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DESIGNING FOR COMFORTABLE COOLING SEASON HUMIDITY IN HOTELS

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ABSTRACT

Many hotels throughout the United States report problems with high humidity, resulting in poor comfort, odor, mold, and the associated costs of decontaminating and repairing building materials and finishes. This paper differentiates the two types of moisture and mold problems experienced in hotels, provides insight into the sources of moisture, and includes a brief discussion of improved air-conditioning and dehumidification systems and equipment that should be considered in designing HVAC systems for new hotels or for retrofitting existing hotels in order to reduce the cooling season relative humidity. A reduction in cooling season relative humidity significantly improves comfort and minimizes or prevents mold formation in hotels. The benefits of comfort, improved guest perception, and loyalty justify the inclusion of improved air-conditioning and dehumidification systems and equipment in hotel designs. Owners should also evaluate the labor and material costs associated with corroded finishes on lamps and hardware, damage to mirrors and pictures, and rusty door frames, lock sets, and air grilles.

INTRODUCTION

Many hotels throughout the United States report problems with high humidity (moisture) and the resultant poor comfort and odor and significant mold treatment and building repair costs. This paper differentiates the two types of moisture (and mold) problems most often experienced in hotels and provides insight into the sources of the moisture. This is followed by brief presentations on systems and equipment that should be considered in designing HVAC systems for new hotels or in retrofitting HVAC systems in existing hotels. These improved air-conditioning and dehumidification systems, used singly or in synergistic combinations, will produce lower space relative humidity, a reduction or elimination of mold problems, improved guest comfort, and lower operating costs. Some of the systems result in lower installed cost. All result in increased cash flow through reduced housekeeping, refurbishing, and repair costs and more repeat guests.

MOISTURE AND MOLD

A hotel invests tens of thousands of dollars annually in advertising, promotion, renovation, and employee training to attract customers and to retain loyal, repeat guests. A hotel experiencing high relative humidity in rooms, corridors, lobbies, lounges, and restaurants is not comfortable and discourages repeat guests. High moisture levels often result in mold and mildew problems that offend the guest, and the guest will probably never return to the hotel, even if the problem is corrected.

Mold experts are straightforward in spelling out the requisites for mold growth:

- Mold spores, which are everywhere.
- Food for the mold spores (paper, adhesives, paste, shellac, paint).
- Temperatures between 50°F (10°C) and 100°F (38°C).
- Liquid moisture or relative humidity above about 70%.

In other words, if a surface has moisture, it will have mold. Since mold spores and food cannot be eliminated, the water vapor must be controlled.

IN-ROOM MOLD VS. MOLD BEHIND VINYL WALL COVERING (VWC)

In-room mold and mold behind vinyl wall covering (VWC) are separate and distinct problems. (VWC, as used herein, represents any interior wall finish material having low water vapor permeability.) The causes of each problem are different, and the two problems must be analyzed separately. Past attempts to group these problems and find a single solution have resulted in confusion and delay in finding solutions.

Mold behind vinyl wall covering is a problem generally confined to Florida, areas within 100 miles (161 km) of the Gulf of Mexico, and other locations in the world having similar climates. It appears as pink or chartreuse splotches on the surface of the VWC. When the VWC is peeled away from the wall, black mold is evident. The pink and chartreuse splotches are evidence of the mold on the back side of the VWC. Mold cannot penetrate the vinyl sheet, but a reaction or secretion allows the mold on the back side of the vinyl to etch pigment from the finish on the front side of the vinyl or cause the dye in the vinyl to run.

The solution to mold occurring behind vinyl wall covering is a forgiving wall system that avoids trapped

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water vapor and prevents both the black mold on the back side and the discoloration on the room side. Unfortunately for the VWC industry, this means that VWC should not be used on the outside walls of guest rooms or on walls that communicate with outside walls if the hotel is in the above locations. The reason behind this is that the VWC is a vapor retarder, and in these climates it is located on the cold (in summer) side of the wall. The dew-point temperature of the outside air is often higher than the temperature of the interior wall surface. Under these conditions, water vapor that infiltrates or permeates through the wall may condense on the back side of the VWC. This is why some insulation manufacturers recommend that the vapor retarder face of the insulation be placed toward the outside of the wall in these climates.

In-room mold is a common problem; however, a regular program of preventive maintenance housekeeping during humid weather can usually control it. The in-room mold problem in new hotels can be significantly reduced, and possibly eliminated, by designing the public space and guest room-corridor make-up air systems with leaving dew-point temperatures well below the room design dew-point temperature. Unconditioned outside make-up air (to replace the air exhausted by the bathroom exhaust fans) is the single largest contributor to high levels of humidity in guest rooms and corridors.

MOISTURE SOURCES

Office buildings seldom experience high indoor relative humidity. Hotels, and in particular, guest room corridors, frequently experience muggy conditions. Hotels require higher outdoor air quantities and 24-hour, seven-day-a-week operation. Many operating hours occur during summer nights and in the spring and fall when the space sensible cooling load is small. This may result in little or no moisture removal as the outside air passes through the cooling coil.

Table 1 illustrates the sensible and latent heat gain load components for a typical guest room corridor serving 24 rooms. (In this load analysis, the air supplied to the corridor is transferred to guest rooms through an undercut door and is then exhausted through the bathroom. This corridor-to-guest-room make-up air system is used in many parts of the country. It is used in this example to illustrate a common application problem, not as an endorsement of the system. Before considering this system, designers should verify that local codes and building officials will accept the system.) Corridors typically have no outside walls. The sensible loads are lighting, possibly some heat gain from ice and vending machines, and, for the top floor only, roof heat gains. The roof heat gains may occur only at the ends of the corridors because the penthouse covers the middle of many corridors.

