and choral rooms, are served by air-handling units with both heating and cooling coils.

Boiler capacity was sufficient to handle either the chiller requirements or the combined heating requirements for the building, the athletic showers, the make-up kitchen wash water, and the swimming pool heating. However, the system design did not provide the flexibility of heating and cooling the classrooms simultaneously, which resulted in very uneven temperatures in the classroom areas. Load analysis revealed that those rooms with an exposure of two walls and the ceiling required cooling down to ambient 37°F (1°C) with full occupancy and lights; those with a single wall and ceiling exposure required cooling down to 29°F (−1°C); and those with only ceiling exposure required cooling to 15°F (−10°C). Thus, except on a handful of days each year, classroom heating is limited to early morning warm-up and the ability to offset variations in occupancy.

LOAD CALCULATIONS

The original system resulted in overall utilities consumption for the school of 360,000 Btu (105,480 W-h) per square foot each year. The first steps in reducing this were to eliminate heating of the swimming pool and to install an energy management system. With this corrective action, consumption was reduced to 166,000 Btu (48,638 W-h) per square foot per year.

For further reduction, the following were examined:

1. Load characteristics
2. System type
3. Chilled-water range
4. Condensing water range
5. Condensing means
6. Heating water range
7. Domestic hot water temperature and source

Load calculations indicated that the original 460-ton capacity was sufficient to handle normal student population with 2 watts per square foot of lighting and 15 cfm of outside air per person. Inside design was set at 75°F (20°C) DB with 55% RH and outside at 92°F (35°C) DB and 78°F (25°C) WB.

Robert B. Stotz, P.E., is a consulting engineer and president of R.B. Stotz Professional Engineers, Inc., Panama City, FL. Robert L. Hanson is an engineering mechanics graduate student at Virginia Polytechnical Institute, Blacksburg.

From an analysis of the classroom load calculations, it was obvious that all occupied classrooms would require some cooling of about 40°F (5°C) ambient; even below that, the heating requirements could be handled with ventilation air heated by water in the condensing water temperature range. Thus, the only heating problem would be in bringing the empty classrooms up to occupancy temperature.

The 460-ton (1,618-kWh) maximum load was being served with chilled water with a 10°F (5°C) range controlled by three-way valves. By changing to two-way modulating valves (installing a bypass to ensure maintenance of sufficient flow) and changing the range to 16°F (10°C), the maximum flow rates and pumping power could be reduced by about 40%.

A change from absorption to centrifugal units would reduce both condensing water quantity and range. However, if a 20°F (10°C) range could be used instead of the normal 10°F (5°C), condenser water quantity could be reduced from 1,380 gpm (87 L/s) to 690 gpm (43.5 L/s). Further, the 20-degree range would favor using condenser water for heating and, if two chillers were piped in series with chilled water and condensing water in counterflow, the two chillers would operate at close to the same differential.

If the condensing water was to be used for heating and reheat, it should be in a closed loop and not go through an open cooling tower. With well water available at 68°F (20°C) to 72°F (22°C), a plate-and-frame heat exchanger could replace the usual cooling tower and provide condensing water at 75°F (24°C) instead of the usual 87°F (30°C) normal for a cooling tower in this area. This would allow the 20°F (10°C) range and simultaneously provide greater efficiency from the low-temperature chiller and save the difference in operating energy between the well pumps and the cooling tower fans.

Examination of the heating load indicated that warm-up heating requirements could be handled by the heat rejected from the high-temperature chiller, operating with leaving chilled water at 52°F (11°C) and with condenser water leaving at 100°F (38°C). This lower temperature water would require additional coil surface from the original installation but seemed to be practical if the cooling coils could be converted to heating coils during the warm-up period.

Domestic hot water would still require heating for use as the cafeteria wash water. But members of the athletic department felt they could operate with shower water in the 95°F (35°C) area.

Accordingly, it was decided that the best solution was to provide cooling with a more efficient system using chillers arranged in series counterflow. Pumping energy requirements would be reduced by changing the chilled-water range from 10°F (5°C) to 16°F (10°C) and the condensing water range from 10°F (5°C) to 20°F (11°C). Simultaneously, all control valves would be changed from three-way modulating to two-way modulating. Thus, only that water required by the instantaneous load would be delivered to each unit.

