INTEGRATED ICE STORAGE/SPRINKLER HVAC SYSTEM SHARPLY CUTS ENERGY COSTS AND AIR-DISTRIBUTION FIRST COSTS

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ABSTRACT

Integrated ice thermal storage/sprinkler HVAC systems developed and applied by the author in several commercial applications shift a major portion of electric utility demand to cheaper off-peak hours, while also reducing significantly the first cost of distribution ductwork. Savings of up to 80% in primary duct and air handler costs (compared to a traditional all-air HVAC system) partially offset the first cost of ice storage equipment, which in turn permits a 50% reduction in utility energy costs for air conditioning in some facilities.

The basic ice storage/sprinkler HVAC system is described as well as optional subsystems, such as cogeneration, that are cost-effective under certain conditions. The system's design parameters, psychrometric process, and thermodynamic characteristics are presented and two installations are described, a 223,000-sq ft shopping center and a 150,000-sq ft, two-story retail store.

Reductions in the size and first cost of primary air-handling equipment and ductwork are achieved by distributing a small quantity (0.1 to 0.2 cfm/sq ft) of very dry, 40°F primary air. All dehumidification is handled by the ice-chilled primary air, which is distributed in variable volume, determined by the space dehumidification requirement, to fan-coil induction terminal units. The primary air is mixed with fan-induced room air in the terminals prior to distribution to the space at a constant volume. The fan-induction terminals contain cooling coils connected to the integrated sprinkler system, which circulates chilled water from the central plant through the coils when additional sensible cooling is required. This chilled water is at a thermodynamically efficient elevated temperature (58-68°F), since it handles sensible cooling load only.

INTRODUCTION

The energy-integrated ice storage HVAC system described here has been developed and applied by the author in several commercial installations in order to reduce the HVAC system life-cycle cost. Various cost-cutting techniques and subsystems are incorporated based on their contribution to (a) thermodynamic efficiency, to accomplish a task with less energy input; (b) shifting utility energy demand to off-peak hours, when lower rates apply; or (c) reducing first cost, so the savings can offset investment in ice storage equipment. The selection and integration of these techniques and subsystems is guided by an analysis of energy flows throughout a facility - the temperature levels as well as quantities of energy required to accomplish tasks - and by the utility rate structure. Performance criteria, never sacrificed, include precise humidity and temperature control and a constant, uniform air flow through occupied spaces to assure good ventilation and air quality.

In most parts of the western, southern and northeastern sections of the country electric utilities offer reduced rates during off-peak (night) hours. In many areas the penalty applied to peak daytime demand is quite severe. In Westminster, Colorado, for example, the utility demand charge is $11.84 per KW, while the usage charge is 2.1¢ per KWH. An ice thermal storage capacity enables a facility to use utility electricity at night to make ice, which is then used for daytime cooling.

Described below are the basic integrated ice storage/sprinkler system, optional subsystems, and two applications - a 223,000-sq ft shopping center installation and a 150,000-sq ft retail store.

INTEGRATED ICE STORAGE/SPRINKLER/VAV INDUCTION TERMINAL HVAC SYSTEM

Ice storage is indicated where the utility rate structure heavily penalizes daytime electric demand - and the facility has a high peak cooling load. Key decisions that determine overall cost-effectiveness and thus the practicality of the system include the optimum size for ice builders, and how the remainder of the cooling load is handled.

DEHUMIDIFICATION VIA ICE-COOLED VAV PRIMARY AIR

In the integrated ice storage/sprinkler/VAV induction terminal HVAC system, ice made at night with utility electricity is used during the day to dehumidify and cool to 40°F a very small quantity of primary air (0.1 to 0.2 cfm/sq ft). This primary air, delivered in variable volume to conserve energy, provides 100% of the space dehumidification requirement. Compared with a traditional all-air HVAC system, this approach achieves significant savings (up to 80%) in the first cost of the primary distribution system - ductwork, air handler, and also fanpower. This approach limits the size of ice builders, while using the ice for a highly appropriate task - producing very cold, dry primary air.

SENSIBLE COOLING AT VAV FAN-COIL INDUCTION TERMINALS

It is then possible to cool sensibly at terminal units with chilled water at an elevated temperature level, 58-68°F, which is an appropriate temperature for the sensible cooling task and which is thermodynamically more efficient than 40-42°F cooling. Thus the vapor compression chillers that produce ice at night, when utility rates are low, operate at a 25% higher efficiency during the day when rates are high.

