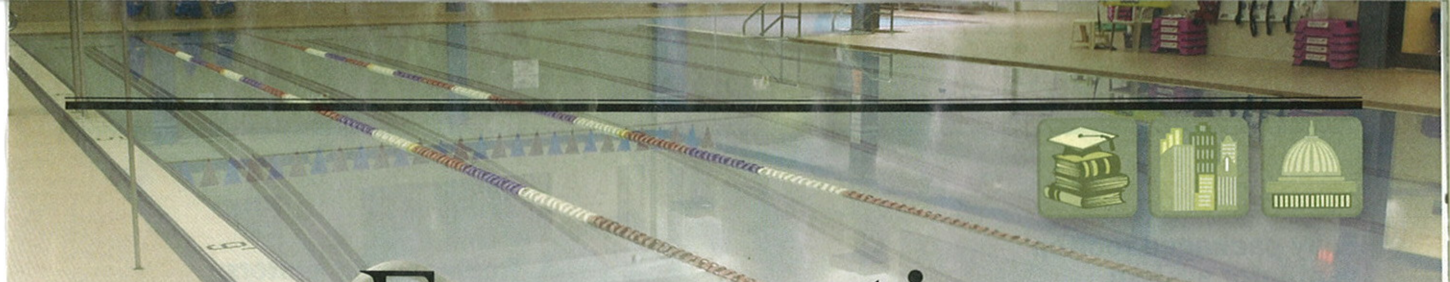

Evaporative Pool Dehumidification

Comparing the actual performance of two approaches to swimming-pool HVAC

By MARK S. LENTZ, PE
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Operated by the Marathon Area Swim Association (MASA), the Ray & Marie Goldbach, Marathon Area Swim Center in Marathon, Wis., is a 10,122-sq-ft facility consisting of a 25-yd-long, six-lane-wide main pool; a small instructional pool; locker rooms; and a small administrative area. It opened in 1988.

The original pool dehumidification unit (DHU) used heat of rejection from the refrigeration system to heat pool water and provide hot-gas reheat. In 2003, it began experiencing operational problems. Those problems became severe in early 2004. The temperature controls failed because of corrosion, causing the service contractor to alter the way the system was controlled. The dehumidification-system condenser lost its ability to heat the pools' water. The cooling-coil fins became oxidized and started to fall off. The hydronic heating coil froze and ruptured on several occasions. By 2009, the unit housing had lost integrity, and MASA concluded it needed to replace the system.

The system MASA chose (IDECVAV) is a direct-fired, variable-volume, 100-percent-outdoor-air unit equipped with an indirect evaporative precooler used to provide sensible cooling and heating-season energy recovery (Figure 1). The system is installed on the roof of the building, where it is subject to winter design conditions of -25°F.

This article compares the performance of the original and replacement systems. The comparison is based on actual performance. Aiding the comparison is the fact the swim center is a stand-alone facility served by dedicated utilities, rather than a small part of a much larger facility without dedicated utilities or submetering, which can obscure, distort, or dilute actual performance.

Strategies

The DHU introduced minimum outdoor air through the mixed-air path. This concept treats ventilation energy as

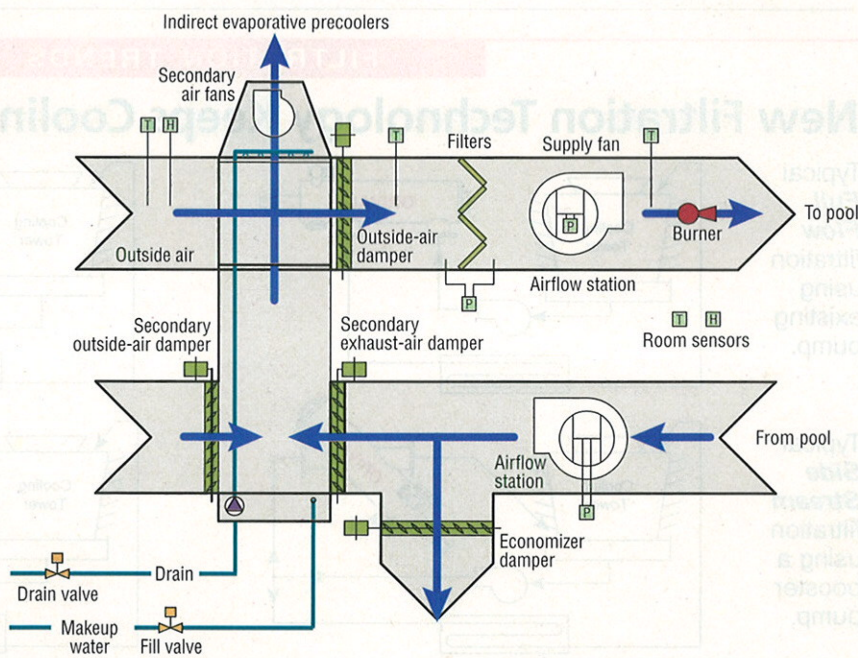

FIGURE 1. IDECVAV system.

