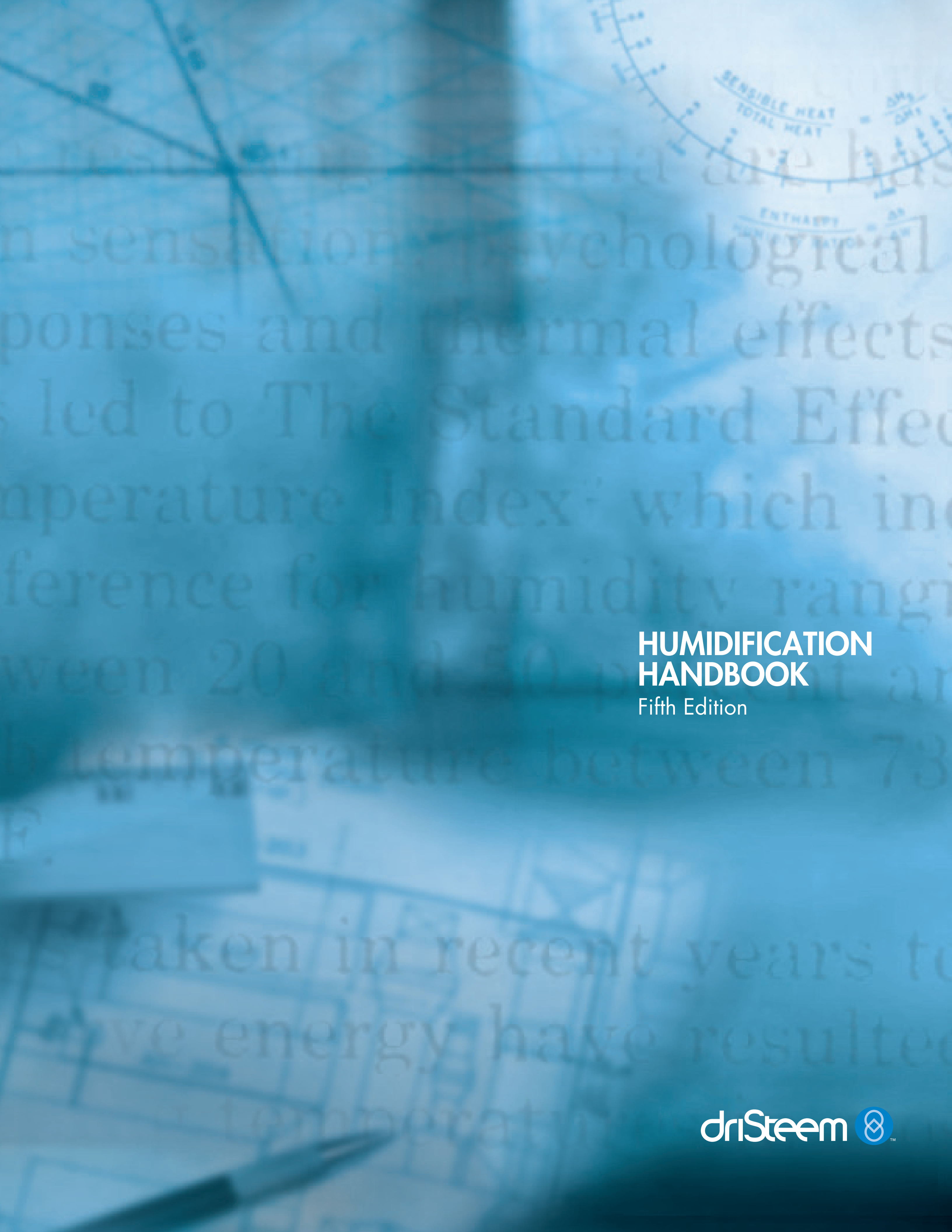

HUMIDIFICATION HANDBOOK

Fifth Edition

driSteem

Humidification Handbook

FIFTH EDITION

BERNARD W. MORTON

AND

DRI-STEEM CORPORATION

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FORWARD

Humidification, being able to add water vapor to the air in controlled amounts, continues to be a growing industry.

What once was considered a healthful luxury to be enjoyed by few has become a necessity to a significant segment of the population. As more and more people around the world discover the health and other benefits to be derived from its use, the growth will continue.

Various high-tech industries, such as semi-conductor, pharmaceutical and electronic, are heavily dependent upon controlled humidification for their success.

A great deal of information exists on this relatively narrow but important subject. It is, however, scattered widely in various technical papers, magazine articles, and technical manuals. The ASHRAE manuals contain much of the information.

This handbook seeks to draw together the information contained in those various sources and to add to it in layman's terms, the knowledge gained by the author in over thirty years of designing, manufacturing, and applying various types of humidification equipment.

It is hoped that the HVAC industry will be better able to serve its clientele by referring to this work for design information and guidance in the creation of humidification.

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Section I: Terminology

Although humidification itself is a simple process, it does work within the confines of certain very specific terminology. To understand the humidification process completely, it is most important that we first have a good understanding of that basic terminology.

HUMIDITY

Air, for the purpose of this discussion, is defined as a gaseous mixture of nitrogen, oxygen, carbon dioxide, water vapor (humidity), several inert gases, and traces of ozone and submicroscopic solid matter, sometimes called permanent atmospheric substances.

All other airborne substances are considered contaminants. The water vapor (humidity), being a gas, occupies space along with the other gases of the air.

In HVAC engineering, air is considered as being made up of only two components-dry air and water vapor. The properties of the dry air, composed of nitrogen, oxygen, carbon dioxide and rare gases, remain relatively unchanged as the temperature of the air rises and falls.

The water vapor, on the other hand, may undergo considerable alteration as the temperature changes, including changes of state (condensing and freezing). Substantial amounts of energy are involved in these transformations.

In measuring these changes, and when working with the processing of air for various air conditioning problems, the engineer is concerned with two basic laws-

Thermodynamics and Psychrometry.

Thermodynamics is the study of heat energy transformations and substances which are affected by them.

Psychrometry deals with the measurement of atmospheric conditions, particularly the moisture mixed with air.

A psychrometric chart is a graphic representation of the thermodynamic tables which the HVAC engineer finds convenient to use when plotting solutions for the various air conditioning processes involving water vapor and temperature changes.

RELATIVE HUMIDITY

When we wish to describe the “wetness” or “dryness” of air at a given temperature and pressure, we use the term RELATIVE HUMIDITY. This tells us the amount of moisture present in the air at a given temperature compared to what the air could hold at that temperature. It is expressed as a percentage.

ABSOLUTE HUMIDITY, HUMIDITY RATIO AND SPECIFIC HUMIDITY

Each of the above terms is expressed as a number which describes a unit weight of water vapor associated with a unit weight of dry air. It is commonly expressed as fractional pounds (or kilograms) of water vapor per pound (or kilogram) of dry air or, if the use of whole numbers is preferred, it is expressed in grains of moisture per pound of dry air. There are 7000 grains in one pound.

DUCT EQUIVALENT RELATIVE HUMIDITY

This is the relative humidity of a duct air stream at a given temperature as compared to the relative humidity of the space served, at a different temperature. For example, a 55°F duct air stream has a “duct equivalent” relative humidity of 80% when compared to a room condition of 72°F and 45% relative humidity. This data is needed when absorption distance evaluations of duct steam dispersers are being made.

DRY BULB TEMPERATURE

is simply the temperature of the air indicated by any type of thermometer or thermocouple that has not been affected by evaporation or radiation.

WET BULB TEMPERATURE

That is an expression of the temperature of the air when a wick or "sock," wetted with water, encases the sensing element of a dry bulb thermometer, and air is passed over it at a velocity of 700 feet per minute or more. The drier the air, the greater is the cooling caused by evaporation and, therefore, the lower the wet bulb temperature.

MEAN RADIANT TEMPERATURE

This is the temperature of a uniform black enclosure in which a solid body or occupant would lose the same amount of heat by radiation only as in a non-uniform environment such as a room that has cool walls.

This factor is one of the components that make up the "Standard Effective Temperature Index."

DEW POINT TEMPERATURE

This is the saturation temperature corresponding to the humidity ratio and pressure of a given moist-air state. In other words, it is the surface temperature at which moisture begins to condense on that surface. The more humid the air, the higher the dew-point temperature. Conversely, the dryer the air, the lower the dew-point temperature.

VAPOR MIGRATION

Water vapor, being a gas, behaves under the laws of low pressure gases. One of these laws, simply stated, says that "in a mixture of gases, the total pressure is the sum of the individual pressures exerted by each of the gases." This means that , in a mixture of water vapor and dry air, the water vapor has its own vapor pressure and will migrate from areas of higher vapor pressure to areas of lower vapor pressure. This migration occurs regardless of air movement, although if the air movement is in the same physical direction, it will be accelerated. This characteristic can sometimes be taken advantage of when designing a humidification system for a large space served by more than one air distribution system. (See Section XV.)

Further, the speed at which this migration takes place is dependent upon the vapor pressure difference between the two areas. The greater the difference, the faster the migration. (See Section VIII for calculation methods.)

It is important to keep this phenomenon in mind when designing humidification for buildings, or spaces within buildings. It may be necessary to consider the use of building materials having vapor barrier qualities in order to prevent loss of moisture, condensation and/or frost formation, within the walls of the structure and its resulting damage.

LATENT HEAT

Latent means "hidden." In HVAC usage it commonly refers to "change of state," which is the heat involved in fusion (freezing water or melting ice) or vaporization (creating water vapor or condensation) with no change in temperature. For water, fusion requires 144 BTU per pound, and vaporization requires 970 BTU per pound. These values are for sea level atmospheric pressure and vary as the pressure changes. Latent heat is not the same for all substances.

SENSIBLE HEAT

Sensible means, in this case, "that which can be sensed." In HVAC usage, it refers to the heat required to cause a change in temperature. The change is detected or "sensed" by the use of a thermometer.

SPECIFIC HEAT

This is the heat required to cause a one degree change in temperature in a unit mass of a substance. Common units are BTUs per pound (Fahrenheit degree) or calories per gram (Celsius degree). The specific heat of water is 1.

Section II: Glossary of Clean Room Terms

Clean Room Terminology

ASEPTIC SPACE

A space controlled such that bacterial growth is contained within acceptable limits; not a sterile space.

CFU (COLONY FORMING UNIT)

A visible growth of microorganisms arising from a single cell or multiple cells.

CHALLENGE

A dispersion of known particle size and concentration used to test filter integrity and efficiency.

CLASS 1

Particle count not to exceed 1 particle* per cubic foot (35 particles/M3).

CLASS 10

Particle count not to exceed 10 particles* per cubic foot.

CLASS 100

Particle count not to exceed 100 particles* per cubic foot.

CLASS 10,000

Particle count not to exceed 10,000 particles* per cubic foot.

CLASS 100,000

Particle count not to exceed 100,000 particles* per cubic foot.

CRITICAL PARAMETER

A room variable (such as temperature, humidity, air changes, room pressure, particulates, viables, etc.) that affects product strength, identity, safety, purity, or quality (SISPQ).

** Particle as defined as .5 micron.*

CRITICAL SURFACE

The surface of the work part to be protected from particulate contamination.

EXFILTRATION

Leakage of air out of a room through pass-throughs and cracks in doors, through material transfer openings, etc., due to a difference in space pressures.

FIRST AIR

The air that issues directly from the HEPA filter before it passes over any work location.

HIGH EFFICIENCY PARTICULATE ARRESTER (HEPA)

A filter with an efficiency in excess of 99.97% of 0.3 micron particles.

PARENTERAL PRODUCT

A pharmaceutical product normally meant to be injected into the patient. Parenterals are manufactured under aseptic conditions or are terminally sterilized to destroy bacteria and meet aseptic requirements.

POLYDISPERSED PARTICLES

A filter challenge utilizing particles of different sizes to determine the cleanliness of a space.

PRIMARY AIR

Air that recirculates through the work space.

SECONDARY AIR

That portion of the primary air circulated through the air-conditioning equipment.

ULTRA LOW PENETRATION AIR (ULPA) FILTER.

A filter with a minimum of 99.999% efficiency on .12 micron.

Section III: Humidity Measurement and Control

HUMIDITY MEASUREMENT

Many instruments are available for measuring the moisture content of air. The sensors used on the instruments respond to different moisture content properties that are related to factors such as wet-bulb temperature, relative humidity, humidity (mixing) ratio, dew point, and frost point.

The hygrometer is an instrument that has the capability of measuring the moisture content of air, as well as other physical materials.

The following sections describe various instruments used to measure humidity, their principle of operation, and the quality and speed of data produced by the instrument.

MECHANICAL HYGROMETERS

Many organic materials change in dimension with changes in humidity; this action is used in a number of simple and effective humidity indicators, recorders, and controllers (see Chapter 41 of the 1995 ASHRAE Handbook—Systems and Equipment). They are coupled to pneumatic leakports, mechanical linkages, or electrical transduction elements to form hygrometers.

Commonly used organic materials are human hair, nylon, dacron, animal membrane, animal horn, wood and paper. Their inherent nonlinearity and hysteresis must be compensated within the hygrometer. These devices are considered unreliable below 32°F and response is generally inadequate for monitoring a changing process. Responses can be affected significantly by exposure to humidity extremes. Such devices require initial and frequent calibration; however, they are useful, because they can be arranged to read directly in terms of relative humidity, and they are simpler and less expensive than most other types.

INDUSTRIAL PSYCRHOMETER

A typical industrial psychrometer consists of a pair of matched electrical or mechanical temperature sensors, one of which is kept wet with a moistened wick. A blower aspirates the sensors, which lowers the temperature at the moistened temperature sensor. The accuracy of the process depends on the purity of the water, cleanliness of the wick, ventilation rate, radiation effects, size and accuracy of the temperature sensors.

FIGURE 18-1: DIMENSIONAL CHANGE HYGROMETER WHICH USES HAIR AS THE SENSING ELEMENT

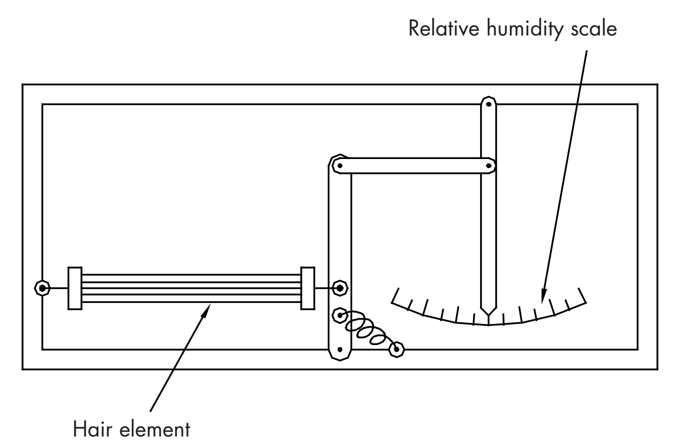
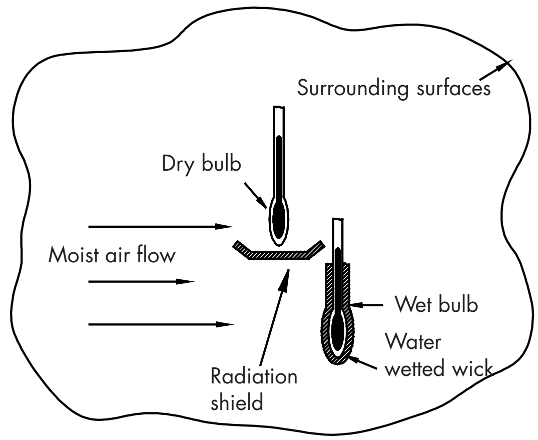


FIGURE 18-2: SCHEMATIC PSYCHROMETER

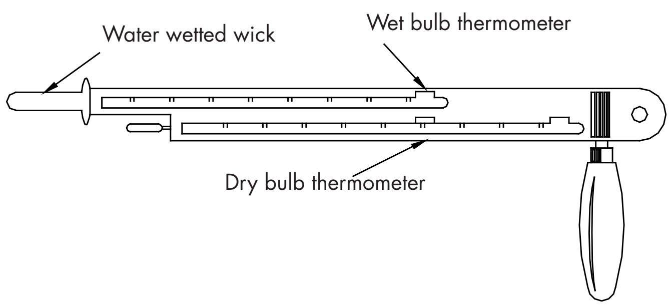


SLING PSYCHROMETER

The sling psychrometer consists of two thermometers mounted side by side in a frame fitted with a handle for twirling the device through the air. The device is spun until the readings of the thermometers stabilize.

ASHRAE Standard 41.6-1992R recommends an airflow over both the wet and dry bulbs of 600 to 1000 fpm for transverse ventilation and 300 to 500 fpm for axial ventilation.

FIGURE 19-1: TYPICAL HAND-HELD SLING PSYCHROMETER



ELECTRICAL IMPEDANCE HYGROMETERS

Many substances adsorb or lose moisture with changing relative humidity and exhibit corresponding changes in electrical impedance. The following are examples of this type of hygrometers.

POLYMER FILM ELECTRONIC HYGROMETERS

The sensor in these devices have hygroscopic organic polymer deposited with thin or thick film-processing technology on a water permeable substrate. Both capacitance and impedance sensors are available. The impedance devices are composed of either ionic or electronic conduction types. These hygrometers typically have integrated circuits that provide temperature correction and signal conditioning. The primary advantages of this sensor technology are small size, low cost, and fast response times in the order of 1 to 120 seconds (s) for 64% change in relative humidity.

ION EXCHANGE RESIN ELECTRIC HYGROMETERS

A conventional ion exchange resin consists of a polymer, with a high relative molecular mass, and polar groups of positive or negative charge in cross-link structure. Associated with these polar groups are ions of opposite charge that are held by electrostatic forces to the fixed polar groups. In the presence of water or water vapor, the electrostatically held ions become mobile; thus, when a voltage is impressed across the resin, the ions are capable of electrolytic conduction. The Pope cell is one example of an ion exchange element. It is a wide-range sensor, typically covering 15 to 95% RH. Therefore, one sensor can be used where several Dunmore elements would be required. The Pope cell, however, has a nonlinear characteristic from approximately 1000 ohms at 100% RH to several megohms at 10% RH.

FIGURE 19-2: A PSYCHROMETER INSTALLED IN A DUCT

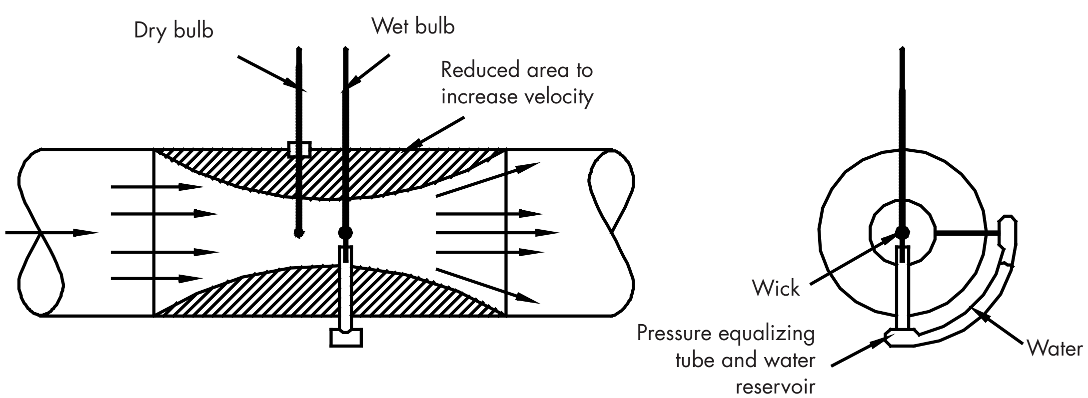
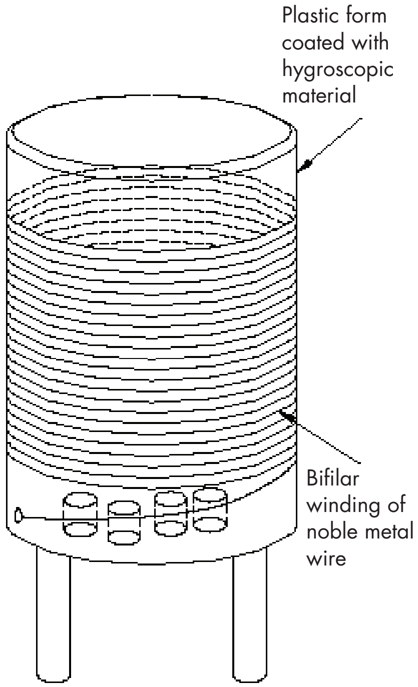


FIGURE 20-1: ELECTRICAL HYGROMETER



DNMORE HYGROMETERS

This device utilizes a sensor consisting of dual electrodes on a tubular or flat substrate; it is coated with a film containing salt, such as lithium chloride, in a binder to form an electrical connection between windings. The relation of sensor resistance to humidity is usually represented by graphs provided by the manufacturer. Since the sensor is highly sensitive, the graphs are a series of curves, each for a given temperature, with intermediate values found by interpolation. Several Dunmore resistance elements are required to cover a normal measurement range. Systematic calibration is essential, since the resistance grid varies with time and contamination as well as with exposure to temperature and humidity extremes.

IMPEDANCE-BASED POROUS CERAMIC ELECTRONIC HYGROMETERS

Utilizing the adsorption characteristics of oxides, humidity-sensitive ceramic oxide devices use either ionic or electronic measurement techniques to relate adsorbed water to relative humidity. Ionic conduction is produced by dissociation of water molecules forming surface hydroxyls. The dissociation causes migration of protons such that device impedance decreases with increasing water content. The ceramic oxide is sandwiched between porous metal electrodes that connect the device to an impedance-measuring circuit for linearizing and signal conditioning. These sensors have excellent sensitivity, are resistant to contamination and high temperature (up to 400°F), and may get fully wet without sensor degradation. They are capable of ±1% RH accuracy when temperature compensated. Their cost is moderate.

ALUMINUM OXIDE CAPACITIVE SENSOR

This sensor consists of a aluminum strip that is anodized by a process that forms a porous oxide layer. This structure is then coated with a very thin layer of gold. The aluminum base and the gold layer form the two electrodes of what is essentially an aluminum oxide capacitor.

Water vapor is rapidly transported through the gold layer and equilibrates on the pore walls in a manner functionally related to the vapor pressure of water in the atmosphere surrounding the sensor. The number of water molecules adsorbed on the oxide structure determines the capacitance between the two electrodes.

CHILLED MIRROR DEW POINT HYGROMETER

The condensation-type (chilled mirror) dew-point hygrometer is an accurate and reliable instrument with a wide humidity measurement range. These features are obtained through complex and costly means when compared to the psychrometer. In the condensation-type hygrometer, a surface is cooled (thermoelectrically, mechanically, or chemically) until dew or frost begins to condense on that surface. The condensing surface is maintained electronically in vapor pressure equilibrium with the surrounding gas, while surface condensation is detected by optical, electrical, or nuclear techniques.

ELECTROLYTIC HYGROMETERS

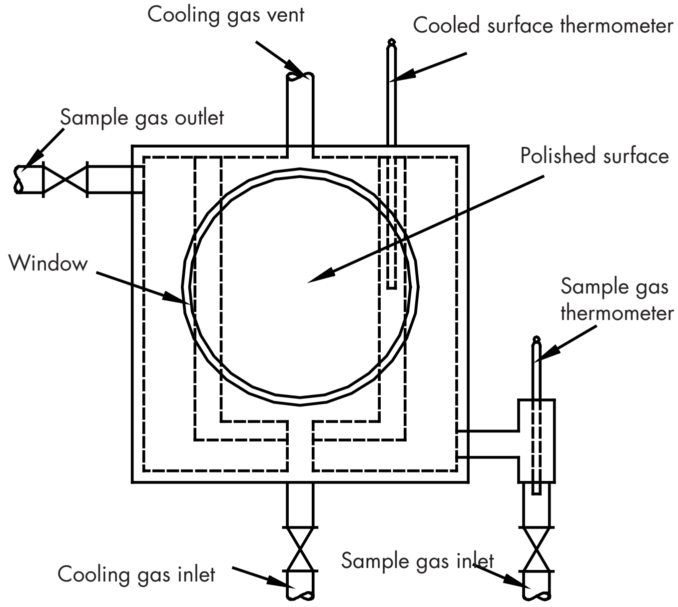
In electrolytic hygrometers, air is passed through a tube where moisture is adsorbed by a highly effective desiccant, usually phosphorous pentoxide and electrolyzed. The airflow is regulated to 0.0035s cfm. As the incoming water vapor is absorbed by the desiccant and electrolyzed into hydrogen and oxygen, the current of electrolysis determines the mass of water vapor entering the sensor. The flow rate of the entering gas is controlled precisely to maintain a standard sample mass flow rate into the sensor. The instrument is usually designed for use with moisture-air rations in the range of less than 1 to 1000 ppm, but can be used with higher humidities.

PIEZOELECTRIC SORPTION

This hygrometer compares the changes in frequency of two hygroscopically coated quartz crystal oscillators. As the mass of the crystal changes due to the absorption of water vapor, the frequency changes. The amount of water absorbed on the sensor is a function of relative humidity, i.e., partial pressure of water as well as the ambient temperature.

A commercial instrument uses a hygroscopic polymer coating on the crystal. The humidity is measured by monitoring the vibration frequency change of the quartz crystal when the crystal is alternately exposed to wet and dry gas.

FIGURE 21-1: DEW POINT HYGROMETER



RADIATION ABSORPTION (INFRARED/ULTRAVIOLET)

Radiation absorption devices operate on the principle that selective absorption of radiation is a function of frequency for different mediums. Water vapor absorbs infrared radiation at 2 to 3 mm wavelengths and ultraviolet radiation centered about the Lyman-alpha line at 0.122 mm . The amount of absorbed radiation is directly related to the absolute humidity or water vapor content in the gas mixture according to Beer's law. The basic unit consists of an energy source and optical system for isolating wavelengths in the spectral region of interest, and a measurement system for determining the attenuation of radiant energy caused by the water vapor in the optical path. The absorbed radiation is measured extremely fast and independent of the degree of saturation of the gas mixture. Response times of 0.1 to 1 for 90% change in moisture content is common. Spectroscopic hygrometers are primarily used where a noncontact application is required; this may include atmospheric studies, industrial drying ovens, and harsh environments. The primary disadvantages of this device are its high cost and relatively large size.

SALT-PHASE HEATED HYGROMETER

Another instrument in which the temperature varies with ambient dew-point temperature is referred to as a self-heating, salt-phase transition or heated electrical hygrometer. This device usually consists of a tubular substrate covered by glass fiber fabric, with a spiral bifilar winding for electrodes. The surface is covered with a salt solution—usually lithium chloride. The sensor is connected in series with a ballast and a 24-V(ac) supply voltage. When in operation, electrical current flowing through the salt film heats the sensor. The electrical resistance characteristics of the salt are such that a balance is reached with the salt and its critical moisture content, corresponding to a saturated solution. The sensor temperature adjusts automatically so that the water vapor pressures of the salt film and ambient atmosphere are equal. With lithium chloride, this sensor cannot be used to measure relative humidity below approximately 12% (the equilibrium relative humidity of this salt), and it has an upper dew-point limit of about 160°F. The regions of highest precision are between -10 and 93°F, and above 105°F dew point. Another problem is that of the lithium chloride solution being washed off when exposed to water. In addition, this type of sensor is subject to contamination problems, which limits its accuracy. Its response time is also very slow; it takes approximately 2 min for a 67% step change.

Table 23-1:
Humidity sensor properties

Type of Sensor	Sensor Category	Method Of Operation	Approx. Range	Uses	Approx. Accuracy
Psychrometer	Evaporative cooling	Temperature measurement 01 wet bulb	32 to 1 80°F	Measurement, standard	±3 to ±7% RH
Adiabatic Saturation Psychrometer	Evaporative Wing	Temperature measurement of thermodynamic wet bulb	40 to 85°F	Measurement, standard	±0.2 to ±2% RH
Cilled Mirror	Dew point	Optical determination of moisture formation	-1 10 to 200°F dp	Measurement, control, meteorology	±0.4 to ±4°F
Heated Saturated Salt Solution	Water vapor pressure	Vapor pressure depression in salt solution	-20 to 160°F dp	Measurement, control, meteorology	±3°F
Hair	Mechanical	Dimensional change	5 to 100% RH	Measurement, control	±5% RH
Nylon	Mechanical	Dimensional change	5 to 100% RH	Measurement, control	±5% RH
Dacron Thread	Mechanical	Dimensional change	5 to 100% RH	Measurement, control	±7% RH
Goldbeater's skin	Mechanical	Dimensional change	5 to 100% RH	Measurement, control	±7% RH
Cellulosic materials	Mechanical	Dimensional change	5 to 100% RH	Measurement, control	±5% RH
Carbon	Mechanical	Dimensional change	5 to 100% RH	Measurement	±5% RH
Dunmore type	Electrical	Impedence	7 to 98% RH at 40 to 140°F	Measurement, control	±1.5% RH
Ion exchange resin	Electrical	Impedence or capacitance	10 to 100% RH at -40 to 190°F	Measurement, control	±5% RH
Porous ceramic	Electrical	Impedence or capacitance	Up to 400°F	Measurement, control	±1 to ±1.5% RH
Aluminum oxide	Electrical	Capacitance	5 to 100% RH	Measurement, control	±3% RH
Aluminum oxide	Electrical	Capacitance	-110 to 140°F dp	Trace moisture measurement, control	±2°F dp
Electrolytic hygrometer	Electrical	Capacitance			
Coulometric	Electrical cell	Electrolyzes due absorbed moisture	1 to 1000 ppm	Measurement	
Infrared laser diode	Electrical	Optical diodes	0.1 to 100 ppm	Trace moisture measurement	±0.1 ppm
Surface acoustic wave	Electrical	SAW attenuation	85 to 98% RH	Measurement, control	±1% RH
Piezoelectric	Mass sensitive	Mass changes due to absorbed moisture	-100 to 0°F	Trace moisture measurement, control	±2 to ±10°F dp
Radiation absorption	Moisture absorption	Moisture absorption of UV or IR radiation	0 to 180°F dp	Measurement control, meteorology	±4°F dp, ±5% RH
Gravimetric	Direct measurement of mixing ratio	Comparison of sample gas with dry airstream	120 to 20,000 ppm mixing ratio	Primary standard, research and laboratory	±0.13% of reading
Color change	Physical	Color changes	10 to 80% RH	Warning device	±10% RH
References: ASHRAE handbook fundamentals (1997)					

Section IV: Humidity Affects Our Comfort, Health and Surroundings

This section will deal with some of the changes that take place in our habitat as a result of changes in humidity.

HUMIDITY AND COMFORT

When all other factors affecting body heat loss remain constant, increasing the humidity of the surrounding air makes us feel warmer.

Perspiration absorbs heat from our skin as it evaporates thus cooling us. By controlling the humidity we can affect how warm or cool we feel.

Studies have been done on human comfort under controlled conditions. The resulting criteria are based on skin sensation, psychological responses and thermal effects. ASHRAE Comfort Standard 55-75 indicates a preference for humidity ranging between 20 and 50 percent and a dry bulb temperature between 73 and 77°F.

A comfort analysis was made on groups of subjects under standard conditions (at rest or doing light office work) and wearing standard clothing (long sleeved shirts and trousers). These individuals submitted comfort votes which established a range of temperatures compatible with "thermal comfort." Eighty percent were described as "thermally comfortable" when the following six conditions were all present:

1. When activity is light office work or its equivalent.
2. When "standard" clothing is worn.
3. When air motion is 40 feet per minute.
4. When relative humidity is 40 percent.
5. When mean radiant temperature is equal to the air temperature.
6. When air temperature is between 72° and 78°F.

This study found that a 20 percent change in relative humidity was equal to a one degree F change in dry bulb temperature. Indoor relative humidity maintained between 35 and 40 percent is recommended for optimum comfort during the heating season.

ASHRAE STANDARD 62-1989 recommends: Relative humidity in habitable spaces preferably should be maintained between 30% and 60% RH to minimize growth of allergenic or pathogenic organisms.

Everyone has an ideal range of temperature and humidity within which they perform best and feel most comfortable. Excessively high or low humidity causes significant environmental and physiological changes.

Even at higher temperatures, dry air feels colder. When people feel cold, they are uncomfortable and less productive. When the humidity level is ideal, building temperatures can actually be lowered without changing the comfort or productivity levels.

It has been stated that the energy savings derived from a reduction in room temperature, as a result of proper humidification, will directly offset the cost of the energy consumed in producing that humidity.

HEALTH

The following statements are excerpts from an ASHRAE paper entitled "*The Effect of Indoor Relative Humidity on Colds*," prepared by the late Dr. George Green, Prof. Of Engineering of the College of Engineering, University of Saskatchewan.

RECOMMENDED INDOOR RELATIVE HUMIDITY IN WINTER

"The studies of the influence of relative humidity quoted in this paper demonstrates a reduction in respiratory illness as the relative humidity is increased up to 50%. This would appear to be the optimum for health with our present knowledge."

"Most buildings in cold climates will have excessive window and wall condensation at -10°C, 14°F outdoor temperature with 50% RH; therefore, this level cannot be maintained constantly. For reasons of health, it would be desirable to hold humidities as high as can be maintained without excessive building condensation. For many buildings with high occupancy load, controls which keep the maximum humidity consistent with the outdoor temperature are justified in view of the high cost of absenteeism resulting from the common cold."

CONCLUSIONS

1. An increase of indoor relative humidity by humidification in winter significantly decreases the occurrences and/or absenteeism due to colds.
2. With preschool children, the investigation reports reduction of 50% in absenteeism in humidified over non-humidified schools. The reductions in absenteeism of adults was from 6 to 18% in similar investigations.
3. It is recommended that winter indoor humidities should be kept as high as possible without causing building damage by condensation, but not to exceed 50% RH.
4. The reduction in absenteeism by winter humidification justifies its capital, operating and energy costs.

HUMIDITY AND SKIN DISORDERS

Winter itch, dermatitis, chapping and painful skin cracking at knees and elbows and where the skin meets the fingernails, plus brittleness and splitting of hair and nails can affect almost everyone, but elderly people in particular. Maintaining a relative humidity of 35 to 50 percent may be significantly effective in reducing these troubles.

HAYFEVER AND ASTHMA

Humidification does little or nothing to alleviate the reaction of the body to specific allergens. It can, however, minimize house dust, feathers, animal hair, insect scales, etc., which irritate persons with allergies.

Some asthmatic people have attacks that come on as a result of sudden temperature and/or humidity changes. A constant humidity level seems to help those individuals.

STERLING STUDY

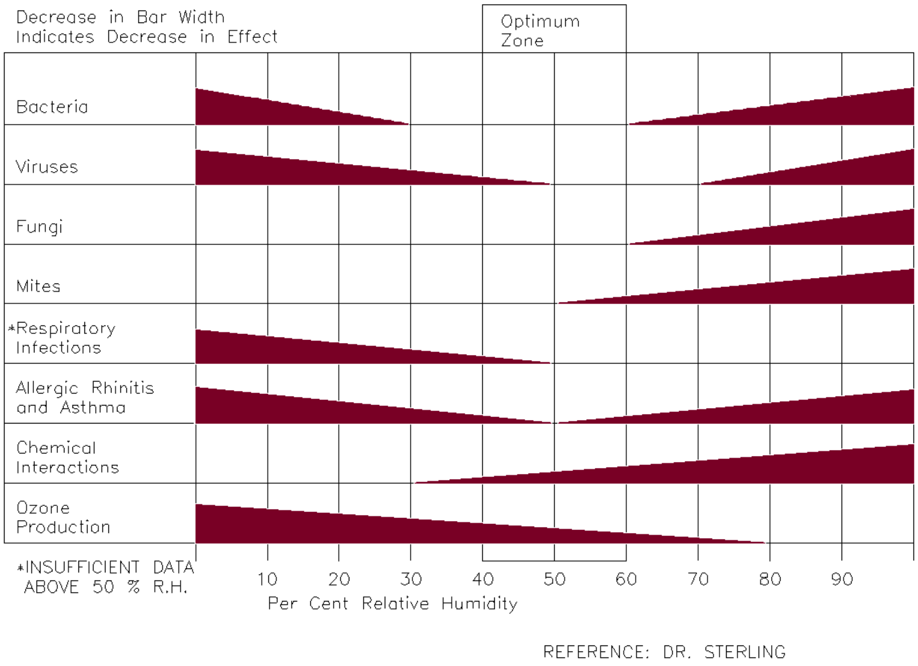
As further evidence of the health benefits of humidification is the following information excerpted from an ASHRAE paper prepared by E.M. Sterling, A. Arundel and T.D. Sterling, Ph.D.

Unlike most gaseous and particulate contaminants that are primarily affected by indoor and outdoor sources and sinks, relative humidity is also a function of air temperature. In addition to the effect of temperature, the selection of the most desirable range of humidity is complicated by the conflicting effects of an increase or decrease in humidity levels. For example, increasing humidity may reduce the incidence of common respiratory infections and provide relief for asthmatics. On the other hand, an increase in humidity may increase the prevalence of microorganisms that cause allergies. Criteria for indoor exposure must balance both effects.

The ideal humidity guideline should specify a relative humidity range that minimizes deleterious effects on human health and comfort as well as reducing, as much as possible, the speed of chemical reactions or the growth of biological contaminants (which will impact human health and comfort).

The Sterling bar graph summarizes the apparent association between relative humidity ranges and factors that affect health of occupants at normal room temperatures. The figure is constructed as a bar graph relating relative humidity levels from 0% to 100% (shown along the horizontal axis) to (1) biological organisms (bacterial, viruses, fungi and mites), (2) pathogens causing respiratory problems (respiratory infections, asthma and allergies), and (3) chemical interactions and ozone production. The decreasing width of the bar represents decreasing effects.

FIGURE 28-1: THE STERLING BAR GRAPH



EXCERPTED COMMENTS:

- The bacterial population increases below 30% and above 60% relative humidity. The viral population increases at relative humidity below 50% and above 70%. Fungi do not cause a problem at low humidity; however, growth becomes apparent at 60%, increases between 80% and 90%, and shows a dramatic rise above 90%. Mites require humidity for survival. Growth in the mite population responds directly to humidity levels in excess of 50%.
- Respiratory infections increase at relative humidity below 40%; however, there is little information on effects of humidity in excess of 50%. The incidence of allergic rhinitis due to exposure to allergens increases at relative humidities above 60% and the severity of asthmatic reactions increases at relative humidities below 40%.
- Most chemical interactions increase as the relative humidity rises above 30% though ozone production is inversely proportional to the relative humidity.
- The evidence suggests that the optimal conditions to enhance human health by minimizing the growth of biological organisms and the speed of chemical interactions occur in the narrow range between 40% and 60% relative humidity at normal room temperature. The narrow range is represented by the optimum zone in the shaded region of the graph. Although keeping indoor humidity levels within this region will minimize health problems, there is probably no level of humidity at which some biological or chemical factor that affects health negatively does not flourish. (Note that for many factors, most prominently chemical interactions, effects are still shown within the optimum zone.)

FIGURE 30-1: FLOORBOARDS BUCKLE

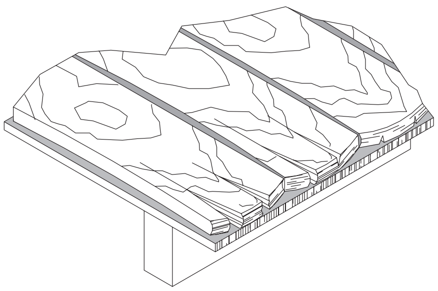
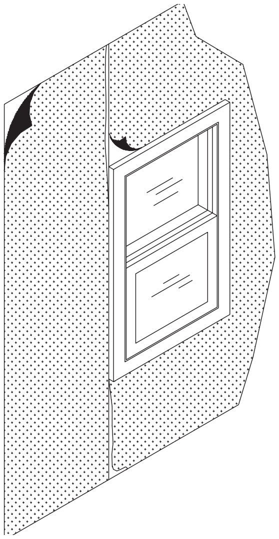


FIGURE 30-2: WALLPAPER SEAMS OPEN AND GAPS FORM



DEGRADATION OF THE BUILDING FURNISHINGS

There can be little doubt that the interiors of buildings are subjected to expensive damages when the air is not humidified. The fibers of carpeting and upholstered furniture become brittle causing them to weaken prematurely. Wall coverings shrink creating unsightly gaps at the seams. Wall paneling, expensive wood furniture, and musical instruments shrink causing cracks and failed joints. Wooden floor boards shrink, the cracks fill with soil causing the boards to buckle upward when the moisture returns.

Controlling humidity eliminates the brittleness of fibers and the cycles of shrinking and expanding that occur as the moisture content changes and greatly reduces the deterioration of the furnishings in the building.

STATIC ELECTRICITY

At one time or another, almost everyone has experienced “a shock” upon touching a door knob after walking across a carpeted room. This “shock” is static electricity caused, in part, by low humidity. The phenomenon was first recorded in 600 B.C. but it was not until 1800 A.D. that its cause was identified. It was found that whenever materials of high electrical resistance are moved or rubbed against each other, causing friction, a build-up of a static electrical charge occurs.

Static cling of clothing is another example of static charges. Anti-static treatment of carpeting and clothing reduces but does not eliminate static build-up.

Static electricity may be a nuisance in our clothing or in our homes, but it can create more serious problems in certain controlled environments such as computer rooms, research laboratories and industrial clean rooms.

Hospitals require specific humidity levels, among other reasons, to eliminate static electricity in the presence of mixtures of high concentrations of oxygen and other potentially explosive gases.

Building maintenance is also affected by static electricity. Minute dust particles are present in the air, even in buildings containing high efficiency filters. Some of the room dust is never captured by the air stream to be drawn into the system for filtration. These particles can become “space charged” and adhere to walls, draperies and other furnishings. Ceiling “stains” surrounding air diffusers are static charged room dust particles induced by the air flow pattern and deposited on the ceiling.

Another form of static dust deposit is found where parallel heat transfer takes place, such as nails or screws securing gypsum board to the studding in the exterior walls of buildings. It has been found that the cold nail head becomes negatively charged, due to a thermal-electrical static build-up, causing it to attract positively charged dust particles.

All of these static-producing conditions are greatly reduced, if not completely eliminated, by maintaining a relative humidity in the 30 to 60 percent range.