The variable volume of primary air distributed to terminals, 0.1 to 0.2 cfm/sq ft, is determined by

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the space conditioning load. Fan-induced room air makes up the balance of the mixed air that is delivered at a constant volume of 1.4 cfm/sq ft to occupied spaces to maintain uniform air circulation and comfort. When additional cooling is required, with the primary air at maximum quantity, high-temperature chilled water is admitted to terminal coils; fan-induced room air is cooled prior to mixture with the primary air in the terminal boxes.

MULTIPLE BENEFITS TO SYSTEM FROM COOLING CAPABILITY IN VAV FAN-INDUCTION TERMINALS

The use of cooling coils in the VAV fan-induction units is a key factor in the overall cost-effectiveness of the integrated system for the following reasons:

1) Energy: Terminal coils increase energy efficiency by permitting high-temperature sensible cooling of secondary air at terminals (with primary air providing dehumidification).
2) First cost: Terminal sensible cooling permits the quantity of cold, dry primary air to be reduced to the minimum required for dehumidification - 0.1 to 0.2 cfm/sq ft. The result is a savings of some 80% in the first cost of the primary air distribution system and operating fanpower (compared with a standard all-air VAV system supplying 1.0 cfm/sq ft).
3) Compared with VAV fan-induction terminals without coils: A cold-primary-air system using the same terminals without coils would require three times as much primary air (0.45 cfm/sq ft) because
   a) the primary air would supply all sensible cooling as well as dehumidification; and
   b) the primary air would have to be at a higher temperature level (45-48°F instead of 40-42°F) to prevent over-dry conditions in the occupied space.

In the first system configuration - cold primary air with terminal sensible cooling - space humidity and temperature can be controlled precisely and primary distribution costs minimized.

INTEGRATED SPRINKLER/VAV FAN-COIL INDUCTION TERMINALS

Instead of installing a separate chilled water piping system from the central plant to terminals for secondary air cooling, most of this first cost is eliminated by using integrated fire sprinkler piping. The integrated sprinkler system[1] and VAV fan-coil induction terminals,[2] which are used together to be able to minimize distribution costs (both primary air and secondary chilled water) while recirculating and cooling air locally to maintain precise temperature and humidity conditions as well as uniform air flow in spaces, are proprietary systems developed by the author.

In the early 1970's, the author pioneered the development of sprinkler systems as a heat transfer medium, designing the first integrated HVAC/sprinkler systems installed and operated for space cooling.[3,4,5,6] Some of these systems have been operational for more than a decade. The VAV fan-coil induction terminal was developed to have the flexibility, in multizone buildings, to control separately humidity (via a small quantity of primary air) and temperature (via chilled water at terminals) within each zone, while supplying air to spaces at a constant volume.

The National Fire Protection Association (NFPA) since 1978 has permitted the use of sprinkler piping for HVAC heating and cooling (NFPA 13 - Standard for the Installation of Sprinkler Systems). Based on the author's experience over the past 10 years, such use supports the integrity of the sprinkler system since the continual flow of water for HVAC purposes ensures that fire water is available at the sprinkler heads.

OPTIONAL SUBSYSTEMS - COGENERATOR, INDIRECT EVAPORATIVE COOLING

In the integrated ice storage/sprinkler system, the vapor compression refrigeration subsystem, which produces ice at night and 58-68°F chilled water during the day, is generally sized to produce the quantity of ice needed to chill to 40°F the small quantity of primary air (the quantity required to handle the space dehumidification load, 0.1 to 0.2 cfm/sq ft). When a facility's daytime cooling load is very high - as in a shopping center with highly variable occupancy and tenant requirements, or a modern office building with heavy concentrations of heat-producing electronic equipment - there may not be a balance between the day and night refrigeration requirement. That is, the chillers sized for the ice-making (primary-air dehumidification) load may not be able to handle the day sensible-cooling requirement for 58-68°F chilled water. In such a case the following options are considered:

1) Gas cogeneration plus absorption chiller. This option is viable when electric utility demand rates are very high in relation to gas rates, and the daytime cooling load is very high. Cogenerated heat powers the absorption chiller, which produces 58-68°F chilled water for sensible cooling; simultaneously, cogenerated electricity powers the HVAC system fans, pumps and, when additional cooling is needed, the vapor compression chiller, as well as contributing to other electrical requirements. This system configuration eliminates all daytime electric utility demand for HVAC, as in the Westminster Mall example described below.
2) Gas-fired absorption chiller. This option is viable when electric utility demand rates are very high in relation to gas rates, but the daytime cooling load is not heavy enough to justify the cogenerator.
3) Enlarging the primary vapor compression refrigeration system. This option is preferable when the electric utility demand-charge penalty is not high enough to justify the cogenerator or absorption chiller. In this case, the optimum arrangement, in terms of overall cost-effectiveness, may include enlarging somewhat the ice-building capacity and quantity of cold primary air distributed.

In dry climates, an option that can make a major contribution to the cost-effectiveness of sensible cooling in the integrated system being described here is indirect evaporative cooling via a cooling tower and a plate heat exchanger. This is practical in a system where sensible cooling is at an elevated temperature (58-68°F). As much as 50% of the annual terminal cooling requirement can be met by indirect evaporative cooling in some locations.

APPLICATION: COLORADO SHOPPING CENTER

An energy-integrated ice storage/sprinkler/cogeneration/indirect evaporative cooling system has been designed for the 223,000-sq ft Westminster Mall addition in Westminster, Colorado, which is currently under construction (Fig. 1). The system has been structured to sharply reduce energy operating costs by eliminating all daytime electric utility demand charges for the HVAC system. In Westminster the utility rate structure is such that, had a conventional all-air HVAC system been used, 73% of the total annual utility cost would have been demand charges (established by peak daytime demand), and only 27% attributable to total annual usage (Table 1). In the new system, ice made at night for day cooling cuts this daytime demand by some 45%, and cogenerated electricity replaces the remainder of the daytime utility demand. As shown in Table 1, based on design projections the new system, using gas for cogeneration plus off-peak utility electricity, will save approximately $150,000 in annual energy costs. The anticipated payback on the incremental cost of this system over a conventional system is less than four years.

PRIMARY DEHUMIDIFICATION/SECONDARY SENSIBLE COOLING

In the mall HVAC system a small, variable quantity of very cold primary air (0.1-0.2 cfm/sq ft at 40°F) handles 100% of mall dehumidification and 15-35% of the sensible cooling load. The size of primary ductwork and air handler is reduced approximately 80% compared with a conventional all-air HVAC system. The balance of the sensible cooling, 65-85%, is done at terminals circulating 58-68°F chilled water from the central plant via the integrated fire sprinkler system.

Ice thermal storage produces 34°F chilled water which cools primary air to 40°F. Terminal sensible cooling via 58-68°F water is provided by indirect evaporative cooling (50%), ice (25% - whenever primary air does not require maximum cooling), and the compressor powered by cogenerated electricity (25%).

REFRIGERATION/ICE BUILDING PLANT

The central refrigeration/ice building thermal storage plant, operating at night using off-peak utility electricity, produces sufficient ice thermal storage to meet the maximum requirement for daytime cooling/dehumidifying of the primary air. The plant is a pumped refrigeration circulation system utilizing R-22 refrigerant. The refrigeration/ice thermal storage plant consists of two 150 bhp refrigeration screw compressors, two pipe coil type ice builder storage units with a combined ice storage capacity of 220,000 lb (based on a 2 in. ice thickness), a water chiller, and a heat recovery unit.

Each night during summer operation the compressors, operating at a +15 F suction temperature, produce 220,000 lb of ice in the ice builders. During the day, 34°F chilled water circulates from the thermal storage system to a coil in the primary air handling unit, where primary air is cooled and dehumidified to a 40°F saturated condition, then distributed