Heating would, in most periods, involve transferring recovered heat from areas requiring cooling to those requiring heating. Supplemental heat would be furnished by aquifer well water transferred through a plate-and-frame heat exchanger. Thus, the aquifer water would provide a heat source/sink by transferring heat to the chilled water during the heating cycle and absorbing heat from the condenser water during the cooling cycle. To successfully effect changes, three control schemes would be developed—normal cooling with reheat, normal heating, and early morning warm-up. Although all changes in water flow would be accomplished with automatic valves and could be programmed into the EMS, the authors preferred initiating the changes by selection buttons to allow for variables and to ensure complete shutdown between modes.

With the lead chiller supplying 44°F (7°C) leaving chilled water at a leaving condenser water temperature of 85°F (20°C) and the lag chiller operating at 52°F (11°C) leaving chilled-water temperature and 95°F (35°C) leaving condenser water temperature, the system produced 460 tons (1,618 kWh) at an EER of 20.79 or a coefficient of performance (COP) of 6.09. Therefore, using the accepted efficiency of 27% of delivered electrical power, the overall efficiency of the centrifugal system for cooling would be 27% × 6.09 or 164% versus the absorption system, where the boiler has an efficiency in the 90% to 95% range reduced to less than 60% with the absorber included. Similarly, on the heating cycle and using the lag chiller cooling from 60°F (15°C) to 52°F (11°C) and with a 100°F (38°C) leaving condenser water temperature, the COP was 7.62. The resulting efficiency of 27% × 7.62 or 205% again far exceeds the boiler efficiency of 90% to 95%.

CONTROL SEQUENCE SCHEMES

The next step was to develop a control sequence that would satisfy classroom variations in lighting and occupancy as well as outdoor variations in temperature and solar insolation. Control scheme A (Figure 1) was established for use whenever outside temperatures exceed 55°F (13°C).

Under control scheme A, both chillers are activated and return chilled water; temperature is the controlling medium. The lag chiller controls to provide leaving chilled water at 51°F (11°C). When chilled water returning from the system drops to 52°F (11°C), this chiller secures and the lead chiller carries the load to produce the constant leaving chilled-water temperature of 44°F (7°C). Closed-loop condenser water passes through the second side (treated-water side) of the plate-and-frame heat exchanger in counterflow with the aquifer (well) water. This aquifer water is controlled in quantity so that only enough well water is pumped to produce return condenser water to the

