#4648

ENERGY IMPACT OF VARIOUS INSIDE AIR TEMPERATURES AND HUMIDITIES IN A MUSEUM WHEN LOCATED IN FIVE U.S. CITIES

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ABSTRACT

The art conservation literature presents a wide range of recommended temperatures and relative humidities required to protect the safety of collections in museums, but the operating energy costs for specific criteria have not been identified. The Scott Gallery at the Huntington Library and Art Gallery in San Marino, CA, was selected for a detailed study of energy costs associated with recommended environmental levels for museums. The results of computer simulations of the Scott Gallery when located in Albuquerque, NM; Burbank, CA; Minneapolis, MN; New Orleans, LA; and New York, NY are presented. The simulations were performed using the DOE-2 building energy analysis computer program. The peak heating and cooling load components are identified, thermal zone loads quantified, and psychrometric analysis of the annual energy requirements with fixed and variable inside air temperature and relative humidity (RH) setpoints are presented. In all five climate regions the minimum energy consumption occurred with a 70°F and 50% RH setpoint.

BACKGROUND

The maintenance of appropriate environmental conditions in museums requires the resolution of several conflicts of interest. On the one hand, the art collections must be protected from deterioration and damage due to improper artificial lighting or exposure to sunlight, fluctuating air temperatures and humidities, particulate and gaseous pollutants, and physical damage from earthquakes and other disasters. On the other hand, and of equal importance, the collection must be displayed and lighted to meet important educational criteria, the public areas must be aesthetically pleasing and comfortable, and the museum must be economical to operate. To obtain a balance of these often conflicting needs, compromises are made that unknowingly result in the design and construction of new or rehabilitated museum buildings that often do not operate as intended and incur unmanageable utility bills.

In the past, conservators involved in architectural planning of new construction or modifications to existing museums have been limited to the role of specifying safe levels of temperature, relative humidity, and lighting. However, a dialogue often is missing between museum professionals (conservators, curators, and museum directors) and the traditional design team (architects, designers, and engineers) on the energy impact of alternative building designs and inside air temperatures and relative humidities. Since the conservation literature, taken as a whole, provides an array of acceptable levels (Ayres et al. 1988), the museum professional is often left with an incomplete understanding for selecting the best of those recommended levels for the situation.

This study takes a well-characterized museum building of modern construction and, with computer simulations, determines the cost sensitivity of operating that structure at various locations around the United States with different baseline environmental parameters. For example, the following questions were asked:

1. What are the energy cost implications when changing the museum's relative humidity (RH) setpoint from 50% to 40% or from 50% to 60% ±2%? In the United States, some pressure has been exerted toward establishing archival standards within the 25% to 35% range for paper and leather, and 40% to 45% for parchment and vellum (Wilson 1988). Yet, earlier limits for the same materials are suggested at 40% to 50% for paper and 55% to 60% for parchment and vellum (Stolow 1977).
2. Maintaining the setpoint relative humidity at 50% and the temperature at 70°F (21°C), can savings be made by reducing the acceptable control from ±2% to ±5% and ±7%?
3. What savings might be realized by permitting the temperature to be set at 65°F (18°C) in the winter, 75°F (24°C) in the summer, and 70°F (21°C) in the fall and spring? In a location such as New York City, engineers and conservators alike intuitively suspect that by relaxing the seasonal temperature requirements for the museum environment, the energy costs might be reduced.
4. What savings can be achieved by using energy-saving heat recovery chillers with indoor air setpoints of 70°F (21°C) and 50% RH ±2% in New York (colder) and Burbank, CA (milder)?

BUILDING MODEL

Several museums in southern California were identified by a conservation institute as candidates for field trips to meet with conservators, building operating engineers, and administrative planners. The intent of the field trips was to familiarize the authors with the history of the building design; the problems with the installed heating, ventilating, and air-conditioning (HVAC) systems; corrective measures taken (if any); and the role of the owners, administrators, and building operating engineers. In addition, each museum was examined to determine if the building could be classified as "typical" for use in subsequent computer simulation studies. The Virginia Steel Scott Gallery at the Huntington Library and Art Gallery in San Marino, CA–being small, of relatively simple architecture, and recently constructed–was selected for detailed studies. In addition, the original construction

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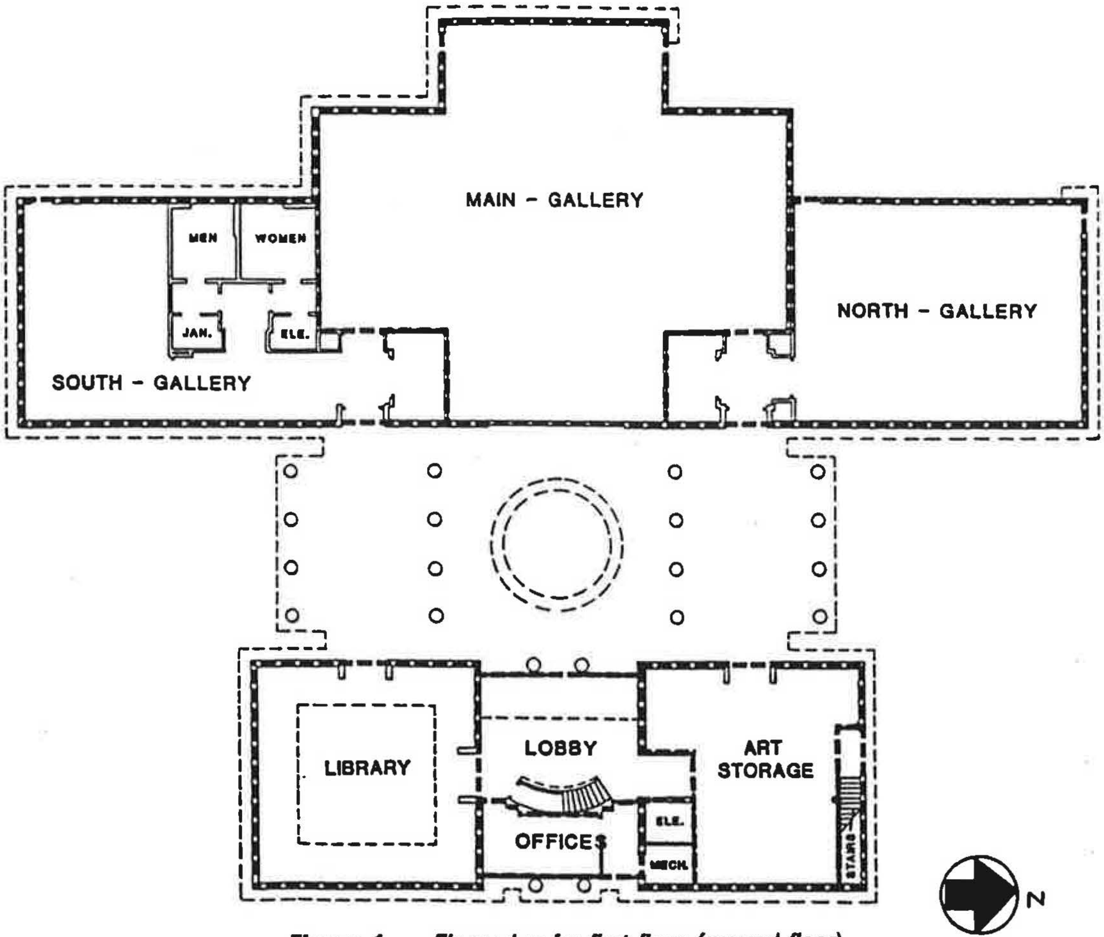

Figure 1 Floor plan for first floor (ground floor)

