Conditioning Swimming Pool Areas

In enclosed spaces containing a swimming pool or hot tub, there is need to control humidity. If humidity is not controlled, condensation — and possibly freezing condensation — may severely damage the structure.

A major problem with conditioning swimming pool (and hot tub) enclosures is the relatively high operating cost of maintaining acceptable conditions. So, some type of cost effective energy recovery device should be considered when the heating, ventilating and/or cooling system is designed.

The least expensive — first cost — method to control humidity is just a ventilation system, using dryer outdoor air to replace humid air. However:

- a considerable amount of heat is needed to temper the outdoor air during cold weather
- when temperatures are intermediate, this system may not control humidity
- during hot weather the system cannot control temperature.

Ventilation system efficiency can be improved — at a higher first cost — if the design includes a provision for some type of energy recovery device to reduce the amount of heat required to temper induced outdoor air.

Remember, recoverable energy consists of both the sensible *and latent* heat of the exhaust air. The latent component can be a very sizable part of total recoverable energy, therefore a heat recovery device that can extract both sensible and latent heat should be considered.

Since a ventilation system can only provide effective humidity control if the outdoor air is reasonably dryer than the indoor air, it will not perform well in humid weather. Therefore, a mechanical cooling or dehumidification system will be required to control humidity in the envelope when the outdoor air approaches the moisture content of the indoor air. During hot weather, a mechanical cooling system will be required to maintain temperature and humidity.

Control strategies

Table 1 shows typical pool and hot tub design conditions and considerations. Table 2 can be used to estimate the evaporation rate from private (negligible splashing) pool surfaces. Table 3 can be used to adjust these values for semi-private and public pools. Use Table 4 for therapeutic baths, hot tubs and whirlpools.

Example: A public pool has 1,800 sq. ft. of surface area. The water temperature is 80F and the entering indoor air is 84F/db, 50% rh. Note:

- Table 1 indicates .0226 pounds of water/sq. ft. of surface area will evaporate
- Table 2 indicates this value should be multiplied by 2. The design value for evaporation from this pool will be —

$$2 \times .0226_{\text{lbs./sq. ft.}} \times 1{,}800_{\text{sq. ft.}} = 81.36_{\text{lbs./hr.}}$$

A value of 1,000 Btuh/lb. of water can be used to estimate the latent load. For the example given, the corresponding latent load is:

$$1{,}000_{\text{Btu/lb.}} \times 81.36_{\text{lbs./hr.}} = 81{,}360_{\text{Btuh}}$$

Estimating outdoor air cfm

The amount of outdoor air required to

continued

Pool & Hot Tub Design Conditions

Pleasure swimming or hot tub	— 74 to 84 db; 50 to 60% rh
Therapeutic applications	— 80 to 85 db; 50 to 60% rh
Pool water temperature:	
Pleasure swimming	— 75 to 80 db
Competitive swimming	— 72 to 75 db
Therapuetic applications	— 85 to 95 db
Hot tubs	— 100 to 105 db

Table 1
Pool and hot tub design conditions.

	RELATIVE HUMIDITY %													
AIR TEMP	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%	50%	60%
	EVAPORATION RATE IN LBS WATER PER SQ. FT. OF POOL SURFACE													
92											.0298	.0215	.0341	.0259
90									.0281	.0217	.0322	.0258	.0366	.0302
88							.0254	.0192	.0293	.0231	.0334	.0272	.0377	.0315
86					.0240	.0181	.0277	.0218	.0315	.0257	.0356	.0290	.0400	.0341
84			.0226	.0171	.0261	.0206	.0298	.0243	.0336	.0281	.0377	.0320	.0410	.0361
82	.0203	.0161	.0236	.0194	.0271	.0229	.0307	.0265	.0346	.0304	.0387	.0350	.0430	.0386
80	.0223	.0163	.0256	.0216	.0290	.0250	.0327	.0287	.0366	.0326	.0406	.0370	.0450	.0410
78	.0243	.0205	.0271	.0239	.0307	.0273	.0345	.0311	.0383	.0348	.0424	.0396	.0468	.0435
76	.0263	.0227	.0286	.0261	.0323	.0296	.0364	.0335	.0400	.0370	.0442	.0422	.0486	.0460
74	.0283	.0249	.0301	.0284	.0340	.0319	.0382	.0358	.0417	.0391	.0460	.0448	.0505	.0485
WATER TEMP	78	78	80	80	82	82	84	84	86	86	88	88	90	90

Table 2
Evaporation rates from still swimming pool surfaces.

continued

control space humidity will depend on the difference between the specific humidity (Sp/H) of the outdoor air and that of the pool air.