HUMIDITY AND ODORS

Perception of the odor associated with cigarette smoke (suspension of tobacco tar droplets plus vapor) and pure vapors is affected by temperature and humidity. An increase in humidity, at constant dry-bulb temperature, lowers the intensity level of cigarette smoke odor, as well as that of pure vapors {Kerka and Humphreys 1956, Kuehner 1956}. This effect is more pronounced for some odorants than for others. An increase in temperature at constant specific humidity lowers the odor level of cigarette smoke slightly. Adaptation to odors takes place more rapidly during the initial stage of exposure. While the perceptible odor level of cigarette smoke decreases with exposure time, irritation to the eyes and nose generally increases. The irritation is greatest at low relative humidities.

To keep odor perception and irritation at a minimum, the air conditioned space should be operated at about 45 to 60% RH. Since temperature has only a slight effect on odor level at constant specific humidity, it generally can be ignored; temperature should be maintained at conditions desired for comfort or economy.

HUMIDITY AND SOUND TRANSMISSION

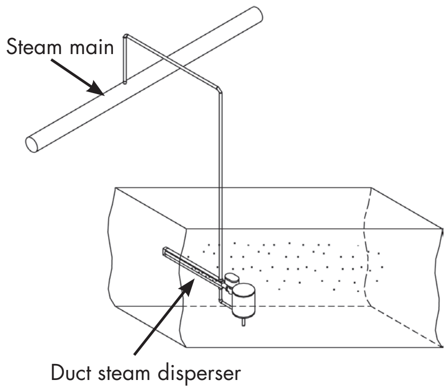
Sound waves are affected by the level of humidity. Maximum air absorption (or loss of sound waves) occurs at 15 to 20 percent relative humidity. High frequencies are affected more than low. Optimum sound transmission occurs in the range of 40 to 50 percent relative humidity.

REFERENCES

1. ASHRAE Handbook Fundamentals, 1977.
2. ASHRAE Journal, June 1981, Are You Comfortable Ralph F. Goldman, PhD.
3. ASHRAE Handbook, Equipment, 1979.
4. ASHRAE Handbook Fundamentals, 1981.
5. Kerka and Humphreys, 1956
6. Kuehner, 1956

Section V: Humidity and Indoor Air Quality

FIGURE 34-1: STEAM INJECTION IN A DUCT



HEALTH CAN BE AFFECTED BY HUMIDIFICATION WITH CHEMICALLY TREATED STEAM

The practice of introducing live steam for humidification sometimes called steam injection, into the air ducts of buildings, has been in use for several decades.

Boiler water treatment chemicals, called neutralizing amines, are necessary for corrosion protection of the boiler and piping system.

In recent years, these chemicals have been found to be potentially injurious to human health. Three netralizing amines: morpholine, chclohexylamine, and diethyaminoethanol (DEAE) are used for this purpose.

Figure 34-1 illustrates one of the most common methods of boiler steam injection. Boiler steam is delivered to the ducted air stream and circulated throughout the building. The neutralizing amines, which are carried with the steam, are circulated as well.

Building occupants can be affected by these chemicals, both by inhalation and by skin contact with fallout that has settled on surfaces in the occupied space.

One such situation occurred in the state of New York. The following statement is an excerpt relating to the instance from a report called *New Onset Asthma After Exposure to the Steam system Additive DEAE*. This descriptive sudy was authored by Margaret E. Gadon, MD; James M. Melius, MD; Gerald J. McDonald, AS; and David Orgel, MD. The report came from the New York State Department of Health, Albany.

"Through a leak in the steam heating system, the anticorrosive agent 2-diethylamincethanol was released into the air of a large office building. Irritative symptoms were experienced by most of the 2500 employees, and 14 workers developed asthma within 3 months of exposure."

"This study was undertaken to review clinical characteristics of these asthmatics. Environmental exposure monitoring data and medical records were reviewed. Seven of 14 cases were defined as "confirmed" and 7 of 14 "suspect," using the National Institute for Occupational Safety and Health surveillance case definition of occupational asthma. Three cases were diagnosed on the basis of work-related symptoms and physical examination alone. The study suggests that acute exposure to the irritating steam additive 2-diethylaminoethanol was a contributing factor in the development of clinical asthma in this population."

Some studies suggest that these chemicals can be powerful skin and lung irritants, and several types can result in a process known as nitrosation to form potent animal (and presumable human) carcinogens. OSHA has developed Permissible Exposure Levels (PEL's) for neutralizing amines.

OSHA has developed Permissible Exposure Levels (PEL's) are guidelines for workplace exposure of average workers in an 8 hour day, 40 hour week environment only. They do not apply to those who may be in ill health, and/or those exposed for extended periods of time such as in hospitals and nursing care facilities. Safe exposure levels of amines have not been determined for these groups, or those who may be particularly sensitive to the effects of these substances. Furthermore, levels for skin-contact exposure to the settled amine particles have not been considered by any regulatory agency.

Symptoms that have been associated with exposure to amines have been documented by the National Institute for Occupational Safety and Health (NIOSH) and include nausea, dizziness, asthma, vomiting, headaches, chest tightness and elevated blood pressure. Skin contact with the amine aerosols has been linked with eye and nose irritation, as well as skin rashes and burning sensations. Studies to determine the potential for the formation of cancer-causing agents are ongoing.

Because of the above, NIOSH, the Environmental Protection Agency (EPA), the American Society of Heating, Refrigerating, and Air-Conditioning Engineers (ASHRAE) and the American Society for Hospitals Engineering of the American Hospital Association, have all strongly cautioned against the use of boiler steam, containing neutralizing amines, for humidification.

In addition to the health concerns mentioned on the previous page, these amines have been associated with deterioration of precious collections of relics and works of art in museums.

CHEMICALS IN HUMIDIFICATION STREAM CAN DAMAGE ARTIFACTS

The following was excerpted from a book entitled, Guide to Environmental Protection of Collections, by Barbara Appelbaum.

"In a well-known case at Cornell University, centrally generated steam intended for closed heating systems was used for museum humidification. It was noted only later that the anti-corrosive, commonly known as DEAE (diethylaminoethanol), was a human health hazard and caused skin and eye irritations, as well as the possibility of long-term health risks.

DEAE was finally detected in a white crystalline haze that was found on the surface of paints in a large museum in the Midwest several weeks after a new humidification system had been turned on the cleaning personnel reported a slimy residue on gallery surfaces. In this case, engineers admitted that DEAE had been used."

The sliminess that probably is the result of salts deposited on surfaces when DEAE reacts with acids in the air is one problem reported in the museum literature. Problems specifically traceable to other chemicals used for this purpose have not been reported, but at present conservators strongly recommend that no anti-corrosives be added to water used for humidification. Alternatives include accepting a certain amount of corrosion in the steam-generating equipment by deleting chemicals or keeping chemical use to a bare minimum.

Note:

Steam humidification equipment that produces chemical-free steam is available and is discussed in later sections of this books.

FIGURE 36-1: PHOTO COURTESY OF THE MCMICHAEL ART GALLERY, TORONTO, CANADA

DUCT WETNESS AND INDOOR AIR QUALITY

Wet areas in ducts must be avoided in order to ensure that indoor air is free of the disease-producing microbes described below.

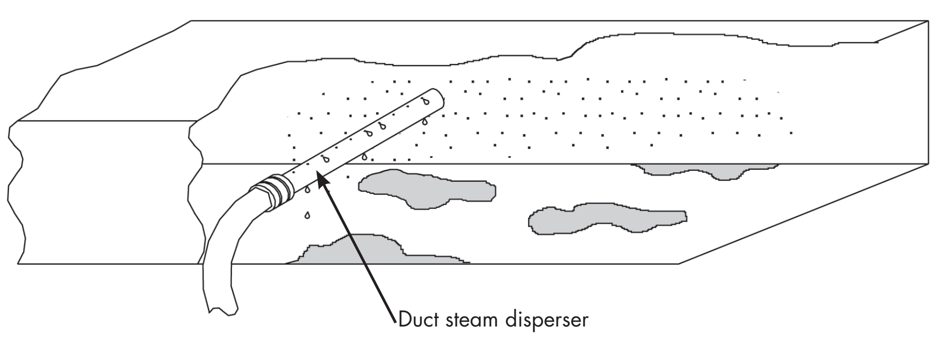
The following excerpt was taken from a paper titled, *Environmental Studies In Moldy Office Buildings*, authored by P.R. Morey, Ph.D.; M.J. Hodgson; W.G. Sorenson, Ph.D.; G.J. Kullman; W.W. Rhodes, P.E.; and G.S. Visvesvara, Ph.D. The study occurred in the 1980s.

"The National Institute for Occupational Safety and Health (NIOSH) has carried out health hazard evaluations in five large office buildings where hypersensitivity pneumonitis (HP) and other respiratory diseases have been alleged or reported. Environmental studies done both in the occupied space and in the heating, ventilating, and air-conditioning (HVAC) system of each building are described. Several buildings were characterized by a history of repeated duct flooding and all contained mechanical systems with pools of stagnant water and microbial slimes. Preventive measures that may be effective in reducing building-associated microbial contamination and building-associated HP illnesses include the following: (A) Prevent moisture incursion into occupied space and HVAC system components; (B) Remove stagnant water and slimes from building mechanical systems; (C) Use steam as a moisture source in humidifiers; (D) Eliminate the use of water sprays as components of office building HVAC systems; (E) Keep relative humidity in occupied space below 70%; (F) Use filters with a 50% rated efficiency; (G) Discard microbially damaged office furnishings; (H) Initiate a fastidious maintenance program for HVAC system air handling units and fan coil units."

The following statements are included as further reinforcement of the significant importance of avoiding wetness in ducts. This is an excerpt from a Code Application Notice given by the Office of Statewide Health Planning and Development (OSHPD) of the State of California dated March 24, 1994. It pertains to hospitals.

"Regardless of humidifier type, all humidifiers shall be specified and installed with proper downstream distances to obstructions and/or restrictions which could be sites for condensation. Such factors as air velocity, airstream temperature, humidification load and relative humidity of the airstream shall be taken into consideration. Air flow proving devices and downstream humidity high limit controls shall be provided. Construction documents shall detail how the distribution tubes are to be installed, indicating minimum distances from changes in direction

FIGURE 37-1: STANDING WATER IN A DUCT DUE TO NON-DRY TYPE STEAM DISPERSER



and other potential points of condensation. See Figure 38-1. Appurtenant piping and accessories shall also be detailed. Psychrometric analysis or other acceptable means, shall be provided to verify that dry steam will be supplied.

Mechanical means of humidification, such as atomizers, and humidifiers requiring direct contact of conditioned air with water or wetted surfaces are not permitted.

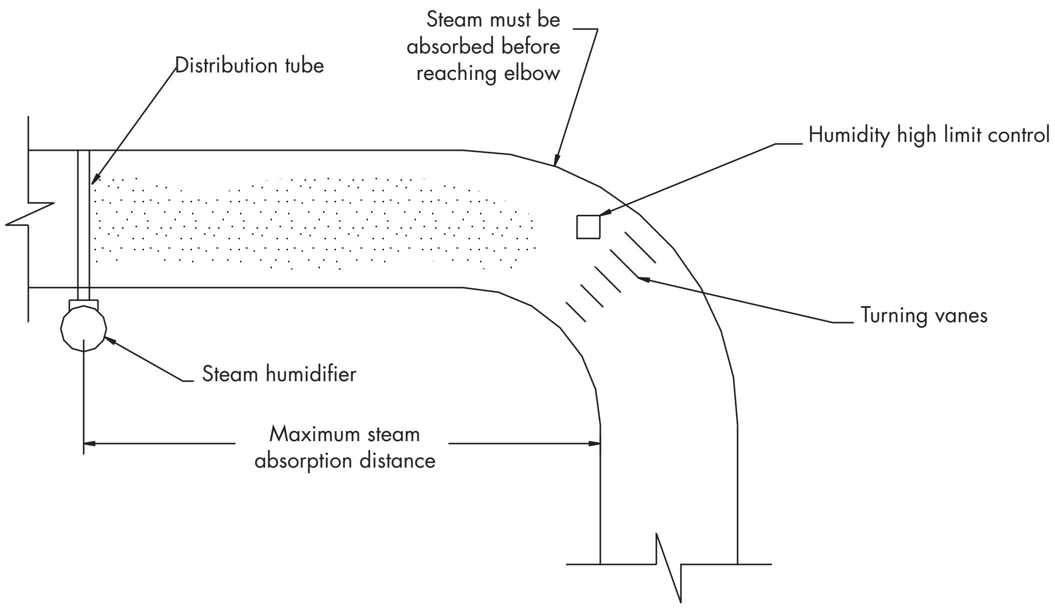
The intent of the requirement for the dry steam type humidifier is to prevent direct contact of conditioned air with water or wetted surfaces which could foster the growth of bacteria (including Legionella) in the HVAC system.

Clean, dry and uncontaminated ductwork is a joint responsibility of the design professional, installing contractor, and the hospital maintenance staff. This Code Application Notice addresses the design and installation considerations necessary to prevent direct contact of conditioned air with wetted surfaces which could become sites for bacterial growth. Proper maintenance of the system is the responsibility of the health care facility and is regulated by other Titles of the California Code of Regulations."

Note:

Humidification equipment is available that will accomplish the above requirements and is described in other parts of this book.

FIGURE 38-1: EXAMPLE OF PLAN VIEW OF REQUIRED INSTALLATION DETAILS



Section VI: Recommended RH for Various Materials and Processes

In certain industrial processes, it is necessary to control humidity due to one or more of the following factors:

- Product moisture content loss or regain
- Rate of chemical reactions
- Rate of biochemical reactions
- Rate of crystallization
- Static electricity
- Accuracy, uniformity and overall product quality

During the manufacturing or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and foodstuffs; air temperature and relative humidity greatly influence production rates as well as product weight, strength, appearance and general quality.

Hygroscopic material absorbs moisture from surrounding air until it reaches equilibrium with the air. This does not mean it will equal the relative humidity of the air. For example, when stored in an atmosphere of 50% R.H., macaroni will assume a maximum moisture content of 11.7%. This is referred to as its Equilibrium Moisture Content (EMC). This is also called regain and is stated as a percentage of its bone dry weight. For example, if a sample is thoroughly dried under standard conditions from 110 to 100 grams, it has given up 10% of its bone dry weight. The regain of the sample is 10%.

Table 41-1 lists the regain of certain materials at various levels of relative humidity. When absorbing moisture from the air, a material will surrender sensible heat to the air equivalent to the latent heat of the moisture that is absorbed. This will cause a rise in air temperature.

In the cooling process, humidity, as well as temperature, plays a major role since it greatly affects evaporation rates. Crystal formations, such as coatings on candies, cereals, and other foods, are affected in terms of size and texture by both the cooling rate and the drying rate.

Bulk handling of various extremely dry, finely powdered materials is affected by moisture content. A slight increase in moisture can reduce the static electricity buildup and allow smooth flow without caking.

In the micro-circuit chip industry, static electricity discharges at critical states in chip making ruin the product.

Table 42-2 lists recommended air temperature and relative humidity levels for various industrial processes.

The industrial processing plant, or mill, is designed for the specific processes it houses. Processing steps, such as machining, assembly, packaging, and materials handling, may dictate the need for additional humidification. However, in many industries the most important consideration is that of maintaining the present moisture content of the material being processed.

Table 41-1:
Regain of various hygroscopic materials

Classification: Natural textile fabrics		Relative humidity								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
Cotton	Sea Island-roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1
Cotton	American cloth	2.6	3.7	4.4	5.5	5.9	6.8	8.1	10.0	14.3
Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8
Wool	Australian Merino - Skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4
Silk	Raw Chevennes - Skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.3
Linen	Table cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2
Linen	Dry spun - yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8
Jute	Average of several grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2
Hemp	Manila and sisal rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7
Classification: Rayons		**Relative humidity**								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
Viscose nitro-cellulose	Average skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0
Cuprammonium cellulose acetate		0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3
Classification: Papers		**Relative humidity**								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
M.F. newspring	Wood pump - 24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6
H.M.F. writing	Wood pump - 3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2
White bond	Rag - 1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2
Comm. ledger	75% rag - 1 % ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9
Kraft wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9
Classification: Misc. organic materials		**Relative humidity**								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
Leather	Sole oak-tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2
Catgut	Racket strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7
Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5
Rubber	Solid tires	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99
Wood	Timber (average)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0
Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8
Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0

Table 42-1:
Regain of various hygroscopic materials

Classification: Foodstuffs		Relative humidity								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
White bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0
Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9
Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.09	22.1
Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1
Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7
Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4
Classification: Misc. Inorganic materials		**Relative humidity**								
Materials	**Description**	**10%**	**20%**	**30%**	**40%**	**50%**	**60%**	**70%**	**80%**	**90%**
Asbestos fiber	Finely divided	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84
Silica gel		5.70	9.80	12.70	15.20	17.20	18.80	20.20	21.50	22.60
Domestic coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89
Activated charcoal	Steam activated	7.10	14.30	22.80	26.20	28.30	29.20	30.00	31.10	32.70
Sulfuric acid	H2SO4	33.00	41.00	47.50	52.50	57.00	61.50	67.00	73.50	82.50

Table 42-2:
Temperature and humidities for industrial air conditioning (continued)

Process		**°C**	**°F**	**RH%**
Abrasives				
	Manufacture	26	79	50
Ceramics				
	Refractory	43 to 66	109 to 150	50 to 90
	Molding room	27	81	60 6o 70
	Clay storage	16 to 27	60 to 80	35 to 65
	Decalcomania production	24 to 27	75 to 80	48
	Decorating room	24 to 27	75 to 80	48
Cereal				
	Packaging	24 to 27	75 to 81	45 to 50
Distilling				
	General manufacturing	16 to 24	61 to 75	45 to 60
	Aging	18 to 22	64 to 72	50 to 60

Table 43-2:
Temperature and humidities for industrial air conditioning (continued)

Process		°C	°F	RH%
Electrical products				
	Electronics and X-rays:			
	Coil and transformer winding	22	72	15
	Tube assembly	20	68	40
	Electrical instruments:			
	Manufacture and laboratory	21	70	50 to 55
	Thermostat assembly and calibration	24	75	50 to 55
	Humidistat assembly and calibration	24	75	50 to 55
	Small mechanisms:			
	Close tolerance assembly	22	72	40 to 45
	Meter assembly and test	24	75	60 to 63
	Switchgear:			
	Fuse and cutout assembly	23	73	50
	Capacitor winding	23	73	50
	Paper storage	23	73	50
	Conductor wrapping with yarn	24	75	65 to 70
	Lightning arrestor assembly	20	68	20 to 40
	Thermal circuit breakers assembly and test	24	75	30 to 60
	High voltage transfer	26	79	5
	Water wheel generators:			
	Thrust runner lapping	21	70	30 to 50
	Rectifiers:			
	Processing selenium and copper	23	73	30 to 40
Gum				
	Manufacturing	25	77	33
	Rolling	20	68	63
	Stripping	22	72	53
	Breaking	23	73	47
	Wrapping	23	73	58

Table 44-2:
Temperature and humidities for industrial air conditioning (continued)

Process		°C	°F	RH%
Leather				
	Drying	20 to 52	68 to 126	75
	Storage, winter room temp	10 to 15	50 to 59	40 to 60
	After leather is moistened in preparation for rolling and stretching, it is placed in an atmosphere held at room temperature with a relative humidity of 95%. Leather is usually stored in warehouses without temperature and humidity control. However, it is necessary to keep humidity sufficiently low to prevent mildew. Air filtration is recommended for fine finish.			
Lenses (optical)				
	Fusing	24	75	45
	Grinding	27	81	80
	Temperature and humidity must be held constant			
Matches				
	Manufacture	22 to 23	72 to 73	50
	Drying	21 to 24	70 to 75	60
	Storage	16 to 17	61 to 63	50
	Water evaporated is 8 to 9 kg (18 to 20 lbs.) per million matches simultaneously with the setting of the glue. The match machine will turn out about 750,000 matches per hour.			
Mushrooms				
	Sweating-out period	50 to 60	122 to 140	–
	Spawn added	16 to 22	61 to 72	nearly sat.
	Growing period	9 to 16	48 to 61	80
	Storage	0 to 2	32 to 36	80 to 85
Paint applications				
	Lacquers: baking	150 to 180	302 to 356	–
	Oil paints: paint spraying	16 to 32	61 to 90	80
	Powder paints			
	Lav VOC paints			
Pharmaceuticals				
	Powder storage (prior to mfg)	*	*	*
	Manufactured powder storage and packing areas	24	75	35
	Milling room	24	75	35
	Tablet compressing	24	75	35
	Tablet coating room	24	75	35
	Effervescent tablets and powders	24	75	20
	Hypodermic tablets	24	75	30

Table 45-2:
Temperature and humidities for industrial air conditioning (continued)

Process		°C	°F	RH%
	Colloids	24	75	30 to 50
	Cough drops	24	75	40
	Glandular products	24	75	5 to 10
	Ampoule manufacturing	24	75	35 to 50
	Gelatin capsules	24	75	35
	Capsule storage	24	75	35
	Microanalysis	24	75	50
	Biological manufacturing	24	75	35
	Liver extracts	24	75	35
	Serums	24	75	50
	Animal rooms	24 to 27	75 to 81	50
	Small animal rooms	24 to 26	75 to 79	50
	*Store in sealed plastic containers in sealed drums			
Photo studio				
	Dressing room	22 to 23	72 to 74	40 to 50
	Studio (camera room)	22 to 23	72 to 74	40 to 50
	Film darkroom	21 to 22	70 to 72	45 to 55
	Print darkroom	21 to 22	70 to 72	45 to 55
	Drying room	32 to 38	90 to 100	35 to 45
	Finishing room	22 to 24	72 to 75	40 to 55
	Storage room (b/w film and paper)	22 to 24	72 to 75	40 to 60
	Storage room (color film and paper)	5 to 10	40 to 50	40 to 50
	Motion picture studio	22	72	40 to 55
The above data pertain to average conditions. In some color processes, elevated temperature as high as 105°F are used, and a higher room temperature is required.				
Conversely, ideal storage conditions for color materials necessitate refrigerated or deep-freeze temperature to ensure quality and color balance when long storage items are anticipated.				
Plastics				
Manufacturing areas:				
	Thermosetting molding compound	27	81	25 to 30
	Cellophane wrapping	24 to 27	75 to 81	45 to 65
In manufacturing areas where plastic is exposed in the liquid state or molded, high efficiency filters may be required. Dust collection and fume control are essential.				

Table 46-2:
Temperature and humidities for industrial air conditioning (continued)

Process		°C	°F	RH%
Plywood				
	Hot pressing (resin)	32	90	60
	Cold pressing	32	90	15 to 25
Rubber-dipped goods				
	Manufacture	32	90	
	Cementing	27	80	25 to 30*
	Dipping surgical articles	24 to 27	75 to 80	25 to 30*
	Storage prior to manufacture	15 to 24	60 to 75	40 to 50*
	Laboratory (ASTM Standard)	23	73.4	50*
*Dew point of air must be below evaporation temperature of solvent.				
Solvents used in manufacturing processes are often explosive and toxic, requiring positive ventilation. Volume manufacturers usually install a solvent-recovery system for area exhaust systems.				
Tea				
	Packaging	18	64	65
Ideal moisture content is 5 to 6% for quality and weight. Low limit moisture content for quality is 4%.				
Tobacco				
	Cigar and cigarette making	21 to 24	70 to 75	55 to 65
	Softening	32	90	85 to 88
	Stemming and stripping	24 to 29	75 to 84	70 to 75
	Packing and shipping	23 to 24	73 to 75	65
	Filler tobacco casing and conditioning	24	75	75
	Filler tobacco storage and preparation	26	77	70
	Wrapper tobacco storage and conditioning	24	75	75

Section VII: Humidity Control in Manufacturing

WOOD AND WOOD PRODUCTS

Since wood is hygroscopic and expands and shrinks with changes in moisture, its Equilibrium Moisture Content (EMC) should not be allowed to change during processing. This means that temperature and humidity must be controlled from storage through final processing.

Table 48-1 shows the EMC of wood at various ambient dry bulb temperatures and relative humidities. It serves as a guide for determining the most efficient and desirable space conditions to be used for wood storage and processing.

For example, at a dry bulb temperature of 18°C (65°F) and a relative humidity of 38%, when the moisture content of wood has stabilized, 7.1% of its bone dry weight will be moisture.

Table 48-1:
EMC% of wood

Dry Bulb		Relative Humidity					
°C	°F	12%	25%	38%	51%	63%	72%
10	50	3	5.0	7.0	8.9	11.0	12.9
12.8	55	3	5.0	7.0	8.9	11.0	12.9
15.5	60	3.1	5.1	7.1	8.9	11.0	13.0
18.3	65	3.2	5.1	7.1	9.0	11.1	13.5
21.1	70	3.2	5.2	7.1	9.0	11.1	13.6
23.9	75	3.3	5.2	7.2	9.0	11.2	14.0
26.7	80	3.3	5.3	7.2	9.1	11.2	14.0

To produce high quality textiles with minimum down time and rejects, it is necessary to properly maintain both temperature and humidity control.

Adequate and constant humidity reduces static electricity and its accompanying difficulties. Also, due to the more uniform friction between adjacent fibers, more precise yarn tension is possible, resulting in a finer product. Proper humidity also increases the abrasion resistance of the warp and allows higher operating speed of the equipment.

Table 49-1 indicates the recommended relative humidities for various steps in processing wool, cotton, and man-made fibers.

Table 49-1:
Recommended RH% for textiles

Departments	Relative Humidity % at 75°F - 80°F (24-27°C)	
Raw wool storage	50 - 55	
Mixing and blending	65 - 70	
Carding - worsted	60 - 70	
Carding - woolen	60 - 75	
Combing, worsted	65 - 75	
Drawing, worsted		
Bradford system	50 - 60	
French system	65 - 70	
Spinning - Bradford, worsted	50 - 55	
French (mule)	75 - 85	
Woolen (mule)	65 - 75	
Winding and spooling	55 - 60	
Warping, worsted	50 - 60	
Perching or clothroom	55 - 60	
Departments	**Cotton**	**Manmade**
Opening and picking	55 - 70	50 - 55
Carding	55 - 60	55 - 65
Silver lapping	55 - 60	55 - 65
Combing	55 - 65	55 - 65
Drawing	50 - 60	50 - 60
Roving	50 - 60	50 - 60
Spinning	35 - 60	50 - 65
Winding and spooling	55 - 65	60 -65
Twisting	60 - 65	50 - 65
Warping	55 - 70	50 - 60
Knitting	60 - 65	50 - 60
Weaving	70 - 85	60 - 70

PHOTOGRAPHIC MATERIALS

In addition to requiring a very high degree of air cleanliness, the manufacture, processing, and storage of photographic products requires precise control of temperature and relative humidity.

Static electricity buildup and subsequent discharge leaves streaks on processed film. It also causes dirt and lint to be attracted to film. Proper humidification is required to control static electricity, curling, and brittleness.

In the printing of motion picture films, the registration of multiple images to produce one clear, sharp, image is adversely affected by improper temperature and relative humidity.

Following are temperature and humidity recommendations for photographic materials:

Table 50-1:
Recommended temperature and RH% for film

Type of film	Temperature		Relative humidity %
	°F	°C	
Film storage (unprocessed)	21 - 24	70 - 75	50 - 55
Film storage (processed)	19 - 24	65 - 75	50
Movie film storage	10	50	40
Printing rooms (photos)	21 - 24	70 - 75	50 - 55

Table 51-1:
Humidity recommendations for paper products

Type of film	Description	Relative humidity %								
		10	20	30	40	50	60	70	80	90
M.F. newsprint	Wood pulp - 24% ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.3	10.8
H.M.F. writing	Wood pulp - 3% ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2
White bonding	Rag - 1% ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2
Comm. ledger	75% rag - 1% ash	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9
Kraft wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9

PAPER

Unless it is tightly rolled or stacked in sheets, paper absorbs and gives off moisture very quickly as the surrounding humidity changes. Paper that contains too much moisture becomes limp and tough, affecting the functions of most processing machines. If too dry, it becomes brittle and loses tensile strength. This causes cracking at creases or folds and breaking of bags and again affects the function of processing machines.

Changes in moisture content at any time between start and completion of a process may cause dimensional changes and curling that are serious enough to cause scrapping of material. For these reasons, it is important that temperature and humidity be held constant at all processing stages and at the appropriate levels.

Generally speaking, a paper moisture content of 5% to 7% provides suitable strength and workability. This condition in turn, requires an indoor relative humidity of 40% to 50%, depending upon the consistency of the product.

Table 51-1 shows the relationship of the EMC of select types of paper at various relative humidities.

PRINTING

Paper, as stated earlier, is a very hygroscopic material that very rapidly absorbs and gives off moisture in accordance with only minor changes in the moisture content and temperature of the surrounding air.

Problems associated with improper humidity control in the printing environment include the following:

- Distorted, curled or buckled finished product
- Wrinkling of sheets or web while printing
- Loss of registration in multi-color operations, resulting in blurry or fuzzy images
- Slow drying of ink
- Swelling or shrinking of composition press rollers
- Light sensitivity changes in the printing surface coating used in photolithography, gravure, and photoengraving

Good quality paper supply houses will store and deliver the paper in moisture-proof wrapping with the correct moisture content for the intended purpose. Unless the paper is then stored in an atmosphere of proper humidity and temperature, the wrapper should not be broken until it is to be used.

Temperature conditioning prior to opening the package will also minimize the problem of change in moisture content.

Paper that is regaining moisture will undergo a dimension change of about 0.1% for each 1% change in moisture content. The critical need for humidity control can best be seen from the fact that multi-color printing requires dimensional accuracy within .005 inches (.127mm).

Figure 52-1 shows the variation in dimension of printing paper caused by changes in moisture content.

Table 52-1 indicates the recommended temperatures and humidities for various printing processes.

CHEESE MAKING

Cheese requires a humid atmosphere at various steps of processing and storage. Steam is a prime source of heat in the making of cheese and is usually readily available for humidification.

The following tables list the optimum temperature and humidity levels for various kinds of cheese.

FIGURE 52-1: EFFECTS OF VARIATION IN MOISTURE CONTENT ON DIMENSIONS OF PRINTING PAPERS

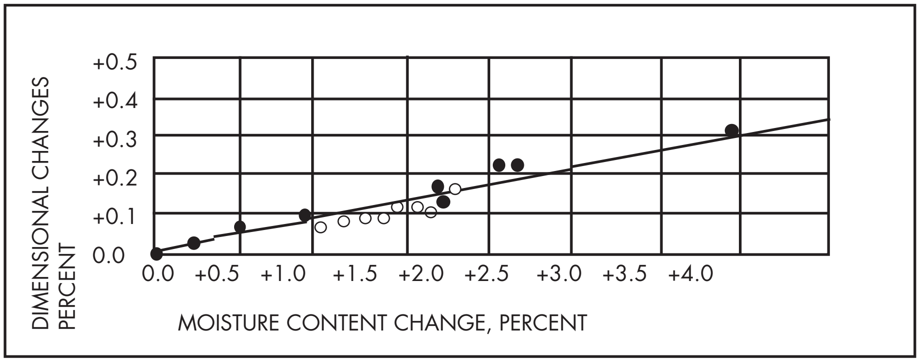


Table 52-1:
Recommended temperature and humidity for various printing processes

Type of process	Temperature °C	Temperature °F	Relative humidity %
Multi color lithographic	77±2	50	45±2
Plate making	77±1	55	45
Rotogravure	77±2	60	47±2
Collotype	80±3	65	85±2
Letterpress	72±2	70	42±1
Paper shaving collection	70 - 75	75	30 - 45
Paper storage	*same as press room		**same as press room
Binding, cutting, drying, folding, and gluing	70 - 80	21 - 27	45 - 50
Roll storage	73 - 80	23 - 27	50

* When paper is stored at other than room temperature, it should be temperature conditioned before the wrapper is opened.
** Multi-color offset and lithographic paper should be 5 to 8% above press room humidity. A sword hygrometer (paper hygroscope) is useful in checking the hygroscopic conditions of paper.

CHEESE MAKING

Cheese requires a humid atmosphere at various steps of processing and storage. Steam is a prime source of heat in the making of cheese and is usually readily available for humidification.

The following tables list the optimum temperature and humidity levels for various kinds of cheese.

Table 53-1: Swiss cheese manufacturing conditions

Processing step	Temperature °C	Temperature °F	Relative humidity %	Time
Setting	35	95	–	0.42 to 0.5 hours
Cooking	50 to 54.4	122 to 130	–	1 to 1.5 hours
Pressing	26.7 to 29.4	80 to 85	–	12 to 15 hours
Salting (brine)	10 to 11.1	50 to 52	–	2 to 3 days
Cool room hold	10 to 15.6	50 to 60	90	10 to 14 days
Warm room hold	21.1 to 23.9	70 to 75	80 to 85	3 to 6 weeks
Cool room hold	4.4 to 7.2	40 to 45	80 to 85	4 to 10 months

Table 53-2: Blue cheese manufacturing conditions

Processing step	Temperature °C	Temperature °F	Relative humidity %	Time
Setting	29.4 to 30	85 to 86	–	1 hour
Acid development	29.4 to 30	85 to 86	–	1 hour
(after cutting)	33.3	92	–	120 seconds
Curd matting	21.1 to 23.9	70 to 75	80 to 90	18 to 24 hours
Dry salting	15.6	60	85	5 days
Curing	10 to 12.8	50 to 55	95	30 days
Additional curing	2.2 to 4.4	36 to 40	80	60 to 120 days

Table 53-3: Other cheese manufacturing conditions

Variety	Temperature °C	Temperature °F	Relative humidity %	Time
Brick	15.6 to 18.3	60 to 65	90	60 days
Romano	10 to 15.6	50 to 60	85	5 to 12 months
Mozzarella	21.1	70	85	24 to 72 hours
Edam	10 to 15.6	50 to 60	85	3 to 4 months
Parmesan	12.8 to 15.6	55 to 60	85 to 90	14 months
Limburger	10 to 15.6	50 to 60	90	2 to 3 months

CANDIES AND NUTS

Temperature and humidity control during raw material storage, processing, and finished product storage and handling are essential to making consistent high quality candies. Table 54-1 shows the optimum design conditions for various candy processes.

The air conditioning system should be designed with a range of flexibility broad enough to encompass all temperatures and humidities expected to be needed.

Humidification steam, from either a vapor generator or in-plant boiler, offers the advantage of rapid moisture absorption by the air with minimum absorption distance. If cooling is needed and equipment space is available, the evaporative cooling method has the advantage of low energy cost cooling and humidification.

The moisture content of the candy, when finished, determines the optimum relative humidity during storage.

Recommendations are as follows:

Marshmallows, gum drops, coconut sticks, jelly beans and fudge are finished at 12% to 16% moisture content and should be stored at 65% relative humidity.

Fine candies, such as nougat bars, nut bars, hard and soft creams, bonbons, and caramels, contain 5% to 9% moisture and should be stored at 50% to 55% RH.

Milk chocolate, hard candies, and others with 2% and lower moisture content should be stored at 45% relative humidity.

Table 55-1 contains a breakdown of the expected storage life for various candies at different storage temperatures. Nuts must be refrigerated during storage to retard staleness, rancidity, and mold. They can be stored at temperatures ranging from 65°F (18°C) to -20°F (-29°C). Generally, the cooler the temperature, the longer the life. During storage, the humidity should be held at 65% to 75% RH. The storage atmosphere must be free of odors, which the oils in nuts readily absorb.

Table 54-1:
Recommended temperature and RH for candies and nuts

Department or process	Temperature °C	Temperature °F	Relative humidity %
Chocolate pan supply air	13-17	55-62	55-45
Enrober room	27-29	80-85	30-25
Chocolate cooling tunnel air supply	4-7	39-45	85-70
Hand dipper	17	62	45
Moulded goods cooling	4-7	39-85	85-70
Chocolate packing room	18	65	50
Chocolate finishing stock storage	18	65	50
Centers tempering room	24-27	75-80	35-30
Marshmallow setting room	24-26	75-78	45-40
Grained marshmallow (deposit in starch) drying	43	110	40
Gum (deposited in starch) drying	52-66	125-150	25-15
Sanded gum drying	52-66	125-150	25-15
Sugar pan supply air (engrossing)	29-41	85-105	30-20
Polishing pan supply air	21-27	70-80	50-40
Pan rooms	24-27	75-80	35-30
Nonpareil supply air	38-49	100-120	20
Hard candy cooling tunnel supply air	16-21	60-70	55-40
Hard candy packing	21-24	70-75	40-35
Hard candy storage	10-21	50-70	40
Caramel rooms	21-27	70-80	40
Raw material storage			
Nuts (insect)	7	45	60-65
Nuts (rancidity)	1-3	33-37	85-80
Eggs	-1	30	85-90
Chocolate (flats)	18	65	50
Butter	-7	20	–
Dates, figs, etc.	4-7	39-45	75-65
Corn syrup†	32-38	90-100	–
Liquid sugar	24-27	75-80	43-30
Comfort air conditions	24-27	75-80	60-50

† Depends on removal system. With higher temperatures, coloration and fluidity are greater.

Table 55-1:
Storage life for candies at various temperature

Department or process	Moisture content %	Relative humidity %	Storage life, months @ storage temperature			
			20°C	9°C	0°C	-18°C
			68°F	48°F	22°F	0°F
Sweet chocolate	0.36	40	3	6	9	12
Milk chocolate	0.52	40	3	2	4	8
Lemon drops	0.76	40	2	4	9	12
Chocolate covered peanuts	0.91	40-45	2	4	6	9
Peanut brittle	1.58	40	1	1.5	3	6
Coated nut roll	5.16	45.50	1.5	3	6	9
Uncoated nut roll	5.89	45-50	1	2	3	6
Nougat bar	6.14	50	1.5	3	6	9
Hard creams	6.56	50	3	6	12	12
Sugar bonbons	7.53	50	3	6	12	12
Coconut squares	7.70	50	2	4	6	9
Peanut butter taffy kisses	8.20	40	2	3	5	10
Chocolate covered creams	8.09	50	1	3	6	9
Plain caramels	9.04	50	3	6	9	12
Fudge	10.21	65	2.5	5	12	12
Gum drops	15.11	65	3	6	12	12
Marshmallows	16.00	65	2	3	6	9

BREAD AND BREAD PRODUCTS

Temperature control is a critical factor in each step of the baking process, from ingredient storage through mixing, fermentation, final proofing, baking, cooling, slicing, wrapping and storage.

To assure a high-quality product, humidity must be controlled during the three critical steps of fermentation, final proofing, and cooling.

The fermentation period is 3 to 5 hours duration, depending on the dough formula. And since water is a major ingredient of the product, uncontrolled evaporation causes significant variations in final weight and quality.

Yeast develops best at 80°F (27°C). A minimum of 75% relative humidity allows the dough surface to remain open so that the gases formed in fermentation are uniformly released. In the baking of breads, this allows the loaf to develop evenly, with a fine texture and no large voids. Since fermentation produces heat, water sprays that are located at the periphery of the fermentation room, are sometimes used to humidify and cool.

During final development or proofing loaves are placed in a proof box for a period of 1 to 1.25 hours.

While in the proof box, to keep the almost exhausted yeast alive, the temperature is held at 95°F to 120°F (35°C to 49°C). To provide a pliable loaf surface, thus assuring proper crust development during baking, the humidity is generally held at 85% to 90% RH.

While the final product is cooled, the humidity is usually held at 80% to 85% R. H. and the temperature at 75°F (24°C).

Steam from a vapor generator, or in-plant boiler if free of volatile amine corrosion protection chemicals (see page 24), can be used for this purpose. The proofing room must be adequately insulated to prevent condensation on the inner wall surfaces, which could result in mold formation. If a vapor generator is used, it may be advisable to consider the use of demineralized water for make-up to ensure clean, odor-free steam. The vapor generator must be designed and constructed of materials suitable for this type of water because of its corrosive nature.

REFERENCES

(1) ASHRAE Handbook, Applications, 1995

Section VIII: High Tech Humidification

ENVIRONMENTAL TEST FACILITIES

Environmental testing is usually of two broad types, climate and dynamic.

Climate testing includes temperature, humidity, salt spray, fungus, and combinations of the preceding.

Dynamic testing includes vibration shock, acceleration, radiation, and other forms of stress.

Multiple usage chambers usually have a wide range of capability in temperature and humidity. Humidity requirements may range from 2% to 98% RH over a temperature range of 35°F to 185°F (2° to 85°C).

Direct injection of steam from an in-plant boiler, assuming boiler chemicals are not objectionable, is suitable for high temperatures and humidities and where quick and accurate response to control is necessary. Modulating control is necessary where a precise humidity level is required.

When high purity water vapor is required, various types of vapor generators using demineralized water for make-up, can be employed.

Care is necessary to avoid causing wet areas in ducts where bacteria could be harbored.

In some cases, where extreme reliability of temperature and humidity control are needed, a dual system (direct steam injection and vapor generator) may be warranted.

LABORATORY HUMIDIFICATION

Because many labs are operating in clean-room environments, they require high levels of air changes. Laboratories generally are of four types: biological, chemical, physical and animal.

Table 58-1:
Recommended ambient temperatures and humidity ranges for laboratory animals

Species	Temperature °F	RH%	Reference
Mouse	64-79	40-70	ILAR (1977)
Hamster	64-79	40-70	ILAR (1977)
Rat	64-79	30-70	ILAR (1977)
Guinea Pig	64-79	40-70	ILAR (1977)
Rabbit	61-70	40-60	–
Cat	64-84	30-70	ILAR (1978)
Dog	64-84	30-70	–
Non-human primate	64-84	30-70	ILAR (1980)

Note: The above ranges permit the scientific personnel who use the facility to select optimum conditions (set points). They do not represent acceptable control range tolerances.

Biological labs support research on scientific disciplines, such as biochemistry, microbiology, cell biology, biotechnology, immunology, botany, pharmacology, and toxicology.

Chemical labs support organic and inorganic synthesis and analytical functions. Some organic and inorganic chemicals are hygroscopically affected by the RH levels. Some chemical reactions are triggered by atmospheric moisture. Humidity must be controlled for elimination of static electricity and for precise dimensional requirements.

Physical labs support experimental processes involving physics including lasers, optics, nuclear, electronics, etc. Humidity must be controlled for the elimination of static electricity and for precise dimensional requirements.

Animal labs support housing of the animals used in medical research and the treatments performed. These labs require narrow ranges of humidity and temperature control.

DATA-PROCESSING SPACES

Heat removal and proper humidity control throughout the year is a requirement of most computer and related equipment installations. Auxiliary devices and supplies (magnetic tape, disc packs, cartridges, data cells, and paper) should be kept in atmospheric conditions similar to those in the computer room. If this is not possible, they may require conditioning or tempering under computer-room atmospheric conditions for a period prior to use.