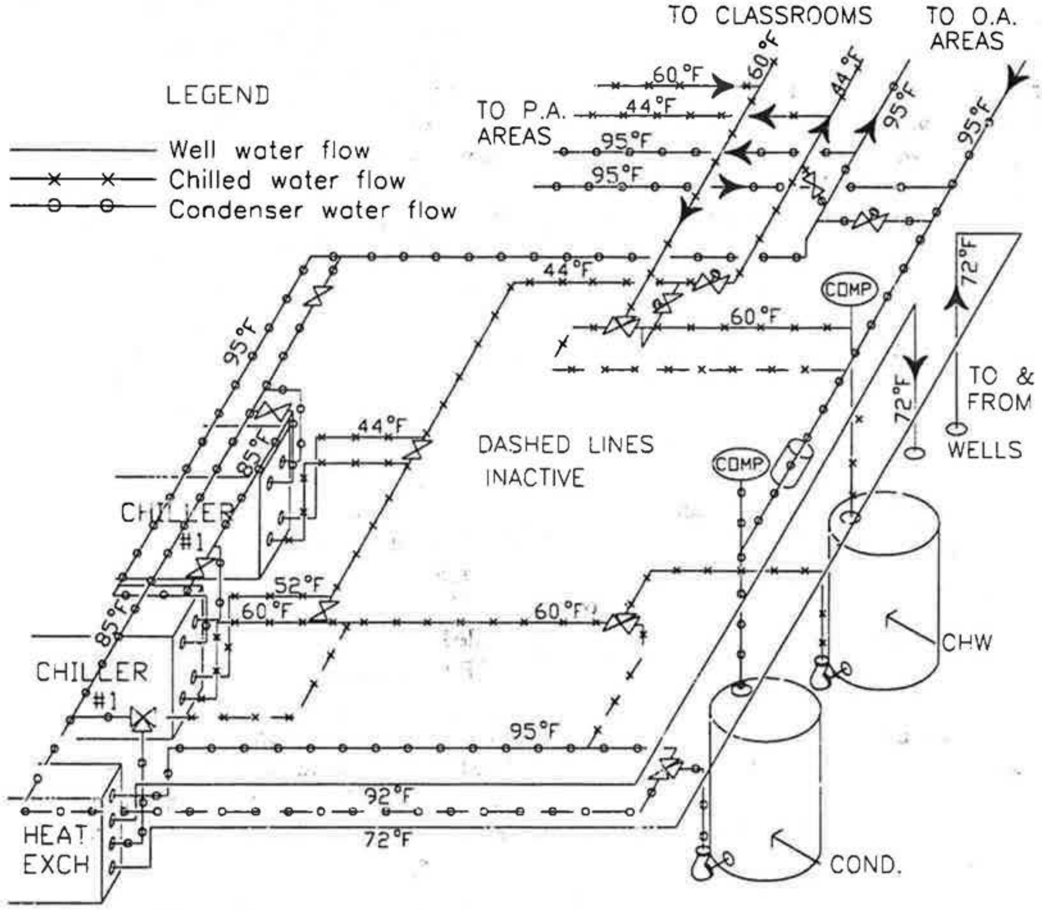

Figure 1 *Control scheme A—normal cooling with reheat.*

storage tank at 95°F (35°C). Thus, the need for a cooling tower is eliminated.

Classroom units are supplied with ambient outside air and chilled water only. All other air-handling systems are provided with chilled water and condenser water for reheat where humidity may be a problem.

Control scheme B (Figure 2) was established for use with outside temperatures ranging between 40°F (5°C) and 55°F (13°C). Under control scheme B, only the lag chiller is activated; the return condensing water temperature to the storage tank is the controlling medium. The treated-water side of the plate-and-frame heat exchanger carries closed-loop chilled water, and the aquifer water is controlled so as to only supplement the returning chilled water with enough heat to produce a leaving condenser water temperature in the 95°F (35°C) to 100°F (38°C) range. Thus the aquifer water becomes a heat source. Classroom units are supplied with outside air at 85°F (30°C) and chilled water. All other air-handling units are provided with both chilled and condenser water.

Control scheme C (Figure 3) was established for early morning warm-up and for days when the outdoor temperature was not expected to exceed 40°F (5°C). Under control scheme C, again only the lag chiller is activated, with return water to the storage tank being the controlling medium. The treated-water side of the heat exchanger carries the closed-loop chilled water, and the aquifer water is controlled to provide only enough heat to maintain the leaving condenser water temperature between 95°F (35°C) and 100°F (38°C). In this mode, classroom units are provided with outside air at 60°F (15°C) and condenser water for heating.

Note that a single plate-and-frame heat exchanger with aquifer water at 68°F (20°C) to 72°F (22°C) passing through the "treating medium" side has been used for all control schemes. Some system improvement might be obtained by using separate heat exchangers for the condensing water and the chilled water so that true counterflow is obtained in each. However, both system cost and well water use would increase if this were done. With a single heat exchanger, careful examination of heating and cooling loads is required to determine the correct changeover between schemes A and B and between schemes B and C.

Under all control schemes, outside air is not introduced during the warm-up or cool-down period. General domestic hot water is maintained by circulation through condenser water storage. A supplemental booster hot water heater is activated from 7 a.m. until 1:30 p.m. to provide the necessary increase for cafeteria wash water. Heating of the swimming pool water is accomplished by manually switching to substitute swimming pool water for well water in the plate-and-frame heat exchanger. This can be ar-