The president of Lentz Engineering Associates Inc. (www.lentzengineering.com) and a member of HPAC Engineering*'s Editorial Advisory Board, Mark S. Lentz, PE, is nationally recognized for having successfully developed, tested, and proven several advanced HVAC-system strategies designed to exceed the performance requirements of ANSI/ASHRAE/IESNA Standard 90.1,* Energy Standard for Buildings Except Low-Rise Residential Buildings, *while meeting or exceeding the requirements of ANSI/ASHRAE Standard 62.1,* Ventilation for Acceptable Indoor Air Quality. *He is the recipient of numerous national engineering awards, including an American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Energy Award, an ASHRAE Symposium Paper of the Year Award, and an ASHRAE Distinguished Service Award.*

IMPACT OF CONTROL PRECISION AND OPERATING CONDITIONS ON COSTS

Swimming pools present unique and substantial challenges to HVAC designers. Occupancy loads are negligible compared with dehumidification loads. While thermal control is important, the precision of humidity control is what drives energy use and ventilation requirements.

Swimming-pool energy use consists of four interrelated components: pool-water heating, ventilation/dehumidification, envelope heating loads, and systemic inefficiencies. Indoor design conditions mimic summer outdoor design conditions. Sensible cooling loads usually are minimal and rarely a consideration in design. Dehumidification and pool-water heating loads are driven by evaporation rates, which are a function of water-surface area and the difference in vapor pressure between water and air.

Air quality. Air quality in swimming-pool installations is driven by pool-water quality. Chemical treatment, usually a form of chlorine, typically is required by health departments to control microorganisms and neutralize metabolic wastes in pool water. Chlorine oxidizes ammonia, a common metabolic-waste byproduct, to form the combined chlorine compounds monochloramine (NH_3Cl), dichloramine (NH_2Cl_2), and trichloramine ($NHCl_3$). These compounds are what produce a strong "chlorine" odor in pools and are noxious, toxic, and corrosive. Being heavier than air, chloramines tend to concentrate in the lower 12 in. of a space, in pool occupants' respiration zone. This necessitates the exhaust of air at floor level.

Chloramines corrode ferrous materials, copper, and aluminum in HVAC systems. They attack copper pipes, electrical circuits, and printed circuit boards used to control HVAC and pool systems. Where air is recirculated, they can cause the rapid deterioration of galvanized-steel ducts.

Recirculation of pool air creates a cycle of concentration that increases background chloramine concentrations until the rate of generation equals the rate of removal by exhaust.

Design objectives. While minimum rates of ventilation are required to limit adverse effects on occupant health, greater-than-minimum rates are desirable when economically viable. One-hundred-percent outdoor

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a tax on the system the owner simply must pay. Refrigeration was used to dehumidify the air. Rejected heat was used to heat supply air and pool water. Recirculation of chloramine-contaminated air caused equipment deterioration and, ultimately, failure. The use of refrigeration for pool-water heating, while more economical than the brute-force use of water heating and ventilation, reduced energy usage, but imposed high parasitic energy losses on the system in the form of compressor energy.

The IDECVAV system is ventilation-based and designed with the following features:

• Outdoor air is free of chloramines and dry under cold weather conditions. The design employs 100-percent outdoor air to maximize air quality. Also, it permits the equipment to be located outdoors, prevents moisture from compromising air-handling-unit insulation during cold-weather operation, allows adjacent areas to be pressurized with fresh air, and limits the potential for corrosion damage.

• Supply-air volume is controlled

to maintain space relative humidity with a low limit set to the minimum-outdoor-air ventilation rate.

• The exhaust fan is a direct-drive plug fan that locates the motor outside of the corrosive air stream. Exhaust airflow is piezometrically tracked to supply airflow to maintain mass balance and a pressure consistent with that of adjacent spaces. Secondary-exhaust-fan speed is slaved to primary-exhaust-fan speed to maintain building pressure under all conditions of operation.

• Exhaust-air, secondary exhaust-air, and secondary outdoor-air dampers are modulated as a mixed-air control.

• The system is designed around an indirect evaporative cooler. The air-to-air heat exchanger is of polymer construction to resist corrosion. The housing is American Iron and Steel Institute Type 304 stainless steel for corrosion resistance. Cooling, on the rare occasions it is needed, is provided by cycling the indirect-evaporative-cooler pump on and off.

• Supplemental heat is provided through a direct-fired gas burner with a 40:1 turndown ratio to maximize fuel efficiency.