In some earlier installations, cooling air was supplied directly into the hardware. In those cases, the air stream must not be allowed to exceed 80% relative humidity because doing so may adversely affect some of the components, particularly those made of gold and silver.

Generally, computer rooms are designed for a temperature of 72°F (22°C) and 45% relative humidity.

To maintain proper relative humidity in a computer room, vapor transmission retarders sufficient to restrain moisture migration during the maximum expected vapor pressure differences between the computer room and surrounding areas, should be installed around the entire envelope. Cable and pipe entrances should be sealed and caulked with a vapor-proof material. Door jambs should fit tightly. Windows in colder climates must be of the insulated glass variety.

Computer-room systems should provide only enough outdoor air for personnel requirements and to maintain the room under a positive pressure relative to surrounding spaces. In most computer rooms, an outdoor air quantity of less than 5% of total supply air will satisfy ventilation requirements and prevent inward leakage. Outdoor air in excess of the required minimum increases the cooling and heating loads and makes control of atmospheric contaminants and winter humidity more difficult.

In order to minimize the dehumidification effect of the cooling coil, the coil should be selected to operate at a sensible heat ratio of .9 to 1.0 (dry coil) at 72°F (22°C), and 45% relative humidity. When applying humidification apparatus, the moisture should be supplied on the leaving side of the cooling coil.

Typically, self contained (packaged) air conditioners are used in these applications. They usually discharge the air vertically downward into a raised floor cavity. Locating the humidifier downstream of the cooling coil usually requires the humidification to be introduced inside the air conditioner since there is no duct space for this purpose.

Various humidification methods are used including direct boiler steam injection, electrode cylinder type vapor generators, electric hot element vapor generators, infrared evaporators and sometimes wetted media types.

TELECOMMUNICATIONS

This industry relies heavily on humidification to minimize malfunctions in electronic switching stations. Inadequate humidity causes equipment failures due to the attraction of very fine dust particles to the sensitive electronic devices in use.

Many of the smaller switching stations are unmanned and are visited on an “as needed” schedule for inspection and periodic maintenance. Such being the case, the humidifiers must be of a type and style that can operate without fail for extended periods. This usually requires types that utilize softened or demineralized water for make-up. Further, they must not add dust particles to the air.

PHARMACEUTICAL AND BIO-MANUFACTURING CLEAN ROOMS

These areas can be either aseptic or non-aseptic. They are typically arranged in suites with specific functions taking place in each suite. Requirements for temperature and humidity are derived from governmental regulations specific to the product.

Variations in ambient temperature or humidity can cause imperfections in the chemical structure of substances, such as penicillin. A temporary variation in humidity can cause rejection of an entire batch of drugs. Time-stamped temperature and humidity recordings that are measured with certifiable instrumentation are a requirement for some pharmaceutical products. Humidification equipment for these areas is typically redundant with multiple back-up units of the type that supply humidity on a continuous basis, such as modulating direct steam injection or, in smaller areas, SCR modulated, electric hot element vapor generators.

ELECTRONIC ASSEMBLY AREAS

Production of large-scale electronic assemblies, such as in the aerospace industry, typically requires a particle-free environment. The production area is frequently pressurized to prevent infiltration of airborne contaminants from other parts of the factory.

One method to reduce particle deposition, also used in clean rooms for semi-conductor fabrication, is to increase the velocity of the airflow within the room. This has been done by using a single large central airhandler to supply humidified air to smaller, secondary airhandlers that are located in the ceiling or floor of the actual clean-room assembly area. These smaller units supply high velocity air flow to only a small portion of the clean room. They can be equipped with heating or cooling coils, high-efficiency filtration, and zone humidifiers to correct for variation in ambient temperature and humidity within the assembly area.

Accessibility of humidification equipment is very important. All servicing must be done without entering or affecting the production area.

ELECTRONIC EQUIPMENT ROOMS IN HIGH DESERT CLIMATES

Humidification is not relegated to colder climates. Static electricity damage can occur in high desert climates due to the low ambient humidities at outdoor ambient temperatures exceeding 90°F (32°C).

One incident took place on a warm day as a service technician descended a stairway after servicing a rooftop air-conditioning unit. Momentarily resting his hand on the steel rack of a card cage, static electricity jumped from him to a highly specialized circuit board. The costs of the resulting damage would have more than compensated for the installation of humidification equipment and all associated maintenance costs for several years.

At risk for this type of damage are electronic equipment rooms with digital components installed in racks. This is typical of radar facilities, telephone switching stations, internet routing centers, web server buildings, microwave relay buildings, and computer training facilities located in the high-altitude, dry climates of the western United States.

Humidification in this type of facility can be challenging due to the remoteness of the locations, many of which are unmanned, and the hardness of the water supply.

An additional factor is the danger of substantial damage caused by any leakage of water. The physical location of the humidifier within the building is very important in a facility that does not have an on-site maintenance staff.

CLEAN ROOM/SPACE HUMIDIFICATION

The design of humidification systems for clean rooms requires an in-depth knowledge of temperature and humidity controls. Since particulate and contaminant control are of such high priority, careful equipment selection and application are needed to assure integrity.

CLEAN ROOM APPLICATIONS

Many different industry groups use clean rooms for their products. Among them some of the most prominent are the semicondustors, pharmaceutical, and aerospace industries.

Semiconductor facilities account for a significant percentage of all clean rooms in operation in the United States, with most of the newer semiconductor clean rooms being Class 100 or cleaner.

Preparation of pharmaceutical, biological, and medical products and genetic engineering research are prime examples of this industry.

Aerospace applications include manufacturing satellites, missiles, and aerospace electronics. Most involve large volume clean spaces with cleanliness levels of Class 10,000 or higher.

Miscellaneous additional clean room applications include aseptic food processing and packaging, manufacture of artificial limbs and joints, automotive paint booths, laser/optic industries and advanced materials research.

FIGURE 61-1: CLEAN ROOM

SEMI-CONDUCTOR CLEAN ROOMS

Precise temperature control is required in most semi-conductor clean rooms. Temperature tolerances of ±1°F are common, and an even tighter tolerance of ±0.1 to 0.5°F is required in some process areas. For example: wafer reticle writing by electron beam technology requires ±0.1°F, while photolithographic projection printers allow ±0.5°F tolerance.

Semi-conductor humidity requirements vary from 30 to 50% RH. Required humidity accuracy tolerances vary from 0.5 to 5% RH.

Photolithographic areas have more precise accuracy needs and the narrowest range of RH. Photoresists are chemicals used in photolithography, and their exposure timing can be affected with varying relative humidities. Negative resists typically require low RH, ranging from 35 to 45%. Positive resists tend to be more stable, so the range can go up to 50% RH, which also reduces the likelihood of a static electricity problem.

Since a small change in temperature has a significant effect on RH, it is imperative that temperature be precisely controlled when close- tolerance RH control is required.

HIGH TECH HUMIDIFICATION SYSTEM CONSIDERATIONS

One of several different types of humidification systems can be used, depending on (1) climate, (2) closeness of control accuracy required, (3) energy/operational cost, (4) size of the job, and (5) purity of humidification needed.

Steam Boiler

A steam boiler, dedicated solely to humidification, supplied with softened, but not chemically treated water and supplying steam duct type dispersers is one option. Obviously, the boiler may require replacement prematurely, due to a lack of chemical treatment but this alternative can sometimes be justified.

High-quality, industrial- grade modulating steam valves are needed to provide a highly accurate control of steam flow.

STEAM HEATED SECONDARY STEAM GENERATOR

This type of system has been installed in pharmaceutical manufacturing clean areas, semiconductor clean spaces and hospitals. The humidification steam is generated in an unfired pressure vessel similar to a scotch marine boiler, but instead of a fuel burner, it contains a tube bundle heat exchanger that is supplied with boiler steam.

Because of the high equipment cost, it is generally used only on large systems requiring high purity water vapor. (See page 110.)

Deionized water is used exclusively for make-up. It offers the dual advantages of providing the high-purity steam required in the manufacturing process and also protects the steam generator from fouling.

Medium-to high-pressure steam is supplied to the tube bundle, creating low-pressure (15 psi or less) secondary steam in the vessel. The low-pressure steam is then distributed via a stainless steel piping system to dry type, duct-injection steam dispersers or steam dispersion panels located at the points of use. All components in contact with the demineralized steam (generator, valves, piping, steam traps, dispersers) are typically constructed of a series 300 alloy, stainless steel.

VAPOR GENERATORS

These devices are designed for use with several forms of heating energy. These include steam, hot water, electricity, and direct-fired gas burners. (See page 111.)

A vapor generator consists of an enclosed water reservoir, usually of stainless steel, that contains the heating device (heat exchanger, electric hot element, etc.). The water to be evaporated surrounds the heating device in the reservoir. Typically, softened or demineralized water is supplied to the unit.

Electric hot element type vapor generators, SCR controlled and supplied with demineralized water will provide the highest level of control accuracy.

The energy costs associated with this type may be higher than other methods of humidification. However, in the cleanroom environment, the most critical concerns are dependability and accurate humidity control. This type of humidification system is highly rated for both needs.

Since they are designed to operate at very low pressure (less than .5 psi), these generators are located at or near the point of use, so the piping conveying the water vapor to the ducts can be kept as short as possible.

Well-designed steam dispersers that do not drip condensed water vapor and that afford rapid and thorough absorption of the water vapor should be used.

COMPRESSED AIR FOGGERS

These systems are in use in many very large semiconductor clean spaces. (See page 108.) Multiple fogger heads are mounted in the main air handler and strategically located to facilitate and accelerate the evaporation of the atomized fog. They are supplied with demineralized water for atomization. The portion of the air-handling system, that could be wetted by the water fallout is typically made of corrosion-resistant stainless steel.

Generally, these systems are designed for maximum dependability and highly accurate RH and temperature control. Energy consumption is a secondary consideration. In operation, the air is first preheated to a temperature that will evaporate a sufficient fog amount, then a cooling coil controlled by a wet bulb sensor condenses out enough moisture to provide the necessary absolute humidity ratio. Finally, a reheat coil heats the air to the required temperature, which then results in the exact desired RH.

Section IX: Health Care Facilities

Continual advances in medicine and technology necessitate constant re-evaluation of the air conditioning needs of hospitals and medical facilities.

A general, acute care hospital has a core of critical care spaces, including operating rooms, labor rooms, delivery rooms, nursery, and, in some cases, a burn unit. The facility also has an emergency room, a kitchen, dining and food service, morgue and central housekeeping support.

AIR CONDITIONING IN THE PREVENTION AND TREATMENT OF DISEASE

Hospital air conditioning assumes a more important role than merely the promotion of comfort. In many cases, proper air conditioning is a factor in patient therapy; in some cases, it is the major treatment.

A hot, dry environment of 90°F dry bulb and 35% RH has been successfully used in treating patients with rheumatoid arthritis. Conversely, excessively dry conditions may constitute a hazard to the ill and debilitated by contributing to a secondary infection or an infection totally unrelated to the clinical condition causing hospitalization.

Clinical areas devoted to upper respiratory disease treatment and acute care, as well as the general clinical areas of the entire hospital, should be maintained at a relative humidity between 30% and 60%. For example, patients with chronic pulmonary disease often have viscous respiratory tract secretions. Under these circumstances, warm, humidified air is essential to prevent further dehydration of them, thus impeding their riddance.

Patients needing oxygen therapy and those who have had a tracheotomy require special attention to ensure a warm, humid supply of air.

Table 66-1:
Recommended temperatures and RH for hospitals

	Temperature		RH%		
	°F	°C	Min. %	Max. %	Avg. %*
Operating room	68-76	20-24	50	60	50
Delivery room	70-76	21-24	50	60	50
Recovery room	75	24	50	60	50
Intensive care	72-78	22-26	30	60	35
Nursery	75	24	30	60	35
Patient room	75	24	30	40	35
Radiology	72-75	22-24	40	50	42
Computer room	72-75	22-24	30	45	42

* Average % means that which is in general usage, or recommended.

Burn patients need a hot environment and high relative humidity. A ward for severe burn victims should have temperature controls that permit adjusting the room temperature up to 90°F dry bulb and the relative humidity up to 95%.

HOSPITAL FACILITIES

Although proper air conditioning is helpful in the prevention and treatment of disease, the application of air conditioning to health facilities presents many problems not encountered in the usual comfort conditioning system.

The basic differences between air conditioning for hospitals (and related health facilities) and that for other building types stem from (1) the need to restrict air movement and isolate treatment rooms; (2) the specific requirements for ventilation and filtration to dilute and remove contamination in the forms of odor, airborne microorganisms, viruses, hazardous chemicals and radioactive substances; (3) the differing temperature and humidity requirements for various areas; and (4) the design sophistication needed to permit accurate control of environmental conditions.

TEMPERATURE AND HUMIDITY

Codes and guidelines specify temperature and humidity range criteria in some hospital areas as measures for infection control and comfort.

AIR MOVEMENT

In general, outlets supplying air to sensitive, ultraclean areas and highly contaminated areas should be located on the ceiling, with exhaust or returns located on the perimeter near the floor. This arrangement provides a downward movement of the cleanest air through the breathing and working zones to the floor area for exhaust. The bottoms of return or exhaust openings should be at least 3 inches above the floor.

However, the laminar airflow concept, developed for industrial clean room use, has attracted the interest of some medical authorities. In this system, the supply air enters the space through outlets located along one wall. It then flows horizontally across the room to return inlets located on the oposite wall. There are advocates of both vertical and horizontal laminar airflow systems, with and without fixed or movable walls around the surgical team.

Laminar airflow systems have shown promise for rooms used for the treatment of patients who are highly susceptible to infection. Among these are burn patients and those undergoing radiation therapy, concentrated chemotherapy, organ transplants, amputations, and joint replacements.

In critical patient-care areas, constant volume systems should be employed to assure proper pressure relationships and ventilation. In noncritical patient-care areas and staff rooms, variable air volume (VAV) systems may be considered for energy conservation. When using VAV systems within the hospital, special care should be taken to ensure that minimum ventilation rates (as required by codes) are maintained and that proper pressure relationships between various departments are maintained.

When humidifying VAV systems, the broad range of humidification load swings requires special RH control. Otherwise, under- and over- shooting of the control point, with moisture fallout in the air duct will likely occur. (See page 134.)

SURGERY AND CRITICAL CARE

No area of the hospital requires more careful control of the sterile condition of the environment than does the surgical suite. Below is a list of factors to consider when designing for a surgical suite.

1. There should be a variable range temperature capability of 68° to 76°F or more.
2. Relative humidity should be kept between 50 and 60%.
3. Humidity and temperature indicators and thermometers should be located for easy observation.
4. Sufficient lengths of watertight, drained, stainless steel duct should be installed downstream of humidification equipment to prevent leakage in the event of accidental water discharge or condensation.
5. Preferably, the duct-mounted steam disperser should be of a design that produces nearly instantaneous evaporation of the steam/mist mixture. If so, a final, high-efficiency type filter bed, downstream from the humidifier, can be utilized with no concern of it becoming wet.

RECOVERY ROOMS

Postoperative recovery rooms used in conjunction with the operating rooms should be maintained at a temperature of 75°F and a relative humidity between 50 and 60%.

NURSERY SUITES

Nurseries should be designed for 75°F and 30 to 60% RH.

INTENSIVE CARE UNIT

This unit serves seriously ill patients, from the postoperative to the coronary patient. A variable range temperature capability of 75° to 80°F, a relative humidity of 30% minimum and 60% maximum, and positive air pressure are recommended.

NURSING CARE FACILITIES

Nursing homes are divided into three general classifications: extended care, skilled care, and residential, or maintenance care.

Extended care homes are used mainly by persons recuperating after leaving the hospital. This facility is used by people of all ages whose stay is usually less than 60 days.

Skilled care homes are intended for patients who are dependent on assistance for all or most daily activities. Characteristics of such patients could include disorientation, incontinence, and immobility. Some forms of therapeutic services may be provided. The clientele is usually of advanced age (average 80 years), and their stay ranges from months to years.

Residential, or maintenance care homes are intended for elderly persons who cannot perform housekeeping duties on a daily basis but who can attend to their personal needs. These persons are able to move freely about the residence and the immediate community. Highly skilled nursing care is not usually provided, and their stay can last years.

Generally, the occupants of all these facilities are senior citizens. They prefer temperatures at the high end of the comfort range. They frequently benefit from a humid atmosphere by reduced respiratory irritation, reduced skin disorders and reduced static electricity, control of odors and lowered air bacterial counts.

The humidifier design and application should address the potential hazard of wet areas in ductwork that would foster the growth of bacteria and algae. A constant temperature of 75°F (24°C) and 30% to 50% relative humidity is recommended for all areas except hydrotherapy and baths, where 80°F (27°C) and 50% relative humidity is preferred.

FIGURE 69-1: HEALTH CARE FACILITIES

Section X: Libraries and Museums

LIBRARIES AND MUSEUMS

Many priceless manuscripts, books, works of art, and other exhibited objects have been destroyed or badly damaged because they were not kept in a properly controlled atmosphere. The environmental conditions most suitable for these objects do not generally fall within the human comfort range.

In addition to housing books, libraries handle microfilms and many forms of electronic media records. Some also handle works of art. Microfilms and magnetic tape become too dry below 37% relative humidity, while paper survives best at an upper limit of 40%.

A variety of museum pieces are organic in nature and survive best at lower temperatures. Some museums rotate their collections between storage and display. While in storage, the temperature should be kept cooler than the comfort range.

Many art conservators firmly believe that maintaining a constant relative humidity is far more important than maintaining a constant temperature. Changes in RH, especially for such hygroscopic materials as wood and paper, can cause dramatic dimensional changes to objects composed of these materials. In fact, a change of 4% RH for certain moisture-absorbent wooden materials can cause an expansion across the grain equivalent to a change of 18°F, at a constant relative humidity. A change of 18°F would be extraordinary in a building with a working air conditioning system, whereas a 4% change in RH is usually considered within spec for most museum environmental control systems.

FIGURE 72-1: LIBRARIES AND MUSEUMS

Changes in the surface temperature of a display object due to heat conduction (an outside wall location) or to radiation (a spotlight intermittently turned on and off) can produce injurious moisture content changes in the object even though the surrounding atmosphere is held constant. If the objects are allowed to cool off overnight, a moisture content change will occur during warm-up the next day. During warm-up, layers of air of exceedingly high moisture content will in extreme cases occur on the surface of the object, causing condensation on the object. This moisture change accelerates the deterioration process.

Direct injection of steam into the air of museums and libraries was once a fairly common method of humidification. In most cases, this humidification practice is currently being scrutinized because of the presence of boiler- treatment chemicals usually present in the steam. These chemicals, consisting of neutralizing amines, have been found to accelerate the aging process of certain artifacts. (See page 36.)

The desired atmospheric conditions of both libraries and museums are generally similar. The following temperature and relative humidity levels are recommended.

Table 73-1:
Recommended temperatures and humidities for libraries/museums

	Temperature		Humidity
	°F	°C	%
Average library or museum	68-72	20-22	45-55
Archival library or museum	55-65	12-18	35
Art storage	65-72	18-22	50±2
Stuffed furred animals	45-50	4.510	50
Fossil and bones keep better at higher, rather than lower, humidities.			

Section XI: Humidity and Structure Design

When humidification is contemplated in a structure being designed, certain precautions should be kept in mind.

MOIST AIR IS LIGHTER THAN DRY AIR

Contrary to common belief, water vapor is lighter than dry air. A cubic foot of 72°F, 100% saturated water vapor weighs less than a cubic foot of dry air at the same temperature and pressure. This fact should be borne in mind when trying to force warm, humid air down into a space from overhead.

VAPOR BARRIERS

As stated in an earlier section, water vapor migrates from areas of high vapor pressure to areas of low vapor pressure. Temperature and vapor pressure go hand in hand. The cooler the temperature of an area, the lower the vapor pressure and the greater the tendency for water vapor to move to that area.

As a consequence, insulation that is not properly protected by a vapor barrier on the warm side will likely become wet due to condensed water vapor. In freezing climates, frost may form within outside walls.

Doors exposed to the outside, if wood, should either be properly sealed against vapor penetration, or they should be constructed of hollow core steel or some other non-hygroscopic material and vapor sealed.

Areas where high parallel heat transfer could occur, such as steel roof and wall supports, should be well insulated and vapor-protected on the warm side.

The use of multiple thickness, welded type window glass minimizes condensation and frost formation.

In an existing building not equipped with vapor barriers, the use of foil type wall covering or low-permeability paints on walls and ceilings will help to contain the water vapor.

In mild climates, structural protection against condensation within walls may not be required. However, vapor barriers may still be warranted to reduce the amount of make-up moisture required.

Table 77-1:
Window condensation at various indoor humidity levels 70°F (21°C) room temperature

Material	Outdoor temperature						
	30 °F	20 °F	10 °F	0 °F	-10 °F	-20 °F	-30 °F
	-1 °C	-7 °C	-12 °C	-18 °C	-21 °C	-27 °C	-32 °C
Single glass	29%	21%	15%	12%	8%	5%	3%
Double glass 3/16″ air space	52%	43%	38%	33%	28%	25%	20%
Double glass 1/4″ air space	54%	48%	41%	36%	30%	27%	23%
Triple glass 1/4″ air space	72%	63%	57%	52%	48%	44%	29%
Triple glass 1/2″ air space	77%	72%	63%	60%	55%	51%	48%

CONDENSATION ON WINDOWS

Condensation flowing down window glass can permanently damage decorative trim and wall finishes. It occurs whenever the temperature of the affected surface falls below the dew-point of the adjacent air.

Various types of windows with varying abilities to resist condensation are available. Table 77-1 serves as a guide in choosing the proper type for the expected conditions of temperature and humidity.

COLD SNAP OFFSET

Some humidification applications, such as those of a comfort only nature, are not highly demanding in control accuracy. These systems may permit a short term, reduced indoor RH during a cold snap. When such is the case, the use of an over-riding device that will automatically reset the humidity control point downward and upward in accordance with changes in the outside temperature should be considered. By making this short-term compromise of lowered RH, it may be possible to reduce construction costs and eliminate window condensation problems while still providing a sufficiently comfortable overall environment.

Section XII: Humidity Load Calculations

A humidified building constantly loses moisture for several reasons. The principal reason is that of air change. Stack effects (in tall buildings) and wind pressure cause warm, humidified air to leave a building while a corresponding amount of cold, dry air enters to take its place. This is called natural ventilation.

The air that passes through a building because of a fan-operated ventilation system taking in outside air and exhausting it is another cause. This is called mechanical ventilation.

Hygroscopic materials or products moving through a building, and absorbing moisture as they go, can be still another cause.

Water vapor that migrates through the cracks, or passes through the walls themselves, forms yet another.

Certain year-round cooling applications impose a humidity loss due to condensation on the air-conditioning cooling coil. Computer rooms are a prime example. This subject is discussed on page 59.

On the other hand, these same buildings gain water vapor from activities taking place within and from the occupants themselves. Generally this gain is so small it can be ignored when calculating the humidification load.

As a general rule, the determination of load is based only on the amount of outside air entering and leaving a building or space. In buildings not having mechanical ventilation systems, the load is usually calculated strictly on the basis of natural infiltration-exfiltration.

It is usually sufficiently accurate to estimate air changes as following:

- Enclosed spaces with one side exposed = 1 air change per hour
- Enclosed spaces with two sides exposed = 1½ air changes per hour
- Enclosed spaces with three sides exposed = 2 air changes per hour
- Enclosed spaces with four sides exposed = 2 air changes per hour

The above estimate will serve as a guide for average construction and average usage. Cases involving unusually poor construction or doors left open for long periods of time should be judged accordingly. In some cases, it may be necessary to double the values shown.

When determining the load for a building that contains a ventilating system but operates with a low percentage of outside air (10% or less), it is wise to calculate the load twice. The first calculation should be made on the basis of air changes due to mechanical ventilation; the other should be made on the basis of natural ventilation as described above. Use the larger of the two results for determining the load.

It is also quite common in cold climates to ignore the moisture contained in the outside air (assume it to be zero) entering the building when making load calculations. The following example, however, has taken it into account.

MECHANICAL VENTILATION METHOD

Assume a printing plant that has a ventilation system which circulates 9000 cfm of total air, of which 25% is outside air. The space design conditions are 50% relative humidity and 70°F (21°C). Outside design conditions are 45% relative humidity and 10°F (-12°C). Determine the load:

From Table 81-1:

Desired moisture content at 70°F and 50% RH = 3.44 lbs/100 cfm.

Outside air moisture content is .30 lbs/100 cfm.

3.44 - .30 = 3.14 lbs/100 cfm to be added

25% of 9000 cfm = 2250 cfm

2250 ÷ 100 (base of Table 81-1) x 3.14 = 70.65 lbs/hour.

Table 81-1:
Pounds of moisture per hour per 100 cfm

Air Temperature		Percentage of saturation											
°C	°F	30%	35%	40%	45%	50%	55%	60%	65%	70%	80%	90%	100%
-29	-20	.043	.05	.057	.064	.071	.078	.064	.093	.099	.114	.13	.14
-23	-10	.074	.085	.097	.11	.121	.134	.147	.159	.171	.20	.22	.24
-18	0	.121	.142	.162	.184	.204	.223	.245	.265	.285	.33	.36	.40
-12	10	.20	.232	.266	.30	.332	.364	.40	.434	.465	.54	.59	.66
-7	20	.32	.374	.430	.494	.535	.583	.635	.695	.758	.86	.96	1.05
-1	30	.50	.585	.67	.75	.84	.92	1.0	1.09	1.17	1.34	1.49	1.65
4	40	.74	.84	.96	1.08	1.20	1.31	1.45	1.53	1.68	1.98	2.20	2.43
10	50	1.05	1.24	1.40	1.58	1.76	1.93	2.12	2.30	2.46	2.83	3.16	3.49
13	55	1.26	1.47	1.68	1.90	2.10	2.30	2.53	2.74	2.94	3.37	3.76	4.16
16	60	1.49	1.74	1.98	2.24	2.50	2.72	2.99	3.24	3.48	4.00	4.46	4.93
18	65	1.75	2.04	2.32	2.63	2.92	3.20	3.50	3.80	4.06	4.73	5.27	5.82
20	68	1.96	2.28	2.60	2.84	3.26	3.56	3.19	4.24	4.55	5.23	5.84	6.05
21.1	70	2.05	2.40	2.74	3.10	3.44	3.75	4.12	4.46	4.80	5.56	6.20	6.45
21.7	71	2.15	2.50	2.85	3.21	3.55	3.90	4.29	4.65	5.00	5.74	6.40	7.07
22.2	72	2.20	2.58	2.94	3.32	3.68	4.03	4.44	4.80	5.15	5.91	6.60	7.29
22.8	72	2.28	2.66	3.03	3.43	3.80	4.16	4.57	4.95	5.31	6.12	6.83	7.54
23.3	74	2.37	2.75	3.13	3.54	3.93	4.31	4.74	5.14	5.51	6.32	7.05	7.78
23.9	75	2.42	2.84	3.23	3.65	4.06	4.45	4.86	5.28	5.65	6.55	7.27	8.03
25.0	77	2.58	3.02	3.42	3.82	4.33	4.73	5.13	5.63	6.04	6.94	7.75	8.55
26.7	80	2.84	3.3	3.75	4.20	4.75	5.19	5.63	6.18	6.62	7.62	8.50	9.38
29.4	85	3.32	3.88	4.39	4.91	5.56	6.07	6.59	7.23	7.75	8.92	9.95	10.98
32.2	90	3.74	4.37	4.95	5.53	6.25	6.84	7.43	8.15	8.73	10.03	11.20	12.37
35.0	95	4.50	5.25	6.00	6.75	7.50	8.25	9.00	9.75	10.50	12.00	13.50	15.00
37.8	100	5.14	5.99	6.85	7.70	8.56	9.42	10.27	11.13	12.00	13.69	15.41	17.12
40.6	105	5.93	6.92	7.90	8.89	9.88	10.87	11.86	12.85	13.83	15.82	17.79	19.77
43.3	110	6.66	8.00	9.14	10.28	11.43	12.57	13.71	14.85	16.00	18.28	20.57	22.85
48.9	120	8.95	10.44	11.93	13.42	14.91	16.40	17.90	19.39	20.88	23.86	26.85	29.83
54.4	130	11.46	13.37	15.28	17.19	19.10	21.01	22.92	24.83	26.74	30.56	34.38	28.20
60.0	140	14.67	17.12	19.56	22.01	24.45	26.89	29.34	31.78	34.26	39.12	44.01	48.90
65.6	150	18.60	21.70	24.80	27.90	31.00	34.10	37.20	40.30	43.40	49.60	55.80	62.00
71.1	160	23.34	27.23	31.12	35.01	38.90	42.79	46.68	50.57	54.46	62.24	70.02	77.80
76.7	170	29.07	33.92	38.76	43.61	48.45	53.29	58.14	63.00	67.80	77.52	87.20	96.90
87.8	190	43.80	51.10	58.40	65.70	73.00	80.30	87.60	94.90	102.20	116.80	131.40	146.00
93.3	200	53.50	62.40	71.32	80.24	89.15	98.07	107.00	115.90	124.80	142.60	160.50	178.30

ECONOMIZER CONTROL

Many year-round air-conditioning systems are equipped with an economizer control. It is called an economizer control because it uses outside air instead of refrigeration to cool the building during those times when the outside temperature is sufficiently low.

There are three ASHRAE ventilation cycles that are used to control heating and cooling units. ASHRAE Cycle II is the most commonly used.

ASHRAE CYCLE I

During warm-up, the outdoor air damper is closed. As the room temperature approaches the thermostat setting, the outdoor air damper is opened fully. The heating source is modulated to maintain space temperature. This cycle is used only where large quantities of outdoor air are required to offset air being exhausted.

The maximum humidification load will be at the maximum air flow and at the minimum outdoor air moisture content.

ASHRAE CYCLE II

During warm-up, the outdoor air damper is closed. As the room temperature approaches the thermostat setting, the outdoor air damper opens to admit a predetermined amount of outside air. The heat source is modulated to maintain the thermostat setting. If the room temperature rises above the thermostat setting, the heat source is turned off and the outdoor air damper opens beyond minimum position to maintain the thermostat setting. The damper can be opened to admit up to 100% outdoor air.

This cycle is the most economical because a minimum amount of outdoor is heated during the heating season, and free outdoor air cooling is available to offset large internal heat gains associated with periodic dense occupancy.

During heating periods the maximum humidification load will be at the minimum outdoor air setting. Minimum outdoor settings range from 10% to as high as 40%, depending on the application. If, however, the internal heat gains are sufficient to require ventilation during the normal heating season, requiring the outside air damper to fully open, the maximum humidification load will occur at that time.

ASHRAE CYCLE III

During warm-up, the outdoor air damper is closed. As the room temperature approaches the thermostat setting, a mixed air thermostat on the entering side of the heating source modulates the outdoor air damper to maintain a constant mixed entering air temperature, usually 55°F. This cycle is economical in operation because the quantity of outside air to be heated, decreases as the outside temperature decreases.

When the outside air temperature has risen to 55°F, the system will be delivering 100% outdoor air. At a predetermined outdoor air temperature, usually 65° to 68°F, the outside air damper returns to the minimum setting, and mechanical cooling, such as chilled water or direct expansion, takes over.

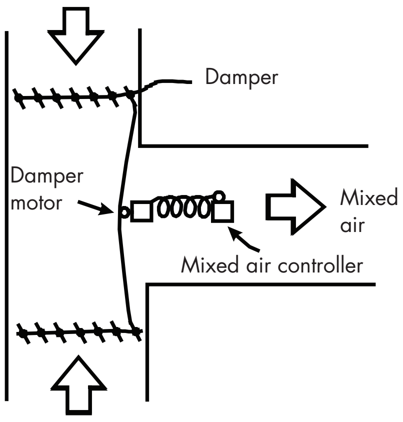
Calculating the maximum loads on ASHRAE Cycles I and II is relatively easy because the maximum quantity of outdoor air is known.

Determining the maximum load for Cycle III, however, can be time consuming because of two opposing variables:

1. As the outside temperature rises, its moisture content increases. This tends to decrease the load.

2. As the outside temperature rises, the ratio of outside air to return air increases. This tends to increase the load.

FIGURE 83-1: TYPICAL ECONOMIZER CONTROL



The formula for determining the ratio of outside air versus return air is:

A = V1

B = V2

Where:

A = temperature difference between mixed and outside air.

B = temperature difference between return air and mixed air.

V1 = volume (cfm) of return air.

V2 = volume (cfm) of outside air.

Example: Determine the outside air quantity in an economizer cycle system where the total air is 12,000 cfm when the outside air is 25°F, the return (room) is 70°F and the mixed air is 55°F.

A = 55° minus 25° = 30°

B = 70° minus 55° = 15°

Table 83-1:
Outside air percentages at various mixed air temperatures

Outside temperature °F	50°F mixed air	55°F mixed air	60°F mixed air
-20	22%	17%	11%
-10	25%	19%	13%
0	29%	21%	15%
10	33%	25%	17%
20	40%	30%	20%
30	50%	38%	25%
40	67%	50%	34%
50	100%	75%	50%
55	–	100%	67%
60	–	–	100%

30 = volume return air (or 2/1 or 2/3 vs. 1/3)

15 volume outside air

12,000 x 1/3 = 4000 cfm outside air

12,000 x 2/3 = 8000 cfm return air

Using the formula, a profile can be developed that shows the outside air quantity at various points (usually 10° intervals), over the temperature range of modulation of the outside air.

Table 83-1 based on the above formula, indicates the percentage of outside air at a 70°F room temperature and three different mixed air temperatures.

Table 84-1:
Outside air RH

Temperature °F	RH%
-20	85
-10	80
0	75
10	70
20	65
30	60
40	55
50	50
60	45

Determining the maximum load will involve the use of the closest of the three mixed-air temperatures in Table 83-1 that fits the problem at hand (or a similar table developed from different mixed and return air temperature conditions).

Also needed will be the difference (in lbs/hr/100 cfm) between the desired moisture content of the air in the space and that contained in the outside air. This difference will have to be made up by the humidifier.

Table 84-1 is an approximation of outside air RH based on Minneapolis weather data. It covers the temperature range between -20°F and 60°F and varies by 5% for each 10°F increment of outside temperature change. Using the same format, the values can be developed for any geographical area by obtaining local weather data.

EXAMPLE

Determine the maximum humidification load for a 12,000 cfm system having a mixed air temperature setting of 55°F when the desired inside conditions are 70°F and 35% RH.

STEP 1

Refer to Table 81-1. Read across the 70°F line to the 35 % column, and find 2.40 lbs/hr/100 cfm.

STEP 2

Determine the moisture to be added at each 10°F increment by using. Tables 83-1 and 84-1. Use the following formula:

{H (space) minus H (outside air)} x % outside air x cfm (total air) = lbs/hr. (load)

Note: H means "lbs/hr/100 cfm."

STEP 3

Now create a load calculation matrix using the method

described in Step 2. Use the temperature and moisture conditions suitable for the climatic conditions of the job location. The maximum load for this system is 65.4 lbs/hr, which occurs at 40°F as shown.

SHORTCUT METHOD

Table 85-2 combines Tables 83-1 and 84-1 and the matrix calculations. It is based on Minneapolis weather conditions and a 70°F humidified space temperature. To use, read across the appropriate mixed air temperature line to desired space RH column. Then multiply this value by the total cfm.

Table 85-1:
Load calculation matrix example

-20°	2.4	less	.12	=	2.28	x	17%	x	$\frac{12000}{100}$	=	46.5
-10°	2.4	less	.2	=	2.2	x	19%	x	$\frac{12000}{100}$	=	50.2
0°	2.4	less	.325	=	2.075	x	21%	x	$\frac{12000}{100}$	=	53.0
10°	2.4	less	.465	=	1.935	x	25%	x	$\frac{12000}{100}$	=	58.2
20°	2.4	less	.695	=	1.7	x	30%	x	$\frac{12000}{100}$	=	61.4
30°	2.4	less	1	=	1.4	x	38%	x	$\frac{12000}{100}$	=	63.8
40°	2.4	less	1.31	=	1.09	x	50%	x	$\frac{12000}{100}$	=	65.4
50°	2.4	less	1.76	=	.64	x	75%	x	$\frac{12000}{100}$	=	57.6
55°	2.4	less	2.0	=	.4	x	100%	x	$\frac{12000}{100}$	=	48.0

Table 85-2:
Shortcut table - maximum load in lbs/hr/100 cfm

Mixed air controller temperature setting		Humidified space desired relative humidity					
°F	**°C**	**30%**	**35%**	**40%**	**45%**	**50%**	**55%**
60	15	0.25	0.36	0.48	0.60	0.71	0.81
55	13	0.37	0.55	0.72	0.90	1.07	1.22
50	10	0.50	0.73	0.96	1.20	1.43	1.63

Note:

Economizer cycle “free cooling” by the use of outside air is not always energy cost effective. The operating cost advantage may be lost when the following operating conditions prevail:

1. The indoor relative humidity requirements are fairly high (40% RH or greater).
2. Electrical energy is to be used to provide the heat for evaporating water for humidification.

A crossover point will occur where the increased humidification energy cost will more than offset the saving, normally derived by using outside air for cooling, instead of operating the mechanical cooling system.

Section XIII: The Psychometric Chart

Psychrometrics deals with the determination of the thermodynamic properties of moist air. These data are used in the analysis of conditions and processes involving moist air.

In the 1940s a research project, to determine moist air properties, was conducted at the University of Pennsylvania by Mssrs. Goff and Gratch. They produced the tables called Thermodynamic Properties of Moist Air. This data was later updated at the National Bureau of standards in an ASHRAE-sponsored project. The later tables are identified in the Addendum section of this book.

The Psychrometric Chart is a graph that depicts the information contained in the tables at the back of this book. The use of the Psychrometric Chart makes it possible to quickly pinpoint the data needed in solving various problems and processes, as opposed to the sometimes lengthy mathematical solution of the data by using the tables.

Two charts, one for normal temperatures and the other for high temperatures, appear in this book. Examples 1 through 6 are based on the normal temperature chart.

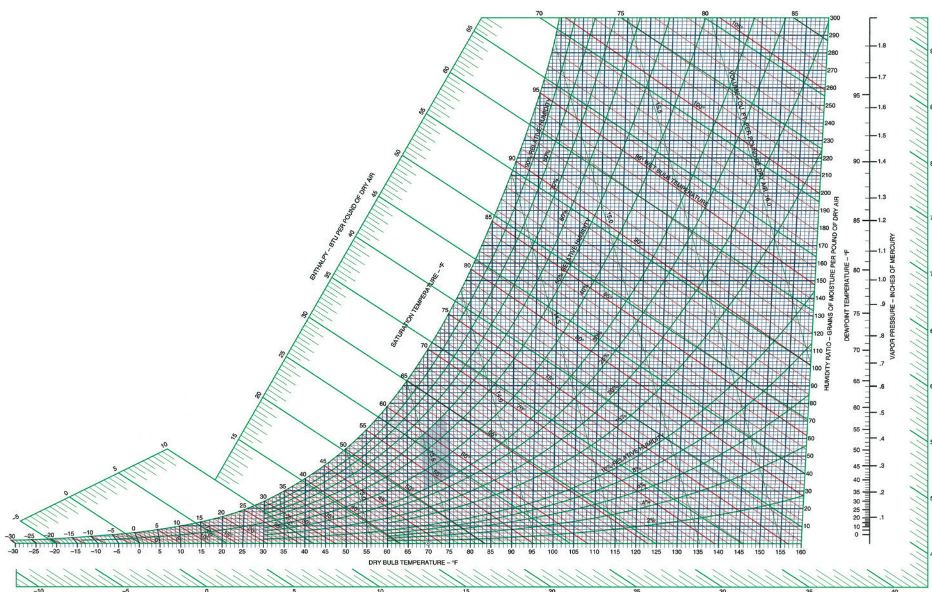
Prior to discussing the use of the Psychrometric Chart, it may be advisable to review the applicable terminology defined in Section I at the beginning of the book.

Water vapor, or moisture is an invisible gas that occupies space along with dry air. The combination, known as moist air, is defined as a binary (two component) mixture of dry air and water vapor. The amount of water vapor in moist air varies from near zero to a maximum of 100% RH, at which point the air is defined as saturated.

The degree of saturation, or relative humidity, is dependent upon the moisture content, the dry bulb temperature, and the atmospheric (barometric) pressure. The following psychrometric charts and examples, which are affected by elevation changes, are all based on sea-level pressure (29.921 inches of mercury).

The specific amount of moisture held by the air is defined as the humidity ratio (w), and is commonly expressed either as a decimal fraction of a pound (or kilogram), or in grains of moisture per pound (or kilogram) of dry air. There are 7000 grains in one pound (or 3175 grains in 1 kilogram). The dew point is a reflection of the humidity ratio, since it is the temperature at which the mixture is saturated.

A great deal more simple than it may at first appear, the Psychrometric Chart consists of the following:



(1) Humidity ratio values (w) –plotted vertically along the right hand margin, beginning with 0 at the bottom and extending to .03 at the top.

(2) Enthalpy, or total heat, (h) –plotted with oblique lines, at intervals of 5 BTU/lb of dry air, extending from upper left to lower right.

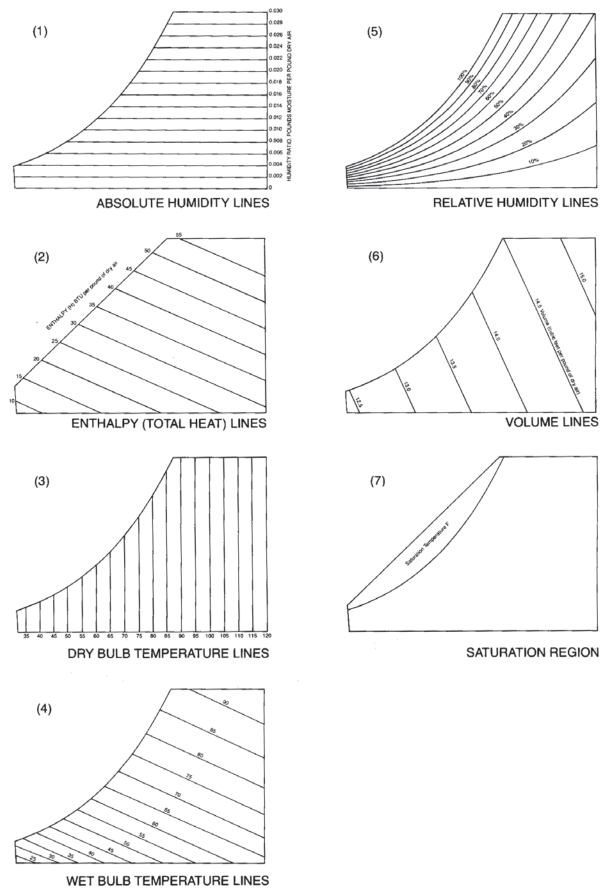
(3) Dry bulb temperature – inclined slightly from the vertical and plotted in 1°F-intervals.

