WHITE PAPER #:

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THE SMART SOLUTION FOR ENERGY EFFICIENCY

CLIMATEMASTER KNOWLEDGE SERIES: TREATMENT OF 100% OUTSIDE AIR

ClimateMaster
WATER-SOURCE HEAT PUMP SYSTEMS

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Introduction

As ASHRAE 62 ventilation codes are implemented for existing or new buildings, many facility managers are encountering new indoor air problems in the form of high humidity, mold, and mildew. This application note reviews the unintended side effects of increasing outside air volumes, and describes a way to solve or prevent these new indoor air problems without a need to change air handlers.

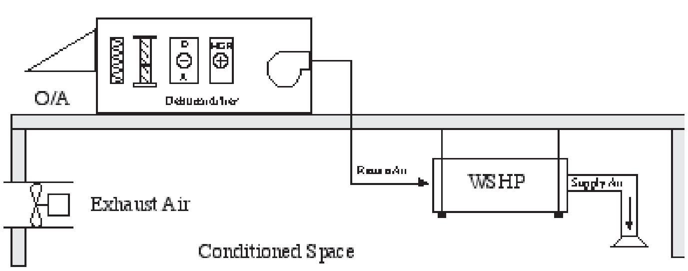
ASHRAE 62 Requirements

The updated ventilation code requires the introduction of 15 to 20 CFM [7.0 to 9.5 l/s] outside air per person for most general applications. This is a three-fold or four-fold increase over the original code requirement of 5 CFM [2.5 l/s] per person. The most common approach to implementing ASHRAE 62 requirements in existing buildings is to simply modify the existing air handler so as to increase the outside air introduced. For new buildings, the first impulse may be to specify more air conditioning capacity to accommodate the added outside air during warm weather.

There is, however, an unintended consequence from these approaches. For an existing system, the original sizing was likely aimed at handling the sensible (indoor) heat load plus only 5 CFM [2.5 l/s] per person of outside load. The significant increase in outside air can result in greatly increased interior humidity during the warm, moist summer months.

For new buildings, even with added cooling capacity the air handler can be inadequate for keeping up with incoming warm, moist air. Usually a certain leaving air dry bulb temperature is targeted, but then excessive moisture is left in the air. (In some cases a particular relative humidity is targeted, in which case the leaving air is far too cold for comfort.) Offices, public facilities, and schools are left with rising interior relative humidity because the air handler design simply cannot remove the additional latent heat load in the summertime.

Figure 1: Pretreatment of Outdoor Air



If humidity is left uncontrolled, new indoor air problems can occur. Occupants complain about working in a "cold swamp" and productivity falls. Viruses, bacteria, mold, and mildew all grow in a humid environment. Increased mold and mildew on interior surfaces cause allergic reactions. Continued high humidity can damage wallboard, metal surfaces, and ultimately the building's structural integrity. Increased outside air solves one indoor air problem only to cause others.

Pretreatment Solution

Is there a way to successfully use existing air handlers, modified to draw additional outside air, to implement the ASHRAE 62 requirements? Can air handlers be applied in new buildings with ASHRAE 62 requirements in a way that prevents moisture problems? Yes! A pretreatment dehumidification system can be used to remove the peak moisture and heat prior to introducing the outside air to the existing air handler. (See Figure 1.)

Ideally, a pretreatment system should emulate the typical return air ("neutral") conditions of 72°F [22°C] and 50% to 60% RH. Then the air handler would see only the level of latent and sensible heat load for which it was originally designed.