With the IDECVAV system, outdoor-air ventilation is used to control space relative humidity and always is provided in quantities that meet or exceed minimum code requirements. Energy is conserved by recovering sensible and latent energy from exhaust air to temper outdoor air. Because much of the energy in the exhaust path is recovered and recycled, the need for refrigeration—and the associated high parasitic energy use—is functionally eliminated.

Figure 2 shows the psychrometric path in heating mode. Both sensible and latent heat (A-B) are converted to sensible heat (X-Y) through energy recovery. Supplemental heat is introduced by a gas burner (Y-Z). Water vapor is absorbed from the space while heat is delivered to offset envelope losses (Z-A). In part because system airflow is controlled from space

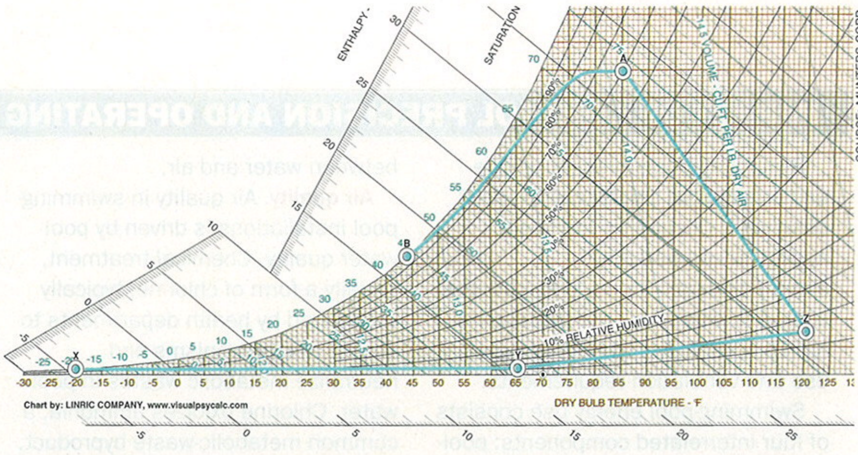

FIGURE 2. IDECVAV heating psychrometrics.

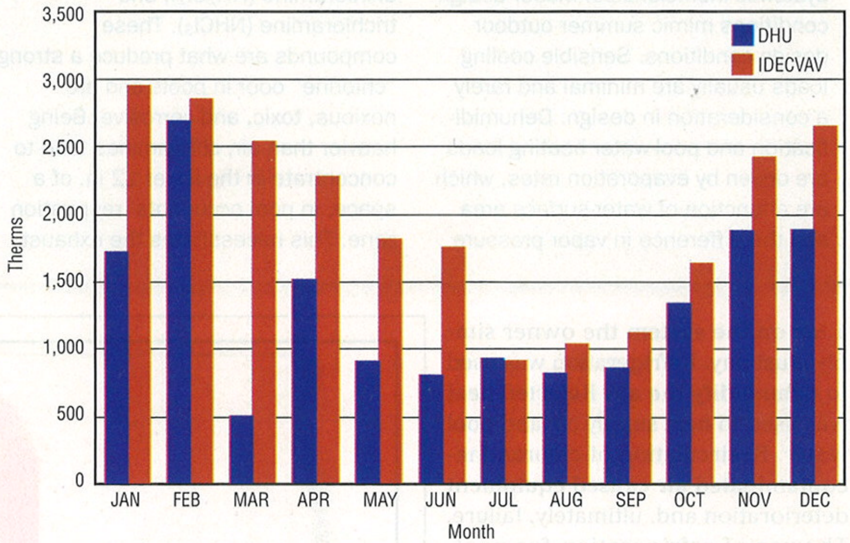

FIGURE 3. Natural-gas energy use.

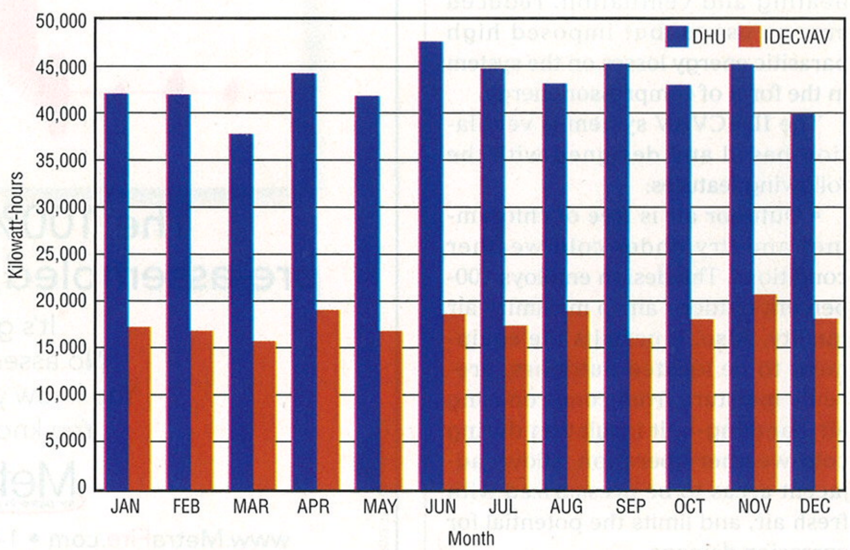

FIGURE 4. Electrical energy use.