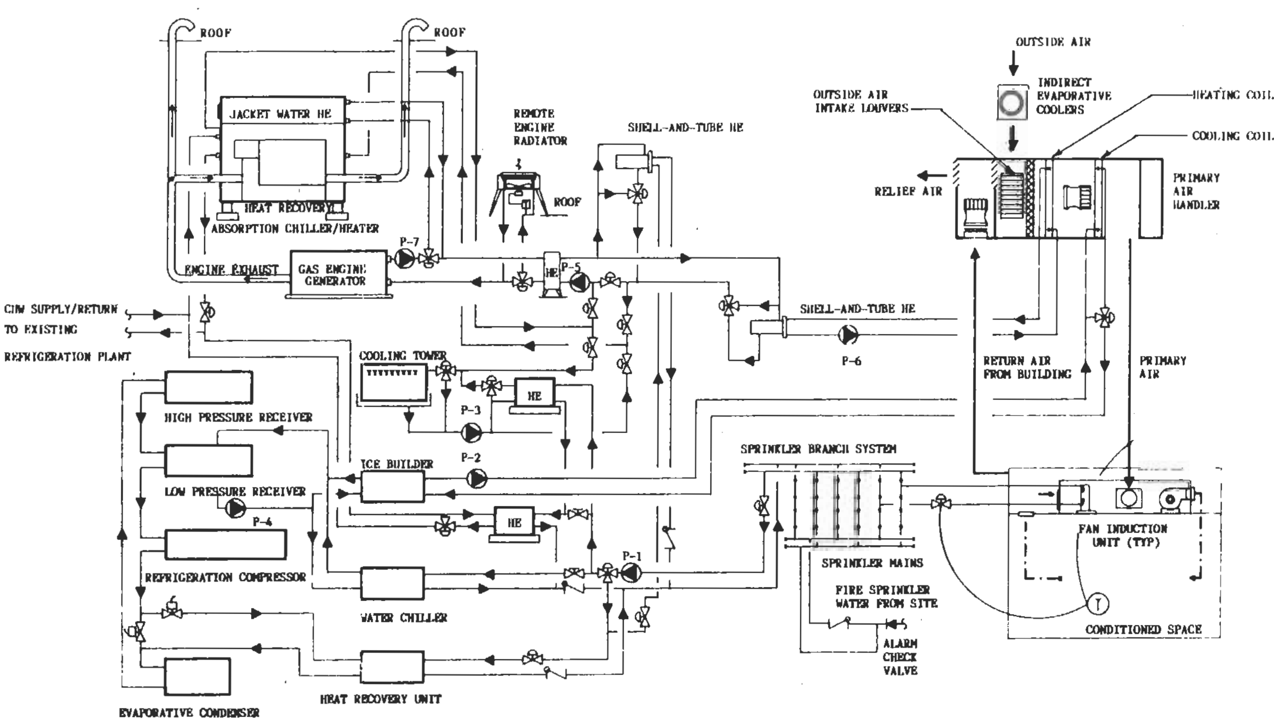


Fig. 1 Ice storage/integrated sprinkler/cogeneration system diagram (Westminster Mall Addition, Westminster, Colorado)

TABLE 1

Projected HVAC System Comparison:
Conventional vs. Ice Storage/Integrated Sprinkler/Cogeneration System
(Westminster Mall Addition, Westminster, Colorado)

Functions provided	Electric utility demand (daytime peak)	Annual demand charge, ($11.84/KW X 12)	Total annual usage (electric utility), KWH	Annual usage charge, 2.1¢ per KWH	Annual gas cost	Total Annual utility cost, electricity and gas	Annual energy cost saving
Conventional system, type used in existing mall							
HVAC for new mall addition	750KW	$106,560.	1,864,283. (12 hr per day operation)	39,150.	0	$145,710.	0
New system: ice storage/integrated sprinkler/cogeneration*							
o HVAC for new mall addition	0	0	773,762.	16,250.	71,145.	87,395.	149,755.**
1. Night ice building for day cooling of primary and secondary air.			(purchased off-peak power to make ice)				
2. Cogenerated electricity runs HVAC fans, pumps refrigeration machine.							
o 54,960. worth of lighting in new mall (additional cogenerated electricity applied to lighting reduces utility costs for lighting).							
o 36,480. worth of cooling in existing mall (cogenerator heat runs absorption chiller, which produces chilled water for existing mall, reduces utility bill of existing mall).							

*223,000 sq ft

**System contribution: $145,710. (HVAC value) + 54,960. (lighting) + 36,480. (cooling to existing mall) = 237,150. - 87,395. (less utility cost) = 149,755. (annual energy cost saving)

to the fan-induction terminals.

When outside conditions are hot and humid, one of the compressors operates during the day at +45 F suction, powered by cogenerated electricity, to produce 220 tons per hr of chilled water at 58°F for use in cooling sprinkler water distributed to terminal coils. When the compressor is not needed to cool sprinkler water, this cogenerated electricity goes instead into lighting to reduce the mall lighting bill.