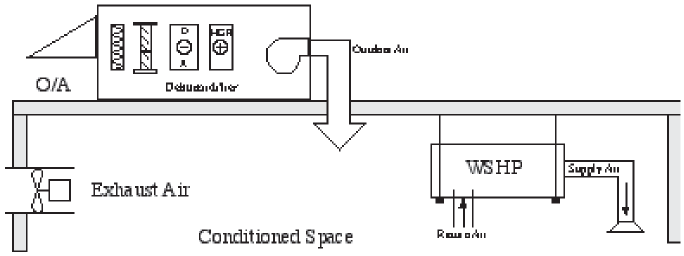
Caution must be applied in choosing the pretreatment system. A standard dehumidification system with full reheat can remove sufficient moisture, but will cause problems because its typical leaving air temperature can rise higher than 95°F [35°C]. A standard air conditioner, meanwhile, cannot remove enough moisture to solve the problem. What is required is a dehumidifier with a partial reheat capability which can consistently ensure that the air leaving the dehumidifier is at 50% RH and neither excessively hot nor cold. In fact, the ideal dehumidifier would not only hit 50% RH, but would have a variable partial reheat capability so that the air passed along to the air handler is consistently at room air design conditions. (See ClimateMaster White Paper #1 for a discussion of options, and the energy consequences of various choices.)

Direct Feed to Space

In some applications it is desirable to have the outside air fed directly into specific rooms, rather than using the indirect method of dumping the outside air into the air handler. Naturally, the issues just described apply in this situation as well. The solution is similar, except that the outside air, pretreated by the partial reheat dehumidifier, now enters directly into the building rather than into the air handler.

When direct feed of outside air into the space is used, it is especially important to specify that the air temperature be controlled to a specific value in all modes of operation: full load, part load and winter. Without specific temperature control, room occupants are likely to be very uncomfortable as temperatures of the air being introduced vary widely. A variable partial reheat dehumidifier is especially useful in this instance in order to achieve temperature control.

Figure 2: Direct Feed of Pretreated Air to Interior Space



Calculating Energy Removal Requirements
The air entering the dehumidification system is 100% outside air. Proper system size is selected by calculating the amount of energy that must be removed from entering air at the maximum design condition to achieve a desired leaving air dewpoint (LAD). The most direct calculation method is known as the total enthalpy method. It is based on the enthalpy difference (BTU/lb) [kJ/kg] between the maximum design condition and the specified leaving air condition, multiplied by the air flow.

I.P. Units:
Rate of energy removal required (BTU/hr) =
Enthalpy difference ΔH (BTU/lb) x air flow (cu ft/min) x 4.5 (min/hr x lb/cu ft)

The 4.5 is a conversion factor of 60 minutes/hour divided by 13.5 cu ft/lb (of air), and CFM is the specified outside air volume.

S.I. Units:
Rate of energy removal required (kW) = Enthalpy difference ΔH (kJ/kg) x air flow (l/s) x 0.0012 kg/l (of air)

The 0.0012 is a conversion factor for air (0.0012 kg per liter), and J/s = Watts, leaving kW as the result. Airflow (l/s) is the specified outside air volume.

Since the weight of air varies with temperature, further accuracy could be gained by using the precise weights for the two different temperatures involved, but this approximation is nearly always sufficient for sizing purposes.

The enthalpy difference is calculated by taking the enthalpy value (BTU/lb) [kJ/kg] at the entering wet bulb temperature and subtracting the enthalpy value at the design dewpoint. Table 1 provides typical design wet bulb values for major cities. (The data in Table 1 is taken from Table 1B of ASHRAE 97 Fundamentals.) Table 2 lists enthalpy values at various dewpoint temperatures.

As an example, suppose we are sizing a pretreatment dehumidifier for a building in St. Louis, with required outside air introduction of 2000 CFM [944 l/s]. Table 1 gives a wet bulb temperature design value of 78°F [26°C], and Table 2 shows an associated enthalpy value of 41.5 BTU/lb [96.5 kJ/kg] (78°F wb = 78°F dewpoint [26°C wb = 26°C dewpoint]). If our air handler expects air at 72°F [22°C] and 55%RH, or 55°F [13°C] dew point, we can look up a corresponding enthalpy from Table 2 of 23.2 BTU/lb [54.0 kJ/kg]. Our dehumidifier will need sufficient capacity to remove energy at the following rate:

I.P. Units:
Rate of energy removal required (BTU/hr) =
(41.5 - 23.2) x 2000 x 4.5 = 164,700 BTU/hr

S.I. Units:
Rate of energy removal required (kW) =
(96.5 - 54.0) x 944 x 0.0012 = 48 kW

This energy removal rate is then compared to the capacities for various dehumidification systems to help determine the best system for the application.