relative humidity, an 8,760-hr typical-meteorological-year psychrometric analysis was employed not only to predict system economics, but to eliminate uncertainty as to which ambient conditions would dictate design airflow rates and to determine which conditions produce both maximum heating and maximum cooling requirements.

The performance of an air-to-air heat exchanger is amplified when moisture condenses on or evaporates from the heat-exchanger surface. This causes boundary-layer resistance to heat transfer to break down. The increased efficiency can become a serious problem at very low ambient conditions. In this application, however, the potential is much reduced, as long as the air in the exhaust path has sufficient energy to prevent the formation of ice in the heat exchanger. For a pool system, that generally is possible if design space-temperature and relative-humidity conditions are maintained. At low-ambient-temperature conditions, that can be compromised by the use of pool covers.

Performance Comparison

The utility use of the DHU in 2003, the last year the DHU was able to perform as designed, was compared with the utility use of the IDECVAV system from July 2010 through June 2011. Both data sets are imperfect: The DHU experienced a shutdown in March 2003; the IDECVAV system had some unresolved startup issues and experienced an unusually cold spring in 2011. However, by any measure, the performance gains realized by the IDECVAV system were significant.

Figure 3 shows aberrations in the DHU's natural-gas utility use from January to March 2003, when operational problems began. The IDECVAV system substituted less expensive natural gas for the benefits of the electrically driven refrigerant compressors in the DHU. As a result, natural-gas use for all purposes increased 33.7 percent, from 15,842 therms with the DHU in 2003 to 23,951 therms with the IDECVAV system in 2011.

Figure 4 shows an aberration in the DHU's electricity use in March 2003. Both systems demonstrate relatively flat electricity-use profiles; however, the IDECVAV system produced substantial reductions: Electricity use for all purposes decreased 58.8 percent, from 776,373 kwh with the DHU in 2003 to 319,641 kwh with the IDECVAV system in 2011. Total combined energy use dropped 17.7 percent, from 455,602 Btu per square foot per year with the DHU to 375,122 Btu per square foot per year with the IDECVAV system (Figure 5). Annual greenhouse-gas emissions dropped 44.6 percent, from 614 to 340 metric tons of carbon dioxide.

Figure 6 shows relative energy costs based on 2011 prices. The DHU's natural-gas costs are $10,268, while the IDECVAV system's are $15,525. The $5,257 difference is more than offset by the difference in electricity costs: The DHU's electricity costs are $54,000.02, while the

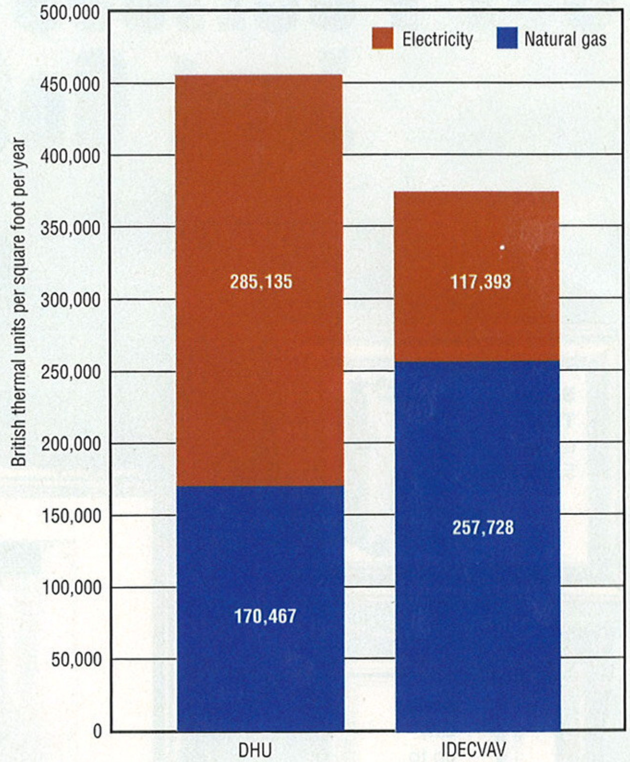

FIGURE 5. Total energy use.