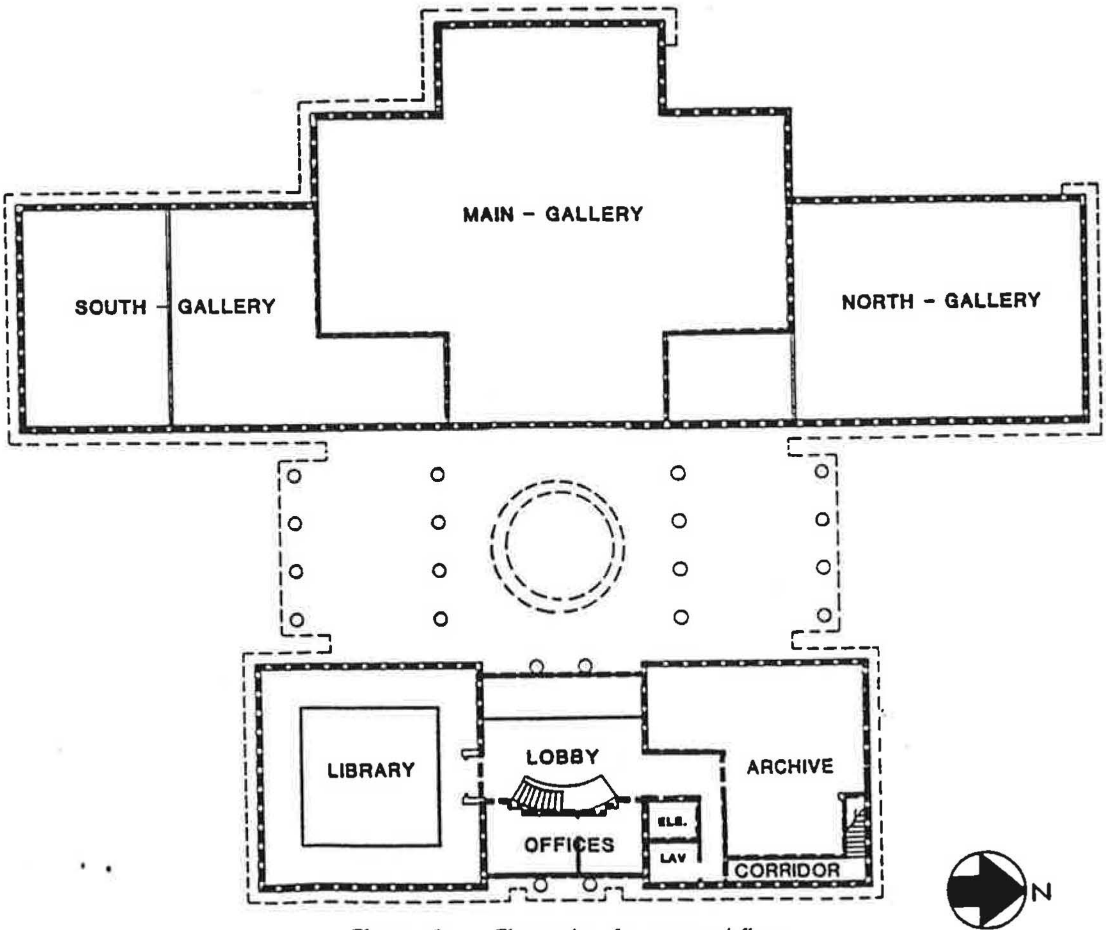

Figure 2 Floor plan for second floor

documents (drawings and specifications) were available, and the building construction manager had become the building operator, so the history of any construction problems and their solutions were known.

The Scott Gallery was field examined several times to obtain detailed historical information, and to confirm the architectural, structural, mechanical, and electrical construction documents. The floor and roof plans of the building are shown in Figures 1 through 3. The single-story west building (gallery) with skylights contains three high-ceilinged galleries (north, south, and main), and the two-story east building (administration) contains office, library, and art storage areas. The space between the buildings is roofed over, but is open to the weather at each end. Note that the amount of glass in the walls is limited to the administration building and only the main gallery in the gallery building has skylights. The facility was constructed in 1983 and was opened to the public in the summer of 1984.

The north gallery is enclosed but was not finished, and is currently being used for storage. The existing HVAC system has duct stub-outs for future service to this area. In the computer simulations, it was assumed that the north gallery was finished in a manner similar to the south gallery (fewer toilets, janitor and electrical rooms). The Scott Gallery plans to finish the north gallery at a later date, so this area was included to complete the "typical" model.

The HVAC system consists of an unhoused central chilled-water and steam plant located at grade to the northwest of the north gallery. Steam and chilled-water lines are extended exposed on the roof to two roof-mounted air-handling units. Low-pressure steam is provided by a gas-fired boiler and chilled water by a packaged electric-driven compressor/chiller condenser. The HVAC systems are constant-air-volume cooling with steam reheat in each thermal zone. Humidifiers are installed in the ducts serving the north, south, and main galleries; art storage; and the library. The supply air downstream from the humidifiers is ducted down through the roof to ceiling supply diffusers, and is returned through ceiling inlets to ceiling plenums. Air is returned to the two air-handling units through above-roof sheet metal ducts. A fixed minimum amount of outside air is mixed with return air, which passes through fiberglass filters and activated carbon filters, and then enters the cooling coil. The temperature of the air leaving the cooling coil is reset and then reheated as required for humidity control in each zone.

SIMULATIONS

The HVAC loads and building energy requirements of the Scott Gallery were obtained by performing hourly computer simulations using the DOE-2.1C computer program (LBL 1984). The building construction elements are shown in Table 1. Note that the walls and roof are well insulated and the skylights are triple glazed.

Location and Climate Data

The cities selected for simulation in this study were based on weather extremes, availability of weather tapes, and the number of significant museums in a locality. Hourly weather data for San Marino are not available in magnetic form, so the California Energy Commission Climate Thermal Zone 09 (CTZ 09 Burbank) was used. The National Oceanic and Atmospheric Administration Test Meteorological Year (TMY) tapes were used for Albuquerque, NM (hot/dry); New Orleans, LA (hot/humid); Minneapolis/St. Paul Airport, MN (cold); and New York City Central Park, NY (significant museums). The ASHRAE outdoor design requirements