Table 1: ASHRAE 1% Design Points

WET BULB TEMPERATURES °F [°C]											
	City	1%		City	1%		City	1%		City	1%
AK	Anchorage	60 [16]	I□	Fort Wayne	77 [25]	□□	Las □egas	71 [22]	T□	Lubbock	73 [23]
AL	Birmingham	78 [26]		Indianapolis	78 [26]		Reno	64 [18]		Odessa	73 [23]
	Mobile	80 [27]	KS	Wichita	77 [25]	□□	Albany	75 [24]		San Antonio	77 [25]
AR	Little Rock	80 [27]	K□	Louisville	79 [26]		Buffalo	74 [23]	UT	Salt Lake City	66 [19]
A□	Phoenix	76 [24]	LA	Baton Rouge	80 [27]		□ew □ork	76 [24]	□A	□orfolk	79 [26]
CA	Long Beach	70 [21]		□ew Orleans	81 [27]		Rochester	75 [24]		Richmond	79 [26]
	Los Angeles AP	70 [21]		Shreveport	79 [26]		Syracuse	75 [24]		Roanoke	75 [24]
	Sacramento	72 [22]	MA	Boston	75 [24]	OH	Cincinnati	77 [25]	□T	Burlington	74 [23]
	San Diego	71 [22]	MD	Baltimore	80 [27]		Cleveland	76 [24]	WA	Seattle	69 [21]
	San Francisco AP	65 [18]	ME	Caribou	71 [22]		Columbus	77 [25]		Spokane	65 [18]
	Santa Barbara	68 [20]		Portland	74 [23]	OK	Oklahoma City	78 [26]		□akima	68 [20]
	Stockton	71 [22]	MI	Detroit	76 [24]	OR	Eugene	69 [21]	WI	□reen Bay	76 [24]
CO	Denver	64 [18]		Flint	76 [24]		Portland	69 [21]		Madison	77 [25]
CT	Hartford	77 [25]		□rand Rapids	75 [24]	PA	Erie	75 [24]		Milwaukee	76 [24]
DC	Washington □at□l	78 [26]		Sault St. Marie	72 [22]		Philadelphia	77 [25]	W□	Charleston	76 [24]
DE	Wilmington	77 [25]	M□	Duluth	72 [22]		Pittsburgh	74 [23]	W□	Cheyenne	65 [18]
FL	Daytona Beach	80 [27]		Rochester	77 [25]		Scranton	74 [23]	CA□ADA		
	Fort Myers	80 [27]		St. Paul	77 [25]	RI	Providence	75 [24]			
	Jacksonville	79 [26]	MO	Kansas City	78 [26]	SC	Charleston	81 [27]	AL	Calgary	65 [18]
	Miami	79 [26]		St. Louis	78 [26]		Columbia	79 [26]	BC	□ancouver	68 [20]
	Orlando	79 [26]	MS	Jackson	79 [26]	SD	Sioux Falls	76 [24]	M□	Winnipeg	75 [24]
	Pensacola	80 [27]		Meridian	80 [27]	T□	Bristol	75 [24]	□B	Saint John	70 [21]
	Tallahassee	79 [26]	MT	Billings	67 [19]		Chattanooga	78 [26]	□F	St. John□s	69 [21]
	Tampa	79 [26]	□C	Wilmington	81 [27]		Knoxville	77 [25]	□S	Halifax	69 [21]
□A	Atlanta	77 [25]		Charlotte	77 [25]		Memphis	80 [27]	O□	Ottawa	75 [24]
	Augusta	79 [26]		Raleigh	78 [26]		□ashville	78 [26]		Sudbury	72 [22]
HI	Honolulu	76 [24]	□D	Fargo	76 [24]	T□	Brownsville	80 [27]		Thunder Bay	72 [22]
IA	Des Moines	78 [26]	□E	Omaha	78 [26]		Corpus Christi	80 [27]		Toronto	75 [24]
	Dubuque	77 [25]	□H	Concord	74 [23]		Dallas	78 [26]		Windsor	77 [25]
ID	Boise	68 [20]	□J	Atlantic City	78 [26]		El Paso	69 [21]	□C	Montreal	75 [24]
IL	Chicago	79 [26]		□ewark	77 [25]		Fort Worth	78 [26]		□uebec	74 [23]
	Rockford	77 [25]	□M	Albuquerque	66 [19]		Houston	80 [27]	SK	Regina	72 [22]