Figure 1 shows the psychrometric process that satisfies the corridor load conditions. The corridor requires a supply air temperature of approximately 70°F (21°C) at a dew-point temperature 2 or 3°F (1 to 2°C) below the desired guest room dew-point temperature. Guest room air conditioners do very little dehumidification, particularly at light loads. A portion of the moisture removed by the room air conditioner's cooling coil in a short cycle of active cooling is reevaporated during the period when the room thermostat is satisfied, there is no refrigerant or chilled-water flow in the coil, and the room air conditioner's fan continues to run. For this reason, designers should consider a corridor make-up air dew-

TABLE 1
Guest-Corridor Cooling Calculations

Design Values

Location	Charlotte, NC
Corridor Area	1,200 ft^2 (111 m^2)
Rooms Served	24
Miscellaneous	Ice and Vending
Room Makeup Air	1,200 cfm (566 L/s)
Serv. Elev. Exh.	300 cfm (142 L/s)
Total Makeup Air	1,500 cfm (708 L/s)
OA Dry-Bulb	93°F (33.9°C)
OA Wet-Bulb	74°F (23.3°C)
Corridor Dry-Bulb	72°F (22.2°C)
Corridor RH	55%
Fan & Duct Rise	4.3°F (2.4°C)

Load Calculations

Item	Btu/h	(kW)
Wall	—	—
Roof	—	—
Lighting	2000	0.59
Equip. & Misc.	1300	0.38
People Sensible	—	—
Room Sensible	3300	0.97
Room Latent (24 Rm)	7200	2.11
Room Total	10500	3.08
OA Sensible	35000	10.26
OA Latent	33000	9.67
Fan & Duct	7000	2.05
Total w/o RH	85500	25.06
Reheat	18500	5.42
Total w/RH	104000	30.48

Psychrometric Properties

Point	Dry-Bulb	Humidity Ratio
OA	93°F (33.9°C)	0.0137
Room	72°F (22.2°C)	0.0092
Required Supply	70°F (21.1°C)	0.0082
------------ Reheat System --------------		
Leaving Coil	54.5°F (12.5°C)	0.0082
After Fan & Duct	58.8°F (14.9°C)	0.0082
After Reheat	70.0°F (21.1°C)	0.0082

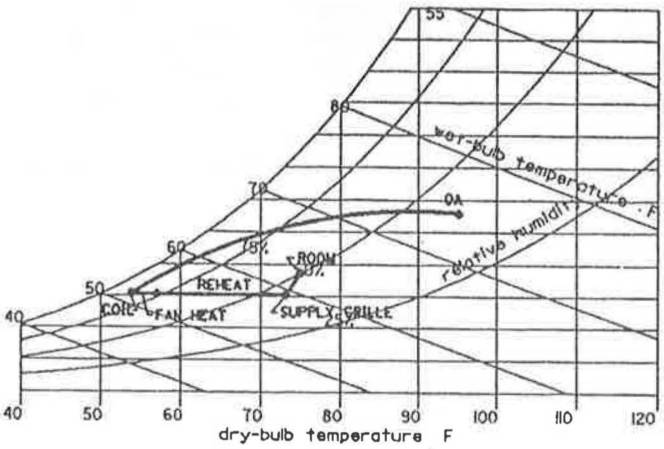

Figure 1 *Corridor psychrometric plot.*

point temperature low enough to satisfy the latent loads of the corridors and the guest rooms.

Moisture and outside air loads in the public spaces (lobby, meeting rooms, lounges, and restaurants) can be estimated using manual or software-based methods. Designers should evaluate part-load moisture removal performance. The systems selected must be capable of delivering air at the correct dew-point temperature at all load conditions; otherwise, space relative humidity may climb to unacceptable levels during the many hours of part-load operation.

The quantity of outside air for the guest tower is usually fixed by the amount of air exhausted from the guest tower. The outside air requirements for public space and back-of-the-house areas can be varied depending on occupancy level and the operation of toilet, range exhaust, and dishwasher exhaust fans. Until the introduction of microprocessor DDC controls, varying the outside air quantity was difficult, but now the outside air quantity can be matched to occupancy, exhaust, and pressurization requirements. It is important that air-handling units be equipped with low-leakage, motorized outside air dampers. This eliminates a potential source of moisture when the air-handling unit is off or in a warm-up or unoccupied mode.

Infiltration moisture loads can be estimated for entries and exits. The hidden infiltration paths are more difficult. The building must receive a thorough infiltration-exfiltration inspection at several intermediate stages of construction. Examples of hidden moisture infiltration paths include

- Barrier between the porte cochere and building
- Barrier between soffited overhangs and building
- Barrier at the underside of metal decking
- Barrier details around structural elements
- Barrier penetrations for pipe, duct, and conduit
- Lighting fixtures in soffits and overhangs and at building entries
- Column covers in outside walls at soffits
- Penthouse and other nontypical walls
- Venting of elevator hoistways
- Gravity dampers on fans
- The interior of exterior walls above the finished ceiling
- The interior of exterior walls within sill height enclosures

Identifying and correcting infiltration problems is vital in order to control relative humidity and comfort. In cold climates, the cost of several construction inspections can easily be paid for by savings in not having to repair the damage resulting from frozen sprinklers and other piping. The benefit of lower summer humidity is free.

Many hotels utilize conventional, non-reheat, packaged direct-expansion or chilled-water rooftop units to condition 100% outside air for the corridors and guest rooms. In some cases, these units are specified to cool the outside air from 95°F EDB, 78°F EWB (35°C EDB, 25.6°C EWB) to approximately 58°F LDB, 57°F LWB (14°C LDB, 13.9°C LWB). This leaving condition is at or below the described space dew-point temperature; however, there are three problems.