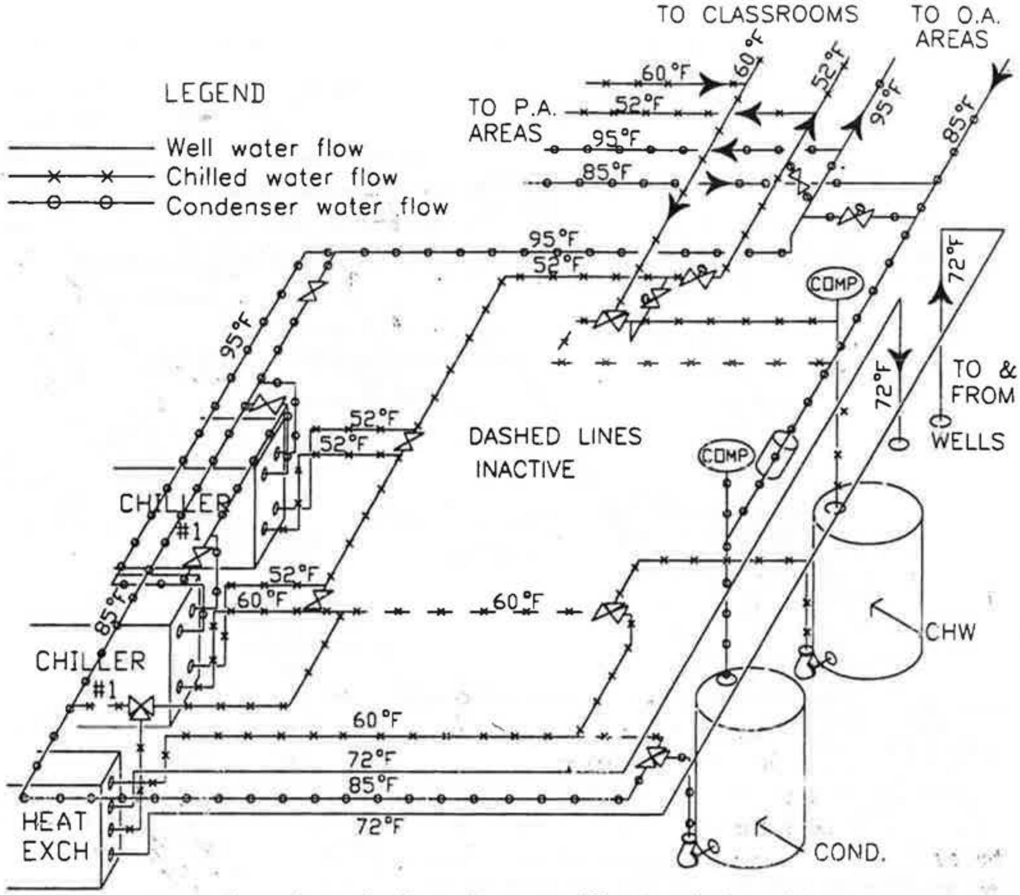

Figure 2 *Control scheme B—normal heating, balanced loads.*

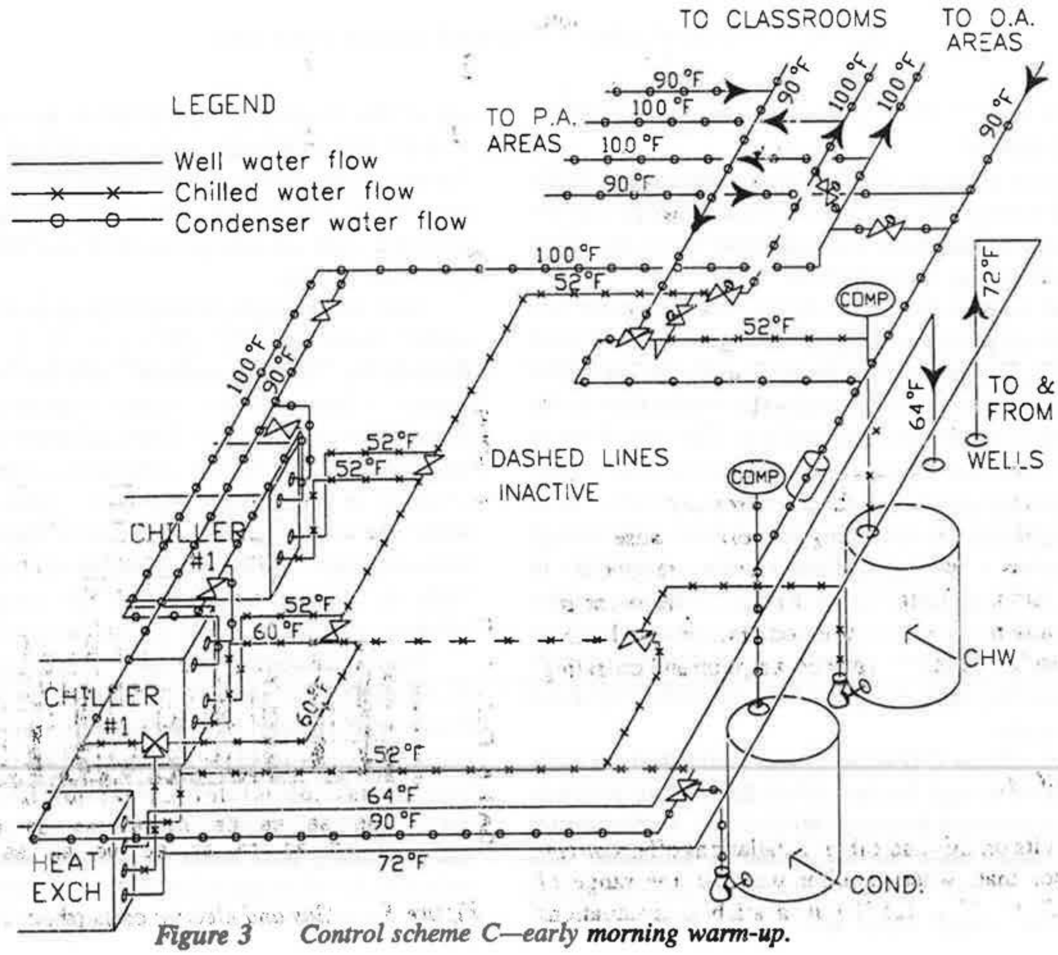

Figure 3 *Control scheme C—early morning warm-up.*

ranged only with the system in the cooling mode but does operate satisfactorily in the late spring and early fall periods (April 15 to November 15). A separate plate-and-frame heat exchanger, piped in parallel with the well water unit, should be used for this to be a reliable alternative.

RESULTS

The high school operated with its renovated HVAC system for the first time during the 1987-88 school year. The new system reduced annual energy consumption from 166,000 Btu (48,638 W-h) per square foot to 63,400 Btu (18,576 W-h) per square foot. Additionally, all classrooms were maintained in a comfortable temperature range.

Figures 4 and 5 offer a comparison of the results obtained with an energy management system and with a system design based on heat recovery and the use of aquifer wells instead of cooling towers and boilers. The original system was operated in the 1980-81 school year at a total utilities cost of $202,032 or $1.15 per square foot. In school year 1986-87, the year before the system change, by eliminating swimming pool heating and the installation of an energy management system, this cost had been reduced to $180,175 or $1.03 per square foot. School year 1987-88 was the first year of operation with the new system, and heating of the swimming pool was not included. The total cost was $115,642 or $0.66 per square foot. Finally, by adding in heating of the pool from April 15 to November 15, the 1988-89 figures reflect a total utilities cost of $125,171 or $0.72 per square foot.

Figure 6 compares monthly costs of utilities between 1986-87 and 1987-88. Gas rates averaged $0.37 per therm in this period. Electric rates averaged $0.051 per kWh including demand charges and increased fuel fees.

Study of the first three years reveals similar savings in energy for each year, but with a gradual increase in energy cost. Comparing these three years with the last year with the original system, savings were more than $180,000. The total cost of the conversion was $266,912, which included the cost of drilling two six-inch supply wells to a depth of 650 feet and two six-inch injection wells to a depth of 350 feet. A similar installation using screw chillers would have cost less than the centrifugal chillers. However, the higher COP with the centrifugal chillers provided a shorter payback period even with the higher cost. Because of the tonnage required, reciprocating chillers were not considered for this installation. However, both reciprocating and screw chillers have been used on smaller installations with good results in both energy consumption and operating cost.

CONCLUSIONS

The results on this and other installations offer convincing evidence that, where aquifer water in the range of 50°F (10°C) to 77°F (25°C) is available, installations