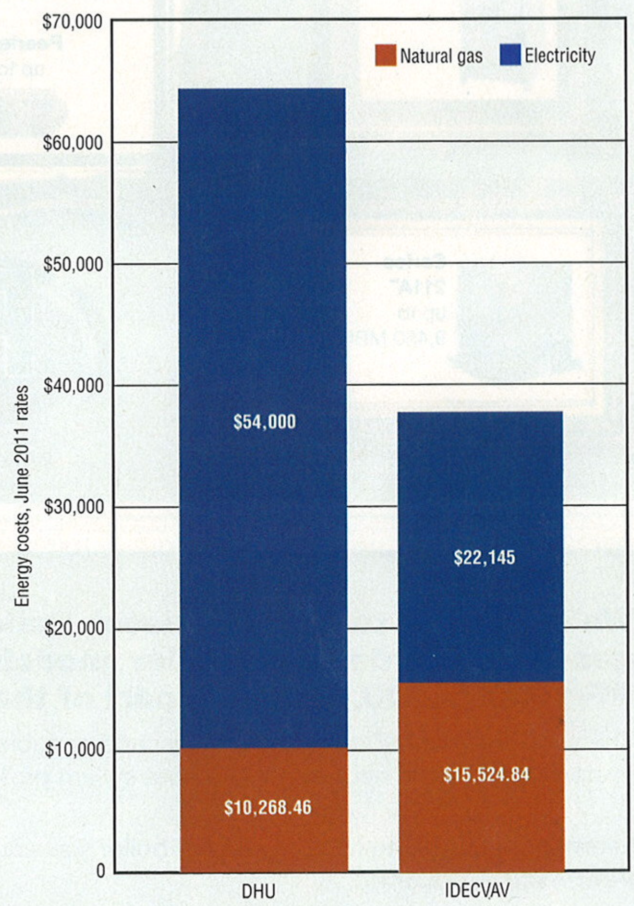

FIGURE 6. Total annual cost.

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IDECVAV system's are $22,145, a difference of $31,855. The net reduction in total energy costs with the IDECVAV system is $26,598, or 41.4 percent.

On a dollars-per-square-foot-of-facility basis, annual energy costs are $4.05 with the IDECVAV system vs. $6.92 with the DHU. On the more reliable dollars-per-square-foot-of-pool-area basis, energy costs are $11.54 with the IDECVAV system vs. $19.68 with the DHU.

Conclusions

Certain aspects of the IDECVAV strategy employed at the Ray & Marie Goldbach, Marathon Area Swim Center, such as the use of an indirect evaporative precooler as both the primary energy-recovery device and the primary cooling system, may seem counterintuitive. However, the efficiency and indoor-air-quality advantages over the dehumidifier solution are significant. Better energy performance comes from the fact that while both act to recover the latent and sensible heat in the pool area, the ventilation-based IDECVAV solution is able to provide them passively, without the need to operate refrigerant compressors.

By all measures, the IDECVAV system outperformed the DHU. It provides superior air quality under most conditions of operation, is less complex, is of more durable construction, consumes less energy, costs substantially less to operate, and reduces greenhouse-gas emissions. Because it uses no hydrofluorocarbons, chlorofluorocarbons, or hydrochlorofluorocarbons, it also is much "greener." Perhaps most importantly, it is affordable. The system costs less to install and can pay for itself quickly with reduced energy costs in retrofit applications.

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IMPACT OF CONTROL PRECISION CONTINUED

Continued from Page 33

air provides the additional advantage of preventing the supply sides of HVAC ductwork, equipment, and controls from being exposed to chloramines. Because it holds very little moisture during cold weather, 100-percent outdoor air also prevents condensation on the supply side of equipment located outdoors in cold weather. Additionally, it breaks the cycle of concentration that occurs when pool air is recirculated.

The amount of outdoor air required to maintain space humidity is dependent on the amount of moisture in the air. Varying outdoor-air volume as required to control humidity is an effective way to satisfy ventilation requirements and displace chloramines.

For purposes of both energy conservation and swimmer comfort, the most desirable space conditions for pool ventilation systems are warm and humid. High levels of air motion create drafts, while low relative humidity increases the rate of evaporation from a swimmer's body, causing discomfort.

For athletic events, water temperature of 81°F to 82°F is preferred. For therapeutic pools, water temperature of 85°F to 87°F is preferred. For spas, water temperature of 104°F to 106°F is preferred.

Pool-water heating loads are driven almost exclusively by evaporation. Figure 7 illustrates the energy implications of small deviations in operating temperature, humidity, and pool-water temperature for a 5,000-sq-ft swimming pool.