Table 2: Enthalpy Values @ Dewpoint

Enthalpy Values BTU/lb [kJ/kg] At Various Dewpoint Temperatures °F [°C] RH => 99.90%					
°F [°C]	BTU/lb [kJ/kg]	°F [°C]	BTU/lb [kJ/kg]	°F [°C]	BTU/lb [kJ/kg]
35 [1.7]	12.9 [30.0]	52 [11.1]	21.4 [49.8]	69 [20.6]	33.2 [77.2]
36 [2.2]	13.4 [31.2]	53 [11.7]	22.0 [51.2]	70 [21.1]	34.0 [79.1]
37 [2.8]	13.8 [32.1]	54 [12.2]	22.6 [52.6]	71 [21.7]	34.9 [81.2]
38 [3.3]	14.3 [33.3]	55 [12.8]	23.2 [54.0]	72 [22.2]	35.8 [83.3]
39 [3.9]	14.7 [34.2]	56 [13.3]	23.8 [55.4]	73 [22.8]	36.7 [85.4]
40 [4.4]	15.2 [35.4]	57 [13.9]	24.5 [57.0]	74 [23.3]	37.6 [87.5]
41 [5.0]	15.7 [36.5]	58 [14.4]	25.1 [58.4]	75 [23.9]	38.5 [89.6]
42 [5.6]	16.1 [37.4]	59 [15.0]	25.8 [60.0]	76 [24.4]	39.5 [91.9]
43 [6.1]	16.6 [38.6]	60 [15.6]	26.4 [61.4]	77 [25.0]	40.5 [94.2]
44 [6.7]	17.1 [39.8]	61 [16.1]	27.1 [63.0]	78 [25.6]	41.5 [96.5]
45 [7.2]	17.6 [40.9]	62 [16.7]	27.8 [64.7]	79 [26.1]	42.5 [98.9]
46 [7.8]	18.1 [42.1]	63 [17.2]	28.5 [66.3]	80 [26.7]	43.6 [101.4]
47 [8.3]	18.7 [43.5]	64 [17.8]	29.3 [68.2]	81 [27.2]	44.6 [103.7]
48 [8.9]	19.2 [44.7]	65 [18.3]	30.0 [69.8]	82 [27.8]	45.7 [106.3]
49 [9.4]	19.7 [45.8]	66 [18.9]	30.8 [71.6]	83 [28.3]	46.9 [109.1]
50 [10.0]	20.3 [47.2]	67 [19.4]	31.6 [73.5]	84 [28.9]	48.1 [111.9]
51 [10.6]	20.8 [48.4]	68 [20.0]	32.4 [75.4]	85 [29.4]	49.3 [114.7]

The ASHRAE guidelines in Table 1 state the design condition simply as a peak wet bulb temperature. Associated with that temperature is a wet bulb line on the psychrometric chart. Sizing for the enthalpy difference between the peak wet bulb and the leaving air dewpoint will ensure that the dehumidifier can handle the wide variety of dry bulb temperature / RH combinations that fall along or beneath the wet bulb line. (See Figure 3.) A dehumidifier sized to remove the necessary energy to meet a 78°F [26°C] wet bulb requirement for St. Louis, for example, will also handle 85°F [29°C] up to 70% RH or 90°F [32°C] up to 60% RH. If the dehumidifier was tested at different points along the wet bulb line, the amounts of latent versus sensible heat removed would change significantly, but the total heat removed would not.