You can use a Psychrometric Chart to establish a threshold value for outdoor air specific humidity. When the condition of the outdoor air approaches this threshold value, mechanical cooling or mechanical dehumidification will be required to maintain the design humidity in the space.

Figure 1 shows the outdoor air must have a specific humidity less than 88 grains if the pool enclosure is to be maintained at 84F and 50% rh. The amount of outdoor air (cfm) required to maintain the space condition can be calculated using Equation #1 below:

Equation #1:

$$\text{cfm} = \frac{\text{latent load}_{\text{Btuh}}}{.68 \times (\text{SpH}_{\text{ia}} - \text{SpH}_{\text{oa}})}$$

Where:

✓ Sp/H_{ia} = grains moisture, indoor air

✓ Sp/H_{oa} = grains moisture, outdoor air.

Table 5 illustrates how outdoor air cfm is affected by its specific humidity. The values in this table were calculated by applying Equation #1 to the conditions outlined in the example.

Inspection of Table 5 indicates shows the specific humidity of the outdoor air must be at least six grains less than the specific humidity of the pool air, or the design cfm will become excessively large. It is important to keep design cfm as low as possible because outdoor air must be heated whenever its temperature is below the pool design, and because the cost of the fan and air distribution system increases as design cfm increases.

In actual practice, design cfm is also related to the Air Loading Factor (ALF). The ALF is the cfm of supply air/sq. ft. of floor (enclosed pool and perimeter) area. When the air loading factor rises above five cfm/sq. ft., it becomes difficult to diffuse the supply air into the space. Table 6 (extracted from SMACNA Manual 15D) serves as a guide for estimating the ALF accommodated by various types of air terminal devices.

Table 7 indicates how a maximum ALF of five can be used to establish the maximum design cfm for pool humidity control. This shows the design airflow for an acceptable ALF, is approximately 14,956-cfm for the public pool in our example. That quantity of outdoor air will offset the latent load whenever its specific humidity is eight or more grains below the pool design specific humidity.

ALF constraints on design cfm

Figure 2 shows maximum design cfm is limited by the (ALF) and this reduces the potential for using outdoor air to maintain design humidity in the space. It also shows how the average db and coincident wb conditions for dry, intermediate and humid climates compare with the operating limits established by the ALF. *Average db and coincident wb weather data was extracted from* Engineering Weather Data, AF 88-8.

Figure 2 also shows that mechanical cooling or dehumidification are needed to maintain desired conditions within the pool envelope, whenever outdoor db is above 65F in humid climates or above 75F in intermediate climates.

Estimating design heating requirement

Capacity of the heating equipment will be determined by adding the building heating transmission loss load to the heating load required to temper outdoor air. Using standard load calculation procedures, the outdoor air heating load is calculated using Equation #2 below.

Equation #2

$$\text{Load} = 1.1 \times \text{cfm}_{\text{oa}} \times (T_{\text{ia}} - T_{\text{oa}})$$

Where:

✓ cfm_{oa} = cfm outdoor air

✓ T_{ia} = indoor air temperature (F db)

✓ T_{oa} = outdoor air temperature (F db).

FOR SEMI-PRIVATE POOLS MULTIPLY THE TABLE ONE VALUES BY	1.00
FOR PUBLIC POOLS MULTIPLY THE TABLE ONE VALUES BY	2.00

Table 3
Semiprivate and public pool correction factors.

AIR TEMP	RELATIVE HUMIDITY 50%	60%
92	.1460	.1300
90	.1500	.1360
88	.1530	.1410
86	.1560	.1460
84	.1620	.1520
82	.1640	.1560
80	.1660	.1580
78	.1693	.1627
76	.1727	.1673
74	.1760	.1720

Table 4
Evaporation rates for therapeutic baths, hot tubs and whirlpools.

