ELECTRIC, GAS, AND ELECTRIC/GAS ENERGY OPTIONS FOR COLD-AIR HVAC SYSTEMS

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ABSTRACT

An important aspect of the design of cost-effective HVAC systems today is (a) sensitivity to the cost impact of the interplay of utility demand charges, time-of-day rates, gas rates, and gas/electric utility incentive programs vis-a-vis HVAC system options; and (b) familiarity with the range of electric, gas, and electric/gas system options available to take advantage of the cost opportunities and avoid the cost penalties in specific utility situations. When these factors are fully evaluated and incorporated into HVAC design for medium-to-large buildings, it is possible to structure system arrangements that reduce energy operating costs very significantly compared to conventional all-air VAV systems and also to all-air VAV ice thermal storage systems, at a comparable first cost. To contribute to the evaluation of these factors, this paper does the following:

1. Describes three cold air-water HVAC systems (electric, gas, electric/gas) developed by the author for office buildings and shopping centers to (a) reduce the first cost and the energy cost of cold air systems, (b) provide cold-air system options suitable for a wide range of gas/electric rate structures, and (c) provide cold-air systems that do not overdry the air as do cold all-air systems in many cases.
2. Presents a study based on a 159,000 ft^2 office building in northern New Jersey which compares five HVAC systems (the three cold air-water systems, a cold all-air/partial ice system, and a conventional 55°F all-air VAV system) in terms of peak refrigeration requirement, primary air distribution, fan demand, peak demand (summer/winter), annual energy usage (on/off-peak), gas input, annual demand cost, annual energy usage cost, total annual energy cost, and system first cost.
3. Shows the impact of various utility rate structures on the energy cost of the five HVAC systems being compared.

INTRODUCTION

Electric utility rate strategies and incentive programs to reduce peak demand have induced a growing number of building owners to install ice thermal storage as a source of cooling. The use of ice storage has led, in turn, to increased use of cold air systems (primarily 40°F all-air VAV systems) to accomplish the following: reduce the quantity of air required to handle the cooling load and, as a consequence, reduce the size and cost of the air distribution system and the net cost of ice thermal storage HVAC systems.

However, the benefits of cold-air distribution are not limited to all-air HVAC systems, thermal storage HVAC systems, or electric HVAC systems. The applicability of cold air systems is much broader than has been recognized.

Cold air-water HVAC systems have been developed by the author in order to expand the cost-effective applicability of cold-air systems in several ways: (1) by reducing the electric demand and energy cost as well as the construction cost of cold-air sysems, (2) by providing gas as well as electric cold-air system options to take advantage of a wide range of utility rate situations, and (3) by resolving the problem of overdry air that results in many buildings from 40°F all-air systems.

The overdry-air problem stems from a characteristic of cold all-air HVAC systems that is energy wasteful and that particularly limits their applicability in humid areas. In 40°F all-air systems, the quantity of cold air distributed is established by the sensible cooling requirement. The result, in a building with normal office occupancy of one person per 100 ft^2 and a sensible load between 20 and 30 Btu/ft^2, is a low relative humidity of 30% to 35% RH at a design dry bulb temperature of 75°F (as compared with the usual design condition of 50% RH at 75°F). The unusually large difference between indoor and outdoor absolute humidity and vapor pressure increases moisture migration through the building envelope into the conditioned space; this can cause a significant increase in the refrigeration load and thus in the size of the ice plant required in 40°F all-air HVAC systems.

Cold Air-Water System Functional Overview

Cold air-water systems resolve the dry-air problem and gain design flexibility by separating dehumidification and sensible cooling. The systems distribute a small quantity of 40°F primary air -- which handles all of a building's dehumidification and up to half of the sensible cooling at design conditions -- and provide the balance of sensible cooling via fan-induction terminal coils that circulate 53° to 58°F chilled water.

By separating dehumidification and sensible cooling -- dehumidifying in the central air handler and cooling primarily at terminal coils -- the air-water systems make it possible to reduce the quantity of 40°F air, raise the temperature of secondary sensible cooling at terminals, increase sensible cooling at terminals without depressing relative humidity (there is no condensation at terminals), and select from a variety of cooling methods and energy sources to produce the cold primary air on the one hand, and the higher-temperature chilled water for sensible cooling on the other.

Cold Air-Water System Benefits

As a result, the 40°F air-water systems, which were developed for shopping centers and medium-to-large office buildings, provide the following benefits:

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1. Cold air-water systems distribute one-third to one-half as much primary air as cold all-air systems, thereby reducing fan energy and the size and cost of the primary air handler and ductwork.
2. Cold air-water systems produce a relative humidity of 38% to 50% RH, given normal office occupancy, as compared with 30% to 35% RH in the cold all-air systems. Thus the air-water systems avoid the all-air system problem, which is particularly acute in humid areas, of excessive moisture migration into the space from the outside and a substantial increase in the refrigeration load and in the size of the ice plant.
3. Annual energy operating cost for the 40°F air-water systems is 25% to 35% less than for the 40°F all-air system and 35% to 45% less than in a conventional all-air VAV system, based on a comparative study described subsequently that was done for a six-story office building in northern New Jersey.
4. The first cost/construction cost of cold air-water systems is less than that of cold all-air systems, due to the savings in primary ducts and air handler.
5. Depending on the utility rate structure in a specific location, cold air-water systems may select from the following options to produce the 40°F primary air and the chilled water for 53° to 58°F secondary cooling:

   (a) 40°F primary air:

   - ice thermal storage
   - efficient refrigeration such as liquid overfeed
   - desiccant-dried air that is air-washed to drop the temperature to 40°F

   (b) Chilled water for 53° to 58°F secondary cooling:

   - vapor compression refrigeration with a chiller barrel (in systems where the compressor makes ice at night)
   - gas absorption machine
   - gas engine-driven chiller (where the engine heat can be used, as for desiccant regeneration in a desiccant dehumidification system)

This paper presents three cold air-water system options, followed by the results of a study comparing the air-water systems with a 40°F all-air system and a conventional 55°F all-air VAV system. The five systems are compared in terms of energy demand, energy usage, and energy cost, as well as first cost. Air distribution is compared in terms of primary fan capacity, maximum primary air distribution (interior, perimeter), and fan demand (both primary and secondary fans). The comparative study is based on a 159,000 ft^2, six-story office building with a maximum cooling load of 510 tons.

40°F AIR-WATER SYSTEM OPTIONS/UTILITY OPTIONS

In the three cold air-water systems presented here, cooling is energized by electricity in one (half off-peak), by a combination of gas and electricity in another, and entirely by gas in the third (excluding pumps and fans). All three incorporate gas heating.

All are VAV systems that distribute 0.16 to 0.27 cfm/ft^2 of 40°F primary air from the central air handler to fan-induction coil terminals (FICUs). In each case the cold primary air provides all dehumidification and 35% to 50% of the sensible cooling. There is no condensation at terminals.

Chilled water for terminal coils is distributed at 53° to 58°F from the central plant via integrated sprinkler piping. FICUs, illustrated in Figure 1, sensibly cool recirculated air as required, mix it with the 40°F ventilation air, and supply the mixed air at a constant volume.

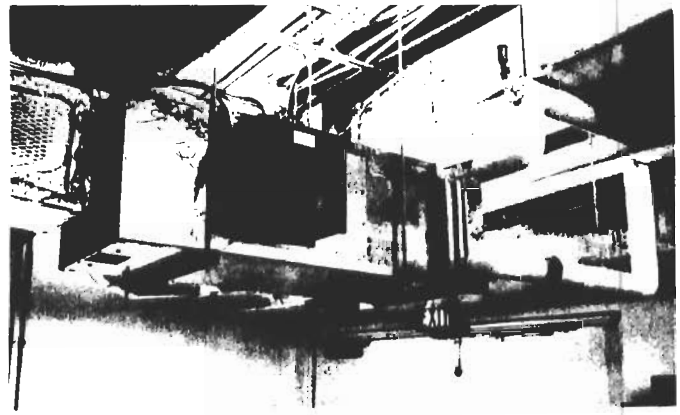

Figure 1 Fan-induction coil terminal (FICU) operating in cold air-water VAV system with partial ice thermal storage at the author's Herndon, Virginia offices. FICU mixes a small, variable quantity of 40°F primary air with recirculated air that is sensibly cooled at terminal coils, as required, with 53° to 58°F chilled water.

<u>1. Electric Cooling: Ice Dehumidification (40°F)/Same Compressor + Chiller Barrel Sensible Cooling (55°F)</u>

In this system, shown schematically in Figure 2 and psychrometrically in Figure 3, compressors produce ice during off-peak hours. During the day melting ice produces 34°F chilled water that cools primary air to 40°F.

One of the same compressors operates at a higher evaporative temperature during the day, in conjunction with a chiller barrel, to produce 55°F chilled water for sensible cooling at FICU terminals.

<u>2. Electric/Gas Cooling: Efficient Vapor Compression Dehumidification (40°F)/Absorption Machine Sensible Cooling (55°F)</u>

Figure 4 shows this system schematically. The psychrometric process is the same as that shown in Figure 3. Efficient vapor compression refrigeration (in this case liquid overfeed) produces 40°F primary air.

A direct-fired gas absorption chiller-heater produces 55°F chilled water for FICU sensible cooling -- as well as hot water for space heating in winter.

<u>3. Gas Cooling: 2-Wheel Desiccant Dehumidification (Air-Washed - 40°F)/Engine-Driven Chiller Sensible Cooling (55°F)</u>