1. The leaving condition may be arbitrary, not the result of a psychrometric analysis of the actual sensible and latent loads of the space. The example sensible load required about 66°F (19°C) LDB. Therefore, the controlling thermostat will cycle the coil on and off or modulate the control valve to obtain a 66°F (19°C) average LDB with little or no moisture removal. Some designs specify the unit at 66°F LDB and 55°F LWB (19°C LDB, 13°C LWB), which may be psychrometrically correct but, unfortunately, no air-conditioning manufacturer has produced a coil that meets those parameters.
2. Most direct-expansion rooftop units are designed to intake or process up to about 30% outside air, not 100% outside air.
3. Direct-expansion units seldom have sufficient compressor modulation range to deal with a load that varies from 100% at 78°F (26°C) EWB to no load at about 55°F (13°C) EWB.

IMPROVED SYSTEMS AND EQUIPMENT FOR LOW HUMIDITY

The following sections discuss: reheat using refrigerant hot gas, treated and recirculated toilet exhaust, heat pipes, run-around coils, dual-path air-conditioning, return air face and bypass, cool storage, cold air distribution, and desiccant dehumidifiers. These technologies may be used independently or synergistically in combination with one another. All these technologies result in lower space relative humidity. Most use less energy, have less electric demand, and save operating cost. Treated and recirculated exhaust and cold air distribution systems (with cool storage) may result in both lower first cost and lower operating cost.

One system conspicuously absent from the preceding list is the "new energy" reheat system in which cooled air is reheated (using electric heat or heat from fossil fuels) and then introduced into the space. New energy reheat systems have a triple penalty: (1) the cooling generation plant, associated auxiliaries, and electrical service that are increased by the amount of reheat; (2) the operating cost to cool the air; and (3) the operating cost to reheat the air. New energy reheat may be necessary as a final supplement to other systems, but it should not be the sole or priority means of humidity control.

Reheat Using Refrigerant Hot Gas

Reheat of the cooled airstream using hot discharge gas from the refrigerant compressor is an option. The thought process is logical. The refrigerant compressor is operating to produce the cooling effect at the direct-expansion cooling coil and, if reheat is required, why not use the hot discharge gas from the refrigerant compressor? Dehumidifiers based on refrigeration cycles are designed using this method. Reheat using hot gas is common in air-conditioning units for supermarkets and natatoriums.

It makes sense to use this "free heat" in lieu of using heating coils deriving their energy from electric, gas, or oil sources. This saves the cost of the reheat energy. Remember that reheat involves three penalties: (1) an increase in the size and cost of the cooling generation plant, (2) the energy cost of cooling the air, and (3) the energy cost of reheating the air. Reheat using refrigerant hot gas mitigates only the third penalty. The refrigeration plant must still be larger by the amount of the reheat, and the owner must pay the energy cost of the cooling cancelled by the reheat.

Designers of refrigeration systems using hot gas for reheat are cautioned that the design is not as simple as it first appears. When the reheat coil is actually reheating, the cycle is obvious. When the controls no longer require reheat, the reheat coil will experience low refrigerant pressure and the coil will fill with liquid refrigerant. When the controls again call for reheat, this liquid refrigerant must be displaced in order for the reheat coil to function. The system must incorporate a larger refrigerant receiver and additional refrigerant charge in order to accommodate the liquid refrigerant that is alternately stored in the reheat coil. Designers have tried numerous ways, including check valves at the reheat coil outlet, to circumvent this problem. If the check valve holds, then the reheat coil will be under very low pressure, inviting leaks. Most valves have a small leakage rate that in this case means the system must still be designed with the extra refrigerant charge and receiver. Some designers suggest bleeding a small amount of hot gas to the reheat coil when reheat is not required. This accomplishes nothing. The coil remains filled with liquid refrigerant. Some designers vent the reheat coil to compressor suction pressure when reheat is not required. This can be an acceptable solution but adds the risk of some liquid floodback to the compressor when the system changes from reheat to no reheat.

A variation of hot gas reheat uses the warm water leaving a water-cooled condenser for reheat. This simplifies the refrigerant cycle and is a better alternative if the refrigerant cycle is water cooled. Penalties 1 and 2 noted above still apply.

Treated and Recirculated Exhaust for Guest Tower Make-Up Air

The outdoor air component of hotel HVAC loads is significant. Also, large quantities of air at room design temperature and humidity are exhausted throughout the year from many hotels. This air contains heat energy—warm or cold—that, if reused, would result in energy and operating cost savings. Various methods are available to recover the heating and cooling potential of exhaust air. Treated and recirculated exhaust is, in effect, similar to rotary wheel, fixed-plate, heat pipe, and run-around heat exchangers. The outside air quantity is reduced to the amount required by ASHRAE Standard 62, and the balance of the air is obtained from treated exhaust air. It is a superior alternative for guest tower make-up air systems because it essentially utilizes all of the sensible and latent heat of the exhaust air. In the humid season, low-moisture exhaust air replaces high-moisture outside air, resulting in significantly lower corridor and guest room humidities.

Central recirculation of the guest room exhaust air permits the recovery of the usable energy in this air. Treatment is required to remove odors and other impurities picked up from the conditioned space and to improve the air quality to meet ASHRAE Standard 62 requirements. The cost of this treatment, however, can be far less than the cost of conditioning the same amount of outdoor air (Gatley 1981). Often these savings are substantial, as shown by the experience at hotels in Atlanta, Austin, Bethesda, Champaign, Charlotte, Dearborn, Ft. Lauderdale, Memphis, Mt. Hood, New Orleans, and Tampa. The accompanying maps show that significant savings can be obtained nationwide.