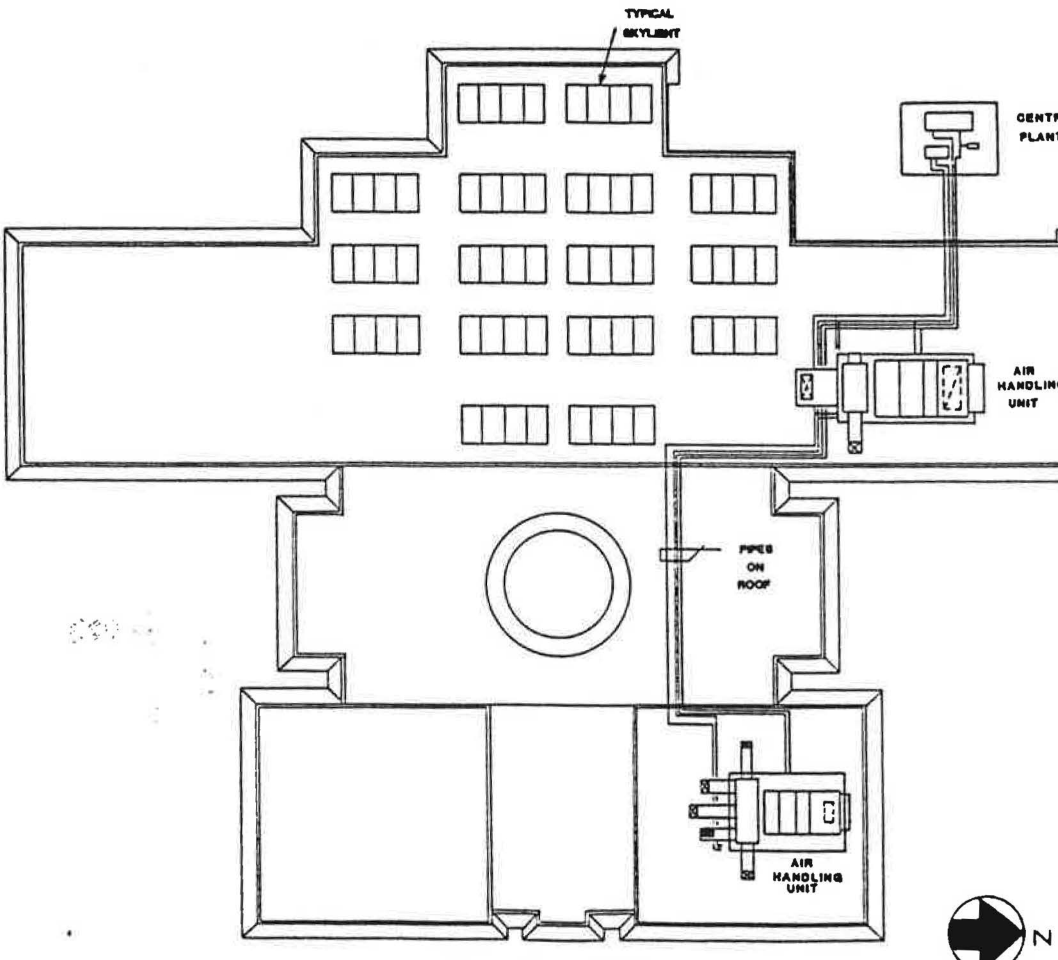

Figure 3 Roof plan

TABLE 1
Construction Elements

Roof (Gallery) Gravel, Built-up Roofing, 12 in. (0.3 m) Lightweight Concrete, 3 in. (0.08 m) Polystyrene Board, Air Space, 1/8-in. (3 mm) Glass.	**Roof (Administration)** Gravel, Built-Up Roofing, 3 in. (0.08 m) Polystyrene Board, R-19 Insulation, Air Space, 5/8-in. (16 mm) Gypsum Board 1/4-in. (6 mm) Clear Glass.
Floor (Gallery) 4 in. (0.10 m) Concrete, 1/2-in. (14 mm) Oak.	**Floor (Administration)** 4 in. (0.10 m) Concrete, Carpet.
Exterior Wall 5/8-in. (16 mm) Portland Cement 12 in. (0.3 m) Concrete Block, 1 1/2 in. (0.04 m) Rigid Insulation, 5/8-in. (16 mm) Gypsum Board.	**Curtain Wall** 1/4-in. (6 mm) Curtain Wall, Air Space, 5/8-in. (16 mm) Gypsum Board.
Metal Door 3/8-in. (10 mm) Steel, Air Space, 3/8-in. (10 mm) Steel.	**Wooden Door** 1 3/4 in. (0.04 m) Hardwood.
Exterior Glass (Lobby) 1/4-in. (6 mm) Tinted Gray, Shading Coefficient = 0.69, U-value = 1.13 Btu/ft^2·h·°F, = 6.42 W/m^2·°C.	**Exterior Glass (Archive)** 1/4-in. (6 mm) Clear Glass, Shading Coefficient = 0.69, U-value = 1.13 Btu/ft^2·h·°F, = 6.42 W/m^2·°C.
Skylights 1/4-in. (6 mm) Clear Glass, 1/2-in. (14 mm) Air Space, 1/4-in. (6 mm) Laminated Translucent White, Air Space, 1/8-in. (3 mm) Glass, Shading Coefficient = 0.49, U-value = 0.32 Btu/ft^2·h·°F, = 1.82 W/m^2·°C.	

(conditions exceeded no more than 21/2% of summer hours and no more than 1% of winter hours) and other data for the five locations are shown in Table 2.

Building Envelope

It was assumed that the building envelope designed for the mild southern California weather would have to be upgraded by adding insulation to meet the weather extremes in the other locations. ASHRAE Standard 90 (ASHRAE 1980) envelope requirements do not treat the entire building envelope as a whole; thus, it does not allow the thermal characteristics (U-value and OTTV) of any building component to be increased and other building components decreased as long as the overall thermal characteristic of the entire building envelope does not exceed the allowable value. To overcome this deficiency, the ASHRAE Standard 90 version contained in the California Title 24 Enerqv Conservation Standards (CEC 1987) was used to establish the envelope requirements.

It was determined that the conservative envelope design met the California Title 24 standards in all five locations. The results of these calculations are shown in Table 3. Note that each of the individual components (walls and roof) of the administration building meet or exceed the Title 24 requirements for heating (U-value) and cooling (OTTV) criteria. In the gallery building, however, the walls (without any windows) easily met the standards, but the roof could not meet the heating criteria in Minneapolis and New York (because of the large skylight area) or the cooling criteria in any of the locations. However, the standards are concerned with the thermal resistance of the entire building envelope, and when all of the components were area-weighted average, the envelope met the standards in all locations.

Internal Loads

The maximum number of people in each thermal zone and the usage schedules were developed from information supplied by the building operator and the conservation institute's review comments on event people densities. The various occupancies in each building are shown in Figures 4 through 6. People in offices and work areas were assumed to be seated and performing a minimum of work, while people in exhibition areas were assumed to be walking slowly. Lighting loads for the various spaces were obtained from the construction drawings, and the number of fixtures in lighting strips were determined by an actual field count.

Thermal Zones

The boundaries of each thermal zone are shown in Figures 1 and 2, and the area, space setpoints, internal loads, usage schedules, and the HVAC systems serving each zone are presented in Table 4. Infiltration of outside air through building cracks and openings was assumed to be zero, which is consistent with the necessary positive pressurization of museums.

HVAC Systems

The HVAC system is constant volume, fixed outside air, with zone reheat and humidification. The roof-mounted equipment, piping, and ductwork were considered to be properly insulated and losses to the outside air were ignored. The simulations were not designed for detailed comparison with field measurements of the performance of the HVAC systems. They were developed for generic comparisons of the model performance when placed in different cities, and were subjected to varying inside air temperature and relative humidity zone control setpoints. The HVAC system parameters in each building when located in the various locations are presented in Table 5.

Design Day Cooling Loads

To calculate the peak cooling load in each zone, the data from the weather tapes were studied to determine the month with the highest dry-bulb temperature. The building was then simulated for five months (the warmest month plus two months before and after) to determine the peak cooling load in each thermal zone. The output of the cooling load simulation provided itemized, sensible, and latent cooling loads for roofs, walls, doors, glass conduction, glass solar, floor slab, and other internal heat gains.

TABLE 2
Design Conditions

	Albuquerque	Burbank	Minneapolis	New Orleans	New York
Longitude, degrees	106	118	93	90	74
Latitude, degrees	35	34	45	30	40
Altitude, ft	5310	699	822	3	132
Summer Dry Bulb, °F	94	91	89	93	89
Summer Wet Bulb, °F	61	68	73	78	73
Winter Dry Bulb, °F	12	37	-16	29	11
Weather Tape	TMY	CEC	TMY	TMY	TMY

TABLE 3
Comparison of Existing Building Envelopes and Title 24 Requirements in Various Locations

Administration Building Envelope												
Location	Existing Envelope						Required Envelope					
	U_{wall}	U_{roof}	U_{enve}	$OTTV_{wall}$	$OTTV_{roof}$	$OTTV_{enve}$	U_{wall}	U_{roof}	U_{enve}	$OTTV_{wall}$	$OTTV_{roof}$	$OTTV_{enve}$
Scott Gallery	0.204	0.042	0.143	9.92	1.15	6.64	0.44	0.100	0.313	31.8	4.10	21.43
Albuquerque, NM	0.204	0.042	0.143	10.43	1.15	6.96	0.37	0.090	0.265	32.1	3.69	21.46
Burbank, CA	0.204	0.042	0.143	9.92	1.15	6.64	0.44	0.100	0.313	31.8	4.10	21.43
Minneapolis, MN	0.204	0.042	0.143	9.98	1.15	6.67	0.26	0.060	0.185	34.9	2.46	22.75
New Orleans, LA	0.204	0.042	0.143	10.00	1.15	6.69	0.45	0.100	0.319	30.7	4.10	20.74
New York, NY	0.204	0.042	0.143	9.82	1.15	6.57	0.36	0.086	0.257	33.8	3.53	22.46