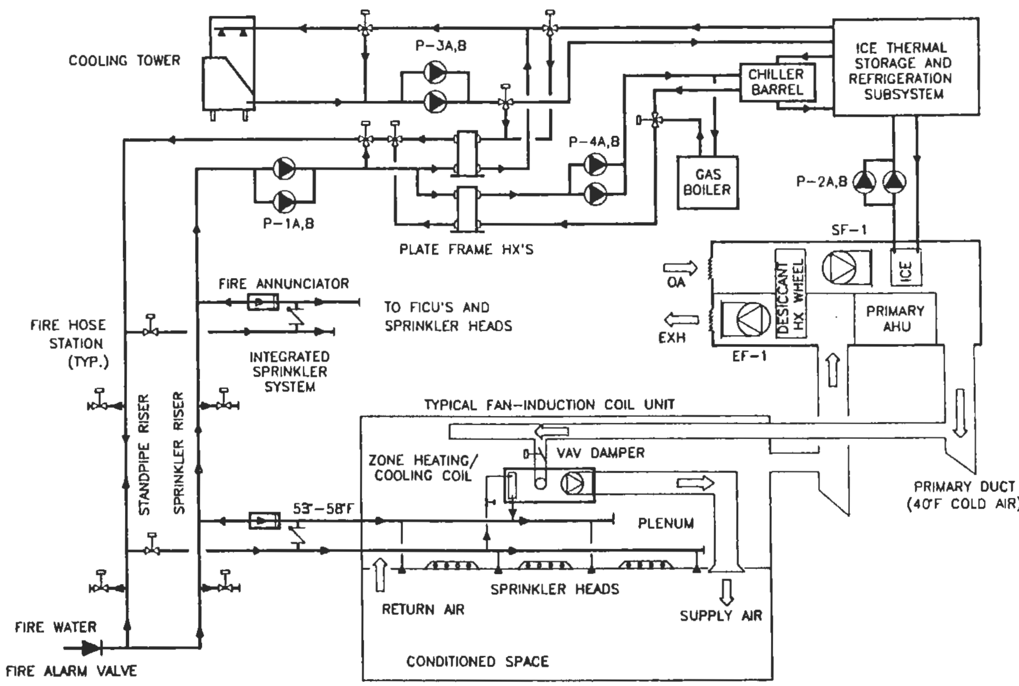

Figure 2 Cold air-water VAV system: Ice thermal storage provides 100% of dehumidification and 35% to 50% of sensible cooling via a small quantity of 40°F primary air; same compressor operates during the day at a higher evaporative temperature, in conjunction with a chiller barrel, to provide 53° to 58°F chilled water for sensible cooling at fan-induction coil terminals (FICUs); gas heat.

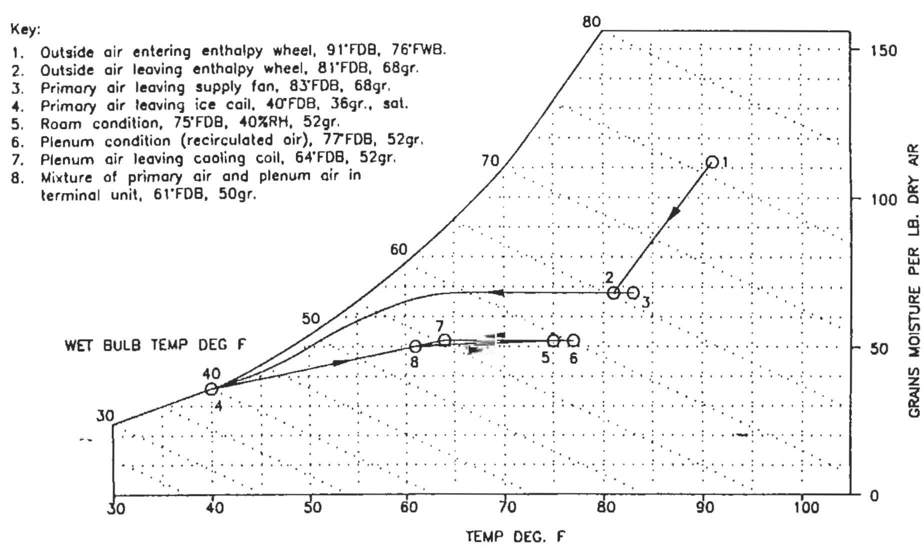

Figure 3 Air conditioning process for Figure 2 cold air-water HVAC system and Figure 4 cold air-water HVAC system.

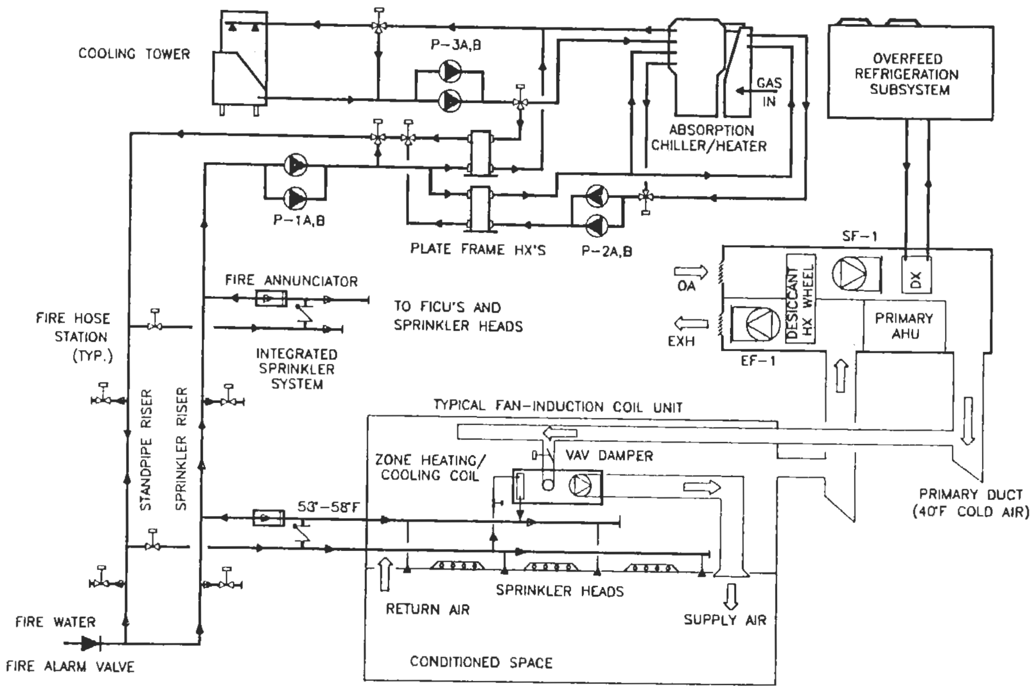

Figure 4 Cold air-water VAV system: Efficient vapor compression refrigeration (e.g., liquid overfeed) dehumidifies the small quantity of 40°F primary air; a direct-fired gas absorption chiller-heater provides the balance of sensible cooling via 53° to 58°F chilled water, which is distributed from the central plant through integrated sprinkler piping to fan-induction coil terminals (FICUs).