The maps show representative savings around the U.S. from treating and recirculating 10,000 cfm (4,720 L/s) of conditioned air rather than exhausting it and replacing it with outdoor air. Figure 2 shows the annual ton-hours of cooling saved. Figure 3 shows savings in annual Btu of heating energy. The ton-hour cooling and Btu heating savings are based on bin method calculations using weather data from Air Force Manual 88-29,

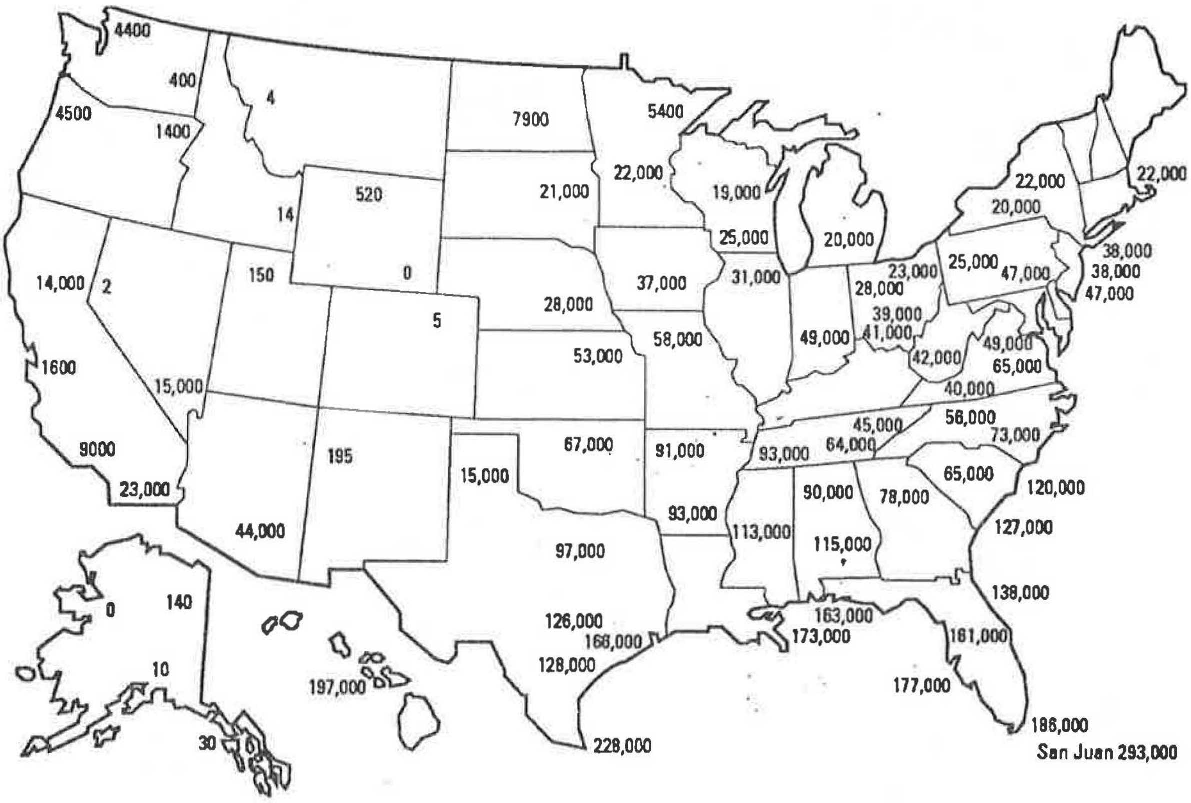

Figure 2 *Annual ton-hour savings treating and recirculating 10,000 cfm. Note: To convert to kWh cooling, multiply by 3.52.*

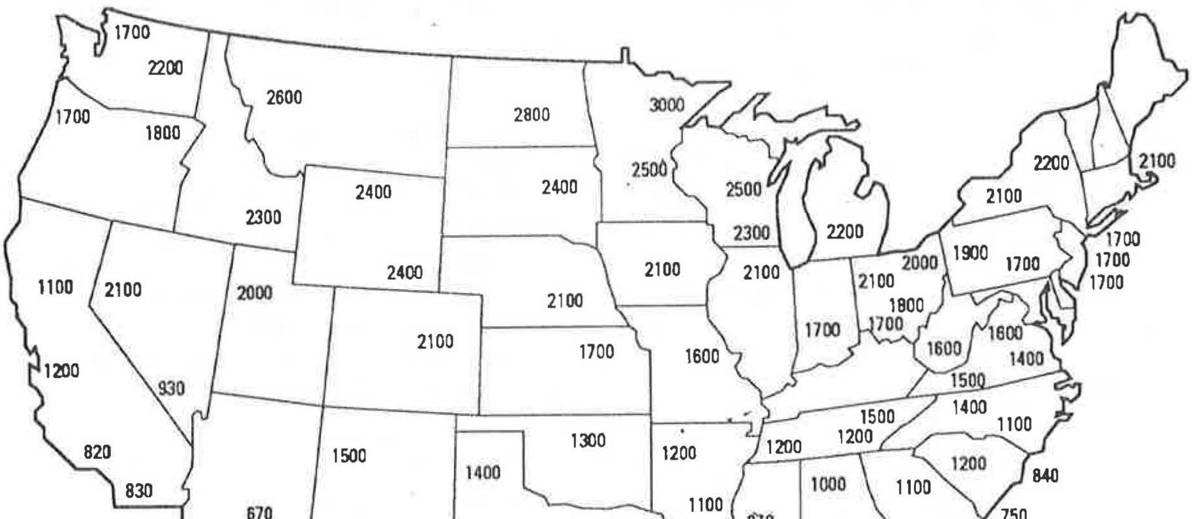

Figure 3 *Annual million Btu heating output savings treating and recirculating 10,000 cfm. Note: To convert to MJ heating output savings, multiply by 1.055.*