(4) Wet bulb temperature – oblique lines that fall almost parallel to the enthalpy lines. They are also shown in intervals of 1°F.

(5) Relative Humidity – lines curving from lower left to upper right in intervals of 10%. They begin at the bottom at 10% and end at the top with the 100% saturation curve.

(6) Volume (cubic feet per pound of dry air – indicated with oblique lines at intervals of .5 cubic foot.

(7) Two – phase region - a narrow, cross-hatched area above and to the left of the saturation curve, indicating fog (a mixture of vapor and condensed water in equilibrium).



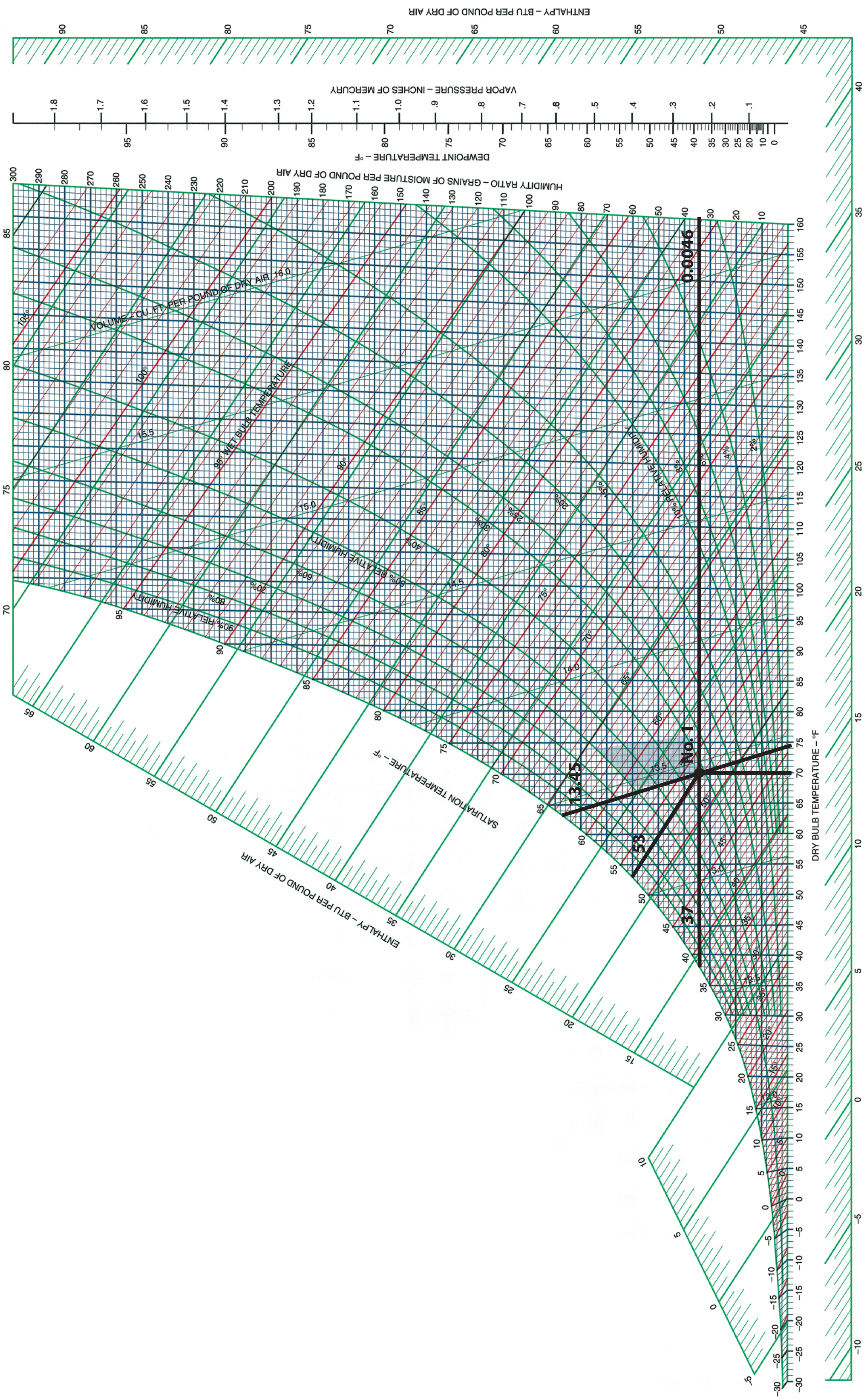
The examples that follow involve using the various elements of information contained in the chart.

EXAMPLE #1

Given the conditions of 70°F dry bulb and 53°F wet bulb temperature, determine the dew point, volume, and humidity ratio in both grains and pounds of dry air.

SOLUTION:

1. Locate the state point where the 70°F dry bulb line intersects the 53°F wet bulb line. Call this state point number 1.
2. Project horizontally to the left to the saturation curve and read 37°F (dew point).
3. Project horizontally to the right and read 32 grains of water per pound of dry air (humidity ratio) (w). To get pounds of water per pound of dry air: 32 / 7,000 = 0.0046.
4. Draw a line through the state point, parallel to the 13.5 volume line, and estimate a volume of 13.45 cubic feet per pound of dry air. This is the volume of the mixture of dry air and water vapor.

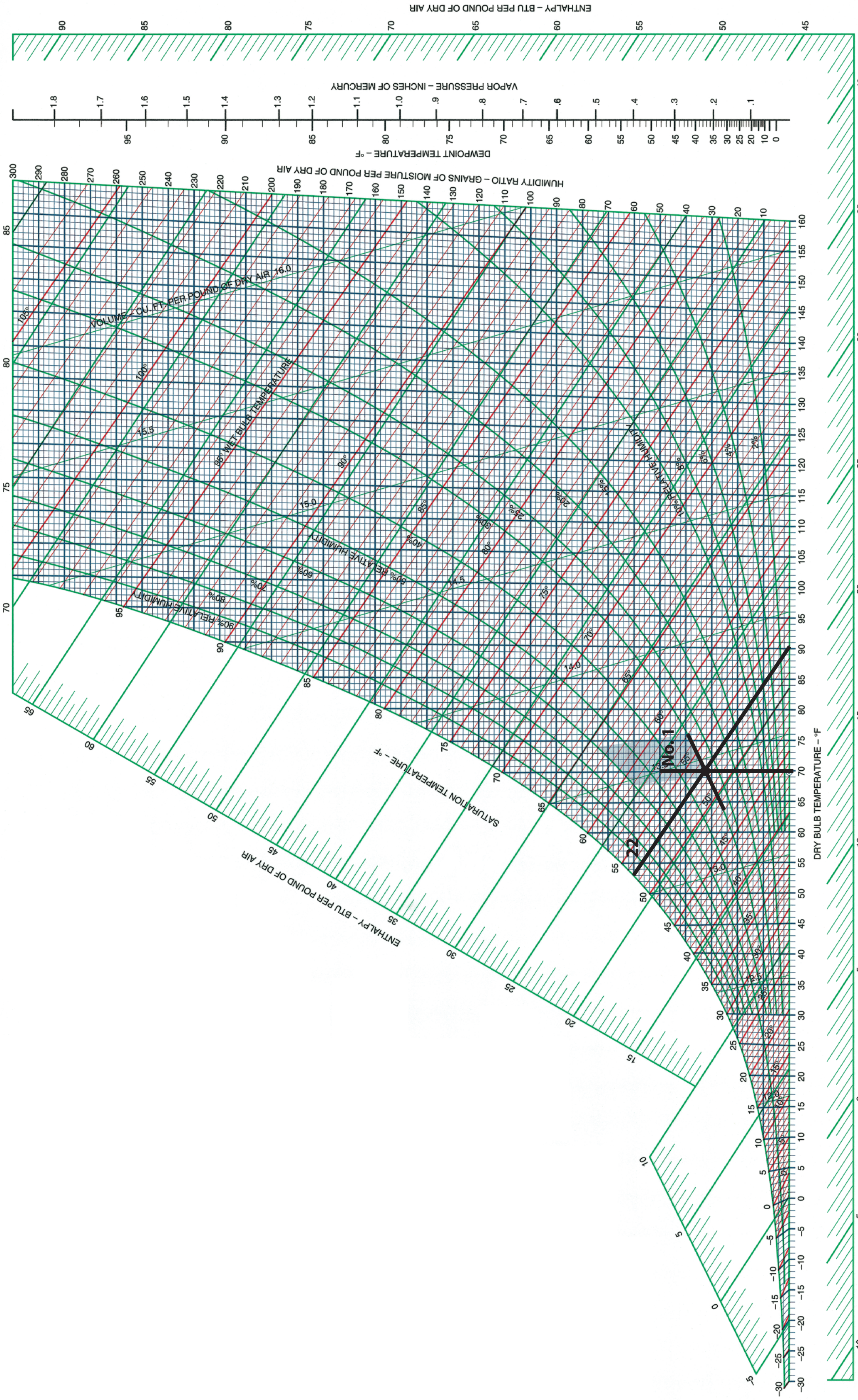

ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
0.0046
No. 1
13.45
53
37

EXAMPLE #2

From state point 1 (intersection of 70°F dry bulb and 53°F wet bulb) in Example No. 1, determine the relative humidity and the enthalpy (total heat) of the mixture.

SOLUTION:

1. Note that state point number 1 falls on the curved relative humidity line labeled 30%.
2. A line drawn through state point 1 and parallel to the nearest enthalpy (total heat) line falls on 22 BTU/pound of dry air.

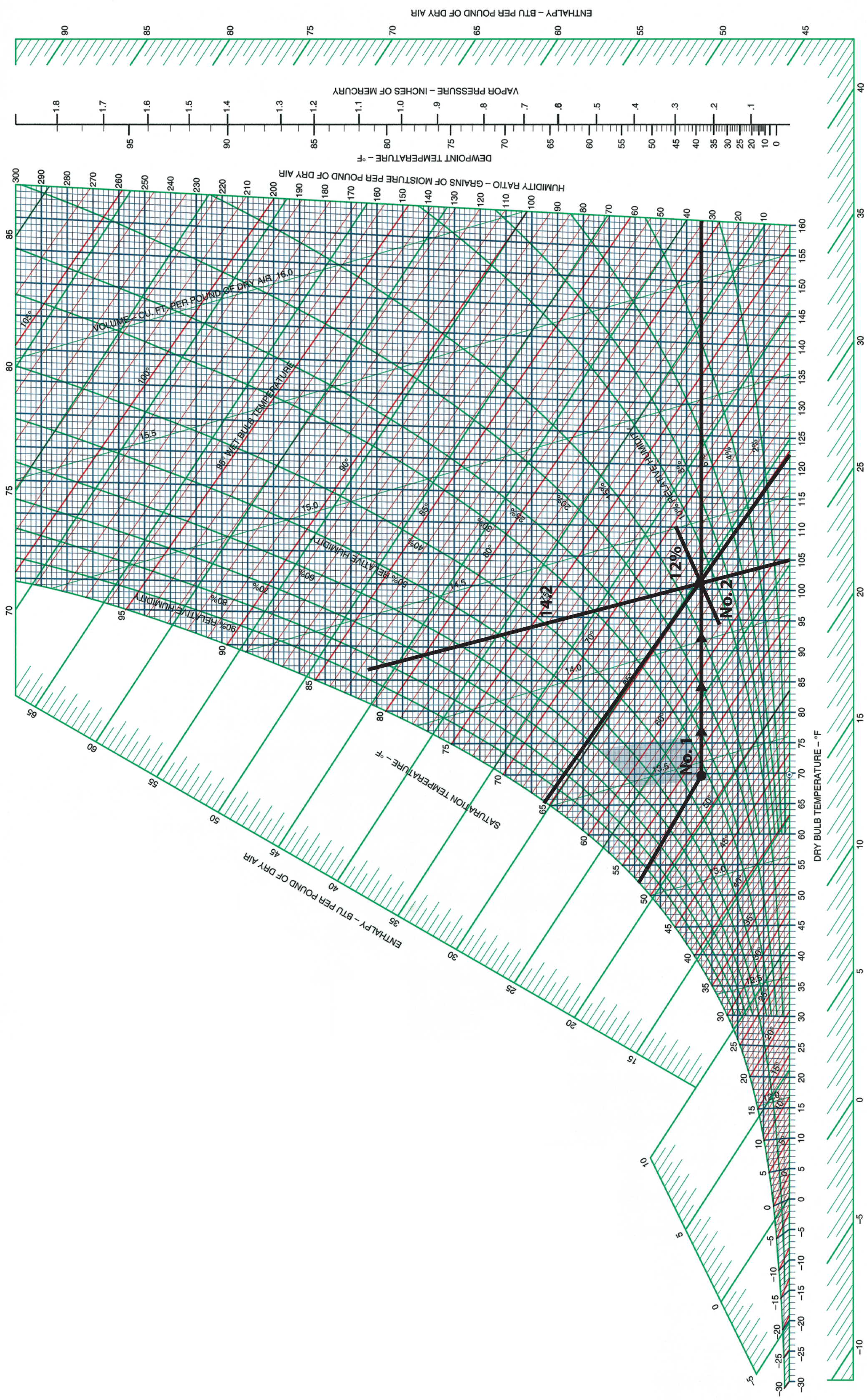

ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
No. 1
22

EXAMPLE #3

Again, from state point 1, determine a second state point that will be the result of adding sufficient heat to bring the moisture to a temperature of 100°F dry bulb, but with no moisture being added. Call this state point number 2, and from it determine the new relative humidity, wet bulb temperature, and enthalpy.

SOLUTION:

1. From state point 1, project to the right on the same .0046 (w) line (no moisture is being added or removed) to the point of intersection with the 100°F dry bulb line. Call this state point number 2.
2. The relative humidity now reads 12%. The same air has dropped 18% in relative humidity. This is because it now is capable of holding more moisture at the higher temperature.
3. Using the same method to determine enthalpy as indicated in example number 2, find 29.2, or a difference of 7.2 BTU/pound. To check this figure, multiply .24 BTU (specific heat of air) by 30 (temperature rise), which equals 7.2 BTU/pound of dry air.
4. Again, following the method shown in example 2, find that the volume of the air has increased to 14.2 cubic feet per pound.
5. The wet bulb temperature, which must be estimated (between the lines), is about 64.2°F.


ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
No. 1
No. 2
12%
142

EXAMPLE #4

HUMIDIFYING WITH UNHEATED ATOMIZED WATER

Assume 6000 cfm of air at a condition of 75°F dry bulb and 20% relative humidity. Moisture is to be added by using unheated water that is atomized into the air stream by the use of fogger nozzles. The process in this case will be 88% efficient, thus producing a final relative humidity of 88%. Note: higher efficiencies are attainable.

Determine the final dry bulb temperature, the pounds of moisture added per pound of dry air, and the amount of moisture added per hour.

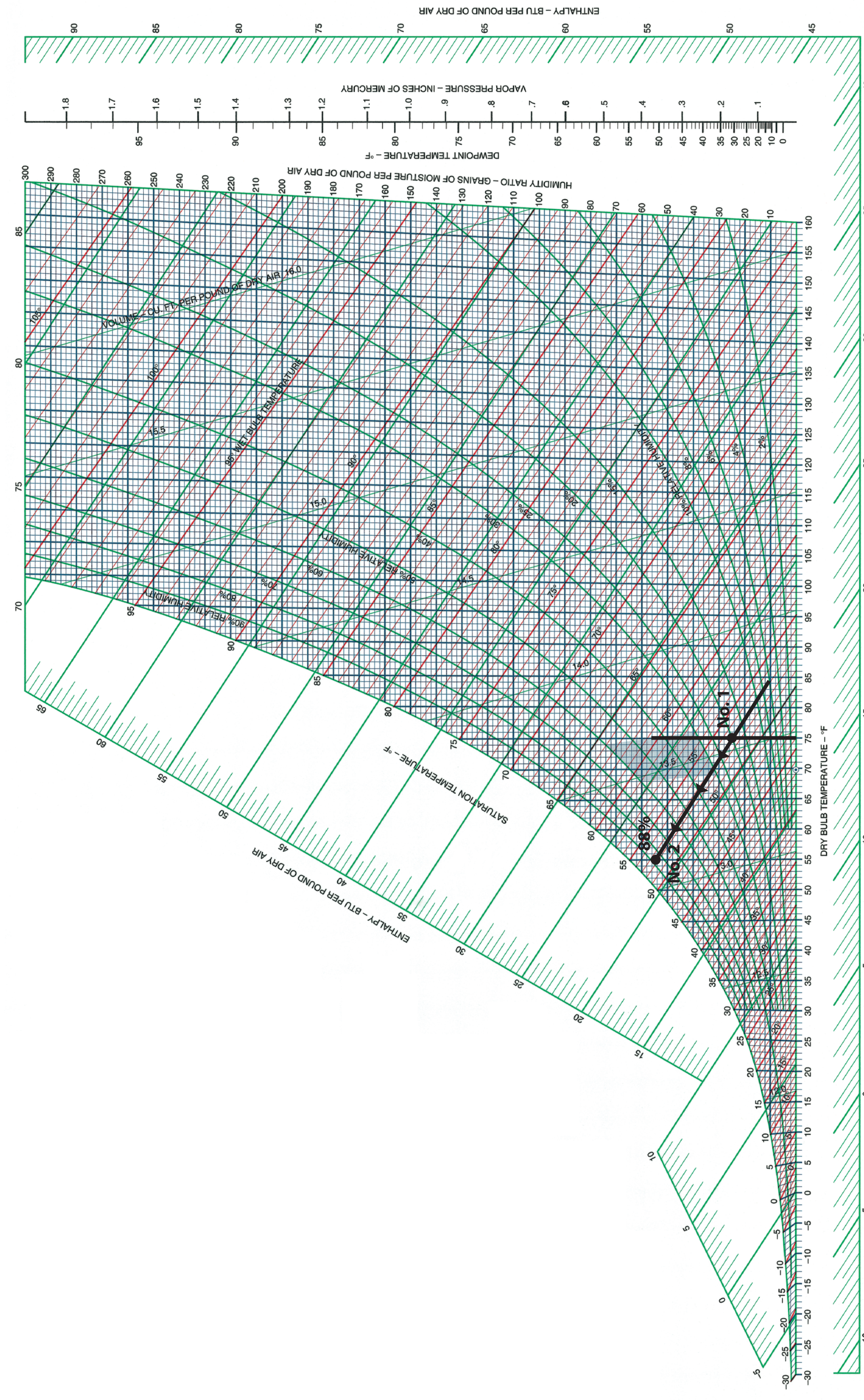
SOLUTION:

1. Locate the beginning state point (Number 1) at the intersection of the 75°F dry bulb line and the 20% relative humidity curve. This point falls on the 53°F wet bulb and the 25 (w) lines.
2. Since no heat is being added, the process will follow the wet bulb line. Locate state point number 2 at the intersection of the 53°F wet bulb and the 88% humidity curve. This point falls on the 55°F dry bulb (final dry bulb) and the 57 (w) lines. The temperature drop of 20°F demonstrates the cooling effect of evaporation.
3. The moisture added is found by subtracting: 57 minus 25 = 32 grains of water added per pound of dry air. Converting: 32 / 7,000 = .0046 lbs. of water per pound of dry air.
4. The volume is estimated at 13.55 cubic feet per pound of dry air. The moisture added is calculated as follows:

   6000 ÷ 13.55 = 443 lbs of dry air

   443 x .0046 = 2.04 lbs of water per minute

   2.04 x 60 min/hr = 122.4 lbs water per hour
   is evaporated by the air stream.


ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
No. 1
No. 2
38%

EXAMPLE #5

HUMIDIFYING WITH HEATED SPRAY WATER

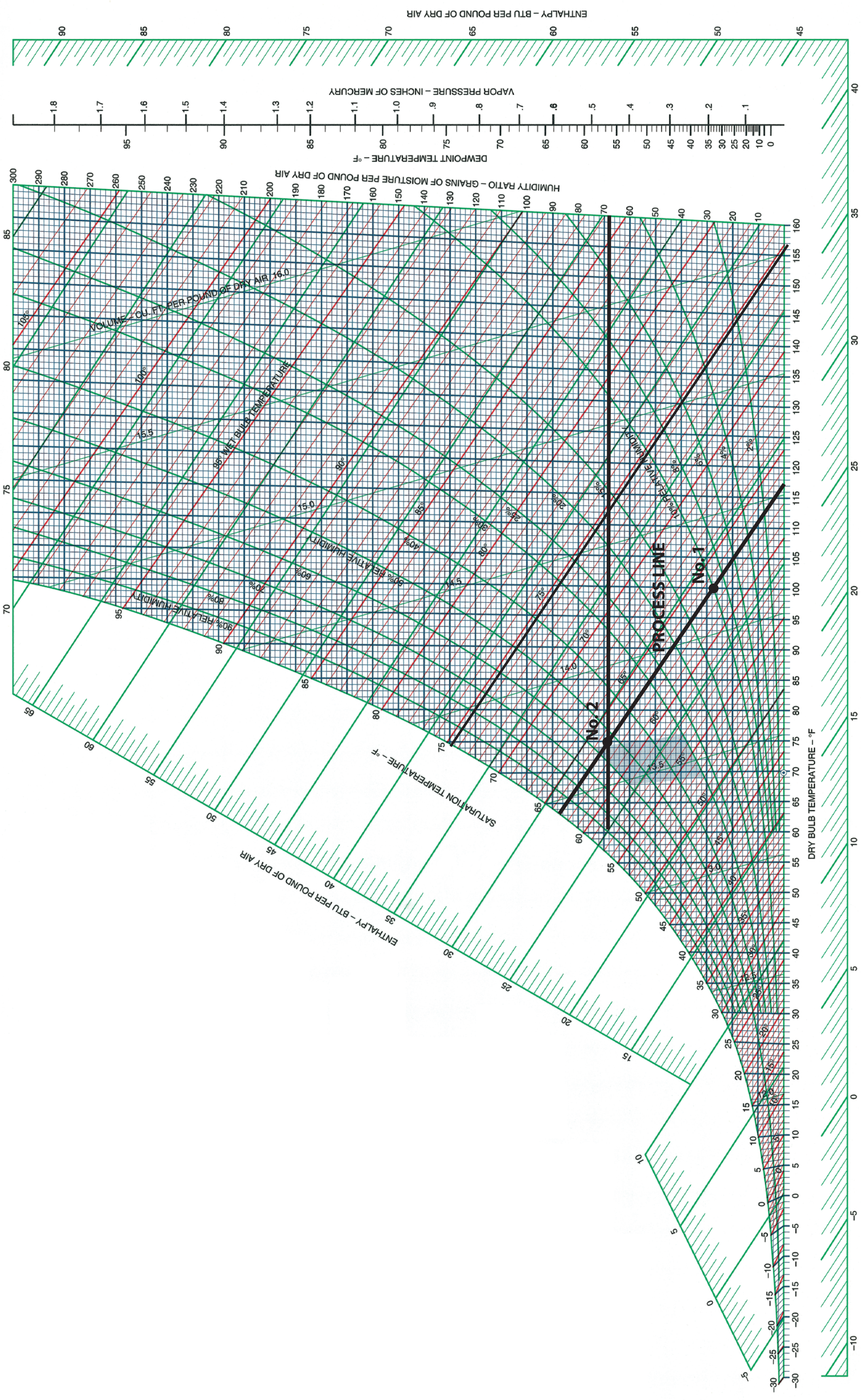
Assume an air stream condition of 100°F dry bulb and 10% relative humidity, which is to be humidified to 50% relative humidity by means of an air washer using recirculated spray water that is heated to 150°F.

Determine the final dry bulb temperature and moisture added per pound of dry air.

SOLUTION:

1. To solve this problem, it is necessary to know the sensible heat content of the 150° water. Refer to Table 163-1 in the Addendum, Thermodynamic Properties of Water at Saturation, page 163. Find 150°F under the column headed Temperature, and read to the right to the triple column headed Enthalpy. Under the hf column, find 117.98 BTU/lb. This is the sensible heat in one pound of 150° water vapor.
2. Locate the beginning state point (Number 1) at the intersection of the 100°F dry bulb line and the 10% relative humidity curve. This point falls on the 28 (w) line.
3. On the protractor, draw a line through the center point and 117.98, continuing the line onto the chart.
4. Now draw a second line (which is called a process line) parallel to the first, and passing through state point Number 1. The point of intersection of the process line with the 50% relative humidity curve establishes state point Number 2, which falls on the 76°F dry bulb (final temperature) and 67 (w) lines.
5. The moisture added is found by subtracting: 67 minus 28 = 39 lbs. per pound of dry air. Converting: 39 / 7,000 = .0056 lbs. of water per pound of dry air.

It should be noted that despite the fact that heated spray water was used, the air temperature still dropped 24°F.


ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
PROCESS LINE
No. 1
No. 2
DRY BULB TEMPERATURE – °F

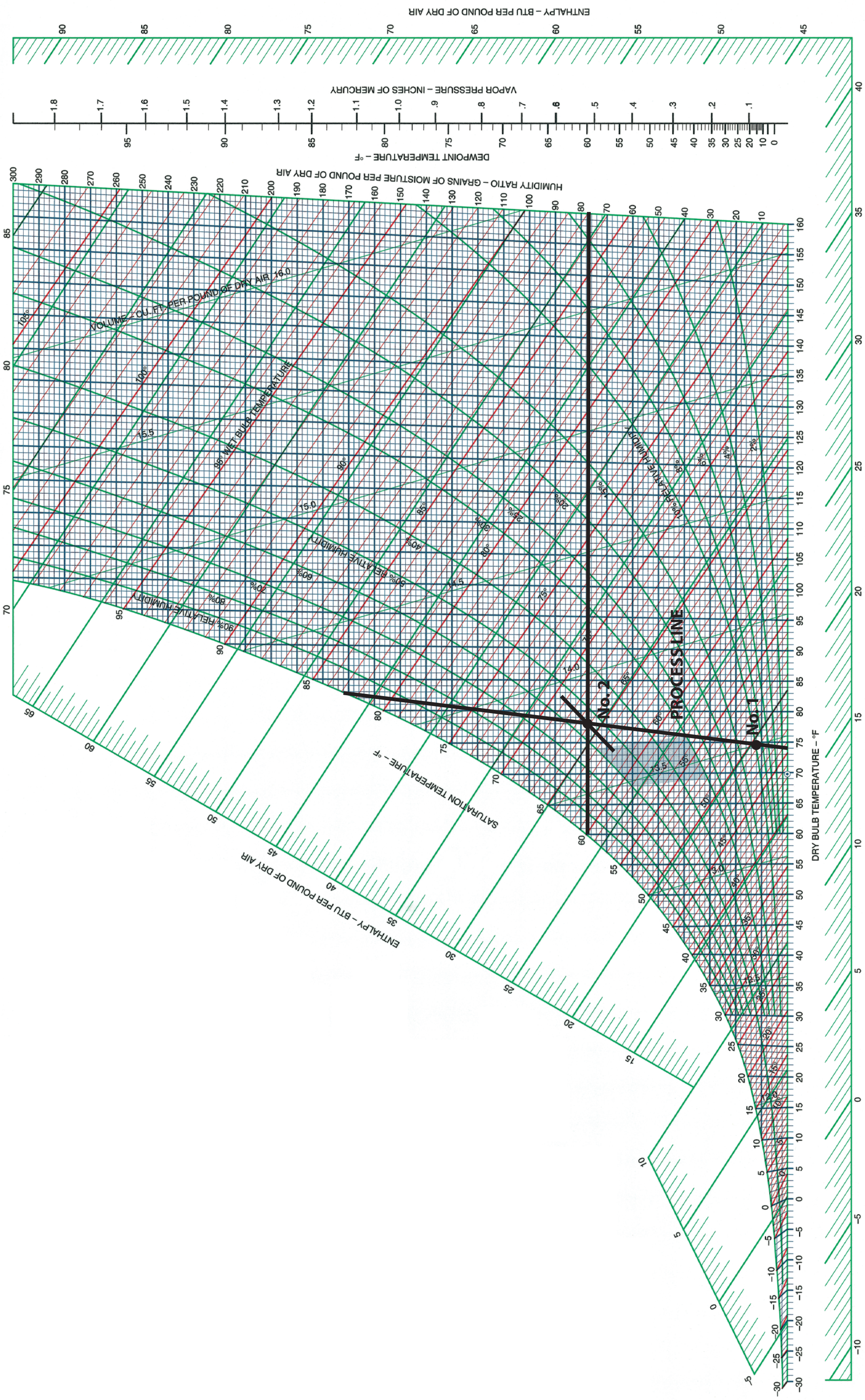
EXAMPLE #6

HUMIDIFYING WITH STEAM

Assume 6000 cfm of air at a condition of 75°F dry bulb and 10% relative humidity. Steam at 10 psi gauge (or 24.7 psi absolute, since absolute = gauge + 14.7) pressure is to be added to produce a final relative humidity of 55%. Determine the temperature rise of the air stream due to the heat of the steam and the quantity of steam to be added in pounds per hour.

SOLUTION:

1. Again refer to the Addendum section, Table 165-1, Thermodynamic Properties of Water at Saturation, page 165. Under the column headed Absolute Pressure, read down to the psi figure closest to 24.7 (24.9869). Then read to the right to the Enthalpy column, and find 1160.48 BTU under saturated vapor. This is the total (latent plus sensible) heat of the steam.
2. Locate the beginning state point (Number1) at the intersection of the 75°F dry bulb line and the 10% relative humidity curve.
3. On the protractor, draw a line from the center through 1160, extending it into the chart.
4. Now draw a second (process) line parallel to the first line, which passes through state point Number 1. This process line intersects the 55% relative humidity curve at the 77.6°F dry bulb line. This, the final condition, is called state point Number 2. Therefore, the temperature rise is 77.6°F - 75°F = 2.6°F.
5. State point No. 2 falls on the 77 (w) line
   State point Number 1 falls on the 13 (w) line
   Converting: 64 / 7,000 = .0091 lbs. of water per pound of dry air.
   6000 ÷ 13.5 (volume) = 444 lbs. dry air
   444 × .0091 × 60 (min/hr) = 242 lbs. steam per hour.


ENTHALPY – BTU PER POUND OF DRY AIR
VAPOR PRESSURE – INCHES OF MERCURY
DEWPOINT TEMPERATURE – °F
HUMIDITY RATIO – GRAINS OF MOISTURE PER POUND OF DRY AIR
VOLUME – CU. FT. PER POUND OF DRY AIR
WET BULB TEMPERATURE
RELATIVE HUMIDITY
SATURATION TEMPERATURE – °F
DRY BULB TEMPERATURE – °F
PROCESS LINE
No. 2
No. 1

Section XIV: Types of Humidification Devices

ADIABATIC AND ISOTHERMAL PROCESSES OF HUMIDIFICATION

Humidification systems fall into one of two distinct process categories, adiabatic and isothermal.

In the creation of water vapor, approximately 1000 BTU's must be absorbed by one pound of water to convert it to one pound of water vapor. This is true regardless of whether it is water in a reservoir, water on the surface of a bank of media, or water in the form of visible mist floating in the air. It is called the latent heat of vaporization.

In the **adiabatic** process, all of the heat for evaporation is taken from the air itself. On the psychrometric chart, this process follows the wet bulb line. (See page 96. Psychrometric Chart example No. 4.) As the process moves up and down the wet bulb line, the RH and the air temperature are both changing, but the total heat content remains unchanged. Stated another way, the air temperature is lowered and the RH is increased by a simple transfer of heat from the air to the water as it evaporates.

This phenomenon of nature is used to reduce energy costs by augmenting mechanical refrigeration in warm, arid climates. For example, in the southwest United States. some buildings are cooled and humidified simultaneously by the use of this principle. It can also be utilized in cold climates where humidification is needed in buildings having surplus heat due to a high internal heat gain.

Isothermal means occurring at constant temperature. This means that the air temperature remains unchanged by humidification. In the strictest sense, this is not true of the humidification systems described below.

In the isothermal process, the 1000 BTU's per pound for evaporating the water is supplied from a source other than the heat of the air. Typically, the water is boiled and the resulting steam is added to the air.

Rather than cooling the air, as in the adiabatic process, the isothermal process can warm the air by up to several degrees. This is due to the temperature difference between the air at room temperature and the temperature of the steam.

Isothermal humidification systems are by far, more popular than the adiabatic type. This is because the majority of the humidification applications occur in small and medium size buildings. The isothermal type offers several advantages in those size ranges.

1. SIMPLER TO CONTROL

A large adiabatic system often requires air preheat control, water output rate modulation, and air reheat control. An isothermal system usually needs only a rate-of-output control.

2. SMALLER PHYSICAL SIZE

The isothermal type, which operates at a much higher temperature, requires a much reduced heating surface in order to accomplish evaporation. Adiabatic types using extended surfaces for evaporation (wetted media) operate at much lower temperatures and therefore need more surface area.

Steam (isothermal) is absorbed more readily by the air than atomized water (adiabatic). This means a shorter distance of straight, unimpeded air duct is needed to complete absorption of the water vapor.

3. ECONOMY

Isothermal systems can be applied more cost effectively in small and medium size buildings.

ADIABATIC PROCESS HUMIDIFICATION SYSTEMS

The following are some examples of adiabatic process humidification systems:

1. Air washer/evaporative cooler
2. Wetted media
3. Pressurized water atomizing
4. Compressed air fogger
5. Electric oscillation atomizing

AIR WASHERS/EVAPORATIVE COOLERS

In comparison to other humidifying methods, the air washer is relatively expensive. The use of this equipment is generally limited to large central air handling systems within institutional, commercial, and industrial buildings, such as those found in the textile, paper, woodworking, and printing industries. The air washer is maintenance-intensive, and servicing is usually performed by only trained personnel.

The humidification capability of this equipment is generally a by-product rather than a primary function. Air cleaning and cooling are most often the primary objectives in the decision to use this method.

There are two basic types of air washers, the spray type and the rigid media cell type.

The spray type air washer consists of a chamber, or section of duct casing, through which the air passes. It contains a grid of spray nozzles and a waterproof tank, or sump, that forms the floor of the unit. The sump is replenished automatically to make up for water lost by evaporation.

The spray nozzle grid forms a curtain of falling water drops through which the air flows. A pump recirculates the water from the sump to the nozzles. The air-entering side of the washer usually has a baffle system to promote uniformity of air flow. The air-leaving side contains an eliminator section upon which unevaporated water drops and foreign material impinge and are removed from the air.

The rigid media type of air washer is similar in design, except that a bank of wetted cells, through which the air passes, is substituted for the curtain of water drops. The fibrous fill or cell media is made of various materials, such as glass, metal, or natural fiber. Water, which is introduced at the top of the bank, cascades down over the cells to keep them wet and is recirculated by a pump in a continuous cycle.

FIGURE 105-1: AIR WASHER

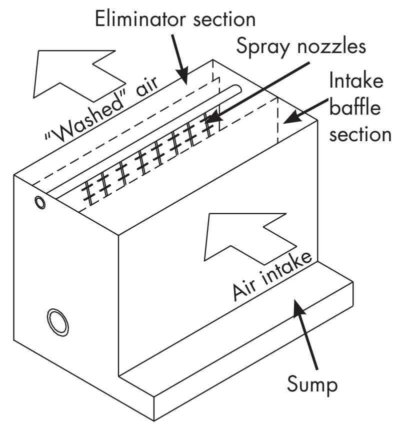
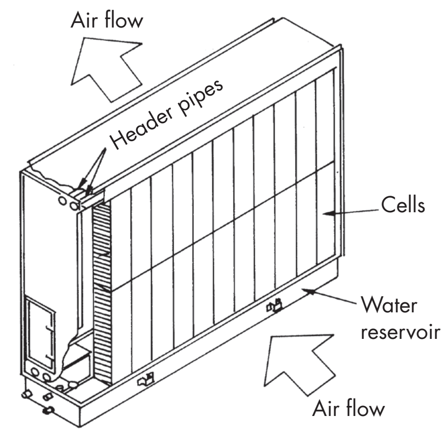


FIGURE 105-2: RIGID MEDIA AIR WASHER



APPLICATION CONSIDERATIONS

a) When used in sub-freezing climates, freeze-up protection should be considered.

b) Well-qualified maintenance personnel are required.

c) Water bleed-off rates are usually quite high; therefore, water costs and availability are a consideration.

d) Accurate control is achievable with the use of a high-integrity control system.

e) Low energy cost humidification is possible, especially when surplus internal building heat is available.

f) Possible algae and microbial growth can be minimized by chlorinization; however, chlorine in the air may not be permissible.

WETTED MEDIA (LARGE RESIDENTIAL)

This is a much smaller unit than the wetted media air washer previously described. This design utilizes a bank of media composed of fibrous glass, metal or natural fiber. The bank is mounted in the air stream.. Water to be evaporated is introduced at the top and cascades down over the bank, keeping the media saturated. Air from an auxiliary source (furnace or air conditioner) or a self-contained fan is forced through the media, causing evaporation. Some variations use heated water in comparatively large volumes to boost capacity and flush mineral deposits to a waste drain. The relatively low output, per unit of space required, limits this equipment to smaller projects.

APPLICATION CONSIDERATIONS

a) Capacity range accommodates large residential and small commercial buildings.

b) Cost of heating water (when used).

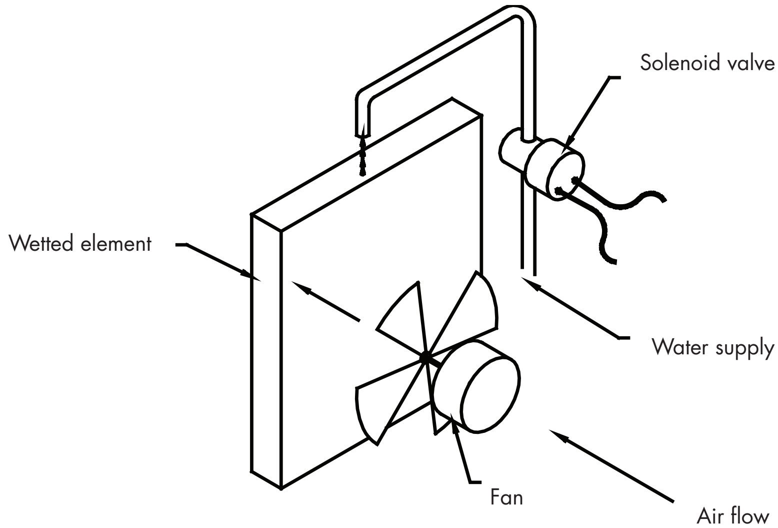
c) Cost of wasting water (when applicable).

d) Uncomplicated, easy to service; however, media may need replacement periodically.

e) Possibility of microbial growth.

f) Does not produce mineral dust.

FIGURE 106-1: WETTED MEDIA HUMIDIFIER



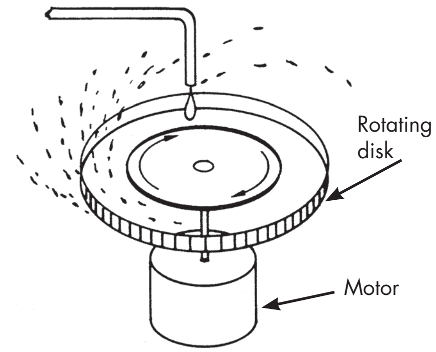
ATOMIZERS AND FOGGERS

The distinction between atomizers and foggers is due to the size of the water particles produced by each.

The atomizer group includes mechanical pressure atomizing nozzles, rotating disks, and rotating cones. These devices are limited in their ability to produce a high percentage of fine particles. Consequently, more fallout of water occurs and longer distances of air/mist travel and/or higher air temperatures are needed to produce satisfactory evaporation results. On the other hand, they are less expensive than foggers.

Foggers utilize either compressed air or electrically driven high-frequency vibrations to produce a fog containing a high percentage of very fine particles that evaporate quickly.

FIGURE 107-1: CENTRIFUGAL ATOMIZING HUMIDIFIER



ATOMIZING - ROTATING DISK OR CONE

Two types are manufactured. One type, small and inexpensive, is used in residential applications. The other type has higher capacity and is ruggedly designed for heavy duty, and long-life in certain industrial applications.

The smaller, inexpensive versions are primarily marketed for residential or small office room type applications, where the device is simply placed in the room and manually refilled. To prevent infection of the user, due to mold and bacteria growth, they should be cleaned and disinfected frequently.

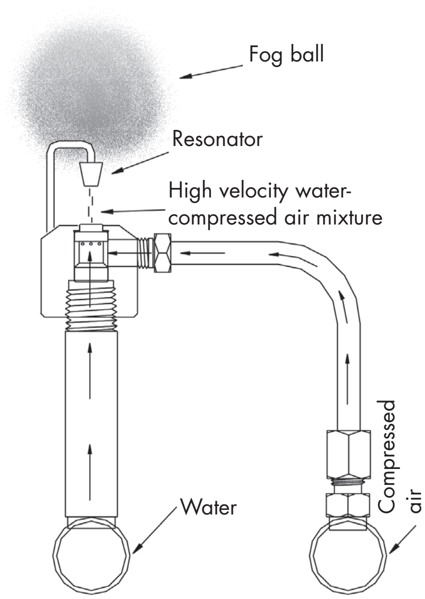
The large-scale industrial models are designed for non-duct application in small-capacity commercial installations where operation is automatic. These types are usually mounted on columns or suspended in space in large, open manufacturing areas such as in paper, textile, and wood processing plants, where their operation is automatic.

In both types the principle of operation involves a disk or cone that rotates at high speed, onto which water is slowly introduced and spun off, becoming atomized. The fine particles of water thus formed are then, in some designs, blown by an integral fan, into the space to be humidified or into a duct, where they evaporate.

ATOMIZING - MECHANICAL PRESSURE

This method utilizes spray nozzles arranged in a bank inside a duct or air handler. The water is supplied by a booster pump. Some systems use pressures up to several hundred psi. Higher pressures result in smaller sized mist particles.