In the gas cooling system, as shown schematically in Figure 5 and psychrometrically in Figure 6, incoming outside air is dried in a two-stage desiccant conditioner. The dehumidified ventilation air is aftercooled to 55°F from a gas engine-driven chiller, then saturated with nonrefrigerated water in an air washer to drop the temperature to 40°F. (Nonrefrigerated water circulates continuously in the air washer.) The cold primary air is distributed in variable volume (0.16 to 0.27 cfm/ft^2) to FICU terminals, as in the other two air-water systems described here.

Chilled water for secondary cooling (55°F) is produced by the engine-driven chiller. Engine heat regenerates the desiccant, with backup from a gas boiler. Engine heat contributes to winter space heating also, when the engine is operating to drive the chiller.

Desiccant Subsystem. The desiccant system includes two rotating desiccant-impregnated wheels. The first wheel is an enthalpy exchanger that handles 30% to 50% of the building's dehumidification without the need for external heat to regenerate the desiccant. This wheel absorbs both heat and moisture from the incoming airstream and transfers them to the drier exhaust airstream.

As a result, the second wheel, which completes the dehumidification process, has a lighter task of moisture removal and requires 30% to 50% less external heat for desiccant regeneration than a one-stage desiccant system. (The thermal coefficient of performance, COP, of the two-wheel regeneration process is 1.5 to 2 at design conditions.)

Dehumidification and regeneration occur as follows. Each desiccant-impregnated wheel rotates through two separate airstreams: the moist, incoming outside airstream and the regeneration airstream that is exhausted to the outside. As the wheels rotate, the desiccant absorbs moisture from the outside airstream and then gives it up to the regeneration airstream. After leaving the first stage and before entering the second-stage dehumidifier, the incoming ventilation air is precooled to 55°F from the engine-driven chiller.

In first-stage regeneration, the outgoing (regeneration) air is the relatively dry, building relief air. The second-stage regeneration airstream may be either relief air or outside air; the air is heated (from the gas engine) prior to flowing through the regeneration chamber. The desiccant in the second-stage dehumidifier is more concentrated than that in the first-stage wheel.

The desiccant used in the enthalpy exchanger can be a silica gel or molecular sieve desiccant; in the second-stage wheel, the desiccant can be a silica gel, molecular sieve, or lithium chloride desiccant, or a combination of two of them.

A two-wheel desiccant system offers another performance advantage over a one-wheel system, in addition to its greater thermal efficiency: Rain or a very humid outside condition does not cause supersaturation in the second wheel; in a one-wheel system, such saturation can significantly impair performance and require a much higher-temperature regeneration heat.