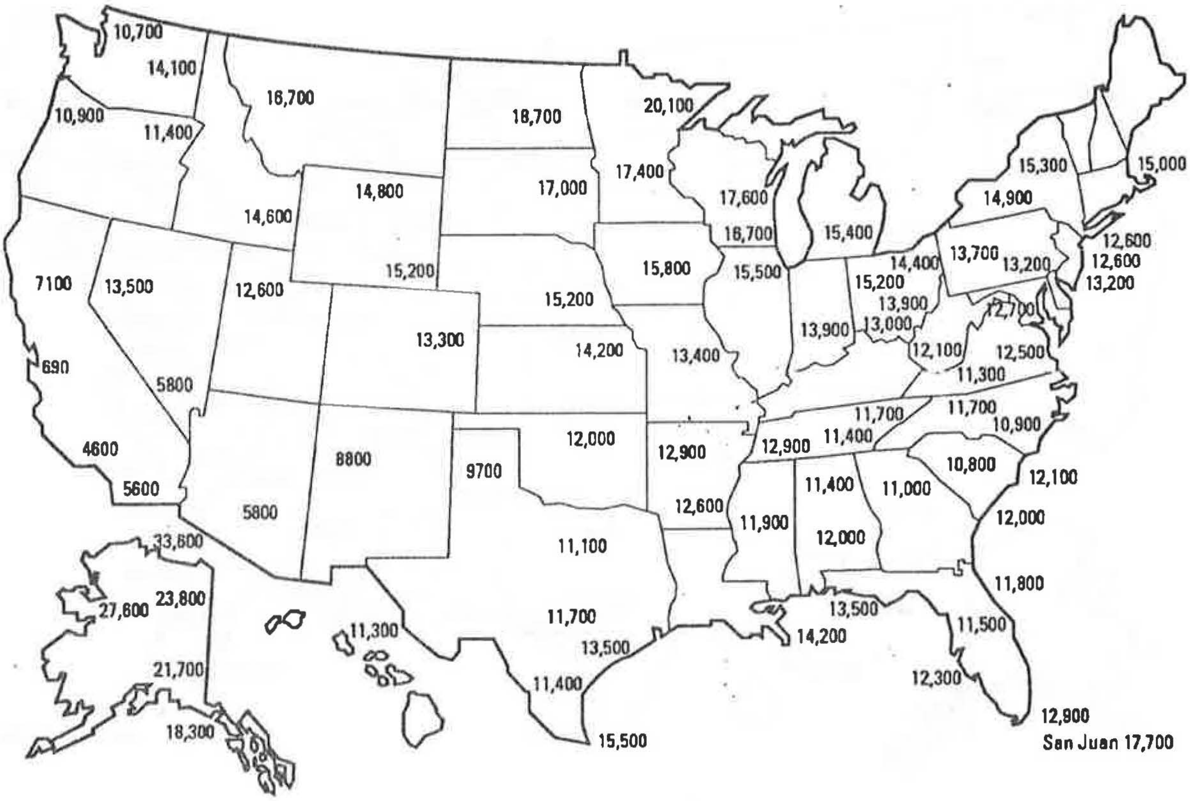

Figure 4 Annual operating cost savings treating and recirculating 10,000 cfm.

Engineered Weather Data. Return air conditions were summer, 75°F (24°C) with 45% RH, and winter, 70°F (21°C). Humidification energy was not included. Figure 4 presents annual operating cost savings. The dollar savings in Figure 4 are net savings. They embrace reductions in heating fuel, cooling energy for electric-drive chillers, cooling tower water, and water treatment chemicals. Two cost penalties are included: the added fan power cost chargeable to the additional pressure drop of the odor oxidant cells used to purify the recirculated air and the annualized replacement cost for the odor oxidant. Annual cost calculations are based on 75% boiler efficiency, $0.55/therm ($5.80/MJ) fuel cost (75% gas, 25% oil), 0.8 kW/ton (0.23 kW_e/kW_c) overall chiller plant energy ratio, 0.39 in. (97 Pa) additional air-side pressure drop, 63% fan efficiency, 88% motor efficiency, 4 gallons (15 L) make-up water per ton-hour cooling, $2.00/1,000 gallon ($0.05/m^3) chemical and water cost, $0.15/cfm ($.31 per L/s) annualized cost of changing the odor oxidant, and $0.074/kWh. Fuel and electricity costs are U.S. averages from NISTIR 85-3273-5 (5/90) including price indices for 1995.

The net annual operating cost savings from treated recirculated exhaust air is approximately $1.50 per cfm ($3.18 per L/s) for most areas in the country, with the exception of California and parts of Arizona, which have low summer wet-bulb temperatures and mild winters. Local utility costs can affect these average figures. Fuel and power costs applicable to local conditions should be used in a detailed analysis.

Figure 5 illustrates a conventional treated and recirculated exhaust system. Figure 6 shows the psychrometric plot and cooling savings.

Treated and recirculated exhaust air is addressed in the BOCA Mechanical Code. Basically the BOCA code requires that the recirculated exhaust be treated to achieve a quality better than that of outside air. This requires particulate filtering and gas-phase absorption through the use of odor oxidant pellets, such as inert expanded alumina saturated with potassium permanganate, or activated charcoal pellets.