Gallery Building Envelope												
Location	Existing Envelope						Required Envelope					
	U_{wall}	U_{roof}	U_{enve}	$OTTV_{wall}$	$OTTV_{roof}$	$OTTV_{enve}$	U_{wall}	U_{roof}	U_{enve}	$OTTV_{wall}$	$OTTV_{roof}$	$OTTV_{enve}$
Scott Galley	0.055	0.089	0.070	2.07	11.22	6.10	0.44	0.100	0.290	31.8	4.10	19.59
Albuquerque, NM	0.055	0.089	0.070	2.07	11.37	6.17	0.37	0.090	0.247	32.1	3.69	19.58
Burbank, CA	0.055	0.089	0.070	2.07	11.22	6.10	0.44	0.100	0.290	31.8	4.10	19.59
Minneapolis, MN	0.055	0.089	0.070	2.07	11.12	6.06	0.26	0.060	0.172	34.9	2.46	20.60
New Orleans, LA	0.055	0.089	0.070	2.07	11.27	6.13	0.45	0.100	0.296	30.7	4.10	18.98
New York, NY	0.055	0.089	0.070	2.07	11.12	6.06	0.36	0.086	0.239	33.8	3.53	20.46

Note: U_{wall} - average thermal transmittance of the gross wall area, Btu/ft^2.h.°F
U_{roof} - average thermal transmittance of the gross roof area, Btu/ft^2.h.°F
U_{enve} - average thermal transmittance of the building envelope, Btu/ft^2.h.°F
$OTTV_{wall}$ - overall thermal transfer value for walls, Btu/ft^2.h.°F
$OTTV_{roof}$ - overall thermal transfer value for roof, Btu/ft^2.h.°F
$OTTV_{enve}$ - overall thraml transfer value for the building envelope, Btu/ft^2.h.°F