HVAC SYSTEMS COMPARATIVE ENERGY/COST STUDY

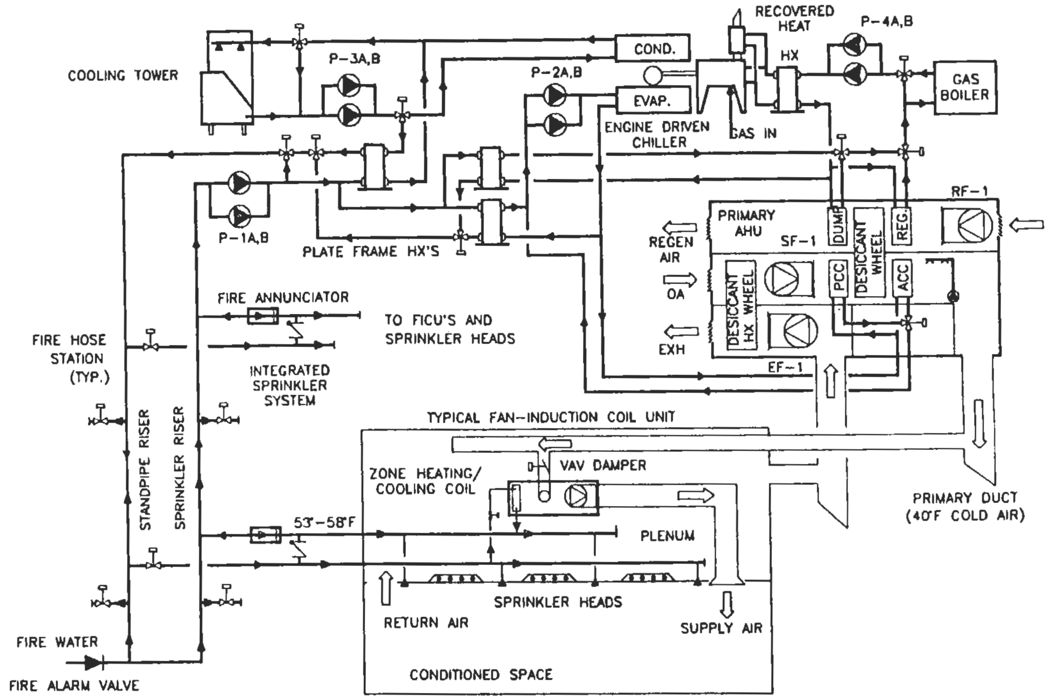

Figure 5 Cold air-water VAV system: 2-stage desiccant dehumidifier dries the small quantity of primary air, which is then aftercooled to 55°F from a gas engine-driven chiller and saturated with nonrefrigerated water to drop the temperature to 40°F; the engine chiller produces 53° to 58°F chilled water for secondary sensible cooling at fan-induction coil terminals (FICUs) as well as cogenerated heat for desiccant regeneration; gas boiler provides backup heat.

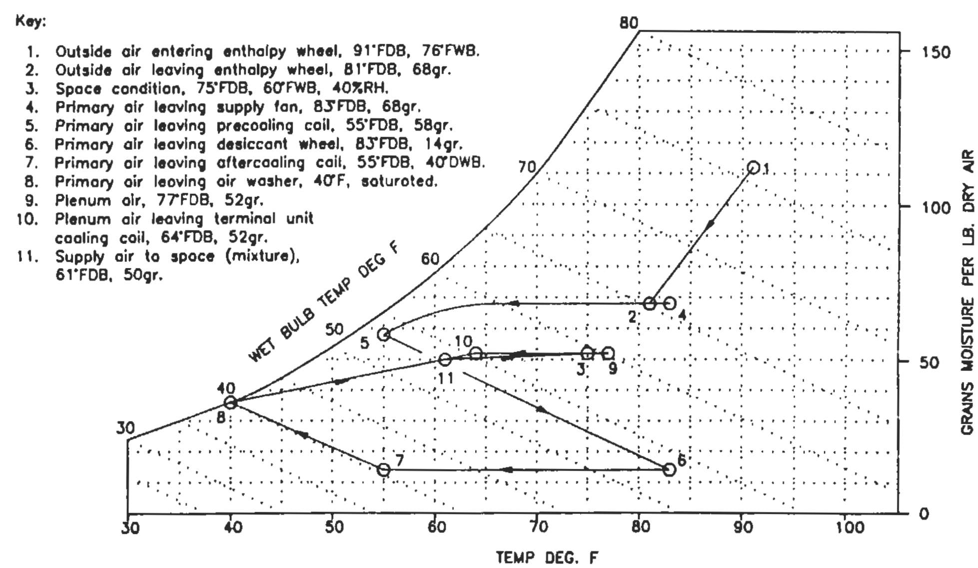

Figure 6 Air conditioning process for Figure 5 cold air-water HVAC system with desiccant dehumidification, air washer to drop temperature to 40°F, and higher temperature secondary sensible cooling at FICU terminals.

An HVAC system energy/cost study based on a six-story, 159,000 ft^2 office building in northern New Jersey compared the following five HVAC systems:

- o System 1 (Electric cooling) - Conventional all-air VAV system (55°F) with vapor compression refrigeration/VAV terminal boxes with electric reheat.
- o System 2 (Electric cooling) - Cold all-air VAV system (40°F) with partial ice thermal storage/ Fan-induction terminals (FIUs) with electric reheat.
- o System 3 (Electric cooling) - Cold air-water VAV system/Ice thermal storage dehumidification via 40°F primary air/Same compressor and chiller barrel for 55°F sensible cooling at fan-induction coil terminals (FICUs)/Gas heat.
- o System 4 (Electric/gas cooling) - Cold air-water VAV system/Vapor compression (liquid overfeed) dehumidification via 40°F primary air/Absorption chiller-heater for 55°F sensible cooling at FICUs -- and winter heating.
- o System 5 (Gas cooling) - Cold air-water VAV system/2-wheel desiccant dehumidification (air-washed to 40°F)/Engine-driven chiller for 55°F sensible cooling and cogenerated heat for desiccant regeneration/Gas boiler backup.

The office building studied has the following physical characteristics, design conditions, and peak loads:

General Building Characteristics

- o Construction - Six-story steel structure with concrete slab, double-glazed perimeter windows, built-up roof deck, suspended acoustic tile ceiling.
- o Gross floor area = 159,000 ft^2
- o Net leasable area = 143,000 ft^2
- o Perimeter zone area = 67,000 ft^2
- o Interior zone area = 76,000 ft^2
- o Common lobbies/corridors = 8,000 ft^2
- o Core area/support services = 8,000 ft^2

Estimate of Peak Cooling and Heating

- o Summer outside design conditions
  - Temperature 91 FDB
  - Mean daily range 21 FDB
  - Relative humidity 43%
  - Summer sunshine 65%
- o Winter outside design conditions
  - Temperature 10 FDB
  - Winter sunshine 45%
- o Summer/Winter occupied temperature 75 FDB
  Winter unoccupied temperature 65 FDB
- o Ventilation rate 0.18 cfm/ft^2
- o Internal loads
  - Number of people 100 ft^2/person
  - Sensible heat per person 250 Btuh
  - Latent heat per person 245 Btuh
  - Average occupancy 90%
  - Lighting load 2 $watts/ft^2$
  - Equipment load 1.5 $watts/ft^2$

Peak Cooling Load Summary, tons

Load Item	Sensible	Latent	Total
Transmission	174	-	174
Internal	179	36	215
Ventilation	36	85	121
TOTAL TONS	389	121	510

Air Distribution/Fan Demand

Table 1 compares the systems in terms of peak refrigeration requirement, minimum primary (ventilation) air, maximum primary fan capacity, maximum primary air distribution, and fan demand (both primary and secondary fans). The primary fan capacity in the air-water systems is 75% less than in the conventional all-air system and 57% less than in the cold all-air system. Maximum primary air distribution in the air-water systems is

- 72% less in the interior and 77% less at the perimeter than in the conventional all-air system and
- 55% less in the interior and 60% less at the perimeter than in the cold all-air system.