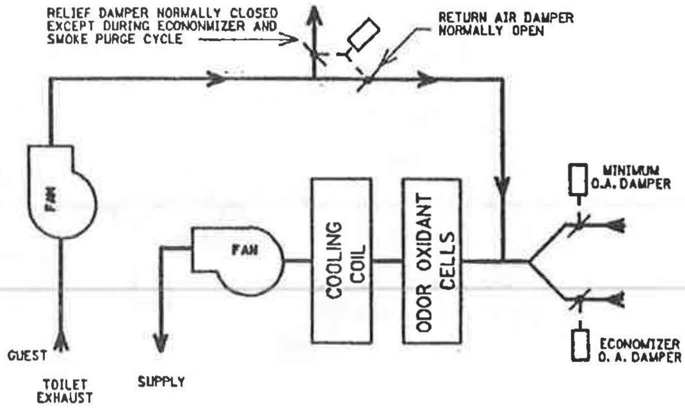

Figure 5 Guest tower—treated and recirculated exhaust.

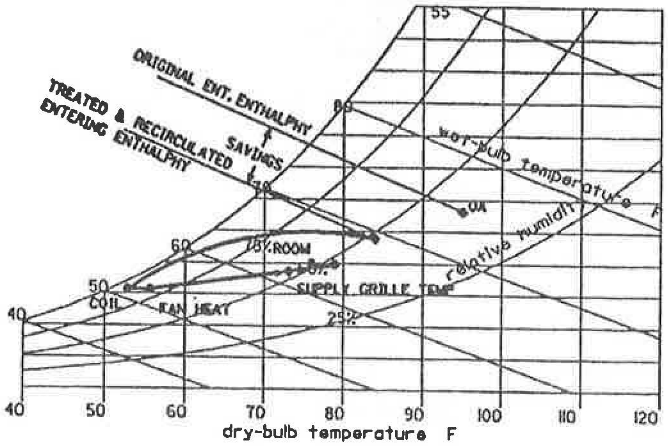

Figure 6 *Treated and recirculated exhaust—conventional.*

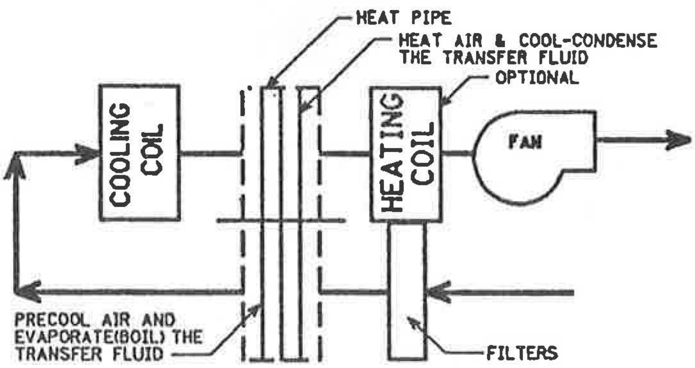

Figure 7 *Heat pipe assisted system.*

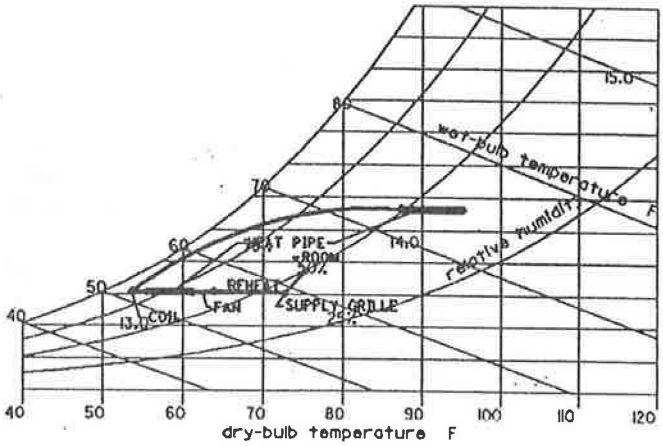

Figure 8 *Heat pipe assisted system.*

For non-BOCA code locations, most authorities will approve a written, well-documented variance request for this system. A variance is required because most codes state that toilet exhaust must go directly to the outside of the building. One code official ruled that toilet exhaust was no longer toilet exhaust after it had passed through particulate and gas-phase filtration.

Heat Pipes

Heat pipes have been used for decades, although their application to improve dehumidification is quite new. The addition of heat-pipe air-to-air heat exchangers can improve the dehumidification capabilities and the energy effectiveness of other technologies.

A heat pipe is a single heat transfer assembly including a cooling (fluid evaporating) section and an adjacent heating (fluid condensing) section. In this application, the cooling section is located in the warm air entering the cooling coil and the heating section is in the cold air leaving the coil. This results in a transfer of heat from the warm, humid incoming air to the cold exiting air, reducing or even eliminating the reheat energy requirement. The air leaving the unit is warmer but drier, and more dehumidification is achieved. Figure 7 illustrates a heat pipe added to a conventional system. Figure 8 shows the psychrometric process.

Use of the heat pipe reduces the cooling and heating energy requirements and can reduce the cooling generation equipment size. However, it increases the fan power needed to move the air over the heat pipe's heat exchange surfaces and may increase first cost. Heat pipes are a good solution for 100% outside air cooling units when treated and recirculated exhaust systems are not used.

The major challenge with heat-pipe heat exchangers is the arrangement of equipment rooms and entering and leaving air ductwork to permit the side-by-side placement of the joined sections of the heat pipe or an alternative arrangement using vertical tubes with the cooling (evaporating fluid) section directly under the heating (condensing fluid) section.

Run-Around Coil Systems

Run-around coils are similar in many respects to the heat pipe. The major differences are that run-around coil systems allow a separation of components and they require a pump, a compression tank, and field installation of the heat transfer fluid.

The function of the two heat transfer sections is psychrometrically similar to that shown for the heat pipe, but the physical location is no longer critical to the function. Because the heat transfer fluid is pumped between and through both coils, the coils can be separated and in any position.