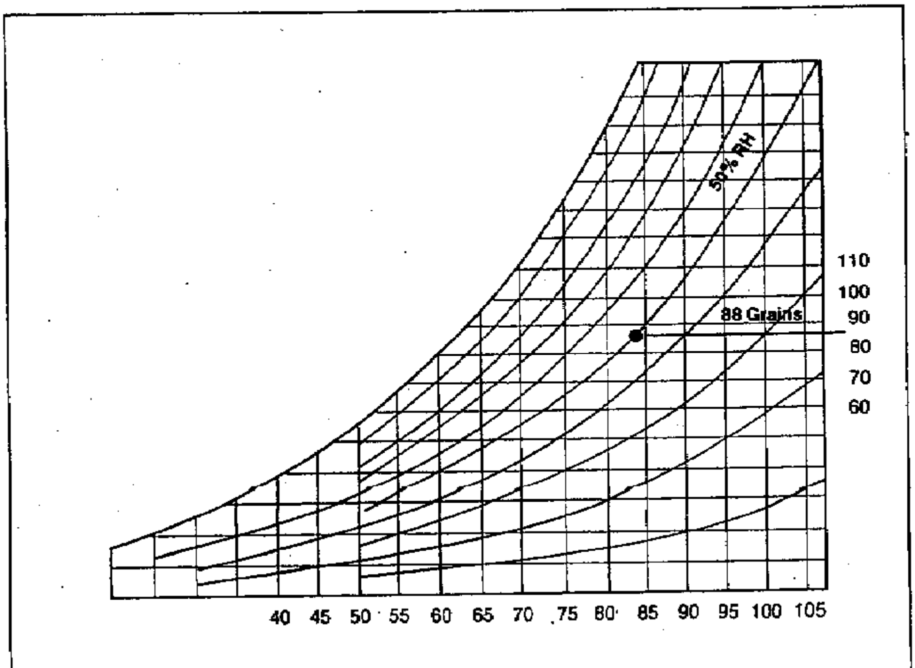

Figure 1
Psychrometric chart showing example pool conditions; 50%$_{rh}$, 88-gr$_{sp/h}$

SPACE GRAINS			GRAINS DIFF.		CFM FOR	CFM FOR
EXAMP NO 1	EXAMP NO 2	OUTDOOR GRAINS	EXAMP NO 1	EXAMP NO 2	EXAMPLE NO. ONE	EXAMPLE NO. TWO
88	80	84	4	N/A	29912	N/A
88	80	82	6	N/A	19941	N/A
88	80	80	8	0	14956	N/A
88	80	78	10	2	11965	9838
88	80	76	12	4	9971	4919
88	80	74	14	6	8546	3279
88	80	72	16	8	7478	2460
88	80	70	18	10	6647	1968
88	80	6	23	15	5202	1312
88	80	60	28	20	4273	984
88	80	55	33	25	3626	787
88	80	50	38	30	3149	656
88	80	40	48	40	2493	492
88	80	20	68	60	1760	328

Table 5

Example of how outdoor air cfm is affected by its specific humidity.

OUTDOOR GRAINS	CFM FOR EXAMPLE NO. ONE	CFM FOR EXAMPLE NO. TWO	ALF FOR EXAMPLE NO. ONE	ALF FOR EXAMPLE NO. TWO
84	29912	N/A	9.35	N/A
82	19941	N/A	6.23	N/A
80	14956	N/A	4.67	N/A
78	11965	9838	3.74	42.77
76	9971	4919	3.12	21.39
74	8546	3279	2.67	14.26
72	7478	2460	2.34	10.69
70	6647	1968	2.08	8.55
65	5202	1312	1.63	5.70
60	4273	984	1.34	4.28
55	3626	787	1.13	3.42
50	3149	656	.98	2.85
40	2493	492	.78	2.14
20	1760	328	.55	1.43

- POOL AREA + PERIMETER
- POOL AREA + PERIMETER AREA = 23

Table 7

Illustration of how a max. ALF of 5 can be used for pool design humidity control.