As a result, the air-water systems provide a major reduction in the size and cost of the primary air handler and distribution ducts, as well as fan energy. Secondary fan demand is greater in the air-water systems; but the reduction in primary fan demand is so dramatic that the net reduction in fan demand for the air-water systems is 24% compared to the conventional all-air system and 30% compared to the cold all-air system.

Off-Peak: Not Always Lowest in System Energy Cost

Table 2 compares the five HVAC systems in terms of system energy demand, energy usage, and energy cost. Utility rates at the site are as follows:

Demand, summer - $10.76/KW
Demand, winter - $ 9.78/KW

Usage, on-peak - $0.0756/KWH
Usage, off-peak - $0.0597/KWH

Gas - $5.30/MMBTU

As indicated in Table 2, the three cold air-water systems are lowest in annual energy cost. System 4 and System 3 are lowest at $0.99/ft^2 and $1.00/ft^2, respectively. System 4 cooling is approximately half electric (on-peak), half gas absorption. System 3 cooling is all-electric -- approximately half on-peak, half off-peak (ice-making). Both the demand charges and the on-peak usage costs are comparable for the two systems. The principal differences are the much lower off-peak usage cost for System 4 plus its higher gas cost. A lower site gas rate would increase the net cost advantage of System 4, as shown in Table 3 ("A" rates).

Another key factor in the choice between System 3 and System 4 is the applicable time-of-day rates. At this site there is only a moderate difference between the on-peak and off-peak usage rates. Where there is a greater difference between the two rates, the ice storage system (air-water) gains the advantage, as shown in Table 3 under "B" and "D" rates.

TABLE 1

COMPARISON OF HVAC SYSTEM AIR DISTRIBUTION/FAN DEMAND

Building: 159,000 ft^2, 6-story office building, northern NJ
Building maximum cooling load: 510 tons

Refrig. Req't/ Air Distrib. Item/ Fan Capacity/ Fan Demand	SYSTEM 1	SYSTEM 2	SYSTEM 3	SYSTEM 4	SYSTEM 5
	Conventional all-air VAV (55°F)/VAV box, electric reheat	Cold all-air VAV (40°F)/ Partial Ice/ FIU	Cold air-water VAV: Ice dehumid. via 40°F prim.air/55°F sensible clg. at FICU coils/Gas heat	Cold air-water VAV: Vapor comp. dehum. via 40°F prim. air/ Gas absorption sensible clg. (55°F) via FICU coils	Cold air-water VAV: Desiccant dehum.prim. air (air-washed to 40°F)/ Gas engine chiller sensible clg. (55°F) via FICU coils
Peak refrigeration requirement, tons	521	370	142	419	389
Minimum primary (ventilation) air, cfm/ft^2	0.16	0.16	0.16	0.16	0.16
Maximum primary fan capacity, cfm/ft^2	1.09	0.63	0.27	0.27	0.27
Maximum primary air distribution, cfm/ft^2	Int. - 0.65 Per. - 1.57	Int. - 0.40 Per. - 0.90	Int. - 0.18 Per. - 0.36	Int. - 0.18 Per. - 0.36	Int. - 0.18 Per. - 0.36
Peak fan demand, KW (summer)					
Primary fan	132	82	33	33	33
Secondary fans	--	60	67	67	67
Total fan demand	132	142	100	100	100

TABLE 2

COMPARISON OF SYSTEM ENERGY DEMAND/USAGE/COST

(Building: 159,000 ft^2, 6-story office building, northern NJ)
(See note)

SYSTEM	ELECTRIC UTILITY DEMAND							ANNUAL ELECTRIC ENERGY USAGE					ANNUAL GAS INPUT		SYSTEM ANNUAL ENERGY COST
	Demand, Summer			Demand, Winter/Interm.			Annual Demand Cost	On-peak		Off-peak		Total Annual Usage			
	KW*		Summer Demand Cost	KW*		Winter/Int. Demand Cost									
	Day	Night		Day	Night			KWH	Cost	KWH	Cost	KWH/Cost	MMBTU	Cost	
System 1 All-air VAV (55°F) FIUs	792	0	$36,400	913	817	$34,600	$71,000	1,517,000	$114,700	1,038,200	62,000	2,555,000/ $176,700	0	0	$247,700 $1.56/$ft^2$
System 2 Cold all-air VAV (40°F)/ Partial ice/ FIU	428	242	$18,100	839	821	$22,400	$40,500	980,400	$ 74,100	1,687,200	$100,700	2,668,000/ $174,800	0	0	$215,300 $1.35/$ft^2$
System 3 Cold air-water VAV (40°F)/Partial ice/Gas heat/FICUs	359	257	$15,400	181	179	$19,700	$35,100	864,000	$65,300	790,000	$ 47,200	1,654,000/ $112,500	2068	$11,000	$158,600 $1.00/$ft^2$
System 4 Cold air-water VAV (40°F)/ Vapor comp.refrig.dehum./Gas absorp.sens.clg./FICUs	432	0	$18,600	209	70	$20,500	$39,100	898,200	$67,900	162,700	$ 9,700	1,061,000/ $ 77,600	7645	$40,500	$157,200 $0.99/$ft^2$
System 5 Cold air-water VAV/Desic.dehum (air-washed)/ engine chiller sens. clg/FICUs	290	0	$13,200	204	74	$19,200	$32,400	843,400	$63,800	176,700	$10,500	1,020,000/ $ 74,300	13,446	$71,300	$178,000 $1.12/$ft^2$