When building ice, each compressor provides 134 tons of refrigeration at 150 bhp (0.89 ton per bhp). On air conditioning duty, compressor output is 256 tons at 141 bhp (1.82 tons per bhp).

At less than design conditions, the cooling tower will provide indirect evaporative cooling (through a plate type heat exchanger) to cool the sprinkler water supplying the VAV induction terminals. This "free" cooling is estimated to be 542,600 ton-hr (Table 2).

TABLE 2

Projected Annual Cooling Load, Free Cooling Contribution, and Mechanical Cooling Required, Ton-hr
(Westminster Mall Addition)

	Primary air refrigeration load	Fan induction terminal cooling coil refrigeration load	Total ton-hr
Total cooling	331,410	945,840	1,277,250
Indirect evaporative sensible cooling:			
Outdoor air*	69,950		
Terminal chilled water**		542,600	
Total free cooling			612,550
Required mechanical cooling			664,700

*Indirect evaporative coolers cool 27,000 scfm of outdoor air 17.6 F at design conditions.
**Cooling tower provides partial or full terminal chilled water cooling at less than design conditions via a plate type heat exchanger.

During milder winter weather when some daytime mechanical cooling is needed, the ice building system acts as a heat pump. Heat rejected from the ice equipment, 2.3 MMBtuh, is heat pumped through the fire sprinkler piping system, which keeps the spaces warm at night (60 to 65°F).

COGENERATION SYSTEM

A 425 KW natural gas driven, turbocharged engine cogenerator operates continuously during the mall's 12-hr day. Cogenerated heat (rejected engine heat) powers a heat recovery absorption refrigeration machine that operates near full load, producing between 150 and 200 tons of refrigeration to help meet the 600 to 800 ton requirement of the existing mall for space cooling.

During the winter, 0.9 MMBtuh of engine exhaust heat are recovered in the absorption unit's condenser water circuit (no chilled water is produced at this time). Another 1.5 MMBtuh are recovered from the engine jacket water. The combined 2.4 MMBtuh provide hot water for heating the primary supply air in the central air handling unit and for space heating at the air terminals via the sprinkler piping loop.

Cogenerated electricity, as already indicated, satisfies all the mall addition's HVAC electrical needs and contributes to the lighting requirement.

HVAC AUTOMATION SYSTEM

A computer based direct digital control system is being installed to optimize energy flow through the various HVAC subsystems so as to minimize energy costs. The fully automatic system will:

1) Control humidities and space temperatures in all

stores throughout the mall addition.

2) Control the functioning of each subsystem: refrigeration plant, cogenerator/absorption chiller, central/terminal VAV air handlers, and the integrated sprinkler chilled/hot water distribution subsystem.

3) Interface these subsystems to optimize energy use.

At any time, the building operator can check the performance and operation of the mall addition's HVAC system and each of its subsystems via the control system microprocessor.

AIR CONDITIONING PROCESS

A roof-mounted VAV central air handling unit supplies to the mall addition 53,600 actual cfm of air (equivalent to 44,600 scfm) consisting of 27,000 scfm of outside air and 17,600 scfm of recirculated air. As shown in the psychrometric chart in Fig. 2, outside air is precooled by means of an indirect evaporative cooler from 92°FDB to 74.4°FDB (1 to 2) and then mixed in the central VAV air handling unit with the recirculated air (5). This air is cooled to 40°F by ice chilled water (6 to 7) and distributed from the central air handler through insulated primary ductwork to the VAV fan-coil induction units located throughout the addition in the ceiling plenum above occupied spaces (one or two terminals per store, depending on store size). At design conditions, the indirect evaporative cooler provides nearly 43 tons of sensible cooling. On an annual basis, 69,940 ton-hr of sensible ventilation-air cooling are done by the indirect evaporative cooler.