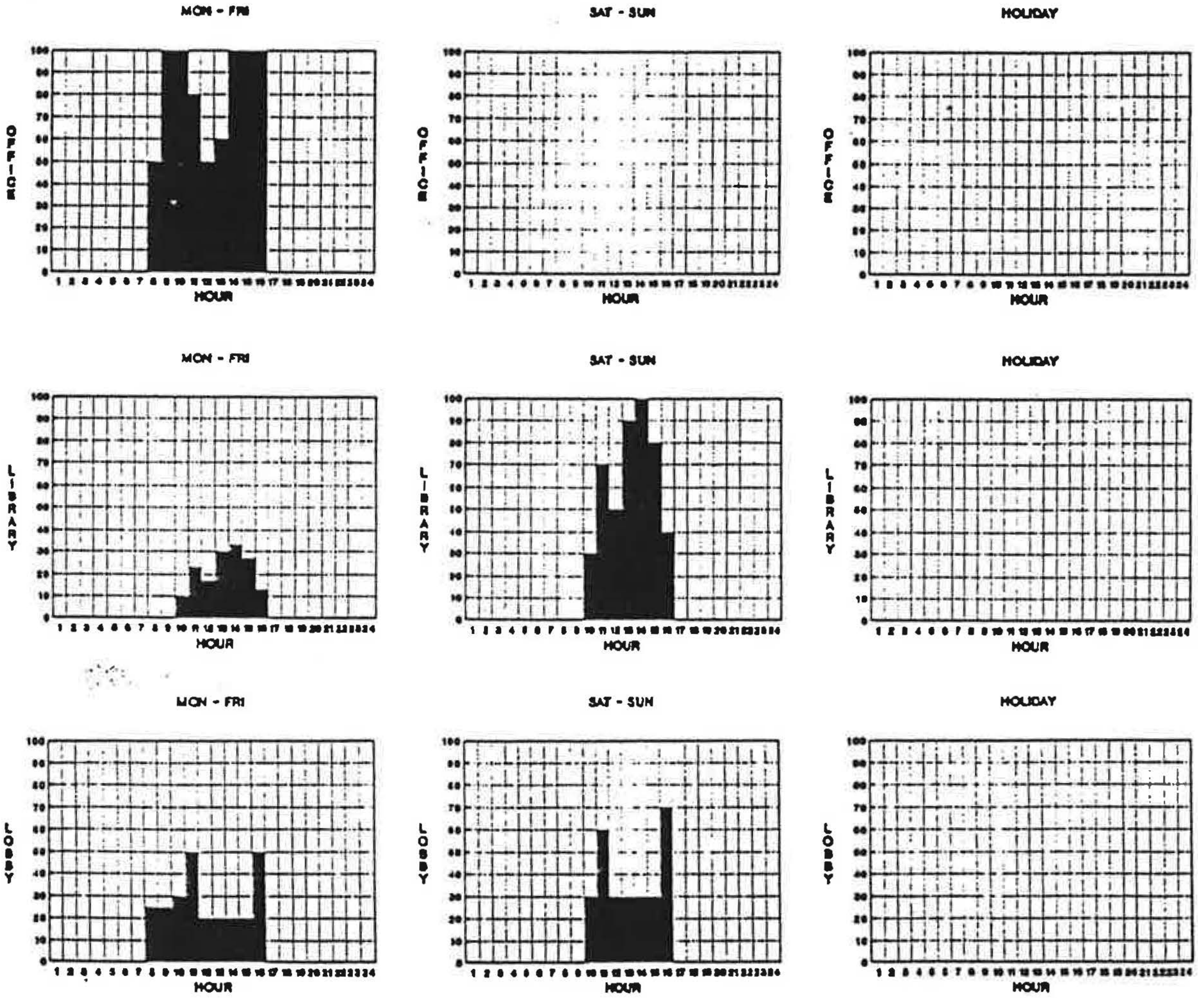

Figure 4 People schedules–administration

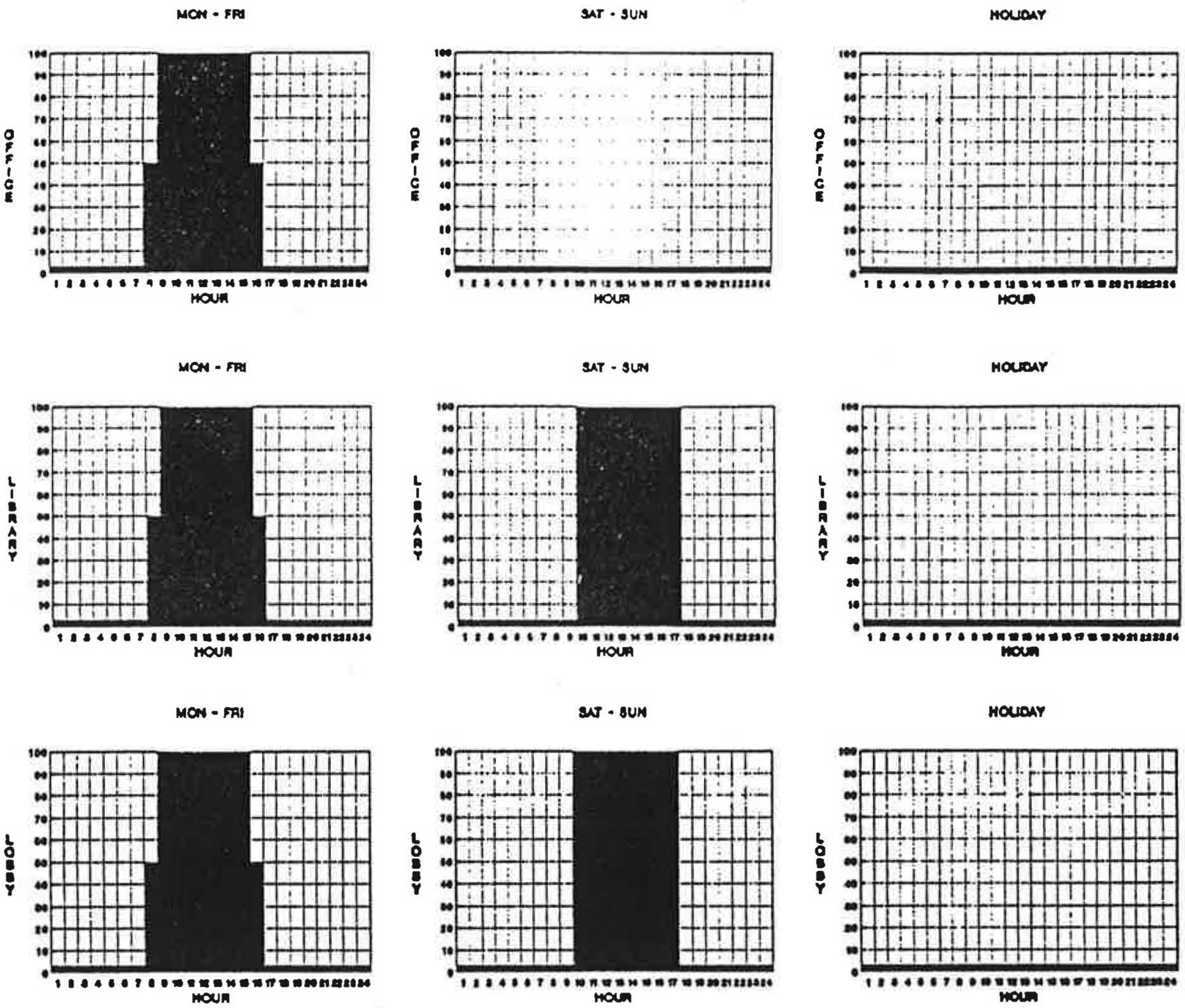

Figure 5 Lights and equipment schedules–administration

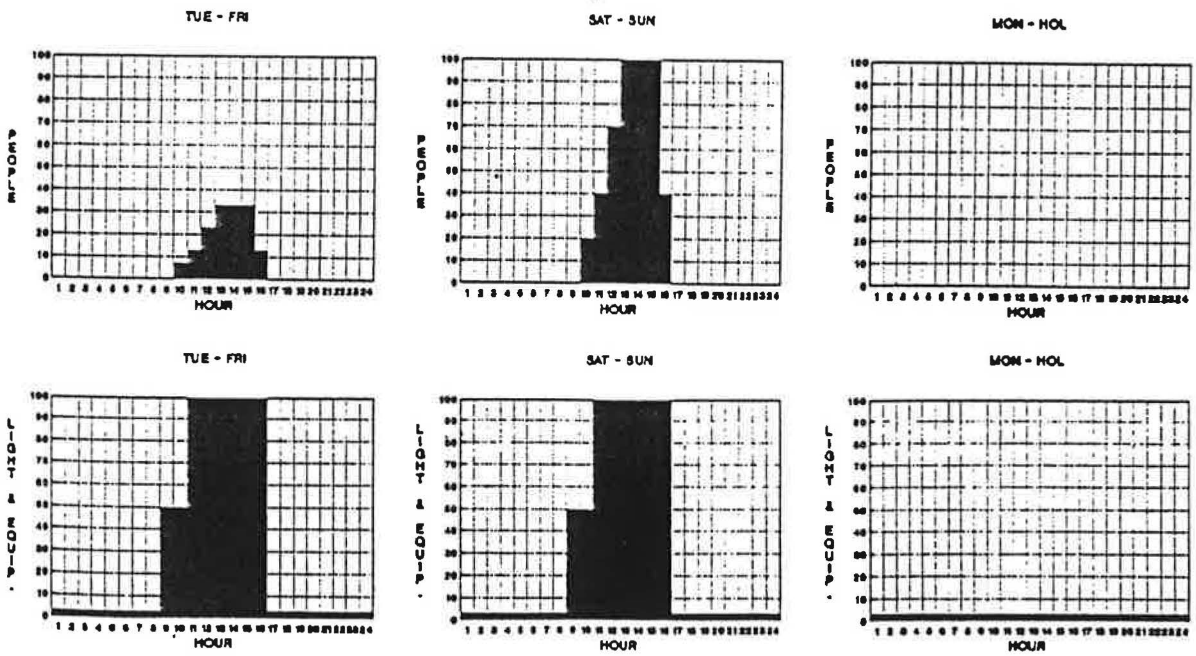

Figure 6 People and light schedules–gallery

TABLE 4
Summary of Zone Data

Thermal Zone Name	Area (ft²)	Space Condition Temp. (°F)	Space Condition RH,2(%)	No. of People	Activity Sen/Lat (Btu/Per)	Light (W/ft²)	Light To Space	Equipment (W/ft²)	Equipment To Space	Usage Schedule People	Usage Schedule Light/Eq.	HVAC System Type	HVAC System Sched	Exh (CFM)
Library	2164	70	50	01	255/255	1.22	100%	0.0	n/a	Library	Library	CVRH	24 hr	0
Art Stg.	1108	70	50	01	255/255	1.78	100%	0.0	n/a	Office	Office	CVRH	24 hr	0
Lobby	1116	70	50	15	255/255	2.04	100%	0.5	100%	Lobby	Lobby	CVRH	24 hr	188
Archive	982	70	50	02	255/255	1.07	100%	0.5	100%	Office	Office	CVRH	24 hr	0
Offices	565	70	50	05	255/255	2.68	100%	0.1	100%	Office	Office	CVRH	24 hr	0
Cor-Str.	208	70	50	01	255/255	2.09	100%	0.0	n/a	Office	Office	CVRH	24 hr	0
S-Gal.	2177	70	50	50	315/325	1.12	100%	0.0	n/a	Gallery	Gallery	CVRH	24 hr	800
N-Gal.	2177	70	50	57	315/325	1.02	100%	0.0	n/a	Gallery	Gallery	CVRH	24 hr	0
M-Gal.	4180	70	50	116	315/325	1.09	100%	0.0	n/a	Gallery	Gallery	CVRH	24 hr	331

Notes: 1. CVRH System = constant-volume reheat system.
1. Parametric runs with RH set at 60% and 40%,2% were also performed.

A typical breakdown of peak cooling loads resulting from the New York simulations, with indoor dry-bulb temperature at 70°F (21°C) and 50% RH, is shown in Tables 6 through 8 for the south gallery, main gallery, and offices. Note that 96% of the total heat gain (sensible + latent) in the south gallery was from internal loads (lights and people), 75% of the total heat gain was from people, and the sensible heat ratio (SHR) was 50%. The north gallery without skylight is similar to the south gallery. In the main gallery, only 47% of the total heat gain was from internal loads because most of the heat gain was through the skylights, and the SHR was 77%. In the offices, the SHR was 94% because the people load was small.

Psychrometric Analysis

An important component of the psychrometric analysis is the SHR, which is shown as a sloping line on a psychrometric chart between the supply air (SA) and the room air (RA). The outside air is mixed with room air, which is passed over the cooling coil and then reheated as required to provide the necessary SA conditions. To determine the required cfm for each space, a straight line at a known SHR is extended from the RA point toward the saturation curve. If the line cannot intersect the saturation line, it means that the air must be overcooled to remove the moisture, then reheated before entering the space. The SA point is determined by extending a horizontal line (representing the addition of sensible heat) from the leaving coil point (CL) to intersect the SHR line at the required SA point (62°F dry-bulb and 53°F wet-bulb in the north and south galleries). With the SA temperature established, the cfm required in each space was calculated. With the total supply cfm determined, an air balance on the building was performed by deducting the air exhausted through the toilets and exfiltrated through doors (due to pressurization) to establish the minimum amount of outside air.

The galleries were also analyzed on weekdays, when the number of visitors is one-third that on weekends. To adjust the cooling load in each of the galleries, the heat gain due to people was reduced, and the sensible heat ratios were recalculated. The ratio in the main gallery increased to 90% and that in the south gallery to 60%. Under these conditions, the cold deck temperature was reset upward to 49°F (9°C).