Table 8 indicates the design heating requirement for our example at various winter outdoor temperatures, assuming:

- the pool envelope heating load is 80,000 Btuh at 0°F$_{oa}$
- the maximum design outdoor cfm is 14,956
- the design indoor temperature is 84F.

This table shows that as outdoor design temperature drops, heating equipment capacity and heating energy (Btuh) — increases dramatically. This occurs because a relatively large amount of outdoor air is required to control humidity at intermediate outdoor temperatures.

Selecting the final design cfm

The final design cfm will be a compromise between first cost, energy cost and range of outdoor weather conditions for which indoor relative humidity control is possible.

In the example, maximum design cfm — 14,956 — was established by air loading limitations (ALF less than 5). But maximum design cfm may not be the best choice when operating and first costs are considered. Table 7 can be used to evaluate the effects of using a design cfm less than ALF limitations.

From table 9, if design cfm for the pool is reduced from the maximum of 14,956 to 9,971-cfm, the operating limit is lowered from 80 to 76 grains.

Figure 2 indicates that using 9,971-cfm (76 grains) to establish the design condition does not significantly affect the range of outdoor weather conditions accommodated by the system. Indoor surface condensation is less likely to occur when outdoor temperatures are above 55F because of lowered thermal transmission, and since the pool system is designed to maintain a relatively low indoor humidity (50%), a swing of 5-10% can be tolerated at intermediate outdoor temperatures.

Type of Outlet	CFM/Square Foot Floor Area
Grilles & Registers	0.6 to 1.2
Slot Diffusers	0.8 to 2.0
Perforated Panels	0.9 to 3.0
Ceiling Diffuser	0.9 to 5.0
Perforated Ceiling	1.0 to 10.0

Table 6

General guide for selecting supply air outlets.

Table 10 indicates the effect on the heating load — and operating cost — if the design cfm is reduced from 14,956 to 9,971-cfm.

Minimum outdoor cfm

The minimum outdoor cfm is determined by the outdoor air specific humidity at winter design temperature.

Figures 2 indicates that, at typical winter design temperatures, specific humidity of the outdoor air is much lower than the outdoor temperatures used to establish the cfm required for humidity control. The value of this minimum cfm can be determined by using Equation # 1 to calculate the cfm corresponding to the specific humidity of saturated air at winter design temperature. *This calculation is somewhat conservative because the outdoor air will not normally be saturated at winter design.*

Table 11 shows the amount of outdoor cfm required to control humidity decreases as the outdoor temperature drops. When these values are compared with the design cfm required to control pool humidity during intermediate outdoor temperatures, it becomes obvious that the total heating capacity and energy requirements will be greatly reduced if outdoor cfm is decreased as the outdoor temperature drops.

Controlling outdoor air cfm

There are a number of ways to control the amount of outdoor air introduced into a pool or hot tub envelope. A few of the more obvious types are discussed below. Remember, the final choice should be made by considering:

- ✓ first cost
- ✓ operating cost
- ✓ return on investment
- ✓ architectural/other considerations.

Constant volume (CV) systems — OA/RA/EA dampers, along with supply and return fans can be used to optimize out-

continued

continued

door air, if the dampers are modulated by a dew point controller located within the conditioned space. This method saves heating energy but it doesn't offer the advantage of potential fan energy savings, because the fans operate at a constant cfm. However, simple CV air terminals can be used for air distribution.

Variable air volume (VAV) systems — Fan cfm can be adjusted using discharge dampers, inlet vanes or fan speed (step or modulation) control. In this case, the dew point controller must operate the OA/RA/EA dampers and the fan modulation device.

Another method is to maintain the outdoor air damper at its minimum position and increase the fan speed until it reaches design cfm, at which point the outdoor damper could begin to modulate open. This method saves heating and fan energy, but the supply air terminals must be selected so they provide adequate distribution at any possible airflow.

Two speed systems — A two speed fan and two position (open or closed) OA/EA dampers can be used to achieve heating and fan energy savings without resorting to the more sophisticated controls required by the systems discussed above. The air terminals used with this system must be selected so they perform adequately at either fan speed.