FIGURE 108-1: COMPRESSED AIR FOGGER



APPLICATION CONSIDERATIONS

a) Dissolved minerals will form and precipitate as dust in the air unless previously removed from the water.

b) When discharging mist into an open space, care must be taken to avoid contact with solid objects which would become wet.

c) Rate of evaporation is dependent on droplet size, air temperature and relative humidity of the air. The lower the air temperature and the higher the RH, the lower the rate of evaporation and the greater the risk of moisture fallout.

d) Where water fallout could cause damage, drip pans should be considered. The tendency for fouling is high, so equipment should be located for convenient service access.

e) If considered for duct applications, provision should be made for eliminating entrained mist and draining of water accumulations in the duct.

f) Possibility of microbial growth in duct applications should be considered.

FOGGER

This type of atomizer, which uses compressed air to accomplish atomization, creates much smaller water particles than the atomizers previously described. The resulting more rapid evaporation means less fallout and a shorter absorption distance.

Generally, the better the design and the higher the degree of precision in manufacturing, the better will be the performance of the fogger head in terms of the fineness of the fog particles.

In the better performing designs of fogger heads, a high-velocity stream of compressed air passes at right angles to a small stream of water. The water is sheared into very fine droplets resembling fog. The resulting mixture of compressed air and mist is then driven against a conical block called a resonator thus forming a “fog ball” in the duct air stream, where it is carried downstream and evaporates. The small size of the mist particles thus created and the mixing action with the duct air stream enables the evaporating performance of the fogger to be superior to that of other atomizing types.

Typically, the fogger heads are mounted in the air handler and arranged in a grid pattern downstream of the pre-heat coil and upstream of the cooling coil. Sufficient distance for evaporation should be allowed between the foggers and in-duct objects downstream. Also, drain pans are recommended to remove the water resulting from fallout.

The use of this equipment is generally limited to large air handling systems used by industries, such as semi-conductor, textile, and printing. (See Section XVIII for fogger system application information).

A compressed air supply and a water supply are required, along with motorized valves and a programming system for control of the system.

Periodic maintenance must be performed by competent technicians in order to keep the system in good operation.

APPLICATION CONSIDERATIONS

a) A high volume of compressed air is a system requirement.
b) These foggers may produce a high-pitched whistle which must be taken into account in application.
c) Installations require ample space and access for maintenance personnel to adjust and maintain the system.
d) Use of non-demineralized water may foul the orifices.
e) When using non-demineralized water, a filter may be required downstream of the foggers, to trap the mineral dust.
f) System characteristics, such as entering air temperature and final duct RH, affect evaporation performance. The higher the duct air temperature, and/or the lower the final RH, the shorter will be the fog travel distance before it becomes evaporated.

FOGGER - ELECTRONIC OSCILLATION

This design sometimes called ultra-sonic, utilizes an atomizing method wherein a piezoelectric transformer causes water particles to oscillate

at high frequency, which breaks the water into a fine mist that readily evaporates.

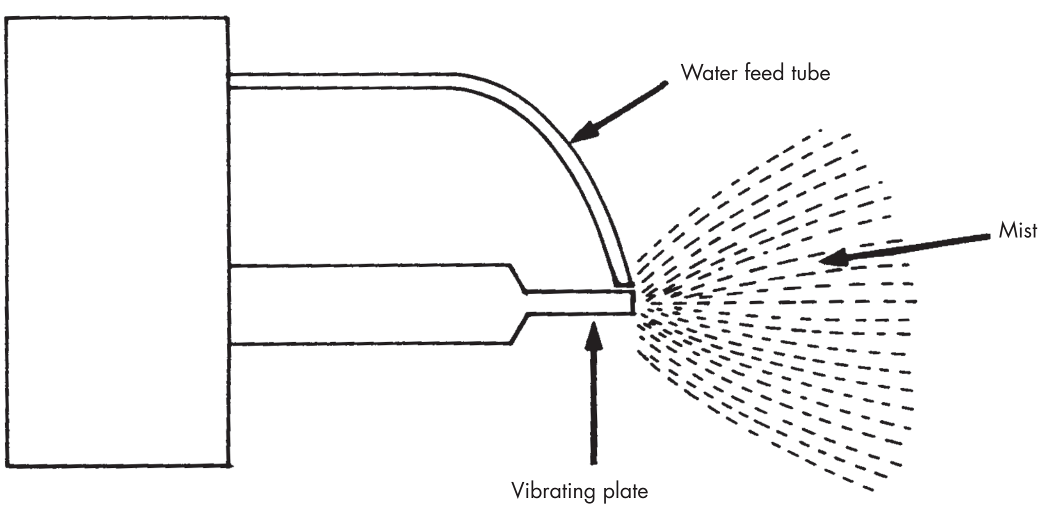
The output capacity compared to the cost of this equipment is quite low, so its use is limited to special- need applications. Some of these needs could be:

1. Low noise level
2. Compact size
3. Low energy consumption (when compared to compressed air foggers)

APPLICATION CONSIDERATIONS

a) Demineralized water must be used.
b) An effective moisture eliminator may be necessary to prevent mist from being carried too far into the duct system.
c) Produces finer mist particles than rotating disk or cone atomizing methods, which results in more rapid evaporation.
d) The small room types should be disinfected regularly.

FIGURE 109-1: ULTRASONIC HUMIDIFIER



ISOTHERMAL HUMIDIFICATION SYSTEMS

This group includes the following:

- Steam Boiler supplies pressurized steam to duct steam dispersers
- Steam Heated Secondary Steam Boiler supplying pressurized steam to duct steam dispersers.
- Heated Vapor Generator supplies non-pressurized vapor to ducts or open spaces and includes the following types:
    - Electric Hot Element
    - Steam Heated
    - Hot Water Heated
    - Gas Burner Heated
    - Replaceable Plastic Evaporator
    - Electric Infrared Pan

STEAM BOILER

This is perhaps the oldest method of providing non-residential humidification.

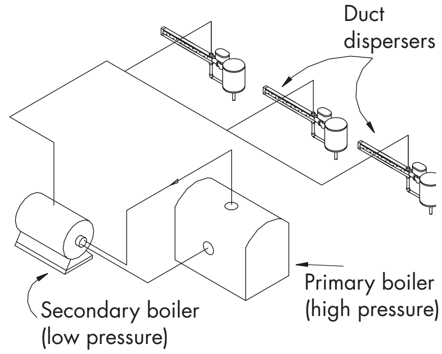
When steam-jacketed dispersers or steam dispersion panels (see pages 139 and 140) are properly applied for introducing the boiler steam to the air in ducts, good absorption of the steam by the air and accurate control of the RH can be expected.

Because of the health-threatening risks associated with boiler steam containing boiler treatment chemicals (see page 34), this method of humidification is declining in popularity especially in health care facilities and other high-occupancy buildings.

APPLICATION CONSIDERATIONS

1. Humidification may be required at times when the boiler may not be needed for heating (spring and fall).
2. No mineral dust.
3. Very little maintenance and when required is confined to one location, i.e. the boiler.
4. Accurate humidity control is achievable.
5. Boiler water treatment chemicals may be objectionable.
6. Duct disperser noise may be objectionable.

FIGURE 110-1: STEAM HEATED SECONDARY STEAM BOILER



Other methods, that eliminate the water treatment chemicals problem and retain the low energy cost advantage of boiler steam are described on the next page.

STEAM HEATED SECONDARY STEAM BOILER

In this system the humidification steam is generated in an unfired pressure vessel similar to a scotch marine boiler. Instead of a fuel burner, a tube bundle heat exchanger, supplied with steam from a high pressure boiler transfers heat to the secondary boiler water and creates low-pressure steam. Because of the relatively high equipment cost, this system is generally used only for large, sophisticated humidification systems, such as those in pharmaceutical manufacturing clean rooms, hospitals, and similar applications requiring large amounts of high- purity steam and close tolerance RH control.

Demineralized water is used for make-up because it offers the dual advantage of providing the high- purity steam required in these manufacturing processes and eliminating mineral buildup on the heating surfaces of the tube bundle.

Medium-to-high pressure steam is supplied to the tube bundle, creating low pressure (15 psi or less) secondary steam in the vessel. The low-pressure steam is then distributed throughout the building via a stainless steel piping system, to steam-jacketed steam dispersers or steam dispersion panels located in the duct air stream. To protect against the corrosive effects of demineralized water, all components in contact with the water and steam (secondary boiler, valves, piping, steam traps, and steam dispersers) are typically constructed of series 300 alloy stainless steel.

APPLICATION CONSIDERATIONS

1. First cost is quite high.
2. Long service life and low maintenance.
3. Accurate humidity control is achievable.
4. Steam is free of contaminants.
5. Duct disperser noise may require extra care.

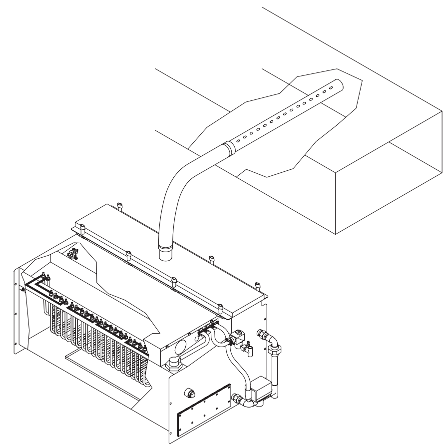
HEATED VAPOR GENERATORS - GENERAL

This equipment consists of a small non-pressurized automatic boiler/water reservoir containing the heating device, which serves as the evaporator. The heating device can be an electric resistance (hot element) heater, electrodes for passing current through the water, a steam heated tube bundle, a hot water heated tube bundle, or a gas burner and firing tubes.

These devices are designed and constructed to produce water vapor at sufficient pressure to overcome the resistance of the vapor flow through the tubing conveying the vapor to the duct- mounted vapor disperser and the flow resistance through the disperser itself. This combined resistance is normally less than 1/2 psi (12 inches water column).

Such being the case, the generators are installed at or very near the point of use, and rigid thin-wall metal tubing or steel pipe (preferably insulated) is used to carry the vapor to the duct mounted disperser. In small capacity, low-budget applications, steam hose is sometimes used instead of metal tubing as the steam conductor.

FIGURE 111-1: ELECTRIC HEATED VAPOR GENERATOR



Special versions for non-duct applications are available with a fan mounted on the vapor generator itself, to disperse the water vapor directly into the surrounding space.

These vapor generators are fully automatic. They have an electronic probe control system that monitors the water level, replenishing it as required, and providing low-water protection. The humidity output controls that are available range from simple on/off control to sophisticated, fully programmable microprocessor types that modulate the vapor output to provide very accurate RH control.

When the hardness of the water for make-up exceeds 10 grains per gallon, ion exchange water softening is recommended. The use of softened water, significantly reduces mineral precipitation in the generator, allowing up to several seasons of service without cleaning.

A bleed-off with overfill function is usually incorporated. This serves to continually dilute the water in the reservoir to minimize fouling caused by precipitation of the mineral content of the water. Each time the make-up valve opens, a slight overfill occurs, which immediately bleeds off. To accommodate unsoftened water, some models incorporate a timer and a drain valve, which periodically drains the reservoir and refills it with fresh water.

FIGURE 112-1: MULTIPLE VAPOR GENERATORS

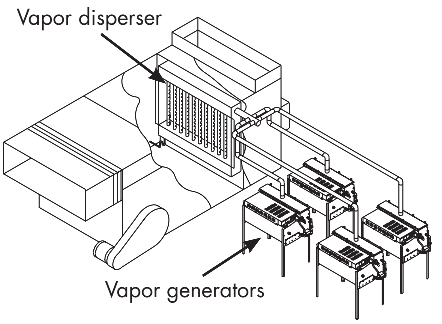


FIGURE 112-2: VAPOR GENERATOR IN AIR HANDLER

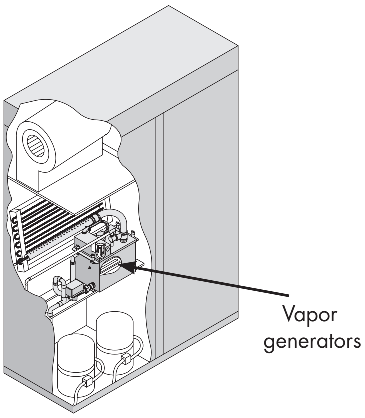
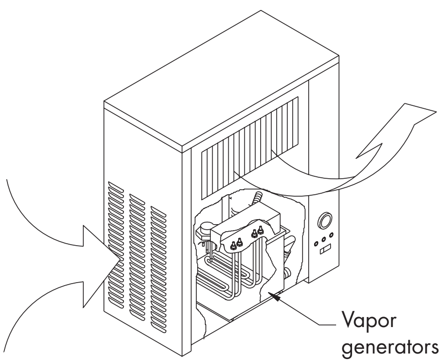


FIGURE 112-3: FAN CABINET UNIT



When zero maintenance (no cleaning) and/or vapor with zero contaminants is necessary, demineralized (DI or RO) make-up water is used. In this version, a float- operated make-up valve and a low water cut-off are substituted for the electronic probe control system, which requires water conductivity for function.

ELECTRIC HOT ELEMENT VAPOR GENERATORS

This version of the previously described vapor generator is perhaps the most widely used simply because there are so many applications for which it answers the need.

These vapor generators usually are in small to medium size buildings, because their small physical size, relatively simple maintenance, and relatively low installed cost provide the best answer, in spite of the comparatively high cost of electrical energy.

These vapor generators are available in a capacity range up to 300 pounds of water vapor output per hour. When larger loads are encountered, multiple units can be ganged together and operated by a single controller.

Small versions of these units are used extensively in computer-room applications, where they are installed within packaged computer room air conditioners as shown in Figure 112-2.

Another version, designed for installation in finished spaces, has a vapor generator mounted in a metal fan cabinet having a circulating fan that draws in room air, humidifies it, and discharges it back into the room.

APPLICATION CONSIDERATIONS

1. Versatile, small physical size, easy to locate.
2. Electricity is the most expensive heat energy source.
3. Relatively simple to maintain, especially when softened or DI/RO water is used.
4. Accurate humidity control within ± 1% RH is achievable.
5. Low first cost.
6. Quiet in operation.

STEAM-HEATED VAPOR GENERATORS

This version is popular in buildings that have heating boiler steam available but do not wish to use it for humidifying because of the presence of boiler treatment chemicals.

This system is usually considerably less expensive than the steam- heated secondary boiler system previously described.

When zero-maintenance or high- purity steam are needed, demineralized make-up water is used.

Steam-heated vapor generators are available in capacities up to 1600 pounds of vapor per hour.

APPLICATION CONSIDERATIONS

Same as electric hot element systems, except that fossil fuel generated steam is a considerably less expensive heat energy source than electricity.

HOT WATER HEATED VAPOR GENERATORS

This version is designed for applications where heating system high temperature hot water (HTHW) (above 240°F) is available. Some college campuses, certain manufacturing facilities, and some district heating municipalities utilize HTHW. Again, energy cost saving is a significant advantage with this method. The units are available in capacities over 500 lbs per hour.

APPLICATION CONSIDERATIONS

1. Same as steam heated.

GAS BURNER HEATED VAPOR GENERATOR

This system, because of its comparatively high overall efficiency and its low-cost fuel, offers significant operating cost savings over other energy sources. This is especially true in larger systems.

In this vapor generator design, depending on capacity, one or up to several firing tubes, each containing an infra-red gas burner provides the heat for evaporation. The burners are of forced-draft design and burn either natural or liquefied petroleum (LP) gas.

The output capacity ranges upward from 75 to several hundred pounds of water vapor output per hour.

APPLICATION CONSIDERATIONS

1. Same as steam-heated, except direct gas firing is usually the least expensive energy source.
2. May be somewhat more difficult to locate within a building because of the need for a gas venting flue.

FIGURE 113-1: STEAM HEATED VAPOR GENERATOR

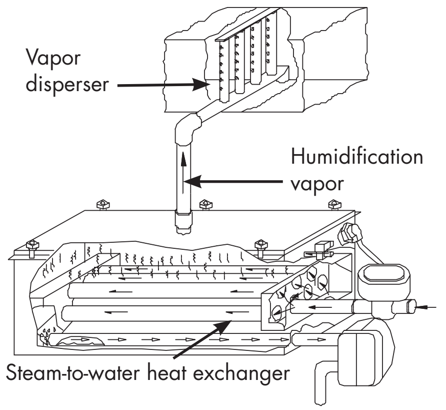


FIGURE 113-2: HIGH TEMPERATURE HOT WATER HEATED VAPOR GENERATOR

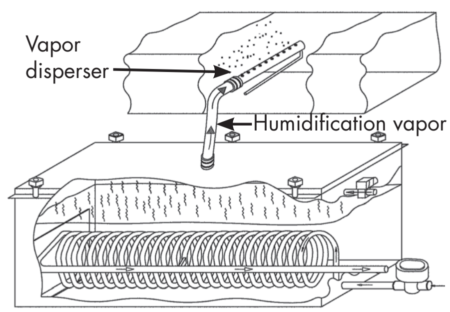
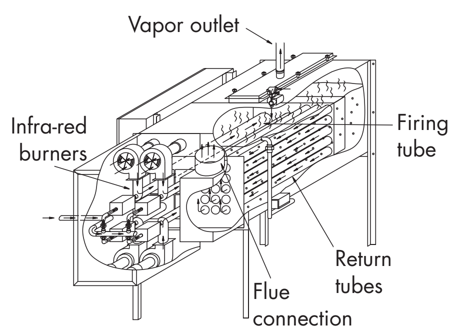


FIGURE 113-3: GAS FIRED VAPOR GENERATOR



ELECTRONIC DISPOSABLE VAPOR GENERATOR

This equipment covers a broad capacity range and is used in commercial and institutional type buildings and in industrial processes. It consists of one or more plastic, bottle-shaped containers called generators, mounted in a metal or plastic cabinet. Each generator maintains a water level by means of a solenoid-operated fill valve and contains internal electrodes. Electric current is passed through the water, via the electrodes, causing it to boil. The resulting vapor is conducted by means of a steam hose from the generator to the air stream where it is carried to the humidified space.

Some ductless versions are provided with a fan unit for dispersing the vapor directly into the space.

Mineral residue collects in the generators eventually rendering them unusable, at which time they are replaced. An automatic drain system is employed to dilute the concentration of the mineral residue, thereby extending the service life of the generator.

The principle of operation depends on water conductivity. Demineralized water, being electrically non-conductive, cannot be used with this system.

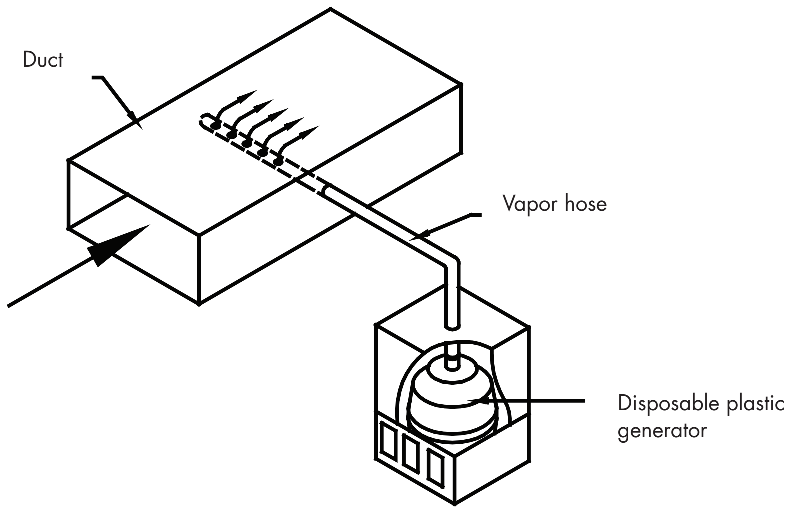
Softened water also does not benefit this equipment because as the concentration of sodium increases, conductivity also increases, which causes electric arcing inside the generator. (See page 120.)

For improved controllability, most versions are available with a form of variable output control.

APPLICATION CONSIDERATIONS

1. Uses most expensive heat energy source.
2. Ongoing cost of replacement generators makes operating cost high.
3. Small physical size means it is easily located.
4. Does not provide highly accurate humidity control.
5. Reduced vapor output when cylinders are nearing need for replacement due to mineral build up.

FIGURE 114-1: VAPOR GENERATOR - DISPOSABLE EVAPORATOR TYPE



INFRA-RED PAN EVAPORATOR

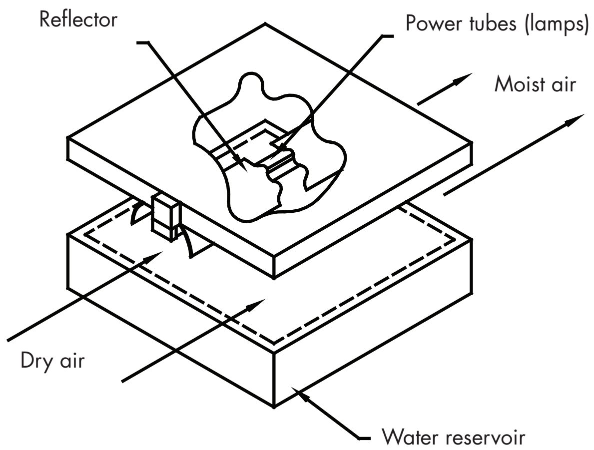
This equipment consists of a pan with the water level maintained by an automatic fill valve. Infra-red lamps, hooded by reflectors, radiate heat downward to the water causing it to evaporate. The vapor is then carried away by the air stream.

These systems are sometimes installed within packaged computer room air conditioners. When not frequently cleaned, a significant loss in operating efficiency occurs. This is due to the formation of mineral scum on the surface of the water in the pan. The scum causes much of the radiant energy to be converted to heat that is carried away by the airstream rather than causing the water to evaporate, as intended. This additional unwanted heat is then added to the cooling load that must be removed by the air conditioner, causing further inefficiency.

APPLICATION CONSIDERATIONS

1. Expensive to operate due to low efficiency and high cost energy.
2. Small physical size.
3. Output and efficiency fall off rapidly as precipitated minerals begin to form.
4. Fairly maintenance intensive.
5. Poor humidity control accuracy.
6. Limited output capacity.

FIGURE 115-1: INFRA-RED EVAPORATOR



Section XV: Water Treatment and Humidifiers

WATER QUALITY AND ITS EFFECTS ON HUMIDIFICATION SYSTEMS

The purity of the water that is supplied to humidification systems, has a major effect on the performance of those systems.

For example, vapor generators create water vapor by evaporating water in a chamber heated by electric heating elements, gas firing, or a heat exchanger heated by steam or hot water. As the vapor is driven off by heating, unless steps are taken to prevent it from happening, the water impurities precipitate and collect in the evaporating chamber. Some type of bleed-off, manual, automatic, or a combination of both is usually employed to minimize the formation of these precipitates, so the system can continue to operate for longer periods between shutdowns for cleaning.

Other types, such as foggers where evaporation occurs in mid-air, create a mineral dust problem. When the mist evaporates, the dust remains and most often it must be removed from the air.

Both of the above troublesome situations can be improved or eliminated by properly treating the water before it enters the humidification system.

In some instances softened or demineralized water is available for the humidification system since it is required for other purposes on the job. When this is not the case and dependable, closely controlled humidification is needed, water treatment should be considered.

WATER IMPURITIES

Water “hardness” is dissolved minerals in water, which are inorganic materials commonly referred to as mineral salts.

Hard water salts are dissolved by the surface water as it flows through rock formations on its way to the underground aquifers that are tapped by wells. The most common mineral ions found in well water are magnesium and calcium. Other mineral ions include iron and silicone. These various mineral ions combine with other common elements, carbon for example, to form mineral salts such as calcium carbonate.

Hardness, by definition, is any mineral that reacts with soap to form a scum or curd. Hard water salts will stay dissolved in water only in weak concentrations. In some cases merely heating (not necessarily boiling) the water will cause these minerals to precipitate (come out of solution) from the water. This explains the scale that accumulates in coffee makers, water heaters, and humidification system evaporators. This is called carbonate, or temporary, hardness.

Non-carbonate, or permanent, hardness is made up of similar mineral ions in the form of sulfate, chloride, or nitrate salts. These forms of salt do not come out of solution nearly as readily and thereby remain dissolved in higher concentrations in heated water.

Most well water has both temporary and permanent hardness. The sum of these two is called the total hardness and is usually expressed in grains per gallon but can also be expressed in parts per million, milligrams per liter, or electrical conductivity.

WATER SOFTENING

Sludge and scale-forming minerals can be removed by ion exchange softening. In this process slightly soluble mineral ions such as calcium, magnesium, and iron are traded for much more soluble sodium ions. This simple ion exchange process dramatically changes the chemistry of water. The newly formed sodium salts do not form soap curds, which means that soaps and detergents are much more effective and less of them are needed.

When softened water is used in them, humidification evaporators are kept much freer of mineral precipitates. By continually supplying a small excess of softened make-up water to the heated evaporator and continually bleeding off an equivalent amount, the concentration of sodium ions can be kept below the point of precipitation, thus greatly reducing the need for cleaning.

REVERSE OSMOSIS (RO)

Another method of water treatment is reverse osmosis. In this process, water is forced at a very high pressure through a semi-permeable membrane. About 95% of the minerals dissolved in the treated water are removed.

Reverse osmosis is also very effective at removing turbidity, dissolved organic material, bacteria, and pyrogens.

DEIONIZATION (DI)

The demineralization of water by the deionization process produces the purest water. In this process, water is passed over a resin bed made of chemically charged material. The resin bed attacks and forms chemical bonds with all dissolved ionized gases and solids. Purity of this type of water is expressed in megohms per centimeter. DI water is extremely hungry, or corrosive, and materials coming in contact with it must be made of high grade stainless steel or special grades of plastic to resist its effects.

In a typical high tech, high capacity water treatment system, three steps are used. First, the water is softened. Second, it is passed through a reverse osmosis process Finally, it is "polished" with a deionization process. Softening is done first because calcium and magnesium ions are more difficult for reverse osmosis machines to remove. Reverse osmosis is performed prior to deionization to extend the useful life of the resin beds between regenerations.

HUMIDIFICATION SYSTEMS THAT BENEFIT BY USING SOFTENED WATER

Generally speaking, most systems employing a reservoir of water that is heated to create water vapor are able to use softened water to advantage.

These systems typically employ a motorized water make-up valve for level control and a bleed-off port. The level control is preadjusted to provide for a slight overfill at each refill and the excess, bleeds off, thus continuously diluting the ion concentration. When naturally soft or softened water is supplied, this system is effective, resulting in up to several seasons of operation without cleaning.

When unsoftened water, harder than 10 grains per gallon, must be used for make-up, a timer operated system to periodically drain and refill the evaporating chamber with new water, is usually recommended. Such a system is, however, not nearly as effective as softened water in supplying a means of minimizing scale formation.

SOME SYSTEMS DO NOT BENEFIT FROM SOFTENED WATER

Wetted media and fogger systems are not benefited because the total dissolved solids remain unchanged by softening. This means that mineral precipitates will continue to build up on the wetted media, and airborne dust will continue to form in the air when softened water is used with these systems.

Electronic or electrode (replaceable plastic jug) vapor generator systems are not helped by water softening either.

Their principle of operation requires electric current to pass through the water to cause boiling. When using unsoftened water, the hardness continually precipitates as scale, building up on the electrodes. In so doing it stabilizes the water conductivity and permits satisfactory operation. With softened water, this precipitation is greatly lessened, but the continually increasing sodium ion content causes the water conductivity to increase to the point where arcing inside the plastic evaporator takes place.

Efforts to reduce this effect by supplying a mixture of softened and raw water to the humidifier typically results in an increased bleed-off of hot water, which means lowered operating efficiency.

HUMDIFICATION SYSTEMS THAT BENEFIT BY USING DEMINERALIZED WATER

In general, the same group of vapor generating systems that benefit from softened water, also benefit from the use of demineralized water. Vapor generators, are benefitted in several ways:

One is cleaning. Eliminating the minerals eliminates the precipitation, so cleaning is seldom, if ever, needed. The high purity of the steam output, while usually not a requirement in any but special types of manufacturing applications, is a virtue of this method as well.

Another benefit is control accuracy. DI water usage eliminates the need for the solenoid type make-up valve that provides the overfill and water bleed-off dilution principle commonly in use with either softened or untreated water.

Instead, a float-operated, modulating type of make-up water valve is used, which allows water to enter the reservoir at the same rate it evaporates. The result is a stable operating temperature with no interruptions of steam output due to cool-downs. This allows the humidifier output to be controlled very accurately, thus allowing it to closely follow changing space conditions.

Again, the electrode type vapor generator does not benefit from demineralized water. In fact, since it operates on the principle of water conductivity, it cannot function using water having no minerals.

All types of atomizers (compressed air foggers, ultrasonic, etc.) benefit by elimination of the mineral dust problem.

With the steam-fired, low-pressure steam generator system described on page 110, using DI/RO make-up water is a must.

All humidifier types using DI water must be constructed of materials that will withstand the corrosive effects of this water.

COSTS OF WATER TREATMENT

The costs associated with water treatment vary, as does the purity of the final product. At this writing, thousands of gallons of softened water can be generated for a dollar. RO treatment water costs about one cent per gallon, and DI water treatment costs about ten cents per gallon.

These costs and the original purchase price of the equipment must be factored into the decision as to what type of humidifier and what type of water treatment process should be considered for any given project.

Generally, degree of accuracy of humidity control, available energy sources, and allowable shutdown time for humidifier maintenance will have a bearing on these decisions as well.

Section XVI: System Design Considerations

TAKING ADVANTAGE OF VAPOR MIGRATION

The design of humidification systems can often be simplified and the installed cost significantly reduced by taking advantage of the principle of water vapor migration.

As mentioned in Section I, water vapor is a gas that behaves under the laws of low-pressure gases. As such, when water vapor is introduced into a room or space, the area in the immediate vicinity of the point(s) of introduction undergoes an increase in vapor pressure. Because of this higher pressure, the water vapor migrates to all parts of the space that are at a lower vapor pressure, whether or not air movement is taking place. The rate of migration, or speed of movement, is proportional to the difference in vapor pressure. However, this migration does not occur rapidly, so if a homogeneous condition is required in a large space, this process alone may not be an adequate method of humidity distribution. The following is an example, based on Figure 122-1, to demonstrate this principal:

An assembly area is served by three air handlers, each supplying 6000 cfm of air at 55°F. The space is to be humidified to 35% RH at 70°F. The total load is calculated to be 54 lbs per hour of water vapor. Since this is an open area with no dividing partitions, water vapor can freely migrate within the space.

In this example, it appears that it would be less expensive to install one large humidifier than three small ones. AC Unit #2 would be the best location for this humidifier, because its air discharge is most centrally located within the space. Before the design can be finalized, it must be determined that the air quantity circulated by AC unit #2 is capable of absorbing the 54 lbs. per hour of water vapor without moisture fallout.

MOISTURE FALLOUT

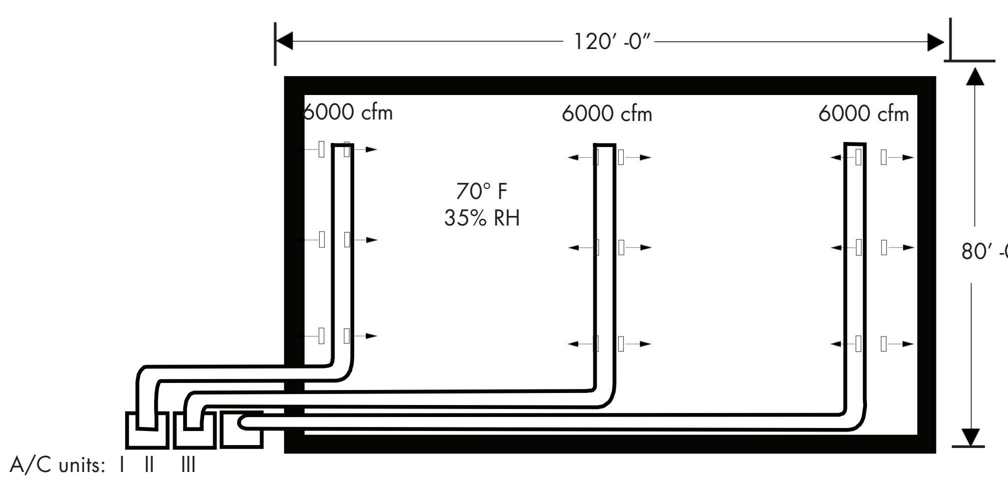
Adding more moisture to an air steam than it can absorb results in downstream condensation inside the air ducts. To eliminate this possibility, the following information must be ascertained:

(A) The moisture-holding capability of the air stream in the duct, at 90% saturation (90% is the recommended upper limit, to avoid moisture fallout in the duct).

(B) The amount of moisture contained in the air stream coming to the humidifier just prior to satisfaction of the humidistat (assuming that "on/off" as opposed to modulating, control is to be used).

(C) The amount of moisture that is being added to the air stream just prior to humidistat satisfaction. (Again, assuming on/off control.)

FIGURE 122-1: VAPOR MIGRATION



EXAMPLE:

For the system to function without moisture fallout, the following conditions must be met: B plus C must not exceed A. Turn to table 81-1, page 81.

SOLUTION:

1. To determine A, read across the 55°F line to the 90% column and find 3.76 lbs/hr/100 cfm
2. To determine B, read across the 70°F line to 35% and find 2.4 lbs/hr/100 cfm.
3. To solve C divide 6000 cfm by 100 to determine the 100s cfm, which in this case is 60. Divide 54 lbs/hr by 60 = .9 lbs/hr/100 cfm.
4. Add B to C (2.4 lbs plus .9 lbs) = 3.3 lbs/hr/100 cfm.
5. A is greater than B plus C, indicating that the air volume of Unit #2 is capable of absorbing all of the moisture required for the entire space.

Note: Had the result been that B + C exceeded A, it would have been necessary to consider one of several alternatives:

1. Add the humidity in two or possibly all three of the air handling unit air streams.
2. Use a secondary, high-limit humidistat set for 90% installed in the duct downstream of the humidifier. (This may result in inadequate humidification during extremely "dry" periods.)
3. Change the control method from on/off to modulating. This would allow the humidifier output to taper downward as the room RH approached the humidistat set-point.

STEAM HUMIDIFIER TEMPERATURE SWITCH

This is an accessory device that is sometimes applied to steam duct dispersers, particularly those types that use a slow-to-warm-up cast iron water-steam separator instead of lightweight stainless steel.

The primary purpose of the switch is to prevent spitting of the humidifier during warm-up. To accomplish this, it delays opening of the humidifier steam valve until the humidifier is heated up to steam temperature. The switch, available in either an electric or pneumatic version, is actuated by its sensor which is usually mounted on the condensate return pipe immediately ahead of the humidifier steam trap. When the sensor detects steam temperature, the humidifier valve is allowed to open.

The switch would also prevent spitting in the event of a flooded steam or return main due to a steam system malfunction.

PRIMARY/SECONDARY VS. SINGLE STAGE HUMIDIFICATION (USING BOILER STEAM DUCT DISPERSERS)

This design technique is commonly applied to hospital surgeries having multiple surgery rooms, served by a single air handler but with individual humidity control required for each surgery room.

This system can be designed in one of two ways. A single stage system would consist of a steam disperser for each surgery room. A primary -secondary system would, in addition to the above, have an additional disperser at the air handler which would maintain a constant RH in the air leaving the air handler.

Surgery rooms represent a difficult humidity control problem. These rooms are usually in the building interior and have no heat loss. As such, they require cooling year around which means the duct air is held at 55° F. Since the room must have a minimum RH of 50% the duct air must be maintained at about 90% RH. This means very little margin for error.

Small capacity steam disperser valves have an inherently low turndown ratio. (See "Modulating Valves and Fluid Flow, page 132).

When a single stage system is used, accurate control of steam flow is difficult and often results in the disperser valve cycling open and closed repeatedly, called hunting, unable to stabilize at an output capacity that equals the demand. In extreme cases, a cloud of steam is periodically seen emerging from the air diffusers in the surgery room.

This is due to the fact that during cold weather the entering outside air contains very little moisture and requires maximum humidification capacity. As the weather moderates the outside air becomes more humid thus reducing the steam needs significantly which leads to the problem. The disperser valve must be capable of covering the full range of the load, which means the valve selected must have the ability to modulate flow, and control accurately, from maximum load down to a near zero load condition. Even very high quality and expensive industrial grade valves may not be able to provide this degree of turndown. (See page 132.)

The primary-secondary method alleviates this condition by minimizing these load swings. In this system the primary steam disperser is set to constantly humidify the supply air to about 45% at 55°F. The zone dispersers now become secondary dispersers, and their valves can be selected to cover a much smaller humidity load range (45% to 90% instead of near zero to 90%). This means that the steam valve is better able to accurately modulate over this a narrower capacity range and the periodic visible steam emissions will not happen.

STEAM PRESSURE FOR BOILER STEAM DUCT DISPERSERS

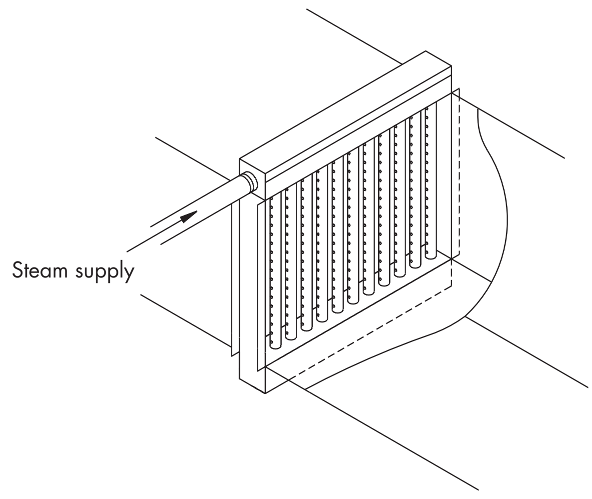
A given steam disperser size has a greater capacity when it is operated at 50 psi than at 5 psi. It is sometimes tempting to economize on equipment cost by designing the humidification system for a higher, rather than a lower, steam pressure, since it will result in smaller duct dispersers and smaller steam piping.

Two important considerations should be weighed as this decision is made: noise level and modulating capability of the steam valve.

Noise. As the steam pressure is elevated, the noise level increases. If the disperser is located in a high background noise level area, such as a mechanical equipment room, the level may be acceptable. If, however, it is located near an occupied area, the noise level may not be acceptable, and a lower steam pressure should be the basis of design.

The second consideration is that of the modulating capability (turndown) requirements of the application. A low flow rate (small Cv) steam valve may not have the turn down capability, as previously explained, to control well during periods of low demand. Raising the steam pressure results in a smaller Cv valve and aggravates this condition.

FIGURE 124-1: STEAM DISPERSION PANEL



CONTROLLING NOISE IN DUCT STEAM DISPERSERS

Hospital surgeries, because of their low background noise level and the need for a distraction-free atmosphere are especially sensitive to steam-disperser noise.

When choosing a steam jacketed duct disperser, the manufacturer's noise level data should be consulted. Some are much quieter than others.

Sound-attenuating measures, such as locating the disperser remotely from the nearest outlet diffuser and providing turns in the duct between the disperser and the diffuser, will help in containing the sound.

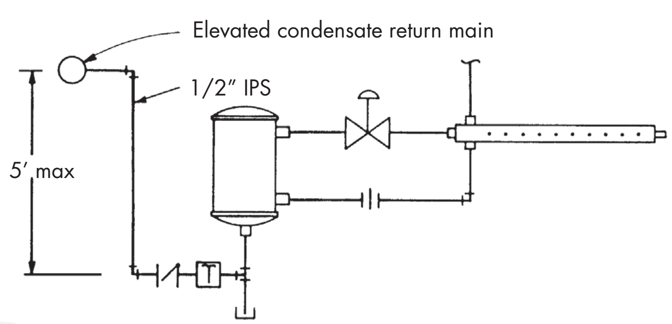
The use of a non-jacketed, panel type disperser as shown in Figure 125-1, will provide the quietest operation of all.

ELEVATING THE CONDENSATE FROM A STEAM DISPERSER

For structural reasons in certain installations of jacketed type boiler steam dispersers, it is not possible to drain the steam trap by gravity. This requires that the condensate must be lifted to the condensate return main. When designing, care must be taken to avoid possible water hammer and incomplete drainage, but when absolutely necessary, it can usually be done successfully by observing the following:

First, adequate steam pressure must be available. Theoretically, one pound of pressure will raise water about 2 feet. In practice, because of pipe friction, pressure drop through a steam trap, back pressure in a return line, etc., a maximum lift of 1/2 foot per pound of steam pressure at the steam trap inlet should be expected. For example, a steam pressure of 5 psi will provide a maximum lift of 2-1/2 feet. Lifts in excess of 5 feet should not be attempted.

FIGURE 125-1: ELEVATING THE CONDENSATE



Second, the proper style of steam trap must be used. The trap should be of an intermittent type that cycles full open and full closed as it operates, such as the inverted bucket or disk types. This type provides the sudden surge of steam pressure that propels the condensate along. Float and thermostatic traps, where the degree of valve opening in the trap varies with the incoming condensate flow rate do not provide the pressure surge.

PIPE SIZE

Do not oversize the vertical pipe. The size of this portion of the piping usually should not be larger than ½" IPS. A surge of steam pressure, when the trap opens, propels the condensate up the vertical pipe. If this pipe is oversized, the steam may bubble up through the water rather than forcing it up.

CHECK VALVE

A check valve should be provided adjacent to the steam trap to prevent backflow of condensate into the steam trap between discharge cycles or during periods of little or no steam pressure. Failure to do so could result in the accumulated backflow entering the steam separator and being discharged into the duct when pressure is resumed.

PLACEMENT OF THE STEAM DISPERSER WITHIN AN AIR HANDLING SYSTEM

Usually, there is no single right or wrong placement for a steam disperser; however, some locations are preferred over others. Much depends on the system design, its uses and its applications.

The following possible locations, along with discussion regarding the appropriateness of each, are presented as a guide for the thought process that will lead to the best selection.

EXAMPLE 1: PLACEMENT IN AN AIR HANDLING UNIT

The primary objectives are to select the proper disperser type and locate it in the system where it will achieve thorough absorption of the water vapor without water forming in the system and do it at a minimum cost.

The source of the humidification water vapor (steam boiler or vapor generator), the final duct RH, and the capacity of the system will influence the style of dispersion system used. The disperser type used will, in turn, influence the point in the system where it is installed.

Vapor Generators, which operate at very low pressure, can be served by two basic styles of dispersers, one or more single tubes (Figure 139-1) or a dispersion panel (Figure 139-2 or Figure 140-1).

Single tubes are usually used in smaller systems that have ample downstream absorption distance while the dispersion panels are used in larger systems and/or applications that have limited steam absorption distance available. The manufacturers engineering data should be consulted for absorption performance.