The run-around coil system requires a separate closed-loop piping system, as shown in Figure 9. It requires a compression tank to handle fluid expansion, a means of fluid makeup and management, a pump, electrical service, and controls. The heat transfer fluid is quite often ethylene glycol when freezing is a possibility; otherwise water is used.

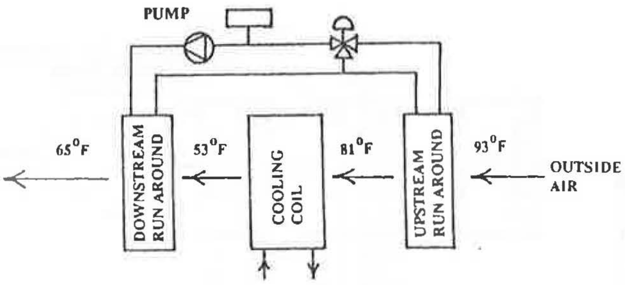

Figure 9 *Run-around coil piping (cooling season conditions).*

Fan power increases with the run-around coil as with the heat pipe. Fluid pumping is a small additional operating cost.

Dual-Path Air-Conditioning Systems

The dual-path air-conditioning system, shown in Figure 10, involves the use of a cooling coil in the outside airstream and a second cooling coil in the return airstream.

The goal is to cool the outside air to a low enough dew point to satisfy the latent load. The low leaving-air temperature of the outside air cooling coil will satisfy a portion of the space sensible load. The coil in the return airstream is used to satisfy the remainder of the sensible load.

The outside air constitutes a major portion of the latent load. Cooling all of the outside air to a low dew-point temperature guarantees that no humid outside air bypasses the cooling coil and enters the space. This is particularly important at part-load conditions, when most air-conditioning systems allow humid outside air to bypass the cooling coil. The two airstreams must be thoroughly mixed prior to any duct branches. Reheat may be required at part-load conditions if the mixture temperature is cooler than required by the space.

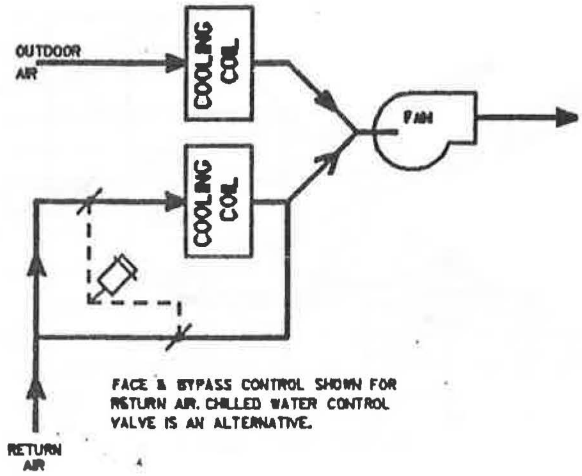

Figure 10 *Dual-path cooling system.*

Return Air Face and Bypass Systems

Return air face and bypass systems are similar in some respects to dual-path air-conditioning systems. Figure 11 illustrates a typical arrangement. Figure 12 shows the psychrometric plot of return air bypass in a treated and recirculated exhaust application. The primary feature is that all of the outside air passes through the cooling coil, even at partial loads. As the space cooling setpoint is approached, more and more return air bypasses the cooling coil. This damper arrangement is sometimes called auditorium bypass and was applied to units supplying auditoriums in schools and other facilities.

An advantage of this arrangement over a dual-path system is the use of a single cooling coil, resulting in a more compact arrangement. Reheat may be required at low space loads if the mixture of dehumidified outside air and bypassed return air is cooler than required to satisfy the space load.

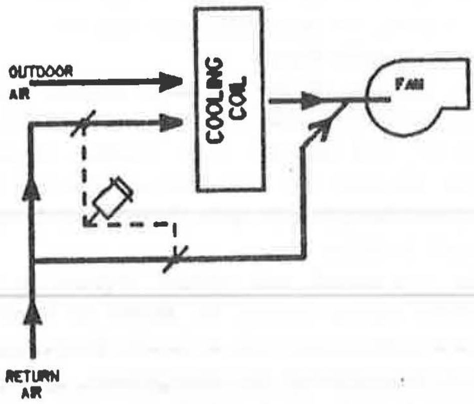

Figure 11 *Return air face and bypass.*

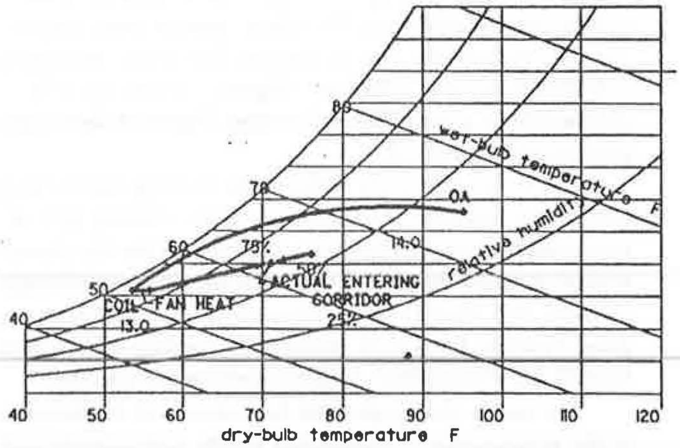

Figure 12 *Treated and recirculated exhaust with dual-path or return air coil bypass.*

Cool Storage

Cool storage involves the generation and storage of cooling during off-peak periods when kWh energy charges are lowest and the subsequent use of the stored cooling during daytime hours when kWh energy and kW demand charges are highest. Cool storage is a 50-year-old technology developed for the dairy industry. This technology received new emphasis in the early 1980s because it reduces operating costs for the owner, reduces the need for new electric generation plants, and significantly improves the air-conditioning load factor for the utility. It means that highly efficient base load plants can be used for the storage portion of the air-conditioning load in lieu of the less efficient oil or gas turbine-powered peaking generators. Cool storage in the U.S. is growing at the rate of 50% per year. More than one dozen manufacturers have developed products tailored for the HVAC industry.