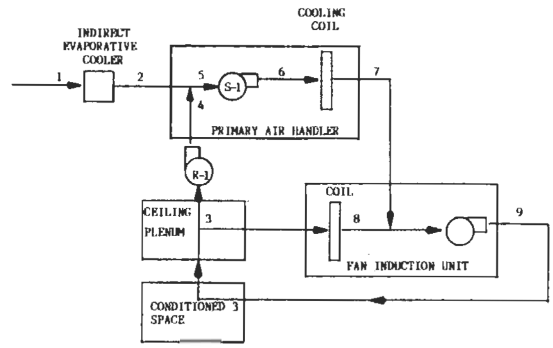
The quantity of cold primary air entering the VAV terminal varies between 0.1 and 0.2 scfm per sq ft, determined by the space air conditioning requirement. Fan-induced room air makes up the balance of the 1.4 cfm per sq ft that is distributed in constant volume from the terminals to the occupied spaces. When the primary air damper is fully open and additional cooling is required to achieve the supply air condition (9), the terminal cooling coil valve opens, admitting 58°F chilled water from the sprinkler piping to cool the fan-induced room air (3 to 8) prior to mixture with the primary air.

APPLICATION: MISSISSIPPI DEPARTMENT STORE

The ice thermal storage HVAC system designed for the two-story, 150,000-sq ft Gayfers Store at Northpark Mall, Ridgeland, Mississippi, includes the following:

1) A refrigeration/ice thermal storage plant consisting of two 80,000-lb ice builders (based on 2 in. thickness) and three 125 bhp reciprocating compressors utilizing R-22 refrigerant. The compressors operate at night, when electric utility rates are lower, to produce sufficient ice thermal storage for daytime cooling/dehumidification of the minimized quantity of primary air (0.1-0.2 cfm/sq ft). During the day in summer, one compressor operates at a higher efficiency providing 58°F chilled water to cool secondary air at terminals.

2) A prefabricated 22,000 cfm rooftop VAV air handling system supplying ice-chilled 43.6°F primary air to all conditioned zones on all floors.



Key:

1. Outside air entering at 92°FDB, 65°FWB.
2. Outside air leaving indirect evaporative cooler at 74.4°FDB, 59.5°FWB.
3. Building air, 75°FDB, 59°FWB.
4. Return air, 77°FDB. 59.5°FWB to central primary air handler.
5. Mixed outside air and return air, 75°FDB. 59.5°FWB at central primary air handler prior to coil cooling.
6. Mixed outside air and return air entering primary cooling coil at 80°FDB, 61°FWB.
7. Mixed outside air and return air leaving primary cooling coil at 40°F saturated.
8. Return air (at VAV FCIU terminals) leaving terminal cooling coil at 64°FDB, 55°FWB.
9. Supply air (mixed primary and terminal-cooled return air), 61°FDB, 53.3°FWB.

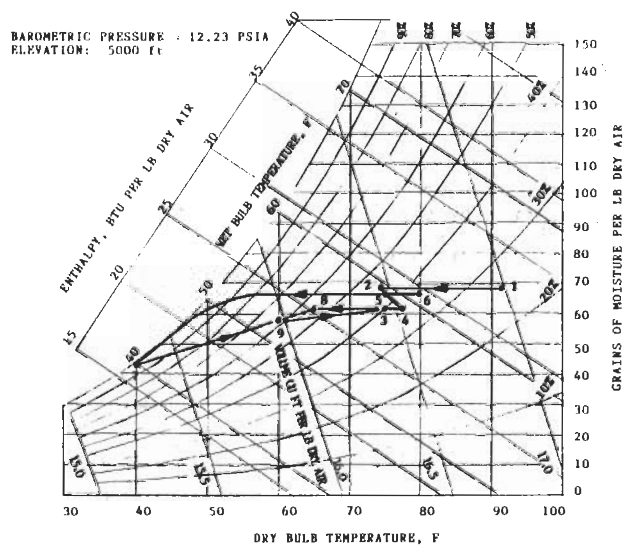


Fig. 2 HVAC system process
(Westminster Mall Addition, Westminster, Colo.)

3) A VAV fan-coil induction terminal in each zone, which mixes the cold primary air with locally recirculated air and supplies the mixed air at a constant volume (1.0 cfm/sq ft) to the occupied zone. The locally recirculated air is cooled in the fan-coil induction unit, with 58°F chilled water, prior to mixture with the primary air.

The small quantity of very cold primary air provides 100% of the dehumidification and, overall, 56% of the HVAC requirement; 44% is handled at terminals with the high-temperature chilled water produced at a higher chiller COP. The system's total cooling capability is 249 tons.

The system is designed to operate during the winter as a heat pump, simultaneously producing ice for day cooling and hot water for heating the building during the unoccupied night hours.