Steam Boilers, which operate at higher pressures, can be served by (a) single steam jacketed dispersers (Figure 138-1), (b) a bank of several steam jacketed disperser tubes (Figure 138-2), or (c) a steam dispersion panel (Figure 140-1).

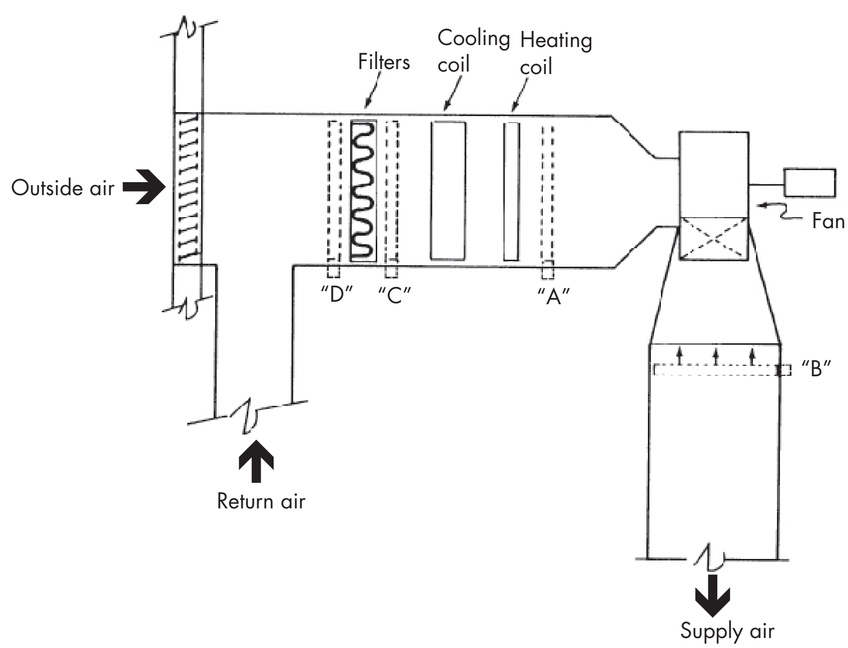
Again, single steam jacketed tubes are used in smaller ducts. In larger systems and/or where absorption space is limited, either the jacketed tube bank or the steam dispersion panel is used. The latter is preferred when the heat given off by the tube jackets of the former (which are always hot) is undesirable.

LOCATION A

In large systems, where either Figure 139-2 or Figure 140-1 is to be used, location A is preferred for several reasons:

1. It is usually the warmest part of the air stream, which facilitates absorption.
2. Access for inspection and service is convenient.
3. Connection points for steam, water, drainage, and power are usually nearby.

FIGURE 126-1: PLACEMENT IN AN AIR HANDLING UNIT. TOP VIEW.



4. The air flow is uniform and straight due to the upstream coil banks. This enhances absorption performance.
5. There is usually sufficient physical space (width and height) to accommodate the dispersion panel when used.
6. The necessary air travel distance to allow complete absorption prior to engaging downstream devices (coils, dampers, fan) is usually available. Note: steam absorption must be sufficiently complete to prevent wetting the fan. Otherwise, microbial growth and rusting are probable.

LOCATION B

This location can be used successfully when absorption distance requirements are easily met (low final duct RH or there is an ample length of unimpeded duct work).

One or more jacketed or unjacketed dispersion tubes can be installed at this location. The absorption performance may be marginal because of the high air velocity at this point. The jets of steam may be unable to penetrate the high velocity air envelope sufficiently to accomplish a high degree of steam-air blending. A dispersion panel probably could not be used because there may be insufficient height for the necessary cross-sectional area to achieve the needed capacity.

LOCATION C

There are several possibilities to study when considering this location:

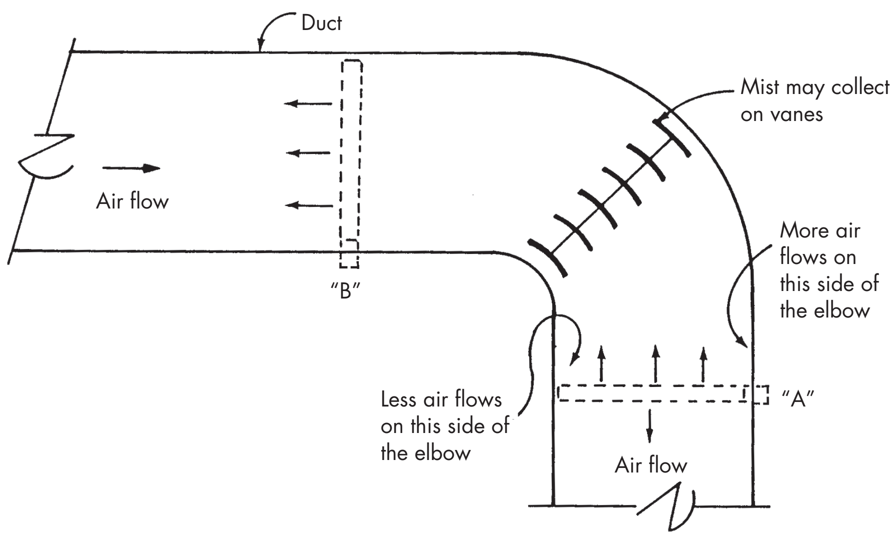
1. The distance from the fan is greater than location A which aids steam absorption.
2. A considerable amount of moisture could be lost when the cooling coil is operational. Even when not cooling, the coil may remove some moisture.
3. Access for inspection and/or service could be limited.
4. Any of the dispersion systems could be installed at C if absorption distance requirements can be met.

LOCATION D (MIXING BOX)

This location should be avoided because:

1. The air is too cool.
2. The filters would become wetted.

FIGURE 127-1: PLACEMENT IN AN ELBOW. TOP VIEW.



EXAMPLE # 2: PLACEMENT IN AN ELBOW

Because of possibly wetting turning vanes, the downstream side of an elbow (location A) is a better choice than B. In cases where it is structurally impossible to avoid location B, a rapid absorption design may be required.

Since more air flows along the outside of a turn, better absorption will result if the humidifier discharges proportionately more steam in that part of the air stream.

EXAMPLE #3: PLACEMENT IN A MULTI-ZONE SYSTEM

The best method of humidifying a multi-zone air handler is that of providing a separate steam disperser in each zone duct. When this is not possible, a single disperser is sometimes mounted within the air handler. When that is done, the following recommendations apply:

Location A is generally the best. It affords the maximum absorption distance ahead of the zone dampers. Some loss of vapor may occur due to condensation on the cooling coil, but since there is a drain pan below the coil, no damage should occur. In cases where the duct relative humidity exceeds 50%, multiple steam dispersion tubes should be used to shorten the absorption distance.

FIGURE 128-1: PLACEMENT IN A MULTI-ZONE SYSTEM. SIDE VIEW.

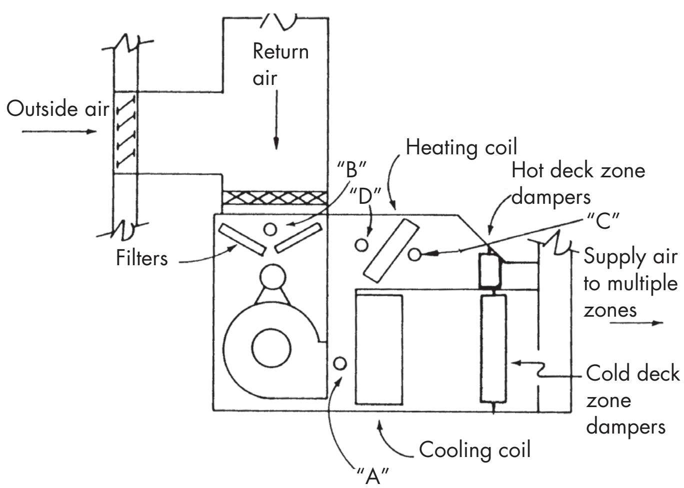
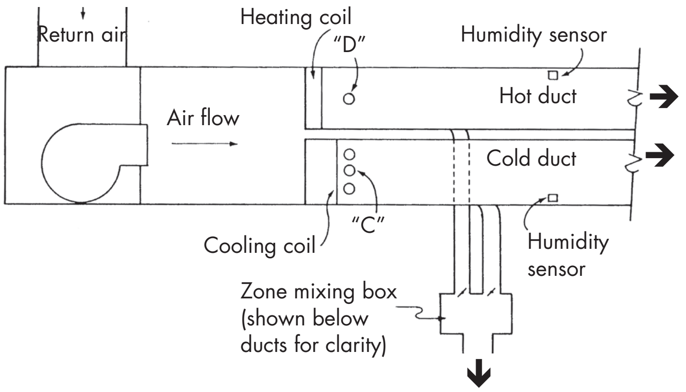


FIGURE 128-2: PLACEMENT IN A DUAL DUCT SYSTEM. SIDE VIEW.



Location C is in a warmer location, but is a shorter distance to the dampers than A. The same is true for location D.

Location B is generally not acceptable because it will likely cause wetting of the filters and the fan.

In some applications, the air quantity of one of the zones will be larger and may be sufficient to absorb all of the humidity required for all of the zones. This is easily checked by use of Table 81-1, page 81. If such is the case, that particular zone duct may be a more desirable location for adding humidity than a location within the air-handling unit itself. While it is true that a differential in relative humidity between zones will exist for a period of time after start-up, it will eventually equalize as the system is operated.

PLACEMENT IN A DUAL DUCT SYSTEM

In this type of system, the hot-and cold- duct air temperatures usually are constant, and temperature control in each zone is accomplished at the zone mixing box in the space by varying the ratio of hot to cold air.

As the heating-cooling load within the building changes, the ratio of the volumes of hot and cold air in the two ducts varies along with it. Neither single location, hot duct or non-cold duct, will be satisfactory at all times for the addition of humidity.

It is sometimes necessary, therefore, to treat each of the two ducts separately, (i.e. provide each with its own humidifier and humidistat).

The maximum load condition should be calculated for both ducts. A short-cut method would be to size each of the two humidifiers for the total building humidification load. In other words, if the total load is 400 pounds per hour, each humidifier should have a capacity of 400 pounds per hour. This is done to assure adequate humidifier capacity at all times in either duct. Special attention should be given to the steam absorption distance, particularly in the cold duct.

The downstream humidistat in each duct would be set to control at the room or space, duct equivalent relative humidity. For example, in a system where the cold duct temperature is a constant 55° F and the desired space conditions are 72° F and 45% RH, the duct humidistat would be set to control at 80% RH.

The humidistat must be far enough downstream to escape the moisture cloud in order to control accurately.

COMPARING FUEL COSTS

Situations arise, when it is necessary to compare energy costs, before deciding upon a humidification method.

To do this, the yield of usable heat per dollar of fuel cost, must be determined. Naturally, fuel costs will vary, and the prevailing prices at the time and place the calculation is to apply, must be substituted for those used in the following examples.

Manufacturers' output ratings, are usually based on 80% efficiency in boilers, furnaces, and other gas-fired appliances. It must be remembered that this rating is fuel-to-the-heated-medium efficiency of the device itself only. The overall efficiency of a heating plant is quite another matter. For example, partial-load efficiencies are much lower than those at maximum design load. Studies show that an overall heating season efficiency of 60% is about the maximum attainable in a well operated system, and the average may well be below 60%. The following examples, however, are all based on 60% overall plant efficiency.

NATURAL GAS EXAMPLE

Assume a natural gas heat content of 1000 BTUs per cubic foot and a cost of $4.90 per 1000 cubic feet. Determine the cost per 10,000 BTUs of usable heat.

Usable Heat: 1000 cu. ft. x 1000 BTU/cu. ft. x .6 (efficiency) = 600,000 BTUs.

Cost per 10,000 BTU: $4.50 ÷ 60 = 10 cents.

LIQUEFIED PETROLEUM GAS EXAMPLE

Assume 91,600 BTU per gallon and 70 cents per gallon.

Usable Heat: 91,600 x .6 (efficiency) = 54,960 BTUs

Cost per 10,000 BTU: $0.70 ÷ 5.5 = 13 cents

FUEL OIL

It is questionable that a fuel-to-heated-medium efficiency of 80% is attainable in practice because oil-combustion products foul heating surfaces more readily than do either natural or L.P. gas. However, the following assumes 80% to be true. Again, apply a heating season overall average of 60% efficiency.

NO. 2 FUEL OIL EXAMPLE

Although the BTU content of No. 2 fuel oil varies, assume a heat content of 143,000 BTU/gallon and a cost of $0.75 per gallon.

Usable Heat: 143,000 BTU X .6 (efficiency) = 85,800 BTUs

Cost per 10,000 BTU: $0.75 ÷ 8.58 = 8.7 cents

NO. 5 FUEL OIL EXAMPLE

Assume 146,600 BTU per gallon and 50 cents per gallon.

Usable Heat: 146,000 X .6 (efficiency) = 87,600 BTU

Cost per 10,000 BTU: $0.50 ÷ 8.76 = 5.7 cents

ELECTRICITY

Electrically heated humidifiers, with the exception of certain radiant types, convert electrical energy to heat energy with 100% efficiency. However, the present state of the art of most commercial size electric humidifiers utilizes a drain-down system of one type or another to minimize mineral build-up in the humidifier. This drain-down principle represents a loss in efficiency because the heated drain-down water is usually wasted.

The following example shows how to calculate that efficiency.

EXAMPLE

Assume an electrically heated humidifier producing 60 lbs. of steam per hour using make-up water at 50°F. Further assume that an additional 30 lbs (or 50% of the total water evaporated) is wasted as drain-down water at 212°F. Determine the loss in efficiency.

Total latent heat required:

60 lbs x 970 = 58,2000 BTU

Total sensible heat required:

60 lbs + 30 lbs = 90 lbs.

90 x Temperature Rise (50°F to 212°F) = 15,480 BTU

Total heat consumed:

58,200 + 15,480 = 73,680 BTU

Sensible heat absorbed by drain-down water:

30 lbs x Temperature Rise of 172°F = 4,860 BTU

Total efficiency loss:

4,860 ÷ 73,680 = .067 (or 6.7%)

With 100% efficiency, a kilowatt of electrical energy converts to 3419 BTU. Obviously, as the percentage of drain-down varies, the efficiency will vary as well. Additional small losses due to conduction are usually usable heat to the heated space and can be ignored.

EXAMPLE

Assume 93% efficiency and a cost of 18 cents per kilowatt hour.

Usable Heat:

3,413 X .93 (efficiency) = 3, 174 BTU

Cost per 10,000 BTU: $0.18 ÷ .3174 = 56.7 cents.

Note: An additional cost, called demand charge must often be taken into account. The amount of this cost varies considerably between electrical utilities, electrical load within the building, etc.

SUMMARY BASED ON PRECEDING EXAMPLES

Natural Gas = 10 cents per 10M BTU

L.P. Gas = 13 cents per 10M

BTU

No. 2 Fuel Oil = 8.7 cents per 10M BTU

No. 5 Fuel Oil = 5.7 cents per 10M BTU

Electricity = 56.7 cents per 10M BTU

CONTROLLING HUMIDIFICATION SYSTEMS

Pressure for higher-performing humidification systems by industry and governmental groups has led to new discoveries and improved equipment. Among the various items of equipment that make up a humidification system, control devices have improved most radically.

Probably the single industry having the most influence in these improvements has been microchip manufacturing. Control of temperature and humidity in their large, clean spaces has progressed to that of requiring the ability to control within a tolerance of ±.2°F DB and 1°F WB.

THE FUNCTION OF THE HUMIDITY CONTROL SYSTEM

RH control consists of maintaining a desired level of humidity in a space, by continually varying the input of water vapor to the space, in accordance with changing demands.

Humidity control systems range from simple, loose-tolerance, on/off type systems that are manually reset, to highly sophisticated, tight-tolerance, fully automatic systems, with several quality levels in between.

Depending on the type of humidification system in use, controlling the water vapor output means controlling one of several possible functions, such as:

- Rate of steam flow to a boiler steam duct disperser
- Rate of steam flow to the heat exchanger in a steam-heated vapor generator
- Rate of heating water flow to a hot water heated vapor generator
- Burner on-off cycling or firing rate modulation in a gas fired vapor generator
- Rater of electrical energy input to a time-proportion modulated electrically heated vapor generator
- Rate of SCR modulated electrical energy input to an electrically heated vapor generator.

The accuracy of RH control in a given humidification system is dependent upon the speed with which the system can react to a trending change in RH within the space.

Obviously, the longer it takes for the system to alter its rate of water vapor input to the space, the further the trend will progress before a correction can have an effect.

HUMIDISTATS AND HUMIDITY SENSORS

These devices are available in several styles and types. In the simplest on-off design, often referred to as a humidistat, a section of hygroscopic material (human hair, wood fibers, cellulose) changes physical dimensions in accordance with changes in RH, causing a switch to open or close.

A simple form of pneumatic modulating humidistat is available that emits a signal of varying compressed air pressure. In this device a hygroscopic material repositions an internal bleed-off valve in accordance with the room RH. This then causes a change in the output air pressure. This pressure is transmitted to a pneumatic valve actuator which repositions the valve causing a change in the fluid flow rate. In the case of an electrically heated vapor generator, the signal can be transmitted to a pneumatic-electric transducer, which in turn, varies or modulates, the cycling rate of a time proportion controlled electric heating element.

A simple electric modulating humidistat can be used for the control of electric motor actuated valves. Changes in RH produce a change in the electrical signal to the valve motor, causing it to reposition, readjusting the fluid flow.

MICROPROCESSOR CONTROL

The humidity transmitter offers the most accurate RH control. In this device, a sensor responds to a change in electrical conductance, which is converted into a signal by a transducer and transmitted to a microprocessor. The higher quality microprocessors have a PID loop (proportional integral derivative). In simple terms, this feature allows the microprocessors to be fine-tuned to the dynamics, or inertia, of the specific humidification system it is controlling. The end result is humidity controlled within very narrow parameters, for example plus or minus 0.5% RH in certain cases.

In addition to controlling the rate of water vapor output from the humidification system, some microprocessors are designed to manage other functions of the system such as:

- Controlling water level
- Periodic drain and refill
- End-of -season drain
- Cold snap reset
- Proof of air flow

Also, some microprocessor systems are equipped with a keypad having a LED display. With this device, the user is able to scroll through numerous readouts of performance of the humidification system and where appropriate, make readjustments. Among the possible items of information displayed are:

- Space or room RH
- RH set point of room controller
- System output (lbs/hr)
- Time-to-go to next servicing
- Fault indicators
- Drain and flush duration (minutes)
- Drain and flush frequency (operating hours)

MODULATING VALVES AND FLUID FLOW

The term modulate when applied to valves, means to regulate or adjust the flow rate of a fluid, such as water, air, or steam.

The degree of accuracy to which a valve can modulate flow rates is determined by several factors among which, two of the most important, are the turndown ratio of the valve and the proper Cv selection for a given maximum flow rate.

Cv means flow coefficient. It is expressed as a number, such as .5, 2, or 20, that is used as a standard and it is a means of stating maximum flow rates, by valve manufacturers. It refers to the gallons per minute of water, at a temperature of 60°F, that will pass through a fully open valve at a pressure differential of 1 psi.

In other words, a valve having a Cv of 8 will pass 8 gpm under the above conditions. The flow rates of other fluids, steam, air, etc., are calculated by applying a correction factor to the Cv.

Modulating valves range in quality from inexpensive, commercial-grade types to finely honed expensive, industrial-grade types. Commercial-grade valves which are comparatively simple and easy to manufacture, are mass produced and therefore built to relatively liberal tolerances. This reduces the cost because the machines do not have to be shut down for readjustment due to cutting tool wear as frequently, and there are fewer inspections.

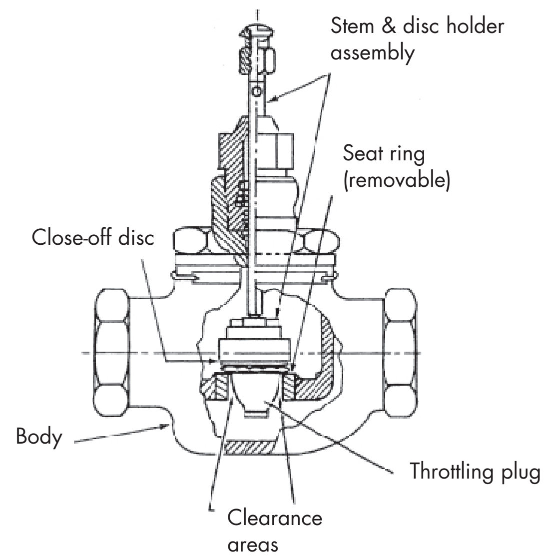
For most humidification systems, the above described valve, when of reasonably good quality, will perform adequately. However, when the requirements of control accuracy become strict, such as a temperature within plus or minus .1°F and an RH within plus or minus 1%, industrial quality valves are called for.

Modulating valves change the flow rate by varying the pressure drop across the valve (see Figure 133-1). A circular, tapered, throttling plug, mounted at the end of a moveable valve stem, is positioned by the actuator within the valve seat area. This creates an annular opening between the plug and the seat through which the fluid flows. The cross sectional area of this opening is varied from maximum to minimum as the actuator varies the insertion depth of the tapered plug. This then varies the pressure drop across the seat-plug combination, thus varying the flow rate. In the commercial grade valves used in most boiler-steam dispersers, a close-off disc (not the throttling plug) engages the seat for final shut off of the flow.

The minimum repeatable controlled flow rate (just before shut off) is governed by the degree of precision to which a valve is manufactured. The tapered plug must not ride against the side of the valve seat as it approaches closing, which may cause it to stick. A clearance, however small, must be maintained if the valve is to operate smoothly and repeatedly. The nearer the large diameter of the tapered circular plug can be made to come to the circular seat without actually touching it, the lower will be the minimum repeatable controlled flow rate and therefore, the greater the turndown ratio.

The turndown ratio of a valve is the ratio of its maximum, or full open flow capacity, to its minimum, repeatable, controlled, flow capacity. If a particular valve has a full open flow rate of 480 lbs/hr and a minimum flow rate of 24 lbs. per hour, it is said to have a turndown ratio of 480 ÷ 24 or 20 to 1.

FIGURE 133-1: MODULATING VALVE



Small Cv factor valves inherently have a lesser turndown ratio than large valves. This is true even though the minimum clearance, or tolerance between the plug and the seat, attainable in manufacturing, is fairly constant. This means that as the plug and seat diameters decrease, the ratio of the clearance area to the full-flow area becomes smaller thus reducing the attainable turn down ratio.

The lowest Cv factor available from good quality, commercial-grade valve manufacturers is approximately .4. A turndown ratio of 5 to 1 is about average for this Cv factor and level of precision. In an industrial grade valve with .4 Cv, a turndown ratio of about 70 to 1 can be expected.

This .4 Cv commercial grade valve, when fully open and at an inlet pressure of 10 psi and zero pressure downstream (100% drop across the valve), will pass about 17 pounds per hour (pph) of steam. At minimum flow, just before close off, it will pass 17 ÷ 5, or 3.4 pph.

Assume these valve and pressure conditions are used in a small hospital surgery room application, where the steam humidification load ranges from a maximum of 15 pph to a minimum of zero. Whenever the load is less than 3.4 pph, the valve will be required to cycle open and closed (called two-positioning) to satisfy the controller.

This type of application will usually require year-round cooling, so the duct air temperature will be in the 55° to 60°F range and the duct relative humidity will have to be about 82% to maintain 50% RH at 72°F in the room. In this example, at minimum flow, just before the disperser valve closes, this air volume at 80% RH can absorb only an additional 4 pph, at which time it becomes 100% saturated. However, the valve can modulate down to 3.4 pph, after which it shuts off. Assuming the disperser design accomplishes rapid steam absorption, the results will be satisfactory.

Let us assume the same example with one change. The steam pressure will be 25 psi instead of 10 psi. In this case, the maximum flow rate will increase to 32 pph, and the minimum flow rate will become

32 ÷ 5, or 6.4 pph. Now, after the addition of 4 pph has produced a 100% saturated air stream, we still have 2.4 pph left over. The valve will now begin to “two-position”, or “hunt” and during the “open” part of the two-position cycle, visible steam most likely will issue from the air outlets into the surgery.

VARIABLE AIR VOLUME (VAV) SYSTEM CONTROL

When this temperature control system is in use, the room or space temperature is controlled primarily by varying the volume of air being supplied. The air entering the space passes through a VAV box mounted in the supply air duct. Inside this box, an air volume control damper modulates in response to a room thermostat, increasing and decreasing the air volume as required, to maintain the desired space temperature.

Typically, a steam or vapor disperser is mounted in the duct on the air leaving (downstream) side of the VAV box. Variations in the air volume due to heating load variations result in variations in the humidity load. At times, these changes can take place quickly which means that quick response by the humidity controls is needed to avoid oversaturation of the air with resultant duct wetting.

A simple room mounted humidistat controlling the humidifier output is not an adequate solution. Instead, two humidity sensors, one in the duct downstream of the duct disperser and the other in the room or space are needed. These two sensor provide input signals to an integrating device, which in turn, modulates the humidity output and allows the humidification system to satisfy the space needs without causing duct wetness and jeopardizing indoor air quality.

Section XVII: Introducing Moisture To The Air Stream

This section deals with the process of absorption of water vapor by the moving air stream in an air handling system. While it is extremely important that the vapor producing device be properly designed and applied successfully, effective delivery of the vapor to the air stream is equally as important. When proper absorption of the vapor does not take place, condensation with resulting moisture fall-out occurs. Wetness in ducts, causing microbial growth or possible rusting, is a condition usually requiring correction. By initially choosing the proper dispersion system for the application, this problem can be eliminated.

In order for the reader to better evaluate water vapor dispersion equipment design and more intelligently apply it to systems, the following detailed explanation of the isothermal process of humidification is offered.

The following sketch is a typical top view representation of water vapor being added to an air stream by means of a steam dispersion tube having multiple steam ports discharging upstream against the airflow.

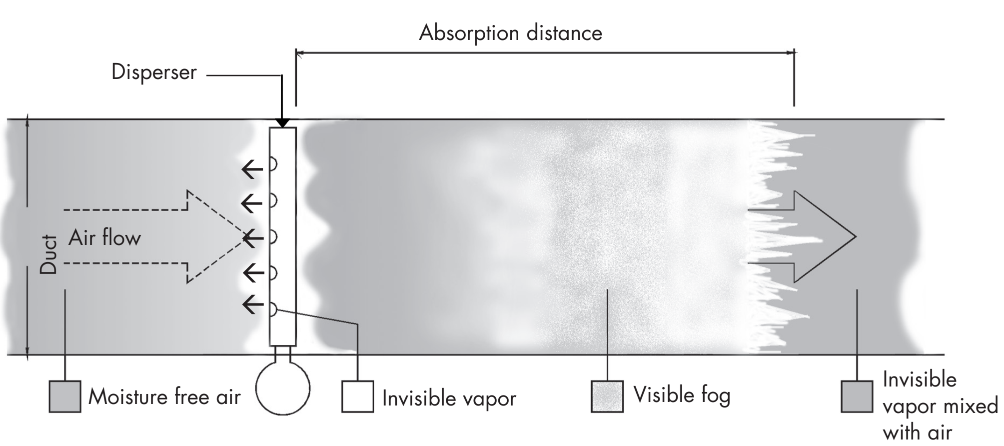
As the steam emerges from the ports of the tube and begins moving downstream, it is invisible for the first inch or so. It then transforms into a fog, expanding in size and becoming less dense as it travels downstream, and finally disappearing. Two changes of state occur when the steam emerges as an invisible gas, condenses into a fog and then returns to an invisible gas. The following is an explanation of what actually takes place.

As the steam emerges from the tube, it is 100% saturated vapor. It immediately begins mixing with air that is lower in temperature. The mixture, at this point, cools and becomes saturated. Some of it condenses. As it condenses, microscopic drops of liquid water known as fog are formed. As this change of state takes place, heat, at the rate of 970 BTU per pound of fog is released, resulting in an immediate increase in the water vapor-air, mixture temperature. (This increase can be significant— as much as 30°F or more. Temperature controls, mounted within this fog area, will be "fooled".)

As the fog moves further downstream, another change of state begins taking place. The plumes of mist fan out and become intermingled with pockets of air that are not saturated. Heat flows back into the mist from these air pockets, causing the mist to evaporate and disappear. When the process is complete, the temperature of the mixture returns almost to the starting point. The total heat of the air/water vapor mixture however, has now been increased by a couple of degrees due to the sensible heat of the steam that was added. Several observations are now in order.

1) The more thorough or uniform the initial mixing of steam with the cross section of the air stream the more rapid will be the two changes of state and the shorter will be the visible vapor travel, or steam absorption distance.

FIGURE 136-1: STEAM DISPERSER IN DUCT



2) In a given system, the lower the duct air temperature for a given quantity of water vapor being added, the higher will be the final duct relative humidity, and the further the mixture must travel in the air stream, to complete absorption.

3) Visible mist will collect on internal devices in ducts with which the mist comes in contact (fans, filters, dampers). This is called fallout. In severe cases, water leakage from the duct may occur. Wet areas in ducts may not be allowable for reasons previously mentioned.

Some manufacturers of steam dispersers have created test laboratories for testing the absorption characteristics of their products. This enables them to measure the air-flow distance required downstream of the disperser for the steam to become absorbed. As a result of testing under varying conditions of air temperature and humidity, upstream and downstream of the disperser, tables have been developed for the various dispersion systems they manufacture. The humidification system designer, through the use of these tables, can engineer a system, that will function as intended, with no wetting of the duct system.

DUCT STEAM DISPERSERS - GENERAL

Dispersers usually consist of one or more metal tubes that span the width of a duct. The tube(s) have small ports along their length, from which the steam is discharged into the passing air stream.

Ideally the metal tube should not drip condensed steam into the duct. Additionally, the disperser design should accomplish thorough absorption of the steam before it can impinge upon downstream internal duct surfaces resulting in fallout with resultant duct wetness.

As was previously explained, the degree of difficulty in achieving proper absorption varies, depending on duct air temperature and required final relative humidity.

Two general types of steam dispersers are in common use. They differ in design because of the pressure of the steam supplied to them. One type, called steam jacketed dispersers is designed for use with steam supplied from a steam boiler at a pressure of 2 psi or more. The other type, called vapor dispersers is designed for use with vapor generators, where the pressure of the vapor supplied to the disperser is very low (mere ounces).

The latter design in a dispersion panel version, is sometimes used for boiler steam applications in cases requiring closely spaced tubes for rapid absorption and where excess jacket heat cannot be tolerated. (See page 140.)

FIGURE 138-1: STEAM JACKETED BOILER STEAM DISPERSER IN A DUCT

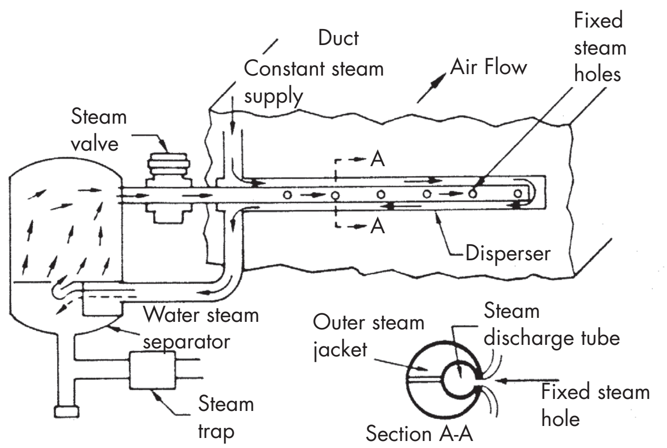
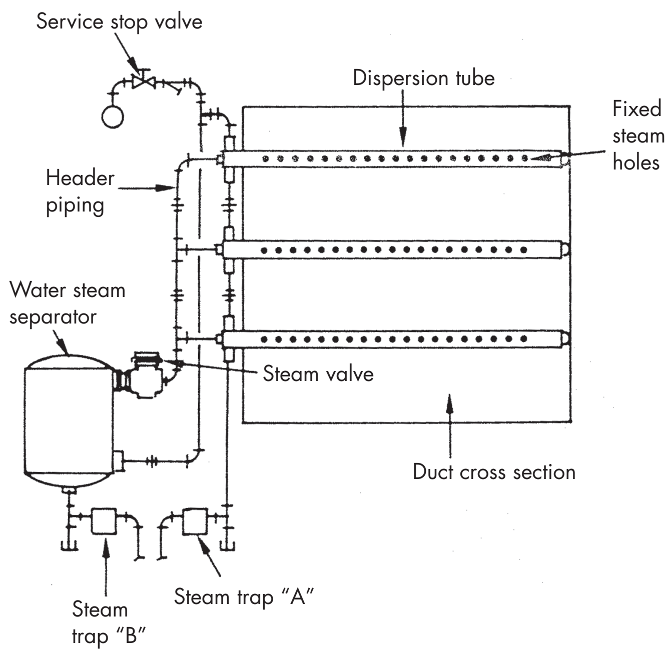


FIGURE 138-2: MULTIPLE TUBE DISPERSER



DUCT MOUNTED STEAM JACKETED BOILER STEAM DISPERSER

This device consists of a water/steam separator, a steam jacketed duct tube, a control valve, and a steam trap.

The steam holes in the duct tube are punched or drilled, and are sized by the manufacturer for the maximum steam flow for which a given tube size can be used. The steam disperser tube, which usually spans the width of the duct, is contained within an outer, or jacketing, tube. The outer jacket contains steam at line pressure, which is several degrees hotter than the zero pressure temperature of the steam being discharged from the inner tube. This temperature difference prevents steam from condensing in the disperser tube. This type usually incorporates a water-steam separator of stainless steel or cast iron for removal of entrained condensate in the steam coming from the steam main.. A normally closed, modulating type valve and filter/noise suppressor downstream of the valve, are usually also parts of this equipment. (See Figure 138-1.)

APPLICATION CONSIDERATIONS:

a) The steam-heated disperser tube constantly gives up heat to the air stream. When this is undesirable, the tube can be factory insulated.
b) The disperser tube should be carefully located in the system to avoid wetting of filters, fans, or dampers.
c) Control of humidity is usually quite accurate.
d) To control noise, locate away from air registers.
e) For optimum performance (noise level and control accuracy), steam pressure to the disperser in the range of 5 to 12 psi produces best results.
f) The condensate formed in the separator and disperser jacket is under line pressure and is usually returned to the boiler.
g) Cast iron separator models, because of their mass, should be equipped with a temperature switch to prevent spitting during warm-up.
h) When a single duct disperser tube is used, adequate absorption performance can be expected in low duct air final RH (60% and lower), or where ample, unobstructed absorption space is present (long distance of straight duct).

MULTIPLE TUBE STEAM JACKETED BOILER STEAM DISPERSER

In this variation, several jacketed tubes are located one above the other, usually in large, tall air streams. (See Figure 138-2.)

The objective is to attain absorption in a shorter distance of air travel, over that provided by a single-tube installation, by dispersing the steam into more planes of the air flow.

APPLICATION CONSIDERATIONS:

Same as for Boiler Steam Dispersers above.

VAPOR DISPERSER (SINGLE TUBE)

This equipment is used to disperse very low pressure vapor into the air stream, such as that from various types of vapor generators that typically operate at less than ½ psi.

Generally, the disperser consists of a metal tube that spans the width of the duct. The vapor is conveyed to the duct disperser from the vapor generator by means of a hose or rigid metal tubing.

The disperser has a line of vapor emission ports that are punched or drilled along one side. The higher quality designs of dispersers utilize high-temperature plastic tubelets as vapor emission ports. These tubelets are pressed into the disperser tube and extend to the center region of the tube. They effectively exclude the condensate that forms from being discharged with the vapor.

The condensate that forms in the duct disperser, must be removed and a drain tube should be provided which will allow it to drain by gravity back to the vapor generator. Some designs emit dryer steam than others.

FIGURE 139-1: VAPOR DISPERSER

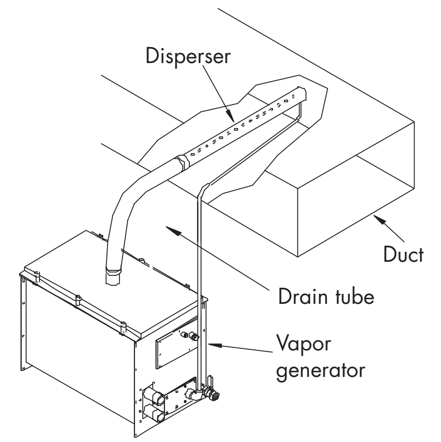


FIGURE 139-2: UPFED DISPERSION PANEL

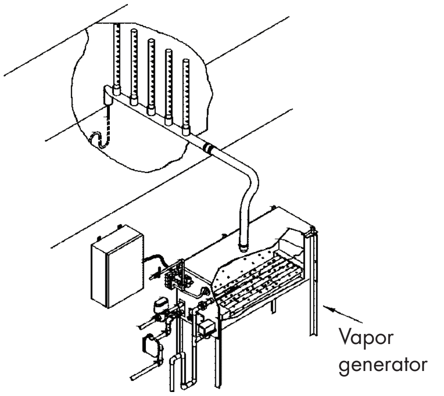


FIGURE 139-3: DISPERSION TUBE DETAILS

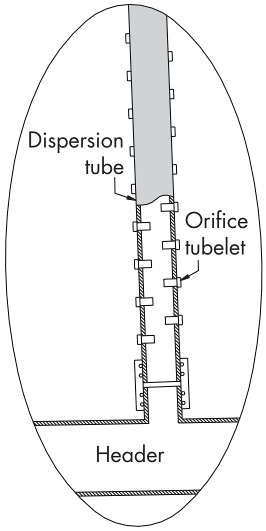


FIGURE 140-1: DOWNFED DISPERSION PANEL

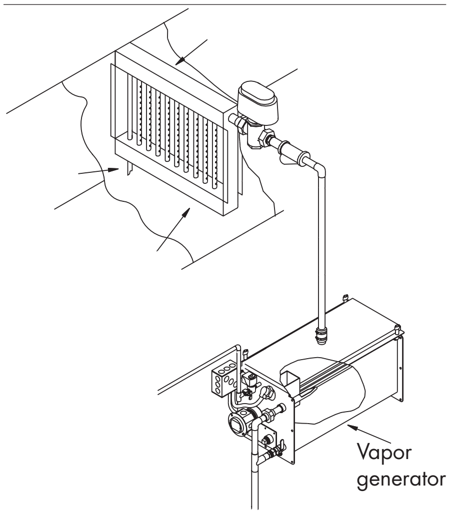


FIGURE 140-2: AREA TYPE VAPOR GENERATOR

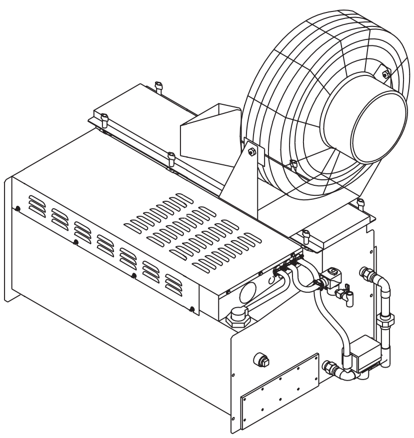
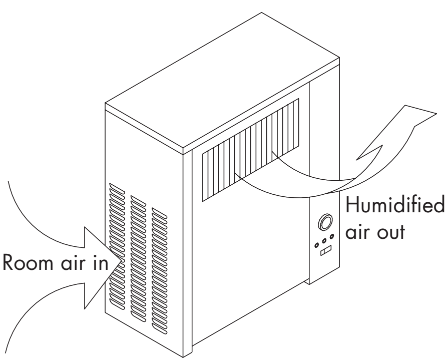


FIGURE 140-3: CABINET TYPE VAPOR DISPERSER



UPFED MULTIPLE TUBE DISPERSION PANEL

As the degree of difficulty in accomplishing absorption within the available absorption distance increases, the sophistication of the dispersion system must also increase.

In this design, a horizontal header at the bottom of the duct, supplies vapor to a row of vertical disperser tubes spanning the duct height. (See Figure 139-2.) The spacing distance between tubes varies according to the degree of absorption difficulty. (The shorter the available absorption distance, the closer together the vertical tubes must be.) Vapor discharge openings in the vertical tubes arranged in two opposing vertical rows discharge water vapor across the air stream. In the better performing designs, each steam discharge opening consists of a non-metallic tubelet inserted through the tube wall and extending to the center region of the vertical tube. (See Figure 139-3.) Since the condensate adheres to the inside tube wall it flows down by gravity to the header for removal. The tubelets allow only steam (no condensate) to be discharged into the air stream.

Since the vapor and the condensate flow counter to one another, the capacity of this design is limited. If, for example, the upward velocity of the vapor were excessive, it would prevent the downward flow of condensate The tube would then become water logged and discharge water along with the vapor. When such is the case the downfed dispersion panel (described below) is called for.

DOWNFED MULTIPLE TUBE DISPERSION PANEL

This is a more sophisticated version of the upfed panel described above. It is used in cases where the absorption distance requirements cannot be satisfied with the simpler upfed design.

This design contains two headers, one for supply and one for return, instead of the single combination supply/return header found in the upfed panel. As a result, the vapor and condensate move in the same direction (downward) instead of opposite each other, which enables higher steam velocities, resulting in much higher capacities. Again, the dual-direction discharge tubelet design shown in Figure 139-3 (on page 139), is utilized. Extremely rapid and thorough steam absorption is achieved with this design.

Additionally, the dual-header arrangement permits the vapor disperser tubes to be mounted either horizontally or vertically. Horizontal dispersers do not drain well in the single header design.

The horizontal tube panel design has an advantage over vertical tube type for a duct that is wider than it is high because fewer tubes are needed, resulting in less fabrication labor and lower overall cost.

This equipment is often used instead of steam jacketed dispersers for boiler-steam applications because it doesn't add unwanted jacket heat to the air stream. Steam jacketed disperser tubes remain hot at all times, and when multiple tubes are required to accomplish rapid steam absorption, the heat gain can become excessive.

VAPOR DISPERSERS (AREA TYPE)

Some spaces to be humidified do not have ducts into which water vapor can be supplied for distribution by the air handling system. Examples could be non-air conditioned manufacturing areas such as woodworking, paper, printing, and textiles.

For these applications, a vapor generator, having an integrally mounted propeller type fan for vapor dispersion, and suspended in the space, can be an effective answer.