NOTE: Utility rates: Demand, summer - $10.76/KW; Demand, einter - $ 9.78/KW; Usage, on-peak - $0.0756/KWH; Usage, off-peak - $0.0597/KWH; Gas - $5.30/MMBTU

Building: 6-story steel structure with concrete slab, double-glazed perimeter windows, built-up roof deck, suspended acoustic tile ceiling, 159,000 ft^2.

*Peak demand at design temperature. Includes refrig. and water distribution (compressors, pumps) plus air distribution (primary and secondary fans).

Energy and First Cost Comparisons

Table 3 illustrates the impact on energy cost of applying several different utility rate structures to the five systems. As Table 3 shows, a cold air-water system is significantly lower in energy operating cost than Systems 1 and 2 (conventional 55°F all-air VAV system and cold all-air/partial ice system) under all utility rate scenarios.

The comparative energy cost and first cost of each of the air-water systems is as follows, based on the energy demand/usage characteristics given in Table 2 and the utility rate structures shown in Table 3:

1. System 3

(a) System 3 Energy Cost

- Of the five HVAC systems, System 3 is lowest in energy cost in 9 of the 15 rate scenarios, and within 1¢ to 3¢/ft2 of the lowest in three more cases.
- The System 3 energy cost is significantly lower than that of System 1 (40% to 43% lower in 10 of the 15 scenarios, 36% to 38% lower in five cases).
- The System 3 energy cost is lower than that of System 2 by 25% to 29% in 8 of the 15 scenarios, and lower by 14% to 23% in 7 cases.

(b) System 3 First Cost

In many locations, including New Jersey, new thermal storage systems are eligible for a one-time utility grant based on avoided KW demand. With such a grant of $1.25/ft2, the first cost of System 3 is $8.25 ($9.50-$1.25).

The System 3 first cost, with the subsidy, is 25¢/ft2 less than that of System 1, and 25¢/ft2 less than that of System 2 with a similar subsidy.

2. System 4

(a) System 4 Energy Cost

- Of the five HVAC systems, System 4 is lowest in energy cost or within 1¢/ft2 of the lowest in 8 of the 15 rate scenarios.
- Compared to System 1, the System 4 energy cost is lower by 34% to 44% in 13 of the 15 scenarios.

TABLE 3

IMPACT OF VARIOUS UTILITY RATES ON ENERGY COST OF SELECTED HVAC SYSTEMS

159,000 ft2, 6-story Office Building, Northern NJ
Building Maximum Cooling Load: 510 Tons
(Based on energy demand/usage characteristics in Table 2)

		ANNUAL ENERGY COST, $/ft2				
		All-Air HVAC Systems		Cold Air-Water HVAC Systems		
ELECTRIC RATES (KW, KWH)/ FIRST COST	GAS RATES (3 shown with each electric rate structure), $/MMBTU	System 1 Conventional All-air VAV (55°F)/VAV box, reheat	System 2 Cold all-air VAV (40°F)/ Partial ice/ FIUs	System 3 Cold air-water VAV: Ice dehum./ 55°F sens.clg./ Gas heat/FICUs	System 4 Cold air-water VAV: Vapor comp. refrig.dehum (40°F)/ Absorption sens.clg-htg/FICUs	System 5 Cold air-water VAV: Desic.dehum (air-washed,40°F) /Eng.chiller sens. clg-htg/FICUs
A. Demand, summer $10.76; Demand, winter $ 9.78; On-Peak KW $0.0756; Off-Peak KW $0.0597	$5.30 (actual)	$1.56	$1.36	$1.00*	$0.99*	$1.12
	$3.50	$1.56	$1.36	$0.98	$0.90*	$0.97
	$3.00	$1.56	$1.36	$0.97	$0.88*	$0.93
B. Demand, summer $10.76; Demand, winter $ 9.78; On-Peak KW $0.0756; Off-Peak KW $0.0200	$5.30	$1.30	$0.94	$0.80*	$0.95	$1.08
	$3.50	$1.30	$0.94	$0.78*	$0.86	$0.93
	$3.00	$1.30	$0.94	$0.77*	$0.84	$0.88
C. Demand, summer $18.00; Demand, winter $17.00; On-Peak KW $0.0756; Off-Peak KW $0.0597	$5.30	$1.88	$1.54	$1.16*	$1.17*	$1.27
	$3.50	$1.88	$1.54	$1.13	$1.08*	$1.12
	$3.00	$1.88	$1.54	$1.12	$1.05*	$1.07*
D. Demand, summer $18.00; Demand, winter $17.00; On-Peak KW $0.0756; Off-Peak KW $0.0200	$5.30	$1.62	$1.12	$0.96*	$1.12	$1.22
	$3.50	$1.62	$1.12	$0.94*	$1.04	$1.07
	$3.00	$1.62	$1.12	$0.93*	$1.01	$1.03
E. Demand, summer $16.90; Demand, winter $ 2.90; On-Peak KW $0.1072; Off-Peak KW $0.0501	$5.30	$1.78	$1.42	$1.09*	$1.14	$1.24
	$3.50	$1.78	$1.42	$1.07*	$1.05*	$1.09
	$3.00	$1.78	$1.42	$1.06	$1.03*	$1.05*
HVAC SYSTEM FIRST COST,** $/ft2		$8.50	$9.75 -1.25 subsidy $8.50	$9.50 -1.25 subsidy $8.25	$9.00	$9.50 -1.25 subsidy $8.25