As shown in the psychrometric chart and diagram in Fig. 3, outside air enters at 95°FDB, 76°FWB (1) and is mixed in the central air handler with return air (3). The mixed air enters the primary cooling coil at 82.8°FDB, 67.4°FWB and is cooled by ice-chilled water to 40°FDB, 40°FWB (4 to 5). Primary air leaves the air handler at 43.6°FDB, 41.5°FWB (6) and is supplied to VAV fan-coil induction terminals. Also entering the terminals is locally recirculated air that is sensibly cooled in the terminals, with 58°F chilled water, to 65°FDB, 59°FWB (7). Primary and recirculated air are mixed in the terminal (8) and supplied to the space (9) at 62.8°FDB, 57°FWB. Inside space conditions are maintained at 75°FDB, 62.5°FWB (2).

Based on design data, the energy-integrated ice storage HVAC system at Gayfers Store achieves a savings of $50,000 in annual energy cost. These savings are based on (1) shifting demand to off-peak hours (ice-making for primary air cooling) and (2) increasing the COP of daytime chiller operation by producing chilled water at an elevated temperature for terminal sensible cooling.

First cost savings of 80% in the primary air distribution system (ductwork, air handler) enhance the practicality of investing in ice-making equipment. Overall, these first cost savings plus the balance between primary and secondary cooling (size of ice-builders versus day utility charges) plus the improved efficiency of daytime sensible cooling achieve a payback of four years on the first cost investment in the ice thermal storage system (over the cost of a conventional all-air HVAC system).

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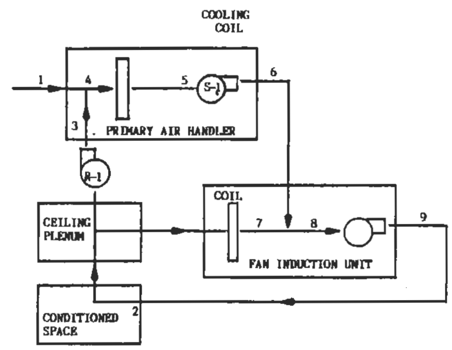

Key:

1. Outside air entering at 95°FDB, 76°FWB.
2. Building air at 75°FDB, 62.5°FWB.
3. Return air, 76.5°FDB, 63°FWB to central primary air handler.
4. Mixed outside air and return air entering primary cooling coil at 82.8°FDB, 67.4°FWB.
5. Primary cooling coil leaving air at 40°FDB, 40°FWB.
6. Primary air supply at 43.6°FDB, 41.5°FWB.
7. Return air (at VAV FCIU terminals) leaving terminal cooling coil at 65°FDB, 59°FWB.
8. Mixed primary and terminal-cooled return air, 61.5°FDB, 56.5°FWB.
9. Supply air to space, 62.8°FDB, 57°FWB.

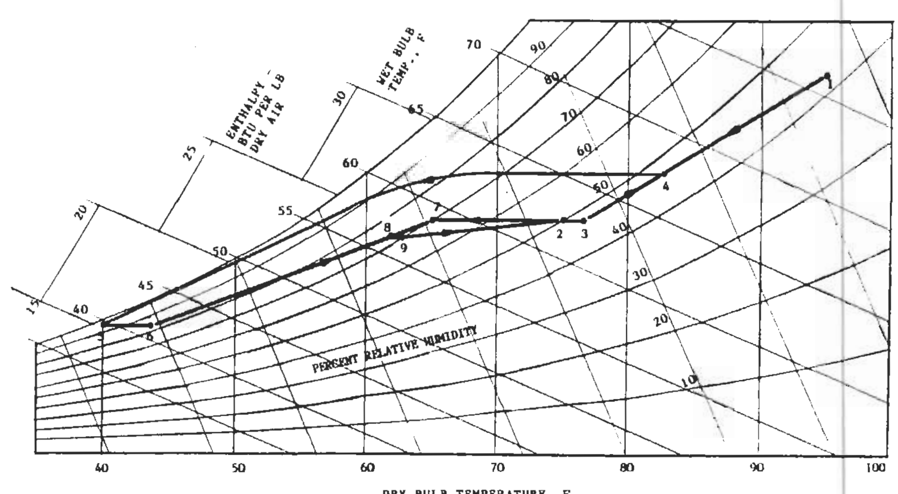

Fig. 3 HVAC system process (Gayfers Store, Northpark Mall, Ridgeland, Mississippi)

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