The ideal cooling load profile for cool storage is shown in Figure 13. The ideal profile has a relatively high daytime load of short duration with little or no nighttime load. The hotel cooling load profile also shown in the figure is a long-duration, relatively flat load from approximately noon until 8 p.m. A portion of the load continues throughout the night. With present utility rates, the entire hotel cooling load is not a good candidate for cool storage, but an opportunity still exists to install cool storage for a portion of the load and use existing cooling technology for the balance of the load. Initially sizing the cool storage to handle the highest 30% of the load and then changing the size up and down in 5% increments to optimize the life-cycle cost are recommended.

Heat recovery from cool storage is an ideal application for hotels. The heat rejection from the refrigeration compressor(s) can be used to preheat domestic hot water in time for the heavy 7 a.m. to 9 a.m. guest demand.

Cold Air Distribution for Public Space and Back-of-the-House Areas

Cold air distribution involves the use of 42 to 46°F (6 to 8°C) supply air in lieu of the 51 to 55°F (11 to 13°C) supply air temperature used in many hotel air-conditioning systems. Cold air distribution is possible with conventional water chillers by using deep cooling coils and operating water chillers at 40°F or lower leaving water temperature. Cold air distribution is becoming the system of choice when used with an ice cool storage system, which takes advantage of the lower water temperatures available from ice storage systems.

Cold air distribution reduces demand kW and in most cases energy kWh. It reduces the cost of ductwork and the space required for ductwork. It may even reduce the number of air-handling units. The reduction in air quantity may also reduce sound level in public spaces or the cost of sound attenuators.

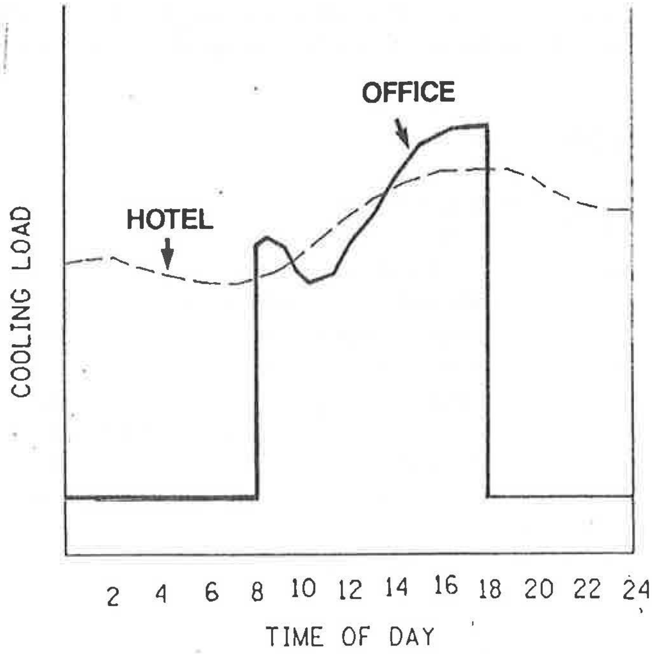

Figure 13 Cooling load profile.

A significant benefit of cold air distribution in the public space and back-of-the-house areas of a hotel is lower space relative humidities. This improves the comfort of guests and employees and could contribute to higher food and beverage sales and greater convention and meeting space sales. Reduced relative humidity will reduce the possibility of surface mold and the associated costs of preventing and removing mold and restoring surface finishes.

Desiccant Dehumidifiers

Desiccant dehumidifiers are a proven technology for drying air. As air passes over the desiccant material, it is dried and the latent heat is converted to sensible heat; therefore, the air leaves the desiccant bed much drier and much warmer than when it entered. The sensible heat must then be removed by heat recovery wheels, indirect evaporative cooling, chilled-water or refrigerant cooling coils, or a combination of methods. Obviously, the overall energy effectiveness of the desiccant dehumidifier cycle is improved when a major portion of the converted latent heat (now sensible heat) is removed without using chilled-water or refrigerated cooling.

The desiccant must be continuously regenerated by heating the desiccant to drive out the absorbed water vapor. Natural gas or steam is the common energy choice for heating the desiccant. Overall energy effectiveness of the regeneration section is improved with heat recovery wheels and the use of waste heat from refrigerant hot gas or other available sources. These low-temperature heat sources can displace a portion of the high-temperature energy required for regeneration. Higher temperature energy sources are required for the last stage of regeneration.

CONCLUSION

Improved dehumidification systems and equipment should be considered in designing HVAC systems for new hotels and for retrofitting existing hotels in order to reduce the cooling season relative humidity. A reduction in cooling season relative humidity will significantly improve comfort and will minimize or eliminate mold formation in hotels. The benefits of comfort, lower operating costs, improved guest perception and loyalty, and value-engineered first costs justify the inclusion of optimized dehumidification systems and equipment in hotel designs. Treated and recirculated exhaust and cold air technologies have demonstrated lower first costs in addition to their other benefits, including lower space relative humidities.

All of the technologies discussed in this paper have been proved in the field for more than 10 years. With any difficult problem, such as the moisture problem in hotels, the key to its solution is first an understanding of the fundamentals behind the problem and then, after correctly stating the problem, listing the possible solutions. This paper has presented a range of options. The synergistic combination of those options and the evaluation of the best option or combinations should be tailored to each project.

REFERENCE

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