Table 12 shows the pool heating load at 50% airflow is approximately half the heating load of design cfm. Table 7 shows 4,986-cfm of outdoor air will provide the desired humidity control within the pool enclosure whenever the specific humidity of the outdoor air is below 64 grains.

The psychrometric chart in Figure 2 shows that 64 grains corresponds to an outdoor temperature a little below 60F for a humid climate and about 65F for intermediate climates. This figure shows half speed operation will be effective for the majority of the heating season. Operating the fan at low speed whenever the outdoor temperature is below 45F db (humid) or 50F db (intermediate) will result in considerable energy savings.

Controlling space temperature

The heater tempering the outdoor air must be capable of tracking variations in the outdoor air heating load. This means the majority of indirect fired furnaces and electric heating coils are not suitable because on/off control is not a satisfactory method of tempering makeup air. Therefore, a heating device which can be staged, or which has a suitable turndown ratio is required to eliminate the possibility of supplying cold outdoor air to pool or hot tub enclosures.

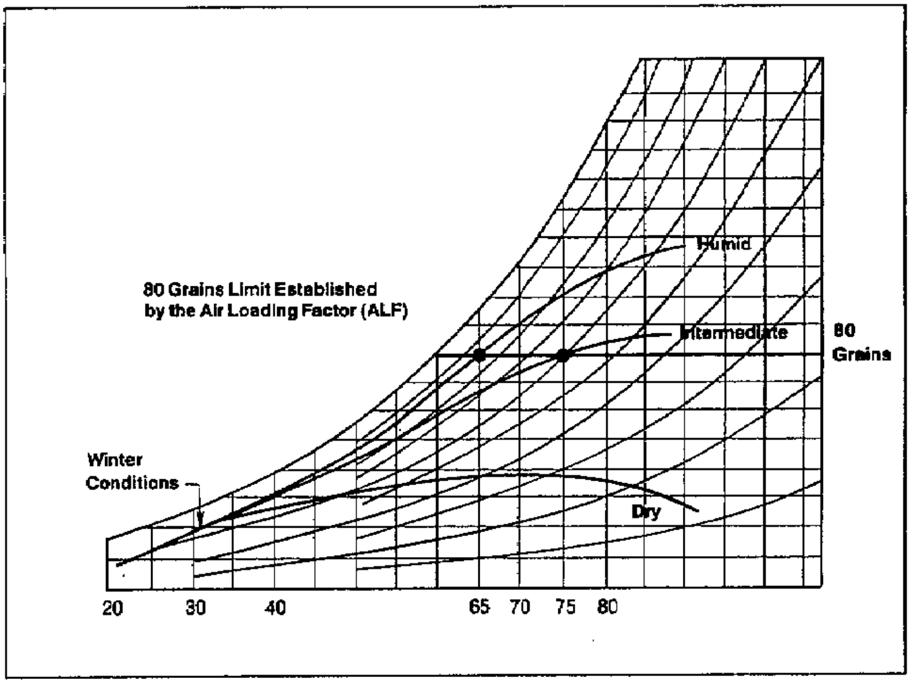

Figure 2
Psychrometric chart plotting differing weather conditions.

Design Outdoor Temperature °F	Envelope Btuh	Outdoor CFM Btuh	Total Btuh
50 °F	32,400	559,354	591,754
40 °F	41,920	723,870	765,790
30 °F	51,440	888,386	939,826
20 °F	60,960	1,052,902	1,113,862
10 °F	70,480	1,217,418	1,287,898
0 °F	80,000	1,381,934	1,461,934
−10 °F	89,520	1,546,450	1,635,970

Table 8
Pool at 14,956-cfm.

If one heating device is used to both offset the transmission load and temper makeup air, it can be controlled by a space thermostat. If a separate heating device is used to temper outdoor air, it can be controlled by a thermostat mounted in the supply duct to control the discharge air temperature.

Energy recovery

Air exhausted from a pool or hot tub enclosure contains a considerable amount of recoverable energy in both sensible and latent forms. Either or both can be recovered and transferred to the makeup air.