Evaporation is driven by the difference in vapor pressure between pool water and air. Vapor pressure in water is a function of water temperature, while vapor pressure in air is a function of air temperature and relative humidity. As the vapor-pressure differential between air and water increases, so does evaporation and energy use. For example, a 4°F reduction (from 82°F to 78°F) in water temperature, a 4°F increase (from 83°F to 87°F) in air temperature, and a 10-percent increase (from 50 percent to 60 percent) in relative humidity effectively reduces pool energy use by 65 percent, from 836 MBH to 296 MBH.

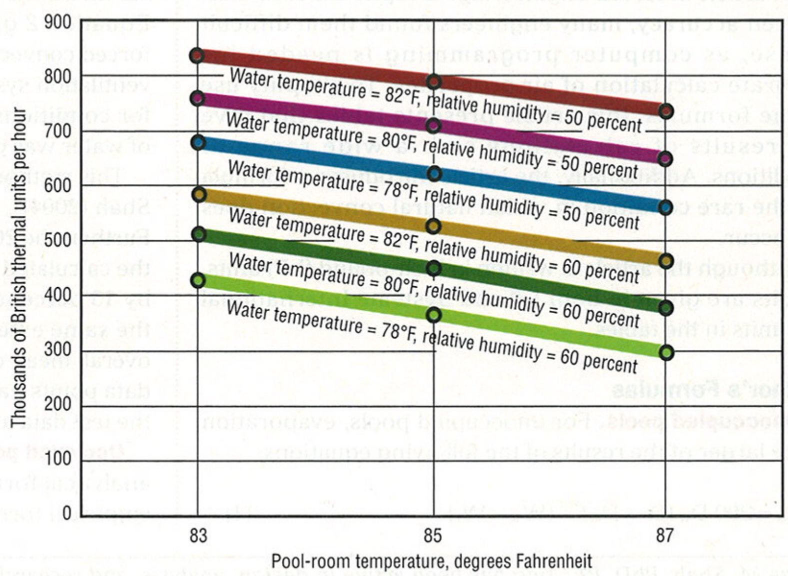

FIGURE 7. Sensitivity to variations in temperature and relative humidity.

Simplified Method of Calculating Evaporation From Swimming Pools

Introducing tables intended to make highly accurate formulas easier to use

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Accurate calculation of evaporation from swimming pools is needed for proper sizing of ventilation and dehumidification equipment. Calculations are needed for both occupied and unoccupied conditions for proper modulation of equipment capacity, as well as estimation of energy consumption. The most common method is the one recommended in ASHRAE Handbook—HVAC Applications.[1] It involves calculation of evaporation using the Carrier equation[2] and correction of the output using activity factors. Comparisons with test data have shown this method is inaccurate. The American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) is undertaking a research program to test it and other methods.

In 2002 and 2003,[3,4] the author developed formulas for occupied and unoccupied pools that were verified with a wide range of data. Those formulas were summarized in an article in *HPAC Engineering*.[5] Despite the formulas' proven accuracy, many engineers found them difficult to use, as computer programming is needed for accurate calculation of air properties. To simplify use of the formulas, this article presents tables that give the results of calculations over a wide range of conditions. Additionally, the article introduces a formula for the rare condition in which natural convection does not occur.

Although the article is written in inch-pound (I-P) units, results are given in both I-P and Systeme International (SI) units in the tables.

Author's Formulas

Unoccupied pools. For unoccupied pools, evaporation is the larger of the results of the following equations:

$$E_0 = 290\, D_w (D_r - D_w)^{1/3} (W_w - W_r) \qquad (1)$$

$$E_0 = 0.125\, (p_w - p_r) \qquad (2)$$

where:

E_0 = evaporation from unoccupied pool, pounds per hour per square foot

D_w = density of air saturated at water temperature, pounds per cubic foot of dry air

D_r = density of air at room condition, pounds per cubic foot of dry air

W_w = humidity ratio, air saturated at water temperature, pounds per pound

W_r = humidity ratio, air at room condition, pounds per pound

p_w = water-vapor pressure in air, air saturated at water temperature, inches of mercury

p_r = water-vapor pressure in air, air at room condition, inches of mercury

Equation 1 gives the rate of evaporation caused by natural convection. It was obtained from the analogy between heat and mass transfer without any empirical factor. Its full derivation can be seen in Shah (2008).[6] Equation 2 gives the rate of evaporation attributed to forced convection by air currents generated by a building ventilation system. It was obtained by analyzing test data for conditions in which the density of air at the surface of water was greater than the density of air in the room.

This method of calculation differs from the one given in Shah (2004)[5], as it includes negative-density differences. Further, the 2004 method uses only Equation 1, increasing the calculated evaporation at very low density differences by 15 percent. The present method was compared with the same extensive database as the earlier method. The overall mean deviation was the same, although individual data points had higher or lower deviations. The ranges of the test data are given in Table 1.