*Lowest energy cost (or within 2¢/ft2 of the lowest) within each rate scenario.

**Many electric utilities provide a one-time grant for avoided KW demand when thermal storage is included in a system. Many gas utilities provide an incentive grant for installation of a gas cooling system. First cost includes piping, ductwork, primary air handler, terminals, automatic temp. controls, refrigeration and, where applicable, ice thermal storage equip., gas engine, desiccant system, and gas boiler.

- Compared to System 2, the System 4 energy cost is lower by 25% to 35% in 7 of the 15 rate scenarios, and lower by 20% and 24% in two more cases.

(b) System 4 First Cost/Payback

- System 4 first cost is approximately $9.00/ft^2.
- Compared to System 1, the System 4 payback is less than one year in 11 of the 15 rate scenarios, and under 18 months in the other four cases.
- Compared to System 2 (when System 2 receives a utility subsidy of $1.25/ft^2), the System 4 payback is less than 18 months in 8 rate scenarios, and 22 months in another.

3. System 5

(a) System 5 Energy Cost

- Of the five HVAC systems compared, System 5's energy cost is comparable to the lowest (within 2¢ or 3¢/ft^2/yr) in two rate scenarios with low gas rates. In each case, System 5's first cost (with a $1.25/ft^2 utility grant) is significantly lower than that of the system with a comparable energy cost.
- Compared to System 1, the System 5 annual energy cost is 40% to 43% less in four rate scenarios, 30% to 39% less in seven cases, and 25% to 28% less in three cases.
- Compared to System 2, the System 5 annual energy cost is 26% to 32% less in five rate scenarios, and 18% to 23% less in three cases.

(b) System 5 First Cost

The System 5 first cost, with a utility incentive grant of $1.25/ft^2, is lower than that of Systems 1, 2, and 4, and equal to that of System 3.

CONCLUSION

Flexible cold air-water HVAC systems have been developed that adapt to a range of cooling techniques and energy sources, depending on the applicable utility rate structure and incentives. The annual energy operating cost of the cold air-water systems is 25% to 35% less than that of a cold all-air VAV system with ice storage, and 35% to 45% less than that of a conventional all-air VAV system. For shopping centers and medium-to-large office buildings, the first cost of the cold air-water systems is (a) lower than that of cold all-air systems with ice storage and (b) *comparable* to that of conventional 55°F all-air VAV systems, in the many locations where utility incentive grants are available.

The cost benefits stem from the basic cold air-water system configuration: The systems distribute a very small quantity of 40°F primary air (one-third to one-half as much as in cold all-air systems), thereby reducing the size and cost of the primary air handler and distribution ducts. The 40°F primary air provides all of the required dehumidification and 35% to 50% of a building's sensible cooling. The balance of sensible cooling is provided by chilled water at an energy efficient elevated temperature (53° to 58°F), which is distributed from the central plant via integrated sprinkler piping to coils in fan-induction terminals.

This configuration permits the use of different energy sources to provide dehumidification (cold primary air) on the one hand, and higher-temperature sensible cooling on the other. The choices are determined based on the most cost-effective combination at the site vis-a-vis the utility rates and available incentive grants. For example, two practical configurations of the cold air-water systems that provide an all-electric and a gas/electric option are (a) off-peak ice-making to provide the dehumidified 40°F primary air/ use of the same compressor operating during the day at a higher evaporative temperature, in conjunction with a chiller barrel, to provide 53° to 58°F chilled water for sensible cooling/ gas heat, and (b) efficient vapor compression refrigeration (such as liquid overfeed) to provide the small quantity of 40°F primary air/ use of a gas absorption chiller-heater to provide 53° to 58°F chilled water for sensible cooling as well as hot water for space heating in winter.

Both of these cold air-water systems are lower in energy cost than all-air HVAC systems, including all-air VAV ice storage systems. The two air-water systems have comparable demand charges and on-peak energy usage costs. The choice between them depends on three cost factors: the incentive grants available from utilities at the location, and two utility rate factors -- the gas rates and the spread between the day and night time-of-day rates. Low gas rates can tip the scale toward the electric/gas system with the absorption machine; a wide spread between day and night electric usage rates favors the all-electric system that uses ice thermal storage for central plant dehumidification.

Comprehensive evaluation of the cost impact of these utility rate/incentive factors on the full range of HVAC system options is an important aspect of HVAC design in today's competitive utility environment. It produces HVAC system arrangements that provide building owners and developers with significantly lower energy operating costs at little or no increase in first cost.