When the gallery is closed at night and on Mondays and holidays, there is no internal load (lights are off and no people are in the building), and the only gain is sensible heat through the envelope. It was assumed that there was no vapor flow through the building envelope. The moisture

TABLE 5
HVAC System Parameters in Various Locations

Location	RH (%)	Administration System[1] Supply Air (CFM)	Supply Air Temp. (°F) Max	Min	Reset	Gallery System[2] Supply Air (CFM)	Supply Air Temp. (°F) Max	Wkday	Wkend	Night
Albuquerque, NM	40	5770	110	43	Warmest	12,200	110	43	38	45
	50	7800	110	50	Warmest	12,450	110	50	47	50
	60	10370	110	55	Warmest	17,450	110	55	53	55
Burbank, CA	40	4635	110	43	Warmest	9850	110	43	38	44
	50	6260	110	50	Warmest	9800	110	49	45	49
	60	8330	110	55	Warmest	13,900	110	55	52	55
Minneapolis, MN	40	4410	110	43	Warmest	11,900	110	43	38	45
	50	5710	110	50	Warmest	10,900	110	49	46	50
	60	7510	110	55	Warmest	13,900	110	55	52	56
New Orleans, LA	40	4130	110	43	Warmest	10,100	110	42	38	44
	50	5550	110	50	Warmest	8650	110	49	44	51
	60	7400	110	55	Warmest	13,050	110	55	52	55
New York, NY	40	4620	110	43	Warmest	9900	110	43	38	45
	50	6210	110	50	Warmest	9550	110	49	45	51
	60	8270	110	55	Warmest	13,450	110	55	53	56

Notes: 1. Administration System:
Supply fan 2.5 in. static pressure with 61% overall efficiency.
Outside air = 10% of supply air.

2. Gallery System:
Supply fan 2.5 in. static pressure with 49% overall efficiency.
Outside air at constant 2310 cfm.
Wkday = Open hours (10 a.m. - 5 p.m.) on weekdays.
Wkend = Open hours (10 a.m. - 5 p.m.) on weekends.
Night = Closed hours (5 p.m. - 10 a.m.) daily and all hours on Mondays and holidays.
Outside air reduced to 300 cfm at closed hours.

TABLE 6
South Gallery, New York-70°F, 50% RH, Itemized Cooling Loads

Item	Sensible, Btu/h	Latent, Btu/h
Walls	2500	0
Roofs	1064	0
Glass Conduction	0	0
Glass Solar	0	0
Doors	0	0
Floor Slab	-3500	0
People to Space	14,731	20,150
Light to Space	3584	0
Equipment to Space	0	0
Total	20,120	20,150

$$\text{Sensible Heat Ratio} = \frac{\text{Sensible Heat}}{\text{Total Heat}} = \frac{20,120}{(20,120 + 20,150)} = 0.5$$

TABLE 7
Main Gallery, New York-70°F, 50% RH, Itemized Cooling Loads

Item	Sensible, Btu/h	Latent, Btu/h
Walls	46	0
Roofs	269	0
Glass Conduction	2625	0
Glass Solar	95,947	0
Doors	696	0
Floor Slab	-6839	0
People to Space	28,830	39,975
Light to Space	12,507	0
Equipment to Space	0	0
Total	134,079	39,975

$$\text{Sensible Heat Ratio} = \frac{\text{Sensible Heat}}{\text{Total Heat}} = \frac{134,079}{(134,079 + 39,975)} = 0.77$$

in the outside air (required to pressurize the building) must be removed before the air is introduced into the space. To accomplish this, the cold deck temperature was set to 51°F (11°C), and the supply air was reheated to satisfy the space sensible load.

The analysis for New York (with inside design conditions of 70°F [21°C] and 40% RH) indicated that a CL temperature of 38°F (3°C) was needed to minimize fan and reheat energy and the chilled-water system had to be replaced with a direct expansion refrigeration system. See Ayres et al. (1988) for detailed analyses of the peak loads, cfm, and psychrometrics.

Simulations

The calculated cfm for each thermal zone at 70°F (21°C), the total cfm provided by each air-handling unit, and the deck temperature schedules determined from the psychrometric analysis were input into DOE-2. The program was then processed for an entire year (8760 hours) of weather simulations. The hourly heating and cooling loads for each thermal zone and the HVAC system coil loads were obtained from the outputs.

The model building annual energy requirements at each of the five locations were determined for room temperatures of 70°F (21°C) and relative humidities of 40%, 50%, and 60% with a tolerance of ±2%. To investigate the impact of the relative humidity tolerance on energy consumption, the New York simulations were rerun with the RA held at 70°F (21°C) and 50%; the tolerances were changed to ±5% and ±7%.

It has been suggested by both conservators and design engineers that one might save energy by lowering the indoor temperature in the winter and raising it in the summer. To evaluate such a suggestion, the New York simulation was rerun with the relative humidity held at 50% ±2% and room temperature set at 65°F (18°C) in the winter, 70°F (21°C) in the spring and fall, and 75°F (24°C) in the summer. The potential energy savings of heat recovery chillers were evaluated in New York and Burbank by rerunning the simulations with room temperatures set at 70°F (21°C) and 50% RH ±2% using a double-bundle heat recovery chiller.

RESULTS

The HVAC system fan energy, chilled water for cooling, and steam for heating and humidification required to maintain 70°F at 40%, 50%, and 60% RH in the administration and gallery buildings in the five locations are presented in Table 9. Note that the air-handling unit serving the

TABLE 8
Offices, New York-70°F, 50% RH, Itemized Cooling Loads

Item	Sensible, Btu/h	Latent, Btu/h
Walls	1541	0
Roofs	125	0
Glass Conduction	3019	0
Glass Solar	14,061	0
Doors	0	0
Floor Slab	-2142	0
People to Space	755	1275
Light to Space	3672	0
Equipment to Space	0	0
Total	21,030	1275

$$\text{Sensible Heat Ratio} = \frac{\text{Sensible Heat}}{\text{Total Heat}} = \frac{21,030}{(21,030 + 1275)} = 0.94$$

TABLE 9
HVAC System Annual Energy Requirements at 70°F Inside Air Temperature

Location	RH (%)	Administration System			Gallery System		
		Fan (kWh)	Chilled Water (MBtu)	Steam (MBtu)	Fan (kWh)	Chilled Water (MBtu)	Steam (MBtu)
Albuquerque, NM	40	19,852	423.093	387.053	55,147	3090.700	2780.649
	50	26,775	370.920	399.171	66,445	2390.353	2247.828
	60	35,569	334.616	465.124	92,387	2556.716	2386.630
Burbank, CA	40	18,891	759.851	542.415	52,950	2633.757	2269.874
	50	25,477	576.029	381.984	52,689	2083.208	1589.174
	60	33,860	434.944	285.375	73,968	2181.371	1692.460
Minneapolis, MN	40	17,975	423.190	584.924	63,592	2809.395	2982.060
	50	23,230	359.847	574.946	58,395	1980.238	2245.526
	60	30,523	303.082	600.714	73,970	1862.926	2158.973
New Orleans, LA	40	17,349	899.169	589.451	54,248	3061.447	2141.259
	50	23,269	874.208	550.683	46,723	2060.389	1275.233
	60	30,984	836.568	506.354	69,556	2349.749	1550.382
New York, NY	40	19,392	577.252	578.913	53,210	2453.540	2372.637
	50	26,023	493.307	535.967	51,389	1786.471	1806.397
	60	34,611	476.328	533.318	71,634	1906.613	1939.761

Note: 1 kWh = 3,413 Btu.