Whether or not to install a recovery system will depend on its cost effectiveness. There are often additional costs associated with a recovery device (fans, ducts, duct transitions and controls) that may drive the price higher than a customer is willing to pay.

Outdoor Grains	CFM	ALF
80	14,956	4.67
78	11,965	3.74
76	9,971	3.12
74	8,546	2.67

Table 9
Pool at 9,971-cfm.

Where DX or chilled water coils are used to mechanically dehumidify:

- some form of reheat is almost always required to insure the design temperature will be maintained
- such systems can use a considerable amount of energy — an important factor
- equipment size will be determined by the latent load — this load will be equal to the latent load of evaporation plus that of the outdoor air when the outdoor air damper is in the minimum position
- design cfm will be established by the

design latent load and the manufacturer's performance ratings — the cooling equipment should be selected for the manufacturer's minimum coil cfm.

Selecting a cooling unit (Example)

Input data — summer design conditions:

- Room — $84F_{db}$, $70F_{wb}$, $50\%_{rh}$, $88\text{-gr.}_{sp/h}$
- Outdoor air — $95F_{db}$, $78F_{wb}$, $114\text{-gr.}_{sp/h}$
- Min. outdoor air (ventilation) — 10%
- Latent load (pool evaporation) — $81{,}360_{Btuh}$
- Sensible load — $31{,}600_{Btuh}$

Step #1: Estimate the approximate capacity of the cooling unit by assuming the equipment will operate at a coil shf of 0.65. This means that 35% of the capacity will be available to offset latent load. Therefore:

$$\frac{81{,}360_{Btuh}}{.35} = 230{,}000_{Btuh}\ (<20\text{ tons})$$

It is important to keep design cfm as low as possible because outdoor air must be heated whenever its temperature is below the pool design, and because the cost of the fan and air distribution system increases as design cfm increases.

Step #2: Refer to the manufacturer's data (example: Table 13) for a nominal twenty ton cooling unit to determine the minimum allowable airflow (4000-cfm for this example).

Step #3: Calculate the latent and sensible loads contributed by the outdoor air (10% for ventilation):

Outdoor air cfm =
$10\% \times 4{,}000_{cfm} = 400_{cfm}$.
Latent load$_{(oa)}$ =
0.68 sensible % $\times 400_{cfm} \times (114_{gr} - 88_{gr}) = 7{,}072_{Btuh}$.
Sensible load$_{(oa)}$ =
$1.1 \times 400_{cfm} \times (95F_{oa} - 84F_{in}) = 4{,}840_{Btuh}$.

Step #4: Calculate the design sensible and latent loads then compare them to the manufacturer's application ratings. (See Table 13 for performance at; $4{,}000_{cfm}$, $85F_{edb}$, $70F_{ewb}$, $95F_{oat}$.)

continued

Design Outdoor Temp. F	Envelope Btuh	Pool at 14,956 CFM Outdoor Btuh	Pool at 14,956 CFM Total Btuh	Pool at 9,971 CFM Outdoor Btuh	Pool at 9,971 CFM Total Btuh
50	32,400	559,354	591,754	372,915	404,315
40	41,920	723,870	765,790	482,596	524,516
30	51,440	888,386	939,826	592,277	643,717
20	60,960	1,052,902	1,113,862	701,958	762,918
10	70,480	1,217,418	1,287,898	811,639	882,119
0	80,000	1,381,934	1,461,934	921,320	1,001,320
−10	89,520	1,546,450	1,635,970	1,031,001	1,119,521

Table 10
Comparing the effects of reducing outdoor cfm.

Outdoor Temp. °F	Space Grains Example #1	Space Grains Example #2	Saturated OA Grains	Minimum CFM Example #1	Minimum CFM Example #2
70	88	80	110	NA	NA
60	88	80	78	11,965	NA
50	88	80	54	3,519	757
40	88	80	36	2,301	447
30	88	80	24	1,869	351
20	88	80	15	1,639	303
10	88	80	9	1,515	277
0	88	80	5	1,442	262
−10	88	80	3	1,408	256
			Design CFM =	14,956	984

Table 11
Control humidity decreases as outdoor temperature drops.