An eye pleasing humidifier design is available in which the vapor generator is mounted within an attractively finished metal cabinet. Within the cabinet is a blower that draws room air in, passes it over an internal vapor disperser where it is humidified and discharges it back into the room. These humidifiers are used in large offices, churches and individual large-room applications.

Section XVIII: Introducing Cold Mist to the Air Stream

This section relates to fogging systems where humidification, not evaporative cooling, is the primary objective.

The fogging apparatus produces water mist particles ranging in size from 10 to 50 microns and uses either compressed air or high mechanical pressure (300 psi approximately) atomizing nozzles.

The application of cold water fogger humidification systems to large central air handlers is an undertaking that requires careful study beforehand. Numerous cases exist where foggers were installed and later replaced with another type of humidification system. This came about because either the foggers could not attain the needed level of RH or they created excessive duct wetness and often both. With thorough planning however, a successfully operating system can be designed.

Humidification system designers sometimes underestimate the degree of difficulty in converting atomizer-produced fog to water vapor in an air handling system. This may be due to a lack of understanding on their part of the difference between the properties of the hot fog that is formed when steam is dispersed in air and condenses and those of the cold fog that emanates from compressed air fogger heads. Although the two fogs are very similar in appearance, there is a world of difference between the requirements of the two in becoming absorbed by the air as water vapor.

As a result, the designer sometimes fails to provide for the large amount of heat required, in the air, to evaporate cold fog.

HOW ATOMIZER FOG AND STEAM DIFFER AS THEY BECOME VAPOR IN AIR

STEAM (ISOTHERMAL PROCESS)

To begin, it must be understood that in a well-designed steam-dispersion system, only a small percentage of the steam discharged into the air duct condenses and forms fog. Most of it is directly absorbed by the air stream as an invisible gaseous vapor without ever going through the fog phase. The portion, however, that does condense into fog, was water vapor containing 970 BTU’s of latent heat per pound of vapor, prior to becoming fog. As it condenses, this latent heat is given up to the adjacent air molecules, resulting in an immediate increase in the temperature of the envelope of the air stream receiving the steam. As the fog-air mixture moves downstream, this heat becomes re-absorbed by the fog particles as they re-evaporate, after which the air temperature of the envelope returns (within 1 or 2 degrees) to what it was prior to receiving the steam. Controlling RH in the above system is merely a matter of controlling the quantity of steam being dispersed.

FOG (ADIABATIC PROCESS)

Fogger mist, on the other hand, is simply finely atomized water that carries no latent heat with it as it enters the air steam. For this fog to become water vapor, 970 BTU’s must be given up by the air molecules of the air stream to evaporate each pound of the mist particles that comprise the fog. This change of state produces a drop in air temperature that is significant. (See Example 4, page 86.) Unless enough excess heat is present in the air being humidified, it must be compensated for by preheating the air. Accurately controlling RH in this system, rather than controlling fogger output amount, requires controlling the evaporation rate. Doing so usually requires controlling the air temperature prior to fogging and in some cases after, as well.

It should be noted that in some of the simpler systems, such as comfort only humidification, which do not require tight control, merely cycling the foggers, on and off can provide acceptable results.

HOW FOG BECOMES WATER VAPOR

“Nature abhors a vacuum” is an old expression that simply says that pressures tend to equalize. This statement can be applied to humidification by the fog-evaporation process as well as to many other natural phenomena. Water evaporates when a difference in vapor pressure exists between the air and the water. The greater the vapor pressure differential, the more rapidly the water will evaporate.

The vapor pressure of air (when its moisture content remains constant) decreases significantly, as its temperature rises, resulting in accelerated evaporation. As a matter of passing interest, the vapor pressure of water increases only slightly as its temperature rises.

In a fogger system the rate, or speed, of evaporation is controlled by three variables, namely

1. The vapor pressure differential between the water and the air.
2. The surface area of the water exposed to the air. The fine fog particles suspended in the air, present a far greater water surface and will evaporate far more readily, than the same pound of water in the form of raindrops (assuming the drops could be made to remain suspended in air).
3. The thoroughness of the intermixing of the fog particles with all of the air molecules in the air stream.

As the air enters the fogger zone and becomes permeated with fog, evaporation of the fog with resultant cooling of the air begins. As the air proceeds through the fogger zone, its temperature continues to trend downward as its RH trends upward. Simultaneously, the vapor pressure difference between the air and water is declining, causing the speed of evaporation to decline as well.

If it were possible to create the ideal fogger system, the entering air temperature and the rate of fog discharge would be exactly correct so that the last fog particle would evaporate as the air left the fogging zone and the air temperature and RH would stabilize exactly where desired.

Unfortunately, it isn't quite that simple. First, the system design must provide for an air temperature entering the fogger zone that is high enough to offset the net amount of heat required to evaporate the necessary amount of fog, and provide for enough heat left over to maintain a sufficiently high vapor pressure difference. Without such provisions, the evaporation rate would tail off to almost nothing and the last few remaining fog particles would have to float in the air for too long a time to become evaporated, requiring an excessively long fogger zone.

Second, the design must provide for more fog than will be evaporated. This is needed to nullify the declining rate of evaporation that would otherwise occur because of the decreasing amount of water surface exposed to the air as the fog rarifies. Again, without this surplus of heating and fog, the time required and consequently the length of the fogging zone to complete evaporation would be excessive.

To reiterate, in an air system humidified by fogging, when a RH above 50% is needed, the design usually must provide for a surplus of heat and fog in the entering air and sufficiently intimate fog-air intermixing in order to accomplish the necessary evaporation within a short enough time period. The result is a practical fogging zone length.

In order to arrive at the desired leaving temperature after humidification, the excess heat must be removed.

By the same token, to prevent wetting the duct system downstream of the fogger zone, the excess fog must also be removed.

Removal of the excess heat usually requires a cooling coil, downstream of the fogger zone. However, as the cooling coil removes heat, it also removes moisture. This means that, in some cases, the preheat coil discharge temperature and the cooling coil discharge temperature must be interrelated so that the final air RH and temperature will stabilize where desired.

In certain cases, to achieve highly accurate temperature and RH control, a reheat coil downstream of the cooling coil is utilized.

Removal of the excess fog is accomplished by the use of a moisture eliminator located at an appropriate distance downstream of the fogger heads. This then forms the fogging zone. Moisture eliminators, being continuously wet, can present an air-contamination problem if proper sanitizing measures are not performed.

FOGGERS DO NOT SATURATE THE AIR

Another major difference in the performance of steam dispersers and foggers lies in their respective abilities to saturate the air. With steam dispersion, 100% RH is easily attainable. However, controlling RH accurately in the high ranges (above 90%) is difficult and requires high-grade instrumentation.

With a fogging system, on the other hand, attaining an RH above 50% without an excessively long fogging zone is difficult. As previously mentioned, to do so usually requires a high enough entering air temperature to achieve the approximate absolute moisture content so that when cooled to the desired temperature, the desired RH is attained.

FOGGER ABSORPTION DATA IS AVAILABLE

The fogging zone is defined as the portion of the air-handling system, beginning at the fogger heads and extending downstream to the point where all of the fog has evaporated or been removed by a moisture eliminator. To avoid the risk of compromising indoor air quality, wetting of downstream ducts should not be allowed to occur. The leading fogger system manufacturers have created duct test systems and offer published design data. The data allows the engineer to design systems that will function without wetting the duct system and control the humidity accurately.

The efficiency of typical fogger systems, in terms of the percentage of the fog that actually evaporates versus the percentage that falls out as water is not known and cannot be calculated. Fogger efficiency varies considerable with the fog particle size, the fog-air intermixing characteristics, the air temperature and the moisture content of the air entering and leaving the fogging zone and it can be determined only by actual test.

Again, leading fogger system manufacturers have tested their equipment and are able to supply this information.

Addenda

Table 148-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entro-py Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
-80	0.0000049	9.553	0.000	9.553	-19.221	0.005	-19.215	-0.04594	0.00001	-0.04592	-193.45	-0.4067	0.000236	-80
-79	0.0000053	9.579	0.000	9.579	-18.980	0.005	-18.975	-0.04531	0.00002	-0.04529	-193.06	-0.4056	0.000255	-79
-78	0.0000057	9.604	0.000	9.604	-18.740	0.006	-18.734	-0.04468	0.00002	-0.04466	-192.66	-0.4046	0.000275	-78
-77	0.0000062	9.629	0.000	9.629	-18.500	0.007	-18.493	-0.04405	0.00002	-0.04403	-192.27	-0.4036	0.000296	-77
-76	0.0000067	9.655	0.000	9.655	-18.259	0.007	-18.252	-0.04342	0.00002	-0.04340	-191.87	-0.4025	0.000319	-76
-75	0.0000072	9.680	0.000	9.680	-18.019	0.007	-18.011	-0.04279	0.00002	-0.04277	-191.47	-0.4015	0.000344	-75
-74	0.0000078	9.705	0.000	9.705	-17.778	0.008	-17.770	-0.04217	0.00002	-0.04215	-191.07	-0.4005	0.000371	-74
-73	0.0000084	9.731	0.000	9.731	-17.538	0.009	-17.529	-0.04155	0.00002	-0.04152	-190.68	-0.3994	0.000400	-73
-72	0.0000090	9.756	0.000	9.756	-17.298	0.010	-17.288	-0.04093	0.00003	-0.04090	-190.27	-0.3984	0.000430	-72
-71	0.0000097	9.781	0.000	9.782	-17.057	0.010	-17.047	-0.04031	0.00003	-0.04028	-189.87	-0.3974	0.000463	-71
-70	0.0000104	9.807	0.000	9.807	-16.806	0.011	-16.817	-0.03969	0.00003	-0.03966	-189.47	-0.3963	0.000498	-70
-69	0.0000112	9.832	0.000	9.832	-16.577	0.012	-16.565	-0.03907	0.00003	-0.03904	-189.07	-0.3953	0.000536	-69
-68	0.0000120	9.857	0.000	9.858	-16.336	0.013	-16.324	-0.03846	0.00003	-0.03843	-188.66	-0.3943	0.000576	-68
-67	0.0000129	9.883	0.000	9.883	-16.096	0.013	-16.083	-0.03785	0.00004	-0.03781	-188.26	-0.3932	0.000619	-67
-66	0.0000139	9.908	0.000	9.908	-15.856	0.015	-15.841	-0.03724	0.00004	-0.03720	-187.85	-0.3922	0.000665	-66
-65	0.0000149	9.933	0.000	9.934	-15.616	0.015	-15.600	0.03663	0.00004	-0.03659	-187.44	-0.3912	0.000714	-65
-64	0.0000160	9.959	0.000	9.959	-15.375	0.017	-15.359	-0.03602	0.00005	-0.03597	-187.04	-0.3901	0.000766	-64
-63	0.0000172	9.984	0.000	9.984	-15.117	0.018	-15.135	-0.03541	0.00005	-0.03536	-186.63	-0.3891	0.000822	-63
-62	0.0000184	10.009	0.000	10.010	-14.895	0.019	-14.876	-0.03481	0.00005	-0.03476	-186.22	-0.3881	0.000882	-62
-61	0.0000198	10.035	0.000	10.035	-14.654	0.021	-14.634	-0.03420	0.00006	-0.03415	-185.81	-0.3870	0.000945	-61
-60	0.0000212	10.060	0.000	10.060	-14.414	0.022	-14.392	-0.03360	0.00006	-0.03354	-185.39	-0.3860	0.001013	-60
-59	0.0000227	10.085	0.000	10.086	-14.174	0.024	-14.150	-0.03300	0.00006	-0.03294	-184.98	-0.3850	0.001086	-59
-58	0.0000243	10.111	0.000	10.111	-13.933	0.025	-13.908	-0.03240	0.00007	-0.03233	-184.57	-0.3839	0.001163	-58
-57	0.0000260	10.136	0.000	10.137	-13.693	0.027	-13.666	-0.03180	0.00007	-0.03173	-184.15	-0.3829	0.001246	-57
-56	0.0000279	10.161	0.000	10.162	-13.453	0.029	-13.424	-0.03121	0.00008	-0.03113	-183.74	-0.3819	0.001333	-56
-55	0.0000298	10.187	0.000	10.187	-13.213	0.031	-13.182	-0.03061	0.00008	-0.03053	-183.32	-0.3808	0.001427	-55
-54	0.0000319	10.212	0.001	10.213	-12.972	0.033	-12.939	-0.03002	0.00009	-0.02993	-182.90	-0.3798	0.001526	-54
-53	0.0000341	10.237	0.001	10.238	-12.732	0.035	-12.697	-0.02943	0.00009	-0.02934	-182.48	-0.3788	0.001632	-53
-52	0.0000365	10.263	0.001	10.263	-12.492	0.038	-12.454	-0.02884	0.00010	-0.02874	-182.06	-0.3778	0.001745	-52
-51	0.0000390	10.288	0.001	10.289	-12.251	0.041	-12.211	-0.02825	0.00011	-0.02814	-181.64	-0.3767	0.001865	-51
-50	0.0000416	10.313	0.001	10.314	-12.011	0.043	-11.968	-0.02766	0.00011	-0.02755	-181.22	-0.3757	0.001992	-50
-49	0.0000445	10.339	0.001	10.340	-11.771	0.046	-11.725	-0.02708	0.00012	-0.02696	-180.80	-0.3747	0.002128	-49
-48	0.0000475	10.364	0.001	10.365	-11.531	0.050	-11.481	-0.02649	0.00013	-0.02636	-180.37	-0.3736	0.002272	-48
-47	0.0000507	10.389	0.001	10.390	-11.290	0.053	-11.237	-0.02591	0.00014	-0.02577	-179.95	-0.3726	0.002425	-47
-46	0.0000541	10.415	0.001	10.416	-11.050	0.056	-10.994	-0.02533	0.00015	-0.02518	-179.52	-0.3716	0.002587	-46

Table 149-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entro-py Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
-45	0.0000577	10.440	0.001	10.441	-10.810	0.060	-10.750	-0.02475	0.00016	-0.02459	-179.10	-0.3705	0.002760	-45
-44	0.0000615	10.465	0.001	10.466	-10.570	0.064	-10.505	-0.02417	0.00017	-0.02400	-178.67	-0.3695	0.002943	-44
-43	0.0000656	10.491	0.001	10.492	-10.329	0.068	-10.261	-0.02359	0.00018	-0.02342	-178.24	-0.3685	0.003137	-43
-42	0.0000699	10.516	0.001	10.517	-10.089	0.073	-10.016	-0.02302	0.00019	-0.02283	-177.81	-0.3675	0.003343	-42
-41	0.0000744	10.541	0.001	10.543	-9.849	0.078	-9.771	-0.02244	0.00020	-0.02224	-177.38	-0.3664	0.003562	-41
-40	0.0000793	10.567	0.001	10.568	-9.609	0.083	-9.526	-0.02187	0.00021	-0.02166	-176.95	-0.3654	0.003793	-40
-39	0.0000844	10.592	0.001	10.593	-9.368	0.088	-9.280	-0.02130	0.00022	-0.02107	-176.52	-0.3644	0.004039	-39
-38	0.0000898	10.617	0.002	10.619	-9.128	0.094	-9.034	-0.02073	0.00024	-0.02049	-176.08	-0.3633	0.004299	-38
-37	0.0000956	10.643	0.002	10.644	-8.888	0.100	-8.788	-0.02016	0.00025	-0.01991	-175.65	-0.3623	0.004575	-37
-36	0.0001017	10.668	0.002	10.670	-8.648	0.106	-8.541	-0.01959	0.00027	-0.01932	-175.21	-0.3613	0.004866	-36
-35	0.0001081	10.693	0.002	10.695	-8.407	0.113	-8.294	-0.01902	0.00028	-0.01874	-174.78	-0.3603	0.005175	-35
-34	0.0001150	10.719	0.002	10.721	-8.167	0.120	-8.047	-0.01846	0.00030	-0.01816	-174.34	-0.3529	0.005502	-34
-33	0.0001222	10.744	0.002	10.746	-7.927	0.128	-7.799	-0.01790	0.00032	-0.01758	-173.90	-0.3582	0.005848	-33
-32	0.0001298	10.769	0.002	10.772	-7.687	0.136	-7.551	-0.01733	0.00034	-0.01699	-173.46	-0.3572	0.006214	-32
-31	0.0001379	10.795	0.002	10.797	-7.447	0.145	-7.302	-0.01677	0.00036	-0.01641	-173.02	-0.3561	0.006601	-31
-30	0.0001465	10.820	0.003	10.822	-7.206	0.154	-7.053	-0.01621	0.00038	-0.01583	-172.58	-0.3551	0.007009	-30
-29	0.0001555	10.845	0.003	10.848	-6.966	0.163	-6.803	-0.01565	0.00040	-0.01525	-172.14	-0.3541	0.007442	-29
-28	0.0001650	10.871	0.003	10.873	-6.726	0.173	-6.553	-0.01510	0.00043	-0.01467	-171.70	-0.3531	0.007898	-28
-27	0.0001751	10.896	0.003	10.899	-6.486	0.184	-6.302	-0.01454	0.00045	-0.01409	-171.25	-0.3520	0.008381	-27
-26	0.0001858	10.921	0.003	10.924	-6.245	0.195	-6.051	-0.01399	0.00048	-0.01351	-170.81	-0.3510	0.008890	-26
-25	0.0001970	10.947	0.003	10.950	-6.005	0.207	-5.798	-0.01343	0.00051	-0.01293	-170.36	-0.3500	0.009428	-25
-24	0.0002088	10.972	0.004	10.976	-5.765	0.220	-5.545	-0.01288	0.00054	-0.01235	-169.92	-0.3489	0.009995	-24
-23	0.0002214	10.997	0.004	11.001	-5.525	0.233	-5.292	-0.01233	0.00057	-0.01176	-169.47	-0.3479	0.010594	-23
-22	0.0002346	11.022	0.004	11.027	-5.284	0.247	-5.038	-0.01178	0.00060	-0.01118	-169.02	-0.3469	0.011226	-22
-21	0.0002485	11.048	0.004	11.052	-5.044	0.261	-4.783	-0.01123	0.00063	-0.01060	-168.57	-0.3459	0.011893	-21
-20	0.0002632	11.073	0.005	11.078	-4.804	0.277	-4.527	-0.01069	0.00067	-0.01002	-168.12	-0.3448	0.012595	-20
-19	0.0002786	11.098	0.005	11.103	-4.564	0.293	-4.271	-0.01014	0.00071	-0.00943	-167.67	-0.3438	0.013336	-19
-18	0.0002950	11.124	0.005	11.129	-4.324	0.311	-4.013	-0.00960	0.00075	-0.00885	-167.21	-0.3428	0.014117	-18
-17	0.0003121	11.149	0.006	11.155	-4.084	0.329	-3.754	-0.00905	0.00079	-0.00826	-166.76	-0.3418	0.014939	-17
-16	0.0003303	11.174	0.006	11.180	-3.843	0.348	-3.495	-0.00851	0.00083	-0.00768	-166.30	-0.3407	0.015806	-16
-15	0.0003493	11.200	0.006	11.206	-3.603	0.368	-3.235	-0.00797	0.00088	-0.00709	-165.85	-0.3397	0.016718	-15
-14	0.0003694	11.225	0.007	11.232	-3.363	0.390	-2.973	-0.00743	0.00093	-0.00650	-165.39	-0.3387	0.017679	-14
-13	0.0003905	11.250	0.007	11.257	-3.123	0.412	-2.710	-0.00689	0.00098	-0.00591	-164.93	-0.3377	0.018690	-13
-12	0.0004128	11.276	0.007	11.283	-2.882	0.436	-2.447	-0.00635	0.00103	-0.00532	-164.47	-0.3366	0.019754	-12
-11	0.0004362	11.301	0.008	11.309	-2.642	0.460	-2.182	-0.00582	0.00109	-0.00473	-164.01	-0.3356	0.020873	-11

Table 150-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entro-py Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
-10	0.0004608	11.326	0.008	11.335	-2.402	0.487	-1.915	-0.00528	0.00115	-0.00414	-163.55	-0.3346	0.022050	-10
-9	0.0004867	11.351	0.009	11.360	-2.162	0.514	-1.647	-0.00475	0.00121	-0.00354	-163.09	-0.3335	0.023289	-9
-8	0.0005139	11.377	0.009	11.386	-1.922	0.543	-1.378	-0.00422	0.00127	-0.00294	-162.63	-0.3325	0.024591	-8
-7	0.0005425	11.402	0.010	11.412	-1.681	0.574	-1.108	-0.00369	0.00134	-0.00234	-162.17	-0.3315	0.025959	-7
-6	0.0005726	11.427	0.010	11.438	-1.441	0.606	-0.835	-0.00316	0.00141	-0.00174	-161.70	-0.3305	0.027397	-6
-5	0.0006041	11.453	0.011	11.464	-1.201	0.640	-0.561	-0.00263	0.00149	-0.00114	-161.23	-0.3294	0.028907	-5
-4	0.0006373	11.478	0.012	11.490	-0.961	0.675	-0.286	-0.00210	0.00157	-0.00053	-160.77	-0.3284	0.030494	-4
-3	0.0006722	11.503	0.012	11.516	-0.721	0.712	-0.008	-0.00157	0.00165	0.00008	-160.30	-0.3274	0.032160	-3
-2	0.0007088	11.529	0.013	11.542	-0.480	0.751	0.271	-0.00105	0.00174	0.00069	-159.83	-0.3264	0.033909	-2
-1	0.0007472	11.554	0.014	11.568	-0.240	0.792	0.552	-0.00052	0.00183	0.00130	-159.36	-0.3253	0.035744	-1
0	0.0007875	11.579	0.015	11.594	0.000	0.835	0.835	0.00000	0.00192	0.00192	-158.89	-0.3243	0.037671	0
1	0.0008298	11.604	0.015	11.620	0.240	0.880	1.121	0.00052	0.00202	0.00254	-158.42	-0.3233	0.039694	1
2	0.0008742	11.630	0.016	11.646	0.480	0.928	1.408	0.00104	0.00212	0.00317	-157.95	-0.3223	0.041814	2
3	0.0009207	11.655	0.017	11.672	0.721	0.978	1.699	0.00156	0.00223	0.00380	-157.47	-0.3212	0.044037	3
4	0.0009695	11.680	0.018	11.699	0.961	1.030	1.991	0.00208	0.00235	0.00443	-157.00	-0.3202	0.046370	4
5	0.0010207	11.706	0.019	11.725	1.201	1.085	2.286	0.00260	0.00247	0.00506	-156.52	-0.3192	0.048814	5
6	0.0010743	11.731	0.020	11.751	1.441	1.143	2.584	0.00311	0.00259	0.00570	-156.05	-0.3182	0.051375	6
7	0.0011306	11.756	0.021	11.778	1.681	1.203	2.884	0.00363	0.00635	0.00272	-155.57	-0.3171	0.054060	7
8	0.0011895	11.782	0.022	11.804	1.922	1.266	3.188	0.00414	0.00286	0.00700	-155.09	-0.3161	0.056872	8
9	0.0012512	11.807	0.024	11.831	2.162	1.332	3.494	0.00466	0.00300	0.00766	-154.61	-0.3151	0.059819	9
10	0.0013158	11.832	0.025	11.857	2.402	1.402	3.804	0.00517	0.00315	0.00832	-154.13	-0.3141	0.062901	10
11	0.0013835	11.857	0.026	11.884	2.642	1.474	4.117	0.00568	0.00330	0.00898	-153.65	-0.3130	0.066131	11
12	0.0014544	11.883	0.028	11.910	2.882	1.550	4.433	0.00619	0.00347	0.00966	-153.17	-0.3120	0.069511	12
13	0.0015286	11.908	0.029	11.937	3.123	1.630	4.753	0.00670	0.00364	0.01033	-152.68	-0.3110	0.073049	13
14	0.0016062	11.933	0.031	11.964	3.363	1.714	5.077	0.00721	0.00381	0.01102	-152.20	-0.3100	0.076751	14
15	0.0016874	11.959	0.032	11.991	3.603	1.801	5.404	0.00771	0.00400	0.01171	-151.71	-0.3089	0.080623	15
16	0.0017724	11.984	0.034	12.018	3.843	1.892	5.736	0.00822	0.00419	0.01241	-151.22	-0.3079	0.084673	16
17	0.0018613	12.009	0.036	12.045	4.084	1.988	6.072	0.00872	0.00439	0.01312	-150.74	-0.3069	0.088907	17
18	0.0019543	12.035	0.038	12.072	4.324	2.088	6.412	0.00923	0.00460	0.01383	-150.25	-0.3059	0.093334	18
19	0.0020515	12.060	0.040	12.099	4.564	2.193	6.757	0.00973	0.00482	0.01455	-149.76	-0.3049	0.097962	19
20	0.0021531	12.085	0.042	12.127	4.804	2.303	7.107	0.01023	0.00505	0.01528	-149.27	-0.3038	0.102798	20
21	0.0022592	12.110	0.044	12.154	5.044	2.417	7.462	0.01073	0.00529	0.01602	-148.78	-0.3028	0.107849	21
22	0.0023703	12.136	0.046	12.182	5.285	2.537	7.822	0.01123	0.00554	0.01677	-148.28	-0.3018	0.113130	22
23	0.0024863	12.161	0.048	12.209	5.525	2.662	8.187	0.01173	0.00580	0.01753	-147.79	-0.3008	0.118645	23
24	0.0026073	12.186	0.051	12.237	5.765	2.793	8.558	0.01223	0.00607	0.01830	-147.30	-0.2997	0.124396	24

Table 151-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			Enthalpy Btu/lb	Entropy Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
25	0.0027339	12.212	0.054	12.265	6.005	2.930	8.935	0.01272	0.00636	0.01908	-146.80	-0.2987	0.130413	25
26	0.0028660	12.237	0.056	12.293	6.246	3.073	9.318	0.01322	0.00665	0.01987	-146.30	-0.2977	0.136684	26
27	0.0030039	12.262	0.059	12.321	6.486	3.222	9.708	0.01371	0.00696	0.02067	-145.81	-0.2967	0.143233	27
28	0.0031480	12.287	0.062	12.349	6.726	3.378	10.104	0.01420	0.00728	0.02148	-145.31	-0.2956	0.150066	28
29	0.0032984	12.313	0.065	12.378	6.966	3.541	10.507	0.01470	0.00761	0.02231	-144.81	-0.2946	0.157198	29
30	0.0034552	12.338	0.068	12.406	7.206	3.711	10.917	0.01519	0.00796	0.02315	-144.31	-0.2936	0.164631	30
31	0.0036190	12.363	0.072	12.435	7.447	3.888	11.335	0.01568	0.00832	0.02400	-143.80	-0.2926	0.172390	31
32	0.0037895	12.389	0.075	12.464	7.687	4.073	11.760	0.01617	0.00870	0.02487	-143.30	-0.2915	0.180479	32
33	0.003947	12.414	0.079	12.492	7.927	4.243	12.170	0.01665	0.00905	0.02570	1.03	0.0020	0.18791	33
34	0.004109	12.439	0.082	12.521	8.167	4.420	12.587	0.01714	0.00940	0.02655	2.04	0.0041	0.19559	34
35	0.004277	12.464	0.085	12.550	8.408	4.603	13.010	0.01763	0.00977	0.02740	3.05	0.0061	0.20356	35
36	0.004452	12.490	0.089	12.579	8.648	4.793	13.441	0.01811	0.01016	0.02827	4.05	0.0081	0.21181	36
37	0.004633	12.515	0.093	12.608	8.888	4.990	13.878	0.01860	0.01055	0.02915	5.06	0.0102	0.22035	37
38	0.004820	12.540	0.097	12.637	9.128	5.194	14.322	0.01908	0.01096	0.03004	6.06	0.0122	0.22920	38
39	0.005014	12.566	0.101	12.667	9.369	5.405	14.773	0.01956	0.01139	0.03095	7.07	0.0142	0.23835	39
40	0.005216	12.591	0.105	12.696	9.609	5.624	15.233	0.02004	0.01183	0.03187	8.07	0.0162	0.24784	40
41	0.005424	12.616	0.110	12.726	9.849	5.851	15.700	0.02052	0.01228	0.03281	9.08	0.0182	0.25765	41
42	0.005640	12.641	0.114	12.756	10.089	6.086	16.175	0.02100	0.01275	0.03375	10.08	0.0202	0.26781	42
43	0.005863	12.667	0.119	12.786	10.330	6.330	16.660	0.02148	0.01324	0.03472	11.09	0.0222	0.27831	43
44	0.006094	12.692	0.124	12.816	10.570	6.582	17.152	0.02196	0.01374	0.03570	12.09	0.0242	0.28918	44
45	0.006334	12.717	0.129	12.846	10.810	6.843	17.653	0.02244	0.01426	0.03669	13.09	0.0262	0.30042	45
46	0.006581	12.743	0.134	12.877	11.050	7.114	18.164	0.02291	0.01479	0.03770	14.10	0.0282	0.31206	46
47	0.006838	12.768	0.140	12.908	11.291	7.394	18.685	0.02339	0.01534	0.03873	15.10	0.0302	0.32408	47
48	0.007103	12.793	0.146	12.939	11.531	7.684	19.215	0.02386	0.01592	0.03978	16.10	0.0321	0.33651	48
49	0.007378	12.818	0.152	12.970	11.771	7.984	19.756	0.02433	0.01651	0.04084	17.10	0.0341	0.34937	49
50	0.007661	12.844	0.158	13.001	12.012	8.295	20.306	0.02480	0.01712	0.04192	18.11	0.0361	0.36264	50
51	0.007955	12.869	0.164	13.033	12.252	8.616	20.868	0.02528	0.01775	0.04302	19.11	0.0381	0.37636	51
52	0.008259	12.894	0.171	13.065	12.492	8.949	21.441	0.02575	0.01840	0.04415	20.11	0.0400	0.39054	52
53	0.008573	12.920	0.178	13.097	12.732	9.293	22.025	0.02622	0.01907	0.04529	21.11	0.0420	0.40518	53
54	0.008897	12.945	0.185	13.129	12.973	9.648	22.621	0.02668	0.01976	0.04645	22.11	0.0439	0.42030	54
55	0.009233	12.970	0.192	13.162	13.213	10.016	23.229	0.02715	0.02048	0.04763	23.11	0.0459	0.43592	55
56	0.009580	12.995	0.200	13.195	13.453	10.397	23.850	0.02762	0.02122	0.04884	24.11	0.0478	0.45205	56
57	0.009938	13.021	0.207	13.228	13.694	10.790	24.484	0.02808	0.02198	0.05006	25.11	0.0497	0.46870	57
58	0.010309	13.046	0.216	13.262	13.934	11.197	25.131	0.02855	0.02277	0.05132	26.11	0.0517	0.48589	58
59	0.010692	13.071	0.224	13.295	14.174	11.618	25.792	0.02901	0.02358	0.05259	27.11	0.0536	0.50363	59

Table 152-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entro-py Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
60	0.011087	13.096	0.233	13.329	14.415	12.052	26.467	0.02947	0.02442	0.05389	28.11	0.0555	0.52193	60
61	0.011496	13.122	0.242	13.364	14.655	12.502	27.157	0.02994	0.02528	0.05522	29.12	0.0575	0.54082	61
62	0.011919	13.147	0.251	13.398	14.895	12.966	27.862	0.03040	0.02617	0.05657	30.11	0.0594	0.56032	62
63	0.012355	13.172	0.261	13.433	15.135	13.446	28.582	0.03086	0.02709	0.05795	31.11	0.0613	0.58041	63
64	0.012805	13.198	0.271	13.468	15.376	13.942	29.318	0.03132	0.02804	0.05936	32.11	0.0632	0.60113	64
65	0.013270	13.223	0.281	13.504	15.616	14.454	30.071	0.03178	0.02902	0.06080	33.11	0.0651	0.62252	65
66	0.013750	13.248	0.292	13.540	15.856	14.983	30.840	0.03223	0.03003	0.06226	34.11	0.0670	0.64454	66
67	0.014246	13.273	0.303	13.577	16.097	15.530	31.626	0.03269	0.03107	0.06376	35.11	0.0689	0.66725	67
68	0.014758	13.299	0.315	13.613	16.337	16.094	32.431	0.03315	0.03214	0.06529	36.11	0.0708	0.69065	68
69	0.015286	13.324	0.326	13.650	16.577	16.677	33.254	0.03360	0.03325	0.06685	37.11	0.0727	0.71479	69
70	0.015832	13.349	0.339	13.688	16.818	17.279	34.097	0.03406	0.03438	0.06844	38.11	0.0746	0.73966	70
71	0.016395	13.375	0.351	13.726	17.058	17.901	34.959	0.03451	0.03556	0.07007	39.11	0.0765	0.76567	71
72	0.016976	13.400	0.365	13.764	17.299	18.543	35.841	0.03496	0.03677	0.07173	40.11	0.0783	0.79167	72
73	0.017575	13.425	0.378	13.803	17.539	19.204	36.743	0.03541	0.03801	0.07343	41.11	0.0802	0.81882	73
74	0.018194	13.450	0.392	13.843	17.779	19.889	37.668	0.03586	0.03930	0.07516	42.11	0.0821	0.84684	74
75	0.018833	13.476	0.407	13.882	18.020	20.595	38.615	0.03631	0.04062	0.07694	43.11	0.0840	0.87567	75
76	0.019491	13.501	0.422	13.923	18.260	21.323	39.583	0.03676	0.04199	0.07875	44.10	0.0858	0.90533	76
77	0.020170	13.526	0.437	13.963	18.500	22.075	40.576	0.03721	0.04339	0.08060	45.10	0.0877	0.93589	77
78	0.020871	13.551	0.453	14.005	18.741	22.851	41.592	0.03766	0.04484	0.08250	46.10	0.0896	0.96733	78
79	0.021594	13.577	0.470	14.046	18.981	23.652	42.633	0.03811	0.04633	0.08444	47.10	0.0914	0.99970	79
80	0.022340	13.602	0.487	14.089	19.222	24.479	43.701	0.03855	0.04787	0.08642	48.10	0.0933	1.03302	80
81	0.023109	13.627	0.505	14.132	19.462	25.332	44.794	0.03900	0.04945	0.08844	49.10	0.0951	1.06728	81
82	0.023902	13.653	0.523	14.175	19.702	26.211	45.913	0.03944	0.05108	0.09052	50.10	0.0970	1.10252	82
83	0.024720	13.678	0.542	14.220	19.943	27.120	47.062	0.03988	0.05276	0.09264	51.09	0.0988	1.13882	83
84	0.025563	13.703	0.561	14.264	20.183	28.055	48.238	0.04033	0.05448	0.09481	52.09	0.1006	1.17608	84
85	0.026433	13.728	0.581	14.310	20.424	29.021	49.445	0.04077	0.05626	0.09703	53.09	0.1025	1.21445	85
86	0.027329	13.754	0.602	14.356	20.664	30.017	50.681	0.04121	0.05809	0.09930	54.09	0.1043	1.25388	86
87	0.028254	13.779	0.624	14.403	20.905	31.045	51.949	0.04165	0.05998	0.10163	55.09	0.1061	1.29443	87
88	0.029208	13.804	0.646	14.450	21.145	32.105	53.250	0.04209	0.06192	0.10401	56.09	0.1080	1.33613	88
89	0.030189	13.829	0.669	14.498	21.385	33.197	54.582	0.04253	0.06392	0.10645	57.09	0.1098	1.37893	89
90	0.031203	13.855	0.692	14.547	21.626	34.325	55.951	0.04297	0.06598	0.10895	58.08	0.1116	1.42298	90
91	0.032247	13.880	0.717	14.597	21.866	35.489	57.355	0.04340	0.06810	0.11150	59.08	0.1134	1.46824	91
92	0.033323	13.905	0.742	14.647	22.107	36.687	58.794	0.04384	0.07028	0.11412	60.08	0.1152	1.51471	92
93	0.034433	13.930	0.768	14.699	22.347	37.924	60.271	0.04427	0.07253	0.11680	61.08	0.1170	1.56248	93
94	0.035577	13.956	0.795	14.751	22.588	39.199	61.787	0.04471	0.07484	0.11955	62.08	0.1188	1.61154	94

Table 153-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entro-py Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
95	0.036757	13.981	0.823	14.804	22.828	40.515	63.343	0.04514	0.07722	0.12237	63.08	0.1206	1.66196	95
96	0.037972	14.006	0.852	14.858	23.069	41.871	64.940	0.04558	0.07968	1.12525	64.07	0.1224	1.71372	96
97	0.039225	14.032	0.881	14.913	23.309	43.269	66.578	0.04601	0.08220	0.12821	65.07	0.1242	1.76685	97
98	0.040516	14.057	0.912	14.969	23.550	44.711	68.260	0.04644	0.08480	0.13124	66.07	0.1260	1.82141	98
99	0.041848	14.082	0.944	15.026	23.790	46.198	69.988	0.04687	0.08747	0.13434	67.07	0.1278	1.87745	99
100	0.043219	14.107	0.976	15.084	24.031	47.730	71.761	0.04730	0.09022	0.13752	68.07	0.1296	1.93492	100
101	0.044634	14.133	1.010	15.143	24.271	49.312	73.583	0.04773	0.09306	0.14079	69.07	0.1314	1.99396	101
102	0.046090	14.158	1.045	15.203	24.512	50.940	75.452	0.04816	0.09597	0.14413	70.06	0.1332	2.05447	102
103	0.047592	14.183	1.081	15.264	24.752	52.621	77.373	0.04859	0.09897	0.14756	71.06	0.1349	2.11661	103
104	0.049140	14.208	1.118	15.326	24.993	54.354	79.346	0.07901	0.10206	0.15108	72.06	0.1367	2.18037	104
105	0.050737	14.234	1.156	15.390	25.233	56.142	81.375	0.04944	0.10525	0.15469	73.06	0.1385	2.24581	105
106	0.052383	14.259	1.196	15.455	25.474	57.986	83.460	0.04987	0.10852	0.15839	74.06	0.1402	2.31297	106
107	0.054077	14.284	1.236	15.521	25.714	59.884	85.599	0.05029	0.11189	0.16218	75.06	0.1420	2.38173	107
108	0.055826	14.309	1.279	15.588	25.955	61.844	87.799	0.05071	0.11537	0.16608	76.05	0.1438	2.45232	108
109	0.057628	14.335	1.322	15.657	26.195	63.866	90.061	0.05114	0.11894	0.17008	77.05	0.1455	2.52473	109
110	0.059486	14.360	1.367	15.727	26.436	65.950	92.386	0.05156	0.12262	0.17418	78.05	0.1473	2.59891	110
111	0.061401	14.385	1.414	15.799	26.677	68.099	94.776	0.05198	0.12641	0.17839	79.05	0.1490	2.67500	111
112	0.063378	14.411	1.462	15.872	26.917	70.319	97.237	0.05240	0.13032	0.18272	80.05	0.1508	2.75310	112
113	0.065411	14.436	1.511	15.947	27.158	72.603	99.760	0.05282	0.13434	0.18716	81.05	0.1525	2.83291	113
114	0.067512	14.461	1.562	16.023	27.398	74.964	102.362	0.05324	0.13847	0.19172	82.04	0.1543	2.91491	114
115	0.069676	14.486	1.615	16.101	27.639	77.396	105.035	0.05366	0.14274	0.19640	83.04	0.1560	2.99883	115
116	0.071908	14.512	1.670	16.181	27.879	79.906	107.786	0.05408	0.14713	0.20121	84.04	0.1577	3.08488	116
117	0.074211	14.537	1.726	16.263	28.120	82.497	110.617	0.05450	0.15165	0.20615	85.04	0.1595	3.17305	117
118	0.076586	14.562	1.784	16.346	28.361	85.169	113.530	0.05492	0.15631	0.21122	86.04	0.1612	3.26335	118
119	0.079036	14.587	1.844	16.432	28.601	87.927	116.528	0.05533	0.16111	0.21644	87.04	0.1629	3.35586	119
120	0.081560	14.613	1.906	16.519	28.842	90.770	119.612	0.05575	0.16605	0.22180	88.04	0.1647	3.45052	120
121	0.084169	14.638	1.971	16.609	29.083	93.709	122.792	0.05616	0.17115	0.22731	89.04	0.1664	3.54764	121
122	0.086860	14.663	2.037	16.700	29.323	96.742	126.065	0.05658	0.17640	0.23298	90.03	0.1681	3.64704	122
123	0.089633	14.688	2.106	16.794	29.564	99.868	129.432	0.05699	0.18181	0.23880	91.03	0.1698	3.74871	123
124	0.092500	14.714	2.176	16.890	29.805	103.102	132.907	0.05740	0.18739	0.24480	92.03	0.1715	3.85298	124
125	0.095456	14.739	2.250	16.989	30.045	106.437	136.482	0.05781	0.19314	0.25096	93.03	0.1732	2.95961	125
126	0.098504	14.764	2.325	17.090	30.286	109.877	140.163	0.05823	0.19907	0.25729	94.03	0.1749	4.06863	126
127	0.101657	14.789	2.404	17.193	30.527	113.438	143.965	0.05864	0.20519	0.26382	95.03	0.1766	4.18046	127
128	0.104910	14.815	2.485	17.299	30.767	117.111	147.878	0.05905	0.21149	0.27054	96.03	0.1783	4.29477	128
129	0.108270	14.840	2.569	17.409	31.008	120.908	151.916	0.21800	0.21810	0.27745	97.03	0.1800	4.41181	129