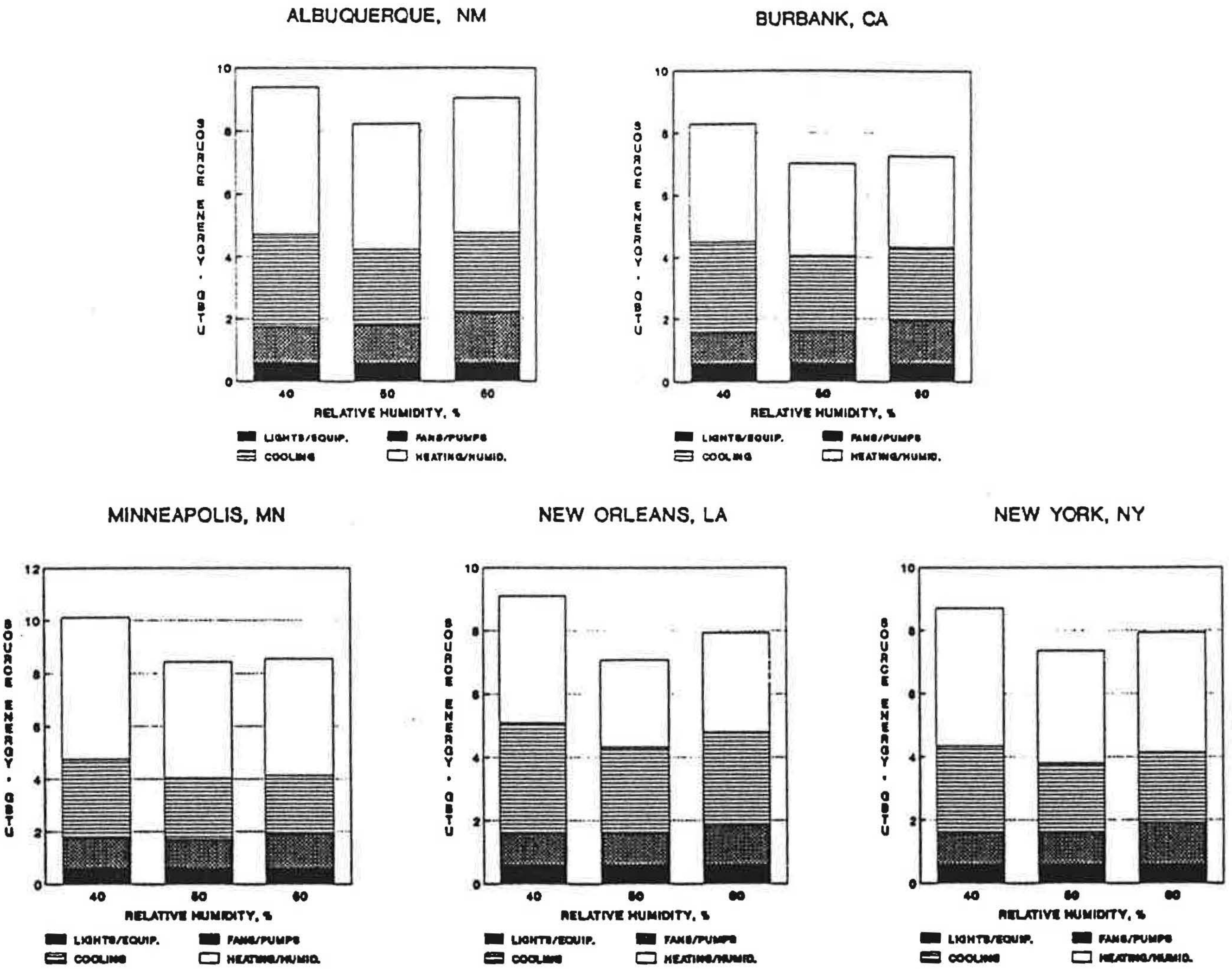

Figure 7 *Itemized annual energy consumption for different relative humidities in various locations*

administration building consumed less fan energy as the relative humidity was reduced from 60% to 40% because less supply air was required at a lower SA temperature. The system was specified to reset the temperature based on the requirement of the worst zone, which results in overcooling of the other zones. This explains why the chilled-water requirements increased as the relative humidity was decreased. Also note that total cooling requirements were considerably higher in New Orleans than in the other locations, primarily because of the warm and humid weather. The amount of steam required depends on the heating, reheat, and humidification loads. In Burbank, New Orleans, and New York the reheat energy was the dominant factor, and the steam requirement increased as the relative humidity was decreased. On the other hand, humidification became the dominant factor in Albuquerque, which is very dry, and the steam requirement increased as the indoor relative humidity was increased. In Minneapolis, where it is extremely cold in the winter, the heating, reheat, and humidification loads all had an influence on the annual steam requirement, which increased as the relative humidity increased from 50% to 60% and also increased as the relative humidity was reduced from 50% to 40%.

Note that the air-handling unit serving the galleries consumed more fan energy at 60% RH than at 50% in all five locations. This was because a higher supply air temperature was required for 60%, and therefore more air was required to meet the space loads. The additional supply air for the 60% RH also resulted in additional fan heat removal and increased cooling energy. As the relative humidity was lowered from 50% to 40%, lower leaving coil temperatures were required. The slope of the SHR line (SHR = 0.77) for the main gallery in New York was so steep that it could not intersect the saturation line, and the space had to be overcooled with added supply air and then reheated to reduce the moisture released by people. Note that the fan energy increased as relative humidity was reduced from 50% to 40% in all locations except Albuquerque, where it is hot and dry, and the SHR was 0.81. Because of overcooling and reheating, additional chilled water and steam were required to maintain 40% RH.

The energy that must be purchased from the electric and gas utilities is provided by the output from the DOE-2 plant simulations. The plant simulations include the inefficiencies of the central heating and cooling plant equipment, distribution losses, and other miscellaneous equipment, and provide monthly and annual summaries of demand and consumption. For comparison purposes, however, the electrical consumptions were converted to source energy based on 1 kWh = 10,239 Btu, which is equivalent to a utility power plant efficiency of 33.3%. Figure 7 graphically compares the energy consumption by components for 70°F (21°C) and 40%, 50%, and 60% RH in the five locations. Note that 50% RH control has the lowest energy consumption in all locations. Lights and equipment contribute 6% to 7% of the total energy consumption, and fan and pumps

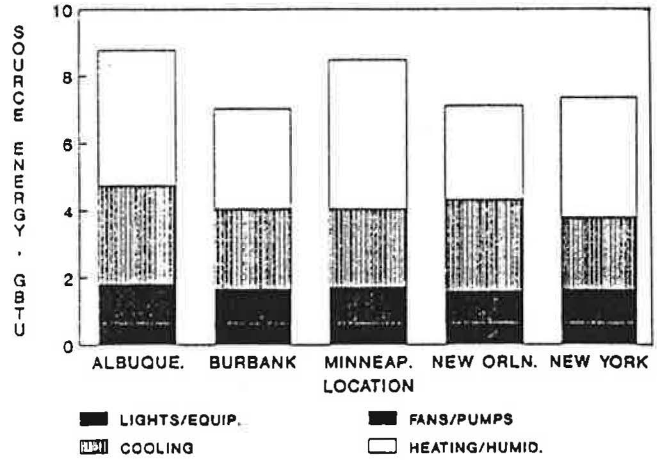

Figure 8 *Itemized annual energy consumption in various locations–70°F 50% RH*

add about 12% in all five locations. However, cooling energy varies between 26% in Minneapolis (with 759 cooling degree-days) and 39% in New Orleans (with 2563 cooling degree-days). The heating varies between 53% in Minneapolis (with 8095 heating degree-days) and 39% in New Orleans (with 1407 heating degree-days). Figure 8 graphically compares the itemized energy consumptions in the various locations with the indoor temperature set at 70°F (21°C) and RH at 50%. As expected, Burbank has the lowest energy consumption because of the mild weather, and Albuquerque has the highest because of the hot and dry summer and cold winter.

The annual electric and gas consumptions, in kWh and therms, at various locations are presented in Table 10. Using an average utility rate of $0.65/therm of gas and $0.10/kWh of electricity, the annual utility costs were also presented. Note that a 50% indoor RH setpoint had the lower utility cost in all five locations. However, the actual utility cost depends heavily on the local utility rate, especially the time-of-use rate with high on-peak demand and energy charges. To determine the actual utility cost, one must use the local utility rates.

Table 11 compares the itemized energy consumptions with an indoor temperature of 70°F (21°C) and relative humidities of 50% ±2%, ±5%, and ±7%. Note that there is only a slight reduction in energy consumption for all items as the humidity control tolerance is allowed to increase.

Table 12 shows that increasing the indoor air temperature from 70°F (21°C) to 75°F (24°C) in the summer and lowering it to 65°F (18°C) in the winter, while maintaining the RH at 50% throughout the year, resulted in a 1% reduction in cooling and 3% in heating/humidification energy consumption. The indoor air setpoint has no impact on energy consumption because most of the energy is used to maintain space RH by cooling and reheating the supply air.

Tables 13 and 14 compare the energy consumption of a conventional reciprocating chiller compressor with a heat recovery double-bundle reciprocating chiller compressor in New York and Burbank with the indoor air at 70°F (21°C) and 50% RH. The heat recovery chillers are less efficient than conventional chillers, so the cooling energy increased by 32% in New York and 38% in Burbank, but the heating and humidification energy was reduced by 88% in New York and 99% in Burbank. This indicates that in a mild climate almost all of the energy required for heating and humidification can be recovered by using double-bundle heat recovery chillers. It is interesting to note, however, that because the amount of heating and humidification energy required in a mild climate is less than in a colder climate, the total energy saved is greater in the cold climate. At $0.10/kWh and $0.65/therm, the saving is equivalent to $13,900 per year in New York and $11,000 in Burbank.