Design Outdoor Temp. °F	Envelope Btuh	Pool at 9,971 CFM Outdoor Btuh	Pool at 9,971 CFM Total Btuh	Pool at 4,986 CFM Outdoor Btuh	Pool at 4,986 CFM Total Btuh
50	32,400	372,915	405,315	186,458	218,858
40	41,920	482,596	524,516	241,298	283,218
30	51,440	592,277	643,717	296,139	347,579
20	60,960	701,958	762,918	350,979	411,939
10	70,480	811,639	882,119	405,820	476,300
0	80,000	921,320	1,001,320	460,660	540,660
−10	89,520	1,031,001	1,120,521	515,501	605,021

Table 12
At 50% cfm, heating loads are approx. half those at design cfm.

CFM STD. AIR	ENT. DRY BULB (F)	AMBIENT TEMPERATURE 85 — ENTERING WET BULB 61 MBh	85 / 61 SHR	85 / 67 MBh	85 / 67 SHR	85 / 73 MBh	85 / 73 SHR	95 / 61 MBh	95 / 61 SHR	95 / 67 MBh	95 / 67 SHR	95 / 73 MBh	95 / 73 SHR
4000	75	196	77	219	58	244	42	186	78	208	58	231	42
	80	196	87	219	67	244	50	186	88	208	68	231	50
	85	197	96	219	76	244	58	187	96	208	77	231	59
	90	202	100	219	85	244	66	194	100	208	87	231	67
6000	75	214	83	238	58	264	41	202	85	225	61	250	41
	80	216	95	238	72	264	52	204	97	225	73	249	52
	85	221	100	238	84	264	62	211	100	225	86	249	64
	90	233	100	240	94	264	72	223	100	227	97	249	74
7000	75	220	86	244	62	270	41	208	88	231	63	255	41
	80	222	99	244	75	270	53	211	100	230	76	255	54
	85	232	100	244	87	270	64	221	100	231	89	255	66
	90	244	100	247	98	270	76	233	100	234	100	255	78

Table 13
Manufacturer's data.

continued

Design latent load =
$81{,}360_{ia/l} + 7{,}072_{oa/l} = 88{,}432_{Btuh}$
Design sensible load =
$31{,}600_{ia/s} + 4{,}840_{oa/s} = 36{,}440_{Btuh}$
Total capacity = $219{,}500_{Btuh}$
Latent capacity =
$(1 - 0.66) \times 219{,}500_{tc} = 74{,}630_{Btuh}$
Sensible capacity =
$0.66 \times 219{,}500_{tc} = 144{,}870_{Btuh}$
Approx. supply t.d. =

$$\frac{144{,}870_{Btuh}}{1.1 \times 4{,}000_{cfm}} = 33F_{db}$$

Approx. supply temp. =
$84_{ra} - 33_{(td)} = 51F_{db}$.

The calculations indicate this unit is approximately 10,000 Btuh short on latent capacity. However, if the space humidity is allowed to drift toward 60%, the design latent load will be reduced. Refer to Tables 2 and 3 to calculate a new evaporation load at 60% rh.

Step #5: Determine the changeover temperature (oa/db) that will allow the system to control humidity without mechanical cooling (100% outdoor air is equal to 4,000-cfm).

Air exhausted from a pool or hot tub enclosure contains a considerable amount of recoverable energy in both sensible and latent forms. Either or both can be recovered and transferred to the makeup air.

Table 14 shows 4,000-cfm of outdoor air will control humidity within the enclosure whenever its specific humidity is less than 58 grains.

Using the psychrometric chart we find the specific humidity will be 58 grains or less whenever the outdoor temperature is less than 57F in humid climates and 60F in average climates. These are the changeover points.

Step #6: Determine the minimum outdoor air requirement. This is the outdoor air cfm required to maintain the enclosure humidity at winter design temperature.