Note that the total enthalpy method simplifies the sizing discussion by focusing on total energy removal (combined latent and sensible) rather than on a moisture load (often presented in lb/hr [kJ/kg]) to be handled by the dehumidifier. Instead of trying to develop a moisture load from dewpoint and wet bulb values, the values are used directly to arrive at the required dehumidifier capacity.

Figure 3: Total Enthalpy Psychrometric Chart

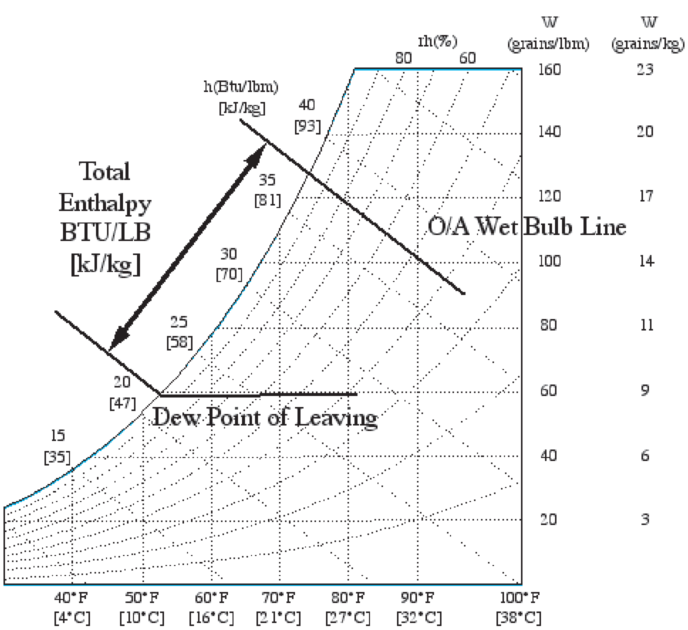


Table 3: Dehumidifier Sizing

Entering °F [°C] wb	Unit Size HP [kw]	LAT dewpoint °F [°C]	Unit Size HP [kw]	LAT dewpoint °F [°C]
80 [27]	14 [49]	55 [13]	10 [35]	60 [16]
78 26]	12 [42]	55 [13]	9 [32]	59 [15]
76 [24]	10 [35]	55 [13]	-	-
74 [23]	9 [32]	54 [12]	7.5 [26]	59 [15]
72 [22]	7.5 [26]	57 [14]	6 [21]	59 [15]
70 [21]	7.5 [26]	55 [13]	6 [21]	57 [14]
68 [20]	6 [21]	55 [13]	-	-
66 [19]	5 [18]	55 [13]	5 [18]	60 [16]

Dehumidifier Selection & Performance

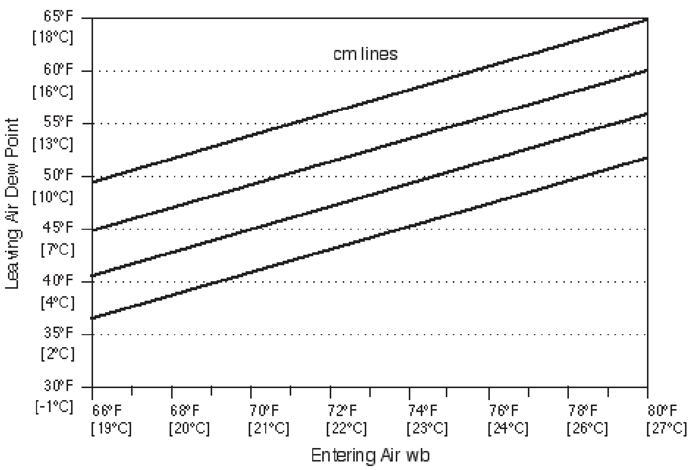
With 100% outside air dehumidifiers, it is important to understand how to select the correct system for the application as well as to understand how the dehumidifier will perform under the varying full and part load conditions it will encounter.