TABLE 10
Annual Utility Consumption and Cost

Location	RH (%)	Electricity (kWh)	Gas (Therm)	Utility Cost ($)
Albuquerque, NM	40	489,905	43,841	77,487
	50	439,093	37,232	68,110
	60	491,629	40,085	75,218
Burbank, CA	40	465,187	35,900	69,854
	50	415,621	27,529	59,456
	60	444,404	27,543	62,343
Minneapolis, MN	40	500,478	50,026	85,565
	50	424,164	40,533	68,763
	60	437,010	39,862	69,611
New Orleans, LA	40	522,568	37,481	76,620
	50	441,677	25,699	60,872
	60	491,152	28,966	67,943
New York, NY	40	452,757	40,671	71,712
	50	395,078	32,934	60,915
	60	432,101	35,057	69,997

Note: Utility cost based on $0.65/therm gas and $0.10/kWh electricity.

TABLE 11
Itemized Energy Consumptions for Various RH Control Tolerances New York–70°F (Source Energy, MBtu)

RH	Light & Equipment	Fans & Pumps	Cooling	Heating & Humidification	Total
50±2%	599	1023	2188	3558	7368
50±5%	599	1019	2141	3480	7239
50±7%	599	1017	2112	3424	7152

Note: 1kWh = 10,239 Btu source energy.

TABLE 12
Itemized Energy Consumptions for Various Indoor Air Temperatures New York–50% RH (Source Energy, MBtu)

Temp.	Light & Equipment	Fans & Pumps	Cooling	Heating & Humidification	Total
70°F	599	1023	2188	3558	7368
65-75°F	599	1023	2164	3459	7245
Diff.	0	0	24	99	123

Note: 1kWh = 10,239 Btu source energy.

TABLE 13
Comparison of Conventional and Heat Recovery Chillers New York–70°F, 50% RH (Source Energy, MBtu)

Chiller	Light & Equipment	Fans & Pumps	Cooling	Heating & Humid.	Total
Conventional	599	1023	2188	3558	7368
Heat Recovery	599	1023	2895	424	4941
Difference	0	0	707	-3134	-2427

Note: 1kWh = 10,239 Btu source energy.

TABLE 14
Comparison of Conventional and Heat Recovery Chillers Burbank–70°F, 50% RH (Source Energy, MBtu)

Chiller	Light & Equipment	Fans & Pumps	Cooling	Heating & Humid.	Total
Conventional	599	1038	2412	2960	7009
Heat Recovery	599	1038	3323	2	4941
Difference	0	0	911	-2958	-2047

Note: 1kWh = 10,239 Btu source energy.

CONCLUSIONS

1. In the north and south galleries, with no infiltration, windows, or skylights, 75% of the peak space cooling load is from people, and significant cooling and reheating of the supply air is required to remove the developed latent heat.

2. Conventional water chillers cannot be used in HVAC systems designed to hold 40% indoor relative humidities; direct-expansion refrigeration or absorbent dehumidifiers must be used.

3. A 70°F (21°C) and 50% RH space setpoint resulted in minimum energy consumption in all five climate zones.

4. Energy consumption was highest in Albuquerque, NM, where the weather is hot in the summer and cold in the winter, and the lowest in Burbank, CA, where the weather is mild throughout the year.

5. The heating and humidification energy was highest in Minneapolis, MN, because of the extreme cold weather in the winter. The heating and dehumidification energy was lowest in New Orleans, LA, because the weather is warm and humid throughout the year.

6. Heat recovery chillers can provide a major portion of the heating and humidification energy, and save approximately $13,500 per year in New York (based on an average $0.10/kWh and $0.65/therm).

ACKNOWLEDGMENTS

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DISCUSSION

Walter F. Spieger, President, Walter F. Spieger, Inc., Jenkintown, PA: How was the air-handling multizone unit configured to control humidity? Was moisture added at central locations or in each zone?

J.M. Ayres: A constant-volume pull-through air-handling unit with cooling coil, filters, and fixed minimum outside air zone ducts extended from the fan discharge plenum with steam reheat coils and a steam humidifier in each duct.

Adrian Tuluca, Principal, Steven Winter Associates, Norwalk, CT: Was the equipment sized previous to inputing the data into DOE-2.1C or was it sized by DOE-2.1C?

Ayres: Space loads from DOE-2 were used to perform the psychrometric analysis and size the equipment. The design cfm and equipment capacities were then reinput for the yearly simulations.

Tseng-Yao Sun, Principal, Hayakawa Associates, Los Angeles, CA: 1. It is unfortunate that the heating and humidification energy are grouped together in Figures 7 and 8 because of a "program limitation." In the air-conditioning process, we know that it requires more reheat energy and less humidification energy to maintain 40% RH but less reheat energy and more humidification energy to maintain 60% RH. What we don't know, and what the computer should be able to tell us, is how significant these differences are. Lumping these two energy uses together cancels the significance in both cases.

To get a more meaningful result for this type of study, perhaps the authors should modify the DOE program or use a simpler program that can differentiate the reheat and humidification energy use. In my opinion, engineering compromises due to "program limitations" are not healthy.

2. I do not understand the statement: "Note that the air-handling unit serving the galleries consumed more fan energy at 60% RH than at 50% in all five locations. This was because a higher supply air temperature was required for 60%, and therefore more air was required to meet the space loads."

Why was a higher supply air temperature REQUIRED? A 40% increase in cfm (Table 5, 13,450 cfm for 60% vs. 9,550 cfm for 50% RH, for example) is too high to be practical for maintaining 60% RH at peak cooling conditions, which occur only a few hours during the year. Under partial load conditions, as the weather gets drier, humidification will be needed anyway. It seems more reasonable to supply a normal amount of conditioned air and use the humidifier to maintain 60% RH at the peak cooling load condition.

3. What is the significance of a maximum cold deck temperature of 110°F in Table 5? This high temperature suggests that there is a heating coil in series with the cooling coil. If this is the case, there is a conflict with the statement on the same page that the cold deck temperature needs to be maintained at 51°F to remove moisture in the outside air during off-hours operation (third paragraph under "Psychrometric Analysis"). The statement also implies that 51°F is maintained during all off-hours periods. It seems unreasonable to cool the air down to 51°F and reheat if the outside air is dry.

4. It would be most helpful if some of the psychrometric plots were included in the paper. I had difficulty plotting the psychrometric process under the peak load condition with SHR = 0.5 and meeting the supply air cfm and temperature stated in the paper.

5. Was a DX system used to simulate the 40% RH case? If so, shouldn't a DX system operating at higher suction temperatures be used to simulate the 50% and 60% RH cases also for a fair comparison? If a DX system was not used for the 40% RH case, how was the chiller performance simulated under the 40% RH conditions?

Ayres: 1. DOE-2 standard reports do not separate humidification energy from reheat energy. Separate hourly reports for a full year could have been requested but were not because of the massive amount of data and budgetary limitations. It would have strengthened the paper to have provided this information.

2. We did not study the energy trade-offs between fan and humidification energy. The cold deck was set as high as possible to avoid excessive dehumidification followed by humidification. In the case of 60% RH, the cold deck was

set at 53°F, which provided approximately 1.5 cfm/ft^2, which is not an impractical design. One should note that in the well-insulated Gallery building, with low lighting levels and minimum outside air, dominant cooling load is caused by the people, and most of the time the space requires dehumidification and not humidification.

3. "Cold Deck Temperature" should read "Supply Air Temperature" and will be corrected before publication.

4. Psychrometric plots were presented in the 1988 referenced report "Energy Conservation and Climate Control."

5. In the 40% RH case, we simulated a reciprocating chiller. We did not simulate a system with a DX cooling coil. If a DX system were used, the cooling energy would have been increased by approximately 10%.