Winter design temp. = 0°F, 5 $gr._{sp/h}$
Latent load (evaporation) = $81{,}360_{Btuh}$

Indoor sp. humidity at $84F_{db}$ & $50\%_{rh}$ = 88_{gr}

$$cfm = \frac{\text{latent load}}{.68 \times (gr_{ia} - gr_{oa})}$$

$$\text{Thus: } cfm = \frac{81{,}360_{Btuh}}{.68 \times (88_{gr} - 5_{gr})}$$

Step #7 — Calculate the heating requirements. It is usually necessary to calculate the heating loads corresponding to the various modes of equipment operation and/or equipment control cycles.

The latent component can be a very sizable part of total recoverable energy, therefore a heat recovery device that can extract both sensible and latent heat should be considered.

The design heating load will be equal to the largest of the following heating loads:

- Provide enough reheat to offset the entire sensible capacity of the cooling unit (145,000 Btuh).
- Determine the point where the unit changes over from mechanical cooling at minimum outdoor cfm to ventilation only at 100% outdoor air — This occurs at approximately 55F in humid climates (see Step 5 mentioned earlier) — then do the following calculation:

Envelope heating load at 55_{oat} =
$27{,}300_{Btuh}$.
Reheat load (before changeover) =
$145{,}000_{Btuh}$.
Heat load (before changeover) =
$27{,}300 + 145{,}000 = 172{,}300_{Btuh}$.
Or:
Envelope heating load (at 55_{oat}) =
$27{,}300_{Btuh}$.
OA load =
$1.1 \times 4{,}000 \times (84 - 55) = 127{,}600_{Btuh}$.
Heat load (just after changeover) =
$27{,}300 + 127{,}600 = 154{,}900_{Btuh}$

Check the design heating load at the winter operating conditions. This load equals the outdoor air load plus the envelope load. ■

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Outdoor Grains	CFM for Example #1	ALF for Example #1
60	4,273	1.34
58	4,000	1.26
55	3,626	1.13

Table 14
In the example, 4,000-cfm will control indoor air humidity whenever outdoor air Sp./H is <58-gr.

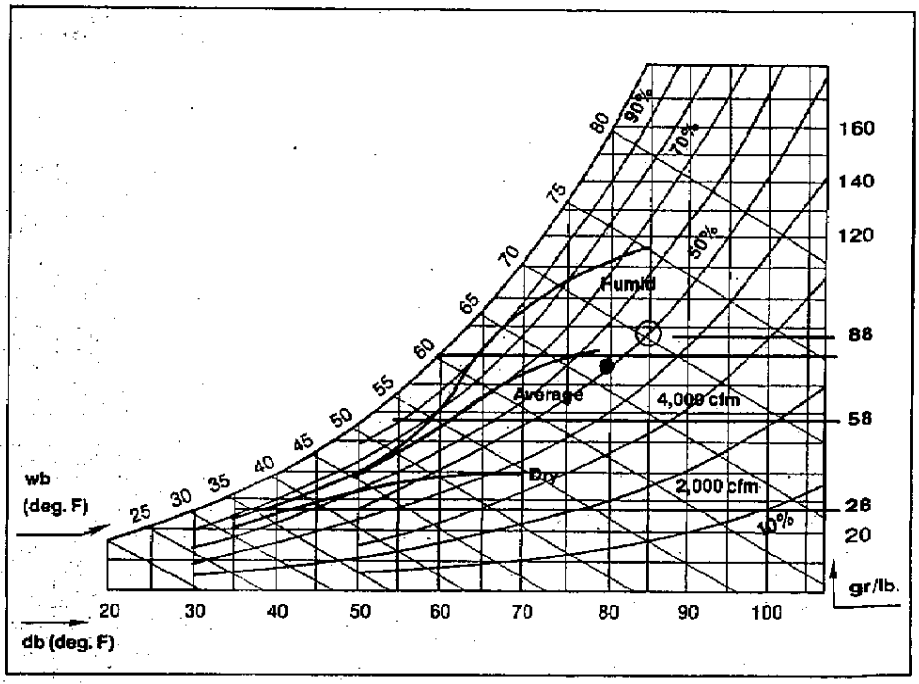

Figure 3
Psychrometric chart shows Sp./H less than 58-gr whenever outdoor temperature is below 57F in humid climates, 60F in average climates.