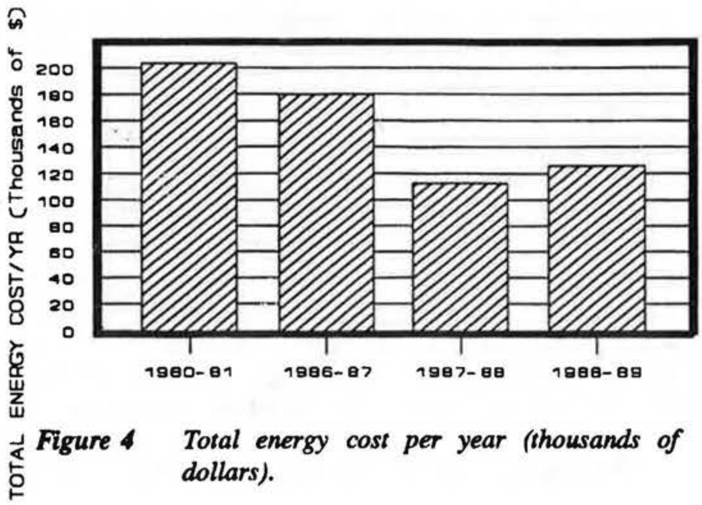

Figure 4 *Total energy cost per year (thousands of dollars).*

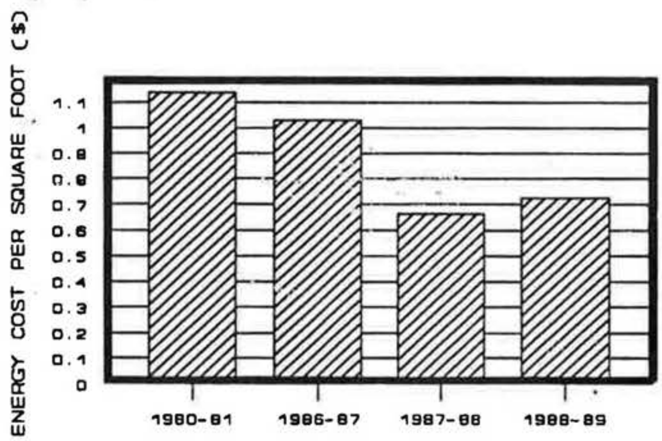

Figure 5 *Energy cost per square foot ($).*

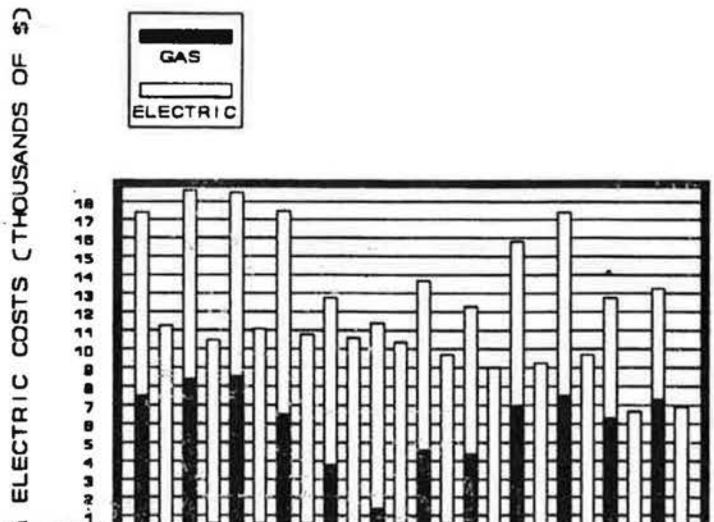

Figure 6 *Gas and electric costs (thousands of dollars).*

using aquifer wells will save energy because of the high COPs produced with chillers operating in the series-counterflow arrangement. The COP will be highest with centrifugal chillers, a little lower with screw machines, and still lower with reciprocating chillers. But with any of these, the combination of COP times the overall efficiency of electrical power generation and distribution exceeds that of any other heating means. The lower the aquifer temperature, the higher the COP during the cooling cycle but it will also be lower during the heating cycle. Thus, the combination with heat recovery on high internal load applications will benefit the energy efficiency of the system. From an economic standpoint, savings in operations costs will depend upon variation in utility rates.

For those applications requiring humidity control, such as libraries, museums, and computer installations, these systems provide warm water for reheat purposes with no additional expenditure of energy. In most new installations, variable-air-volume (VAV) units with coils in the reheat position can be used to control final area heat and humidity as required.

An added plus is the environmental benefit produced. With no boiler or furnace involved, there are no products of combustion released to the atmosphere during the heating cycle. By reinjection of the water into wells, the atmospheric heat pollution that would occur with either a cooling tower or air-cooled condensers is avoided. Thus, systems of this design will save energy and atmospheric pollution and in most cases will also save operating and maintenance costs.