The correct dehumidifier is selected by specifying the following criteria:

Volume of air required
Max. design condition (db/wb)
Leaving air dewpoint required
Desired Leaving Air Temperature

The dehumidifier will be sized to balance the air velocity across the coils, the capacity of the compressor and the condensing temperature of the condensers. A wide range of systems can be selected to meet the criteria above. Table 3 shows the various sizes and their corresponding leaving air dew points for various maximum design ambient wet bulb conditions. The selections are for 2,000 CFM [944 l/s] at a 95°F [35°C] db ambient.

Figure 4: Dehumidifier Performance



The total energy removal required, and therefore the dehumidification capacity needed, is directly proportional to air flow. Conversely, for the same air flow, a lower leaving air dewpoint can be achieved by moving to a dehumidification system with greater capacity.

For example, compare the performance of two dehumidifiers with entering air at 78°F [26°C] wet bulb, a 2000 CFM [944 l/s] air flow requirement to meet ASHRAE 62, and a required leaving air dewpoint of 55°F [13°C] or lower to match the original design conditions for an existing air handler. (See Table 3 for the capacities.) At an air flow of 2000 CFM [944 l/s], the smaller unit can only produce a leaving air dewpoint of 60°F [16°C], which will not meet our 55°F [13°C] requirement. The larger unit, at the 2000 CFM [944 l/s] air flow, can produce a leaving air dewpoint of 55°F [13°C], and would be acceptable for this application.

A convenient way to portray the performance of a dehumidification system over the wide range of ambient conditions is by plotting on a graph with "entering air wet bulb temperatures" on the x axis and "leaving air dewpoint capabilities" on the y axis. The graph shows a family of curves corresponding to different air flow levels. (See Figure 4.) Given the entering wet bulb temperature and the air flow, the leaving air dewpoint can be read off the chart to show the resultant leaving air condition at part load conditions.

Reheat
One of the greatest benefits of using a refrigeration-type mechanical dehumidifier for pretreatment is the availability of free reheat energy. A partial reheat dehumidifier will use energy recovered during moisture removal to produce, via hot gas reheat, leaving air temperatures in a range (typically 65°F to 80°F [18°C to 27°C]) that is likely to be acceptable to the air handler. A variable partial reheat adjusts the amount of hot gas reheat continuously to hit a particular leaving air temperature (e.g., 72°F [22°C]) chosen by the design engineer.

Thus, the designer can specify the dry bulb temperature (or temperature range) and the RH of the pretreated outside air going into the air handler. Any energy required to warm the dehumidified air is recovered from the moisture removal process rather than being added using a heater. In contrast, when a standard air conditioner is used to remove large amounts of moisture from air, the leaving air is unacceptably cold unless a substantial amount of electric reheat is used. The result of using air conditioning for moisture removal is significantly increased operating costs. (Refer to ClimateMaster White Paper #1 for a detailed analysis of reheat technologies and energy savings.)

Conclusion
To allow an existing air handler, modified to meet the ASHRAE 62 ventilation code, to function as it was originally designed, the added outside air must be pretreated to match typical return air conditions. Similarly, in new designs for ASHRAE 62, pretreatment of outside air before it is introduced to the air handler or the space is a necessary part of any practical solution, since simply adding air conditioning capacity is not a desirable method of removing moisture from that air. An effective solution in new and existing buildings is pretreatment by a dehumidifier with partial or variable partial reheat, to remove peak latent heat load and maintain reasonable entering air conditions for the air handler.

Proper dehumidification system sizing can be accomplished by calculating the amount of total (latent and sensible) heat to be removed per hour from the additional outside air, based on ASHRAE wet bulb temperature design values. As a convenience, some manufacturers provide graphs (for each size dehumidification system made) from which the leaving air dewpoint can be obtained for a given entering wet bulb temperature and air flow requirement.

Without pretreatment, increased outside air brought into an air handler solves one indoor air problem only to cause others. By pre-treating outside air with a partial or variable partial reheat dehumidification system, all the benefits of a healthy, productive environment for building occupants can be realized without introducing excessive moisture or improper temperatures.

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