Occupied pools. For occupied pools, the author gave an analytical formula, as well as an empirical formula. The empirical formula produces much closer agreement. For

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fully occupied pools:

$$E = 0.023 - 0.0000162 \div U + 0.041\,(p_w - p_r) \qquad (3)$$

where:

E = evaporation from pool, pounds per hour per square foot

U = utilization factor (number of people in pool area multiplied by 48.4 divided by pool area), the applicable range of which is 0.1 (10-percent occupied) to 1 (fully occupied). The ranges of test data are given in Table 1.

Tables

Table 2 gives values of evaporation from unoccupied pools calculated using the method above. (Table 3 gives the corresponding values in SI units.) Values are given at 2-degree intervals. For in-between temperatures, linear interpolation can be performed. This table is applicable to all types of pools with an undisturbed water surface.

Table 4 gives values of evaporation from fully occupied pools calculated using Equation 3. (Table 5 gives the

	Unoccupied pools	Occupied pools
Pool area, square feet	0.78 to 4,573	687 to 13,008
Water temperature, degrees Fahrenheit	45 to 201	77 to 86
Air temperature, degrees Fahrenheit	43 to 95	80 to 90
Air relative humidity, percent	28 to 98	33 to 72
$(p_w - p_r)$, millimeters of mercury	0.062 to 23.7	0.31 to 0.61
$(p_r - p_w)$, pounds per cubic foot	-0.00025 to +0.062	0.0013 to 0.000081
Number of occupants	0	8 to 180
Utilization factor	0	0.1 to 1.5
Number of data sources	9	4

TABLE 1. Verified ranges of author's formulas.

Water temperature, degrees Fahrenheit	Space air temperature, degrees Fahrenheit, and relative humidity, percent											
	76°F		78°F		80°F		82°F		84°F		86°F	
	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%
76	0.0213	0.0159	0.0176	0.0120	0.0135	0.0099	0.0123	0.0085	0.0110	0.0069	0.0097	0.0053
78	0.0269	0.0211	0.0232	0.0173	0.0193	0.0133	0.0149	0.0106	0.0132	0.0091	0.0118	0.0075
80	0.0327	0.0266	0.0291	0.0228	0.0252	0.0188	0.0212	0.0146	0.0166	0.0114	0.0141	0.0098
82	0.0390	0.0326	0.0353	0.0287	0.0315	0.0246	0.0274	0.0204	0.0232	0.0160	0.0185	0.0110
84	0.0458	0.0391	0.0420	0.0350	0.0381	0.0309	0.0340	0.0266	0.0298	0.0222	0.0253	0.0176
86	0.0530	0.0461	0.0492	0.0419	0.0452	0.0377	0.0410	0.0333	0.0368	0.0288	0.0323	0.0241
88	0.0608	0.0536	0.0569	0.0494	0.0528	0.0450	0.0486	0.0405	0.0442	0.0358	0.0397	0.0311
90	0.0692	0.0618	0.0651	0.0574	0.0610	0.0529	0.0567	0.0482	0.0522	0.0435	0.0476	0.0386
102	0.1335	0.1250	0.1289	0.1199	0.1241	0.1147	0.1192	0.1093	0.1142	0.1037	0.1089	0.0979
104	0.1469	0.1383	0.1422	0.1331	0.1374	0.1277	0.1324	0.1222	0.1272	0.1165	0.1219	0.1106

TABLE 2. Evaporation from unoccupied pools calculated by the author's method, pounds per hour per square foot.

Water temperature, degrees Celsius	Space air temperature, degrees Celsius, and relative humidity, percent											
	25°C		26°C		27°C		28°C		29°C		30°C	
	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%
25	0.1085	0.0809	0.0918	0.0636	0.0732	0.0515	0.0498	0.0450	0.0583	0.0382	0.0523	0.0311
26	0.1355	0.1042	0.1171	0.0872	0.0997	0.0693	0.0806	0.0547	0.0575	0.0479	0.0626	0.0479
27	0.1579	0.1290	0.1433	0.1118	0.1262	0.0941	0.1081	0.0753	0.0885	0.0582	0.0723	0.0510
28	0.1877	0.1556	0.1711	0.1380	0.1539	0.1200	0.1360	0.1014	0.1171	0.0818	0.0960	0.0618
29	0.2174	0.1841	0.2006	0.1661	0.1831	0.1477	0.1651	0.1287	0.1644	0.1092	0.1268	0.0888
30	0.232	0.2146	0.2143	0.1962	0.1960	0.1773	0.1772	0.1579	0.1576	0.1380	0.1372	0.1176
31	0.2831	0.2474	0.2655	0.2285	0.2475	0.2091	0.2289	0.1892	0.2098	0.1688	0.1900	0.1480
32	0.3192	0.2825	0.3013	0.2631	0.2829	0.2432	0.2639	0.2228	0.2445	0.2019	0.2445	0.1805
39	0.6461	0.6035	0.6256	0.5808	0.6045	0.5575	0.5827	0.5334	0.5604	0.5087	0.5379	0.4833
40	0.7051	0.6617	0.6842	0.6386	0.6627	0.6148	0.6405	0.5903	0.6177	0.5655	0.5943	0.5391

TABLE 3. Evaporation from unoccupied pools calculated by the author's method, kilograms per hour per square meter.

Water temperature, °F	Space air temperature, degrees Fahrenheit, and relative humidity, percent: 76°F 50%	76°F 60%	78°F 50%	78°F 60%	80°F 50%	80°F 60%	82°F 50%	82°F 60%	84°F 50%	84°F 60%	86°F 50%	86°F 60%
76	0.0416	0.0379	0.0403	0.0363	0.039	0.0347	0.0375	0.033	0.036	0.0312	0.0344	0.0293
78	0.0441	0.0404	0.0428	0.0389	0.0415	0.0373	0.0401	0.0356	0.0386	0.0337	0.037	0.0318
80	0.0468	0.0431	0.0455	0.0416	0.0442	0.0400	0.0428	0.0382	0.0413	0.0382	0.0397	0.0364
82	0.0497	0.0459	0.0484	0.0444	0.0470	0.0428	0.0456	0.0411	0.0411	0.0393	0.0425	0.0374
84	0.0527	0.0490	0.0514	0.0474	0.0501	0.0458	0.0486	0.0441	0.0471	0.0423	0.0451	0.0404
86	0.0559	0.0521	0.0546	0.0506	0.0532	0.0490	0.0518	0.0473	0.0503	0.0455	0.0487	0.0436

TABLE 4. Evaporation from fully occupied pools calculated using Equation 3, pounds per hour per square foot.

Water temperature, °C	Space air temperature, degrees Celsius, and relative humidity, percent: 25°C 50%	25°C 60%	26°C 50%	26°C 60%	27°C 50%	27°C 60%	28°C 50%	28°C 60%	29°C 50%	29°C 60%	30°C 50%	30°C 60%
25	0.2065	0.1878	0.2007	0.1809	0.1947	0.1737	0.1884	0.1660	0.1817	0.1580	0.1747	0.1496
26	0.2179	0.1992	0.2122	0.1923	0.2062	0.1851	0.1998	0.1775	0.1932	0.1695	0.1861	0.1611
27	0.2300	0.2113	0.2242	0.2044	0.2182	0.1972	0.2119	0.1895	0.2052	0.1815	0.1982	0.1731
28	0.2427	0.2239	0.2369	0.2171	0.2309	0.2098	0.2246	0.2022	0.2179	0.1942	0.2109	0.1858
29	0.2560	0.2373	0.2503	0.2304	0.2442	0.2232	0.2379	0.2156	0.2312	0.2076	0.2242	0.1992
30	0.2700	0.2513	0.2643	0.2445	0.2583	0.2372	0.2519	0.2296	0.2453	0.2216	0.2383	0.2132

TABLE 5. Evaporation from fully occupied pools calculated using Equation 3, kilograms per hour per square meter.

corresponding values in SI units.) The data are applicable to pools with air temperatures of 76°F to 90°F and water temperatures of 76°F to 86°F.

Use of the tables can be illustrated with two examples:

Example 1. A 10,000-sq-ft public swimming pool has a water temperature of 80°F, an air temperature of 78°F, and relative humidity of 50 percent.

From Table 2:

- Evaporation when pool is unoccupied: 0.0291 lb per hour per square foot.
- Total evaporation: 0.0291 × 10,000 = 291 lb per hour.

From Table 4:

- Evaporation when pool is fully occupied: 0.0455 lb per hour per square foot.
- Total evaporation: 0.0455 × 10,000 = 455 lb per hour.

Example 2. A 10,000-sq-ft public swimming pool has a water temperature of 79°F, an air temperature of 78°F, and relative humidity of 50 percent.

Table 2 does not list 79°F water temperature, so interpolation is required. Evaporation at 78°F water temperature is 0.0232 lb per hour per square foot, while evaporation at 80°F water temperature is 0.0291 lb per hour per square foot. Evaporation at 79°F water temperature, then, is:

$$(0.0232 + 0.0291) \div 2 = 0.026 \text{ lb per hour per square foot}$$

Discussion

The method for unoccupied pools presented here has been verified with a wide range of test data and has a firm theoretical foundation. It can be used with confidence for all types of pools.

The method for occupied pools presented here was verified with test data from four public pools. The ASHRAE Handbook[1] method was found to have a mean deviation of 36.9 percent, while the method presented here had a mean deviation of only 16.2 percent.

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