Table 154-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entropy Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
130	0.111738	14.865	2.655	17.520	31.249	124.828	156.076	0.05986	0.22470	0.28457	98.03	0.1817	4.53148	130
131	0.115322	14.891	2.745	17.635	31.489	128.880	160.370	0.06027	0.23162	0.29190	99.02	0.1834	4.65397	131
132	0.119023	14.916	2.837	17.753	31.730	133.066	164.796	0.06068	0.23876	0.29944	100.02	0.1851	4.77919	132
133	0.122855	14.941	2.934	17.875	31.971	137.403	169.374	0.06109	0.24615	0.30723	101.02	0.1868	4.90755	133
134	0.126804	14.966	3.033	17.999	32.212	141.873	174.084	0.06149	0.25375	0.31524	102.02	0.1885	5.03844	134
135	0.130895	14.992	3.136	18.127	32.452	146.504	178.957	0.06190	0.26161	0.32351	103.02	0.1902	5.17258	135
136	0.135124	15.017	3.242	18.259	32.693	151.294	183.987	0.06230	0.26973	0.33203	104.02	0.1919	5.30973	136
137	0.139494	15.042	3.352	18.394	32.934	156.245	189.179	0.06271	0.27811	0.34082	105.02	0.1935	5.44985	137
138	0.144019	15.067	3.467	18.534	33.175	161.374	194.548	0.06311	0.28707	0.35018	106.02	0.1952	5.59324	138
139	0.148696	15.093	3.585	18.678	33.415	166.677	200.092	0.06351	0.29602	0.35954	107.02	0.1969	5.73970	139
140	0.153538	15.118	3.708	18.825	33.656	172.168	205.824	0.06391	0.30498	0.36890	108.02	0.1985	5.88945	140
141	0.158643	15.143	3.835	18.978	33.897	177.857	211.754	0.06431	0.31456	0.37887	109.02	0.2002	6.04256	141
142	0.163748	15.168	3.967	19.135	34.138	183.754	217.892	0.06471	0.32446	0.38918	110.02	0.2019	6.19918	142
143	0.169122	15.194	4.103	19.297	34.379	189.855	244.233	0.06511	0.33470	0.39981	111.02	0.2035	6.35898	143
144	0.174694	15.219	4.245	19.464	34.620	196.183	230.802	0.06551	0.34530	0.41081	112.02	0.2052	6.52241	144
145	0.180467	15.244	4.392	19.637	34.860	202.740	237.600	0.06591	0.35626	0.42218	113.02	0.2068	6.68932	145
146	0.186460	15.269	4.545	19.815	35.101	209.550	244.651	0.06631	0.36764	0.43395	114.02	0.2085	6.86009	146
147	0.192668	15.295	4.704	19.999	35.342	216.607	251.949	0.06671	0.37941	0.44611	115.02	0.2101	7.03435	147
148	0.199110	15.320	4.869	20.189	35.583	223.932	259.514	0.06710	0.39160	0.45871	116.02	0.2118	7.21239	148
149	0.205792	15.345	5.040	20.385	35.824	231.533	267.356	0.06750	0.40424	0.47174	117.02	0.2134	7.39413	149
150	0.212730	15.370	5.218	20.589	36.064	239.426	275.490	0.06790	0.41735	0.48524	118.02	0.2151	7.57977	150
151	0.219945	15.396	5.404	20.799	36.305	247.638	283.943	0.06829	0.43096	0.49925	119.02	0.2167	7.76958	151
152	0.227429	15.421	5.596	21.017	36.546	256.158	292.705	0.06868	0.44507	0.51375	120.02	0.2184	7.96306	152
153	0.235218	15.446	5.797	21.243	36.787	265.028	301.816	0.06908	0.45973	0.52881	121.02	0.2200	8.16087	153
154	0.243309	15.471	6.005	21.477	37.028	274.245	311.273	0.06947	0.47494	0.54441	122.02	0.2216	8.36256	154
155	0.251738	15.497	6.223	21.720	37.269	283.849	321.118	0.06986	0.49077	0.56064	123.02	0.2233	8.56871	155
156	0.260512	15.522	6.450	21.972	37.510	293.849	331.359	0.07025	0.50723	0.57749	124.02	0.2249	8.77915	156
157	0.269644	15.547	6.686	22.233	37.751	304.261	342.012	0.07065	0.52434	0.59499	125.02	0.2265	8.99378	157
158	0.279166	15.572	6.933	22.505	37.992	315.120	353.112	0.07104	0.54217	0.61320	126.02	0.2281	9.21297	158
159	0.289101	15.598	7.190	22.788	38.233	326.452	364.685	0.07143	0.56074	0.63216	127.02	0.2297	9.43677	159
161	0.31027	15.648	7.740	23.388	38.715	350.610	389.325	0.07220	0.60025	0.67245	129.02	0.2330	9.8978	161
162	0.32156	15.673	8.034	23.707	38.956	363.501	402.457	0.07259	0.62128	0.69388	130.03	0.2346	10.1353	162
163	0.33336	15.699	8.341	24.040	39.197	376.979	416.175	0.07298	0.64325	0.71623	131.03	0.2362	10.3776	163
164	0.34572	15.724	8.664	24.388	39.438	391.095	430.533	0.07337	0.66622	0.73959	132.03	0.2378	10.6250	164

Table 155-1:
Thermodynamic Properties of Moist Air, Standard Atmospheric Pressure, 14,696 psi (29.921 in. Hg)

Temp	Humidity Ratio lb_w/lb dry air	Volume ft3/lb dry air			Enthalpy Btu/lb dry air			Entropy Btu/(lb dry air) °F			En-thalpy Btu/lb	Entropy Btu/lb °F	Pressure in. Hg	Temp.
t, °F	W_s	v_a	v_{as}	v_s	h_a	h_{as}	h_s	s_a	s_{as}	s_s	h_w	s_w	p_s	°F
165	0.35865	15.749	9.001	24.750	39.679	405.865	445.544	0.07375	0.69022	0.76397	133.03	0.2394	10.8771	165
166	0.37220	15.774	9.355	25.129	39.920	421.352	461.271	0.07414	0.71535	0.78949	134.03	0.2410	11.1343	166
167	0.38639	15.800	9.726	25.526	40.161	437.578	477.739	0.07452	0.74165	0.81617	135.03	0.2426	11.3965	167
168	0.40131	15.825	10.117	25.942	40.402	454.630	495.032	0.07491	0.76925	0.84415	136.03	0.2442	11.6641	168
169	0.41698	15.850	10.527	26.377	40.643	472.554	513.197	0.07529	0.79521	0.87350	137.04	0.2458	11.9370	169
170	0.43343	15.875	10.959	26.834	40.884	491.372	532.256	0.07567	0.82858	0.90425	138.04	0.2474	12.2149	170
171	0.45079	15.901	11.414	27.315	41.125	511.231	552.356	0.07606	0.86058	0.93664	139.04	0.2490	12.4988	171
172	0.46905	15.926	11.894	27.820	41.366	532.138	573.504	0.07644	0.89423	0.97067	140.04	0.2506	12.7880	172
173	0.48829	15.951	12.400	28.352	41.607	554.160	595.767	0.07682	0.92962	1.00644	141.04	0.2521	13.0823	173
174	0.50867	15.976	12.937	28.913	41.848	577.489	619.337	0.07720	0.96707	1.04427	142.04	0.2537	13.3831	174
175	0.53019	16.002	13.504	29.505	42.089	602.139	644.229	0.07758	1.00657	1.08416	143.05	0.2553	13.6894	175
176	0.55294	16.027	14.103	30.130	42.331	628.197	670.528	0.07796	1.04828	1.12624	144.05	0.2569	14.0010	176
177	0.57710	16.052	14.741	30.793	42.572	655.876	698.448	0.07834	1.09253	1.17087	145.05	0.2585	14.3191	177
178	0.60274	16.078	15.418	31.496	42.813	685.260	728.073	0.07872	1.13943	1.21815	146.05	0.2600	14.6430	178
179	0.63002	16.103	16.139	32.242	43.054	716.524	759.579	0.07910	1.18927	1.26837	147.06	0.2616	14.9731	179
180	0.65911	16.128	16.909	33.037	43.295	749.871	793.166	0.07947	1.24236	1.32183	148.06	0.2632	15.3097	180
181	0.69012	16.153	17.730	33.883	43.536	785.426	828.962	0.07985	1.29888	1.37873	149.06	0.2647	15.6522	181
182	0.72331	16.178	18.609	34.787	43.778	823.487	867.265	0.08023	1.35932	1.43954	150.06	0.2663	16.0014	182
183	0.75885	16.204	19.551	35.755	44.019	864.259	908.278	0.08060	1.42396	1.50457	151.07	0.2679	16.3569	183
184	0.79703	16.229	20.564	36.793	44.260	908.061	952.321	0.08098	1.49332	1.57430	152.07	0.2694	16.7190	184
185	0.83817	16.254	21.656	37.910	44.501	955.261	999.763	0.08135	1.56797	1.64932	153.07	0.2710	17.0880	185
186	0.88251	16.280	22.834	39.113	44.742	1006.149	1050.892	0.08172	1.64834	1.73006	154.08	0.2725	17.4634	186
187	0.93057	16.305	24.111	40.416	44.984	1061.314	1106.298	0.08210	1.73534	1.81744	155.08	0.2741	17.8462	187
188	0.98272	16.330	25.498	41.828	45.225	1121.1754	1166.399	0.08247	1.82963	1.91210	156.08	0.2756	18.2357	188
189	1.03951	16.355	27.010	43.365	45.466	1186.382	1231.848	0.08284	1.93221	2.01505	157.09	0.2772	18.6323	189
190	1.10154	16.381	28.661	45.042	45.707	1257.614	1303.321	0.08321	2.04412	2.12733	158.09	0.2787	19.0358	190
191	1.16965	16.406	30.476	46.882	45.949	1335.834	1381.783	0.08359	2.16684	2.25043	159.09	0.2803	19.4468	191
192	1.24471	16.431	32.477	48.908	46.190	1422.047	1468.238	0.08396	2.30193	2.38589	160.10	0.2818	19.8652	192
193	1.32788	16.456	34.695	51.151	46.431	1517.581	1564.013	0.08433	2.45144	2.53576	161.10	0.2834	20.2913	193
194	1.42029	16.481	37.161	53.642	46.673	1623.758	1670.730	0.08470	2.61738	2.70208	162.11	0.2849	20.7244	194

Table 156-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
-80	0.000116	0.000236	0.01732	1953234	1953234	-193.50	1219.19	1025.69	-0.4067	3.2112	2.8045	-80
-79	0.000125	0.000254	0.01732	1814052	1814052	-193.11	1219.24	1026.13	-0.4056	3.2029	2.7972	-79
-78	0.000135	0.000275	0.01732	1685445	1685445	-192.71	1219.28	1026.57	-0.4046	3.1946	2.7900	-78
-77	0.000145	0.000296	0.01732	1566663	1566663	-192.31	1219.33	1027.02	-0.4036	3.1964	2.7828	-77
-76	0.000157	0.000319	0.01732	1456752	1456752	-191.92	1219.38	1027.46	-0.4025	3.1782	2.7757	-76
-75	0.000169	0.000344	0.01733	1355059	1355059	-191.52	1219.42	1027.90	-0.4015	3.1701	2.7685	-75
-74	0.000182	0.000371	0.01733	1260977	1260977	-191.12	1219.47	1028.34	-0.4005	3.1619	2.7615	-74
-73	0.000196	0.000399	0.01733	1173848	1173848	-190.72	1219.51	1028.79	-0.3994	3.1539	2.7544	-73
-72	0.000211	0.000430	0.01733	1093149	1093149	-190.32	1219.55	1029.23	-0.3984	3.1459	2.7475	-72
-71	0.000227	0.000463	0.01733	1018381	1018381	-189.92	1219.59	1029.67	-0.3974	3.1379	2.7405	-71
-70	0.000245	0.000498	0.01733	949067	949067	-189.52	1219.63	1030.11	-0.3963	3.1299	2.7336	-70
-69	0.000263	0.000536	0.01733	884803	884803	-189.11	1219.67	1030.55	-0.3953	3.1220	2.7267	-69
-68	0.000283	0.000576	0.01733	825187	825187	-188.71	1219.71	1031.00	-0.3943	3.1141	2.7199	-68
-67	0.000304	0.000619	0.01734	769864	769864	-188.30	1219.74	1031.44	-0.3932	3.1063	2.7131	-67
-66	0.000326	0.000664	0.01734	718508	718508	-187.90	1219.78	1031.88	-0.3922	3.0985	2.7063	-66
-65	0.000350	0.000714	0.01734	670800	670800	-187.49	1219.82	1032.32	-0.3912	3.0907	2.6996	-65
-64	0.000376	0.000766	0.01734	626503	626503	-187.08	1219.85	1032.77	-0.3901	3.0830	2.6929	-64
-63	0.000404	0.000822	0.01734	585316	585316	-186.67	1219.88	1033.21	-0.3891	3.0753	2.6862	-63
-62	0.000433	0.000882	0.01734	548041	547041	-186.26	1219.91	1033.65	-0.3881	3.0677	2.6730	-62
-61	0.000464	0.000945	0.01734	511446	511446	-185.85	1219.95	1034.09	-0.3870	3.0601	2.6730	-61
-60	0.000498	0.001013	0.01734	478317	478317	-185.44	1219.98	1034.54	-0.3860	3.0525	2.6665	-60
-59	0.000533	0.001086	0.01735	447495	447495	-185.03	1220.01	1034.98	-0.3850	3.0449	2.6600	-59
-58	0.000571	0.001163	0.01735	418803	418803	-184.61	1220.03	1035.42	-0.3839	3.0374	2.6535	-58
-57	0.000612	0.001246	0.01735	392068	392068	-184.20	1220.06	1035.86	-0.3829	3.0299	2.6470	-57
-56	0.000655	0.001333	0.01735	367172	367172	-183.78	1220.09	1036.30	-0.3819	3.0225	2.6406	-56
-55	0.000701	0.001427	0.01735	343970	343970	-183.37	1220.11	1036.75	-0.3808	3.0151	2.6342	-55
-54	0.000750	0.001526	0.01735	322336	322336	-182.95	1220.14	1037.19	-0.3798	3.0077	2.6279	-54
-53	0.000802	0.001632	0.01735	302157	302157	-182.53	1220.16	1037.63	-0.3788	3.0004	2.6216	-53
-52	0.000857	0.001745	0.01735	283335	283335	-182.11	1220.18	1038.07	-0.3778	2.9931	2.6153	-52
-51	0.000916	0.001865	0.01736	265773	265773	-181.69	1220.21	1038.52	-0.3767	2.9858	2.6091	-51
-50	0.000979	0.001992	0.01736	249381	249381	-181.27	1220.23	1038.96	-0.3757	2.9786	2.6029	-50
-49	0.001045	0.002128	0.01736	234067	234067	-180.85	1220.25	1039.40	-0.3747	2.9714	2.5967	-49
-48	0.001116	0.002272	0.01736	219766	219766	-180.42	1220.26	1039.84	-0.3736	2.9642	2.5906	-48
-47	0.001191	0.002425	0.01736	206398	206398	-180.00	1220.28	1040.28	-0.3726	2.9570	2.5844	-47
-46	0.001271	0.002587	0.01736	193909	193909	-179.57	1220.30	1040.73	-0.3716	2.9499	2.5784	-46
-45	0.001355	0.002760	0.01736	182231	182231	-179.14	1220.31	1041.17	-0.3705	2.9429	2.5723	-45
-44	0.001445	0.002943	0.01736	171304	171304	-178.72	1220.33	1041.61	-0.3695	2.9358	2.5663	-44

Table 157-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
-43	0.001541	0.003137	0.01737	161084	161084	-178.79	1220.34	1042.05	-0.3685	2.9288	2.5603	-43
-42	0.001642	0.003343	0.01737	151518	151518	-177.86	1220.36	1042.50	-0.3675	2.9218	2.5544	-42
-41	0.001749	0.003562	0.01737	142566	142566	-177.43	1220.37	1042.94	-0.3664	2.9149	2.5485	-41
-40	0.001863	0.003793	0.01737	134176	134176	-177.00	1220.38	1043.38	-0.3654	2.9080	2.5426	-40
-39	0.001984	0.004039	0.01737	126322	126322	-176.57	1220.39	1043.82	-0.3644	2.9011	2.5367	-39
-38	0.002111	0.004299	0.01737	118959	118959	-176.13	1220.40	1044.27	-0.3633	2.8942	2.5309	-38
-37	0.002247	0.004574	0.01737	112058	112058	-175.70	1220.40	1044.71	-0.3623	2.8874	2.5251	-37
-36	0.002390	0.004866	0.01738	105592	105592	-175.26	1220.42	1045.15	-0.3613	2.8806	2.5193	-36
-35	0.002542	0.005175	0.01738	99522	99522	-174.83	1220.42	1045.59	-0.3603	2.8738	2.5136	-35
-34	0.002702	0.005502	0.01738	93828	93828	-174.39	1220.42	1046.03	-0.3592	2.8671	2.5078	-34
-33	0.002872	0.005848	0.01738	88489	88489	-173.95	1220.43	1046.48	-0.3582	2.8604	2.5022	-33
-32	0.003052	0.006213	0.01738	83474	83474	-173.51	1220.43	1046.92	-0.3572	2.8537	2.4965	-32
-31	0.003242	0.006600	0.01738	78763	78763	-173.07	1220.43	1047.36	-0.3561	2.8470	2.4909	-31
-30	0.003443	0.007009	0.01738	74341	74341	-172.63	1220.43	1047.80	-0.3551	2.8404	2.4853	-30
-29	0.003655	0.007441	0.01738	70187	70187	-172.19	1220.43	1048.25	-0.3541	2.8338	2.4797	-29
-28	0.003879	0.007898	0.01739	66282	66282	-171.74	1220.43	1048.69	-0.3531	2.8272	2.4742	-28
-27	0.004116	0.008380	0.01739	62613	62613	-171.30	1220.43	1049.13	-0.3520	2.8207	2.4687	-27
-26	0.004366	0.008890	0.01739	59161	59161	-170.86	1220.43	1049.57	-0.3510	2.8142	2.4632	-26
-25	0.004630	0.009428	0.01739	55915	55915	-170.41	1220.42	1050.01	-0.3500	2.8077	2.4577	-25
-24	0.004909	0.009995	0.01739	52861	52861	-169.96	1220.42	1050.46	-0.3489	2.8013	2.4523	-24
-23	0.005203	0.010594	0.01739	49986	49986	-169.51	1220.41	1050.90	-0.3479	2.7948	2.4469	-23
-22	0.005514	0.011226	0.01739	47281	47281	-169.07	1220.41	1051.34	-0.3469	2.7884	2.4415	-22
-21	0.005841	0.011892	0.01740	44733	44733	-168.62	1220.40	1051.78	-0.3459	2.7820	2.4362	-21
-20	0.006186	0.012595	0.01740	42333	42333	-168.16	1220.39	1052.22	-0.3448	2.7757	2.4309	-20
-19	0.006550	0.013336	0.01740	40073	40073	-167.71	1220.38	1052.67	-0.3438	2.7694	2.4256	-19
-18	0.006933	0.014117	0.01740	37943	37943	-167.26	1220.37	1053.11	-0.3428	2.7631	2.4203	-18
-17	0.007337	0.014939	0.01740	35934	35934	-166.81	1220.36	1053.55	-0.3418	2.7568	2.4151	-17
-16	0.007763	0.015806	0.01740	34041	34041	-166.35	1220.34	1053.99	-0.3407	2.7506	2.4098	-16
-15	0.008211	0.016718	0.01740	32256	32256	-165.90	1220.33	1054.43	-0.3397	2.7444	2.4046	-15
-14	0.008683	0.017678	0.01741	30572	30572	-165.44	1220.31	1054.87	-0.3387	2.7382	2.3995	-14
-13	0.009179	0.018689	0.01741	28983	28983	-164.98	1220.30	1055.32	-0.3377	2.7320	2.3943	-13
-12	0.009702	0.019753	0.01741	27483	27483	-165.52	1220.28	1055.76	-0.3366	2.7259	2.3892	-12
-11	0.010252	0.020873	0.01741	26067	26067	-164.06	1220.26	1056.20	-0.3356	2.7197	2.3841	-11
-10	0.010830	0.022050	0.01741	24730	24730	-163.60	1220.24	1056.64	-0.3346	2.7136	2.3791	-10
-9	0.011438	0.023288	0.01741	23467	23467	-163.14	1220.22	1057.08	-0.3335	2.7076	2.3740	-9
-8	0.012077	0.024590	0.01741	22274	22274	-162.68	1220.20	1057.53	-0.3325	2.7015	2.3690	-8
-7	0.012749	0.025958	0.01742	21147	21147	-162.21	1220.18	1057.97	-0.3315	2.6955	2.3640	-7

Table 158-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
-6	0.013456	0.027396	0.01742	20081	20081	-162.75	1220.16	1058.41	-0.3305	2.6895	2.3951	-6
-5	0.014197	0.028906	0.01742	19074	19074	-161.28	1220.13	1058.85	-0.3294	2.6836	2.3541	-5
-4	0.014977	0.030493	0.01742	18121	18121	-160.82	1220.11	1059.29	-0.3284	2.6776	2.3492	-4
-3	0.015795	0.032159	0.01752	17220	17220	-160.35	1220.08	1059.73	-0.3274	2.6717	2.3443	-3
-2	0.011654	0.033908	0.01742	16367	16367	-159.88	1220.05	1060.17	-0.3264	2.6658	2.3394	-2
-1	0.017556	0.035744	0.01742	15561	15561	-159.41	1220.02	1060.62	-0.3253	2.6599	2.3346	-1
0	0.018502	0.037671	0.01743	14797	14797	-158.94	1220.00	1061.06	-0.3243	2.6541	2.3298	0
1	0.019495	0.039693	0.01743	14073	14073	-158.47	1219.96	1061.50	-0.3233	2.6482	2.3249	1
2	0.020537	0.041813	0.01743	13388	13388	-157.99	1219.93	1061.94	-0.3223	2.6424	2.3202	2
3	0.021629	0.044037	0.01743	12740	12740	-157.52	1219.90	1062.38	-0.3212	2.6367	2.3154	3
4	0.022774	0.046369	0.01743	12125	12125	-157.05	1219.87	1062.82	-0.3202	2.6309	2.3107	4
5	0.023975	0.048813	0.01743	11543	11543	-156.57	1219.83	1063.26	-0.3192	2.6252	2.3060	5
6	0.25233	0.051375	0.01743	10991	10991	-156.09	1219.80	1063.70	-0.3182	2.6194	2.3013	6
7	0.026552	0.054059	0.01744	10468	10468	-155.62	1219.76	1064.14	-0.3171	2.6138	2.2966	7
8	0.027933	0.056872	0.01744	9971	9971	-155.14	1219.72	1064.58	-0.3161	2.6081	2.2920	8
9	0.029379	0.059817	0.01744	9500	9500	-154.66	1219.68	1065.03	-0.3151	2.6024	2.2873	9
10	0.030894	0.062901	0.01744	9054	9054	-154.18	1219.64	1065.47	-0.3141	2.5968	2.2827	10
11	0.032480	0.066131	0.01744	8630	8630	-153.70	1219.60	1065.91	-0.3130	2.5912	2.2782	11
12	0.034140	0.069511	0.01744	8228	8228	-153.21	1219.56	1066.35	-0.3120	2.5856	2.2736	12
13	0.035878	0.073047	0.01745	7846	7846	-152.73	1219.52	1066.79	-0.3110	2.5801	2.2691	13
14	0.037696	0.076748	0.01745	7483	7483	-152.24	1219.47	1067.23	-0.3100	2.5745	2.2645	14
15	0.039597	0.080621	0.01745	7139	7139	-151.76	1219.43	1067.67	-0.3089	2.5690	2.2600	15
16	0.041586	0.084671	0.01745	6811	6811	-151.27	1219.38	1068.11	-0.3079	2.5635	2.2556	16
17	0.043666	0.088905	0.01745	6501	6501	-150.78	1219.33	1068.55	-0.3069	2.5580	2.2511	17
18	0.045841	0.093332	0.01745	6205	6205	-150.30	1219.28	1068.99	-0.3059	2.5526	2.2467	18
19	0.048113	0.097960	0.01745	5924	5924	-149.81	1219.23	1069.43	-0.3049	2.5471	2.2423	19
20	0.050489	0.102796	0.01746	5657	5657	-149.32	1219.18	1069.87	-0.3038	2.5417	2.2379	20
21	0.052970	0.107849	0.01746	5404	5404	-148.82	1219.13	1070.31	-0.3028	2.5363	2.2335	21
22	0.055563	0.113128	0.01746	5162	5162	-148.33	1219.08	1070.75	-0.3018	2.5309	2.2292	22
23	0.058271	0.118641	0.01746	4932	4932	-147.84	1219.02	1071.19	-0.3008	2.5256	2.2248	23
24	0.061099	0.124398	0.01746	4714	4714	-147.34	1218.97	1071.63	-0.2997	2.5203	2.2205	24
25	0.064051	0.130408	0.01746	4506	4506	-146.85	1218.91	1072.07	-0.2987	2.5149	2.2162	25
26	0.067133	0.136684	0.01747	4308	4308	-146.35	1218.85	1072.50	-0.2977	2.5096	2.2119	26
27	0.070349	0.143233	0.01747	4119	4119	-145.85	1218.80	1072.94	-0.2967	2.5044	2.2077	27
28	0.073706	0.150066	0.01747	3940	3940	-145.35	1218.74	1073.38	-0.2956	2.4991	2.2035	28
29	0.077207	0.157195	0.01747	3769	3769	-144.85	1218.68	1073.82	-0.2946	2.4939	2.1992	29
30	0.080860	0.164632	0.01747	3606	3606	-144.35	1218.61	1074.26	-0.2936	2.4886	2.1951	30

Table 159-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
31	0.084669	0.172387	0.01747	3450	3450	-143.85	1218.55	1074.70	-0.2926	2.4834	2.1909	31
32	0.088640	0.180474	0.01747	3302	3302	-143.35	1218.49	1075.14	-0.2915	2.4783	2.1867	32
32*	0.08865	0.18049	0.01602	3302.07	3302.09	-0.02	1075.15	1075.14	0.0000	2.1867	2.1867	32
33	0.09229	0.18791	0.01602	3178.15	3178.16	0.99	1074.59	1075.58	0.0020	2.1811	2.1832	33
34	0.09607	0.19559	0.01602	3059.47	3059.49	2.00	1074.02	1076.01	0.0041	2.1756	2.1796	34
35	0.09998	0.20355	0.01602	2945.66	2945.68	3.00	1073.45	1076.45	0.0061	2.1700	2.1761	35
36	0.10403	0.21180	0.01602	2836.60	2836.61	4.01	1072.88	1076.89	0.0081	2.1645	2.1726	36
37	0.10822	0.22035	0.01602	2732.13	2732.15	5.02	1072.32	1077.33	0.0102	2.1590	2.1692	37
38	0.11257	0.22919	0.01602	2631.88	2631.89	6.02	1071.75	1077.77	0.0122	2.1535	2.1657	38
39	0.11707	0.23835	0.01602	2535.88	2535.86	7.03	1071.18	1078.21	0.0142	2.1481	2.1623	39
40	0.12172	0.24783	0.01602	2443.67	2443.69	8.03	1070.62	1078.65	0.0162	2.1426	2.1589	40
41	0.12654	0.25765	0.01602	2355.22	2355.24	9.04	1070.05	1079.09	0.0182	2.1372	2.1554	41
42	0.13153	0.26780	0.01602	2270.42	2270.43	10.04	1069.48	1079.52	0.0202	2.1318	2.1521	42
43	0.13669	0.27831	0.01602	2189.02	2189.04	11.04	1068.92	1079.96	0.0222	2.1265	2.1487	43
44	0.14203	0.28918	0.01602	2110.92	2110.94	12.05	1068.35	1080.40	0.0242	2.1211	2.1454	44
45	0.14755	0.30042	0.01602	2035.91	2035.92	13.05	1067.79	1080.84	0.0262	2.1158	2.1420	45
46	0.15326	0.31205	0.01602	1963.85	1963.87	14.05	1067.22	1081.28	0.0282	2.1105	2.1387	46
47	0.15917	0.32407	0.01602	1894.71	1894.73	15.06	1066.66	1081.71	0.0302	2.1052	2.1354	47
48	0.16527	0.33650	0.01602	1828.28	1828.30	16.06	1066.09	1082.15	0.0321	2.1000	2.1321	48
49	0.17158	0.34935	0.01602	1764.44	1764.46	17.06	1065.53	1082.59	0.0341	2.0947	2.1288	49
50	0.17811	0.36236	0.01602	1703.18	1703.20	18.06	1064.96	1083.03	0.0361	2.0895	2.1256	50
51	0.18484	0.37635	0.01602	1644.25	1644.26	19.06	1064.40	1083.46	0.0381	2.0843	2.1224	51
52	0.19181	0.39053	0.01603	1587.64	1587.65	20.07	1063.83	1083.90	0.0400	2.0791	2.1191	52
53	0.19900	0.40516	0.01603	1533.22	1533.24	21.07	1063.27	1084.34	0.0420	2.0740	2.1159	53
54	0.20643	0.42029	0.01603	1480.89	1480.91	22.07	1062.71	1084.77	0.0439	2.0689	2.1128	54
55	0.21410	0.43591	0.01603	1430.61	1430.62	23.07	1062.14	1085.21	0.0459	2.0637	2.1096	55
56	0.22202	0.45204	0.01603	1382.19	1382.21	24.07	1061.58	1085.65	0.0478	2.0586	2.1064	56
57	0.23020	0.46869	0.01603	1335.65	1335.67	25.07	1061.01	1086.08	0.0497	2.0536	2.1033	57
58	0.23864	0.48588	0.01603	1290.85	1290.87	26.07	1060.45	1086.52	0.0517	2.0485	2.0002	58
59	0.24735	0.50362	0.01603	1247.76	1247.78	27.07	1059.89	1086.96	0.0536	2.0435	2.0971	59
60	0.25635	0.52192	0.01604	1206.30	1206.32	28.07	1059.32	1087.39	0.0555	2.0385	2.0940	60
61	0.26562	0.54081	0.01604	1166.38	1166.40	29.07	1058.76	1087.73	0.0575	2.0334	2.0909	61
62	0.27519	0.56029	0.01604	1127.93	1127.95	30.07	1058.19	1088.27	0.0594	2.0285	2.0878	62
63	0.28506	0.58039	0.01604	1090.94	1090.96	31.07	1057.63	1088.70	0.0613	2.0235	2.0848	63
64	0.29524	0.60112	0.01604	1055.32	1055.33	32.07	1057.07	1089.14	0.0632	2.0186	2.0818	64
65	0.30574	0.62249	0.01604	1020.98	1021.00	33.07	1056.50	1089.57	0.0651	2.0136	2.0787	65
66	0.31656	0.64452	0.01604	987.95	987.97	34.07	1055.94	1090.01	0.0670	2.0087	2.0758	66

Table 160-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
67	0.32772	0.66724	0.01605	956.11	956.12	35.07	1055.37	1090.44	0.0689	2.0039	2.0728	67
68	0.33921	0.69065	0.01605	925.44	925.45	36.07	1054.81	1090.88	0.0708	1.9990	2.0698	68
69	0.35107	0.71478	0.01605	895.86	895.87	37.07	1054.24	1091.31	0.0727	1.9941	2.0668	69
70	0.36328	0.73964	0.01605	867.34	867.36	38.07	1053.68	1091.75	0.0746	1.9893	2.0639	70
71	0.37586	0.76526	0.01605	839.87	839.88	39.07	1053.11	1092.18	0.0765	1.9845	2.0610	71
72	0.38882	0.79164	0.01606	813.37	813.39	40.07	1052.55	1092.61	0.0783	1.9797	2.0580	72
73	0.40217	0.81883	0.01606	787.85	787.87	41.07	1051.98	1093.05	0.0802	1.9749	2.0552	73
74	0.41592	0.84682	0.01606	763.19	763.21	42.06	1051.42	1093.48	0.0821	1.9702	2.0523	74
75	0.43008	0.87564	0.01606	739.42	739.44	43.06	1050.85	1093.92	0.0840	1.9654	2.0494	75
76	0.44465	0.90532	0.01606	716.51	726.53	44.06	1050.29	1094.35	0.0858	1.9607	2.0465	76
77	0.45966	0.93587	0.01607	694.38	794.40	45.06	1049.72	1094.78	0.0877	1.9560	2.0437	77
78	0.47510	0.96732	0.01607	673.05	673.06	46.06	1049.16	1095.22	0.0896	1.9513	2.0409	78
79	0.49100	0.99968	0.01607	652.44	652.46	47.06	1048.59	1095.65	0.0914	1.9466	2.0380	79
80	0.50736	1.03298	0.01607	632.54	632.56	48.06	1048.03	1096.08	0.0933	1.9420	2.0352	80
81	0.52419	1.06725	0.01608	613.35	613.37	49.06	1047.46	1096.51	0.0951	1.9373	2.0324	81
82	0.54150	1.10250	0.01608	594.82	594.84	50.05	1046.89	1096.95	0.0970	1.9327	0.0297	82
83	0.55931	1.13877	0.01608	576.90	576.92	51.05	1046.33	1097.38	0.0988	1.9281	2.0269	83
84	0.57763	1.17606	0.01608	559.63	559.65	52.05	1045.76	1097.81	0.1006	1.9235	2.0242	84
85	0.59647	1.21442	0.01609	542.93	542.94	53.05	1045.19	1098.24	0.1025	1.9189	2.0214	85
86	0.61584	1.25385	0.01609	526.80	526.81	54.05	1044.63	1098.67	0.1043	1.9144	2.0187	86
87	0.63575	1.29440	0.01609	511.21	511.22	55.05	1044.06	1099.11	0.1061	1.9098	2.0160	87
88	0.65622	1.33608	0.01609	496.14	496.15	56.05	1043.49	1099.54	0.1080	1.9053	2.0133	88
89	0.67726	1.37892	0.01610	481.60	481.61	57.04	1042.92	1099.97	0.1098	1.9008	2.0106	89
0	0.69889	1.42295	0.01610	467.52	467.53	58.04	1042.36	1100.40	0.1116	1.8963	2.0079	90
91	0.72111	1.46820	0.01610	453.91	453.93	59.04	1041.79	1100.83	0.1134	1.8918	2.0053	91
92	0.74394	1.51468	0.01611	440.76	440.78	60.04	1041.22	1101.26	0.1152	1.8874	2.0026	92
93	0.76740	1.56244	0.01611	428.04	428.06	61.04	1040.65	1101.69	0.1170	1.8829	2.0000	93
94	0.79150	1.61151	0.01611	415.74	415.76	62.04	1040.08	1102.12	0.1188	1.8785	1.9973	94
95	0.81625	1.66189	0.01612	403.84	403.86	63.03	1039.51	1102.55	0.1206	1.8741	1.9947	95
96	0.84166	1.71364	0.01612	392.33	392.34	64.03	1038.95	1102.98	0.1224	1.8697	1.9921	96
97	0.86776	1.76678	0.01612	381.20	381.21	65.03	1038.38	1103.41	0.1242	1.8653	1.9895	97
98	0.89456	1.82134	0.01612	370.42	370.44	66.03	1037.81	1103.84	0.1260	1.8610	1.9870	98
99	0.92207	1.87736	0.01613	359.99	360.01	67.03	1037.24	1104.26	0.1278	1.8566	1.9844	99
100	0.95031	1.93485	0.01613	349.91	349.92	68.03	1036.67	1104.69	0.1296	1.8523	1.9819	100
101	0.97930	1.99387	0.01613	340.14	340.15	69.03	1036.10	1105.12	0.1314	1.8479	1.9793	101
102	1.00904	2.05443	0.01614	330.69	330.71	70.02	1035.53	1105.55	0.1332	1.8436	1.9768	102
103	1.03956	2.11667	0.01614	321.53	321.55	71.02	1034.95	1105.98	0.1349	1.8393	1.9743	103

Table 161-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
67	0.32772	0.66724	0.01605	956.11	956.12	35.07	1055.37	1090.44	0.0689	2.0039	2.0728	67
68	0.33921	0.69065	0.01605	925.44	925.45	36.07	1054.81	1090.88	0.0708	1.9990	2.0698	68
69	0.35107	0.71478	0.01605	895.86	895.87	37.07	1054.24	1091.31	0.0727	1.9941	2.0668	69
70	0.36328	0.73964	0.01605	867.34	867.36	38.07	1053.68	1091.75	0.0746	1.9893	2.0639	70
71	0.37586	0.76526	0.01605	839.87	839.88	39.07	1053.11	1092.18	0.0765	1.9845	2.0610	71
72	0.38882	0.79164	0.01606	813.37	813.39	40.07	1052.55	1092.61	0.0783	1.9797	2.0580	72
73	0.40217	0.81883	0.01606	787.85	787.87	41.07	1051.98	1093.05	0.0802	1.9749	2.0552	73
74	0.41592	0.84682	0.01606	763.19	763.21	42.06	1051.42	1093.48	0.0821	1.9702	2.0523	74
75	0.43008	0.87564	0.01606	739.42	739.44	43.06	1050.85	1093.92	0.0840	1.9654	2.0494	75
76	0.44465	0.90532	0.01606	716.51	726.53	44.06	1050.29	1094.35	0.0858	1.9607	2.0465	76
77	0.45966	0.93587	0.01607	694.38	794.40	45.06	1049.72	1094.78	0.0877	1.9560	2.0437	77
78	0.47510	0.96732	0.01607	673.05	673.06	46.06	1049.16	1095.22	0.0896	1.9513	2.0409	78
79	0.49100	0.99968	0.01607	652.44	652.46	47.06	1048.59	1095.65	0.0914	1.9466	2.0380	79
80	0.50736	1.03298	0.01607	632.54	632.56	48.06	1048.03	1096.08	0.0933	1.9420	2.0352	80
81	0.52419	1.06725	0.01608	613.35	613.37	49.06	1047.46	1096.51	0.0951	1.9373	2.0324	81
82	0.54150	1.10250	0.01608	594.82	594.84	50.05	1046.89	1096.95	0.0970	1.9327	0.0297	82
83	0.55931	1.13877	0.01608	576.90	576.92	51.05	1046.33	1097.38	0.0988	1.9281	2.0269	83
84	0.57763	1.17606	0.01608	559.63	559.65	52.05	1045.76	1097.81	0.1006	1.9235	2.0242	84
85	0.59647	1.21442	0.01609	542.93	542.94	53.05	1045.19	1098.24	0.1025	1.9189	2.0214	85
86	0.61584	1.25385	0.01609	526.80	526.81	54.05	1044.63	1098.67	0.1043	1.9144	2.0187	86
87	0.63575	1.29440	0.01609	511.21	511.22	55.05	1044.06	1099.11	0.1061	1.9098	2.0160	87
88	0.65622	1.33608	0.01609	496.14	496.15	56.05	1043.49	1099.54	0.1080	1.9053	2.0133	88
89	0.67726	1.37892	0.01610	481.60	481.61	57.04	1042.92	1099.97	0.1098	1.9008	2.0106	89
0	0.69889	1.42295	0.01610	467.52	467.53	58.04	1042.36	1100.40	0.1116	1.8963	2.0079	90
91	0.72111	1.46820	0.01610	453.91	453.93	59.04	1041.79	1100.83	0.1134	1.8918	2.0053	91
92	0.74394	1.51468	0.01611	440.76	440.78	60.04	1041.22	1101.26	0.1152	1.8874	2.0026	92
93	0.76740	1.56244	0.01611	428.04	428.06	61.04	1040.65	1101.69	0.1170	1.8829	2.0000	93
94	0.79150	1.61151	0.01611	415.74	415.76	62.04	1040.08	1102.12	0.1188	1.8785	1.9973	94
95	0.81625	1.66189	0.01612	403.84	403.86	63.03	1039.51	1102.55	0.1206	1.8741	1.9947	95
96	0.84166	1.71364	0.01612	392.33	392.34	64.03	1038.95	1102.98	0.1224	1.8697	1.9921	96
97	0.86776	1.76678	0.01612	381.20	381.21	65.03	1038.38	1103.41	0.1242	1.8653	1.9895	97
98	0.89456	1.82134	0.01612	370.42	370.44	66.03	1037.81	1103.84	0.1260	1.8610	1.9870	98
99	0.92207	1.87736	0.01613	359.99	360.01	67.03	1037.24	1104.26	0.1278	1.8566	1.9844	99
100	0.95031	1.93485	0.01613	349.91	349.92	68.03	1036.67	1104.69	0.1296	1.8523	1.9819	100
101	0.97930	1.99387	0.01613	340.14	340.15	69.03	1036.10	1105.12	0.1314	1.8479	1.9793	101
102	1.00904	2.05443	0.01614	330.69	330.71	70.02	1035.53	1105.55	0.1332	1.8436	1.9768	102
103	1.03956	2.11667	0.01614	321.53	321.55	71.02	1034.95	1105.98	0.1349	1.8393	1.9743	103

Table 162-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
104	1.07088	2.18034	0.01614	312.67	312.69	72.02	1034.38	1106.40	0.1367	1.8351	1.9718	104
105	1.10301	2.24575	0.01615	304.08	304.10	73.02	1033.81	1106.83	0.1385	1.8308	1.9693	105
106	1.13597	2.31285	0.01615	295.76	295.77	74.02	1033.24	1107.26	0.1402	1.8266	1.9668	106
107	1.16977	2.38168	0.01616	287.71	287.73	75.01	1032.67	1107.68	0.1420	1.8223	1.9643	107
108	1.20444	2.45226	0.01616	279.91	279.92	76.01	1032.10	1108.11	0.1438	1.8181	1.9619	108
109	1.23999	2.52464	0.01616	272.34	272.36	77.01	1031.52	1108.54	0.1455	1.8139	1.9594	109
110	1.27644	2.59885	0.01617	265.02	265.03	78.01	1030.95	1108.96	0.1473	1.8097	1.9570	110
111	1.31381	2.67494	0.01617	257.91	257.93	79.01	1030.38	1109.39	0.1490	1.8055	1.9546	111
112	1.35212	2.75293	0.01617	251.02	251.04	80.01	1029.80	1109.81	0.1508	1.8014	1.9521	112
113	1.39138	2.83288	0.01618	244.36	244.38	81.01	1029.23	1110.24	0.1525	1.7972	1.9497	113
114	1.43162	2.91481	0.01618	237.89	237.90	82.00	1028.66	1110.66	0.1543	1.7931	1.9474	114
115	1.47286	2.99878	0.01619	231.62	231.63	83.00	1028.08	1111.09	0.1560	1.7890	1.9450	115
116	1.51512	3.08481	0.01619	225.53	225.55	84.00	1027.51	1111.51	0.1577	1.7849	1.9426	116
117	1.55842	3.17296	0.01619	219.63	219.65	85.00	1026.93	1111.93	0.1595	1.7808	1.9402	117
118	1.60277	3.26327	0.01620	213.91	213.93	86.00	1026.36	1112.36	0.1612	1.7767	1.9379	118
119	1.64820	3.35577	0.01620	208.36	208.37	87.00	1025.78	1112.78	0.1629	1.7726	1.9356	119
120	1.69474	3.45052	0.01620	202.98	202.99	88.00	1025.20	1113.20	0.1647	1.7686	1.9332	120
121	1.74240	3.54755	0.01621	197.76	197.76	89.00	1023.62	1113.62	0.1664	1.7645	1.9309	121
122	1.79117	3.64691	0.01621	192.69	192.69	90.00	1024.05	1114.05	0.1681	1.7605	1.9286	122
123	1.84117	3.74863	0.01622	187.78	187.78	90.99	1024.47	1114.47	0.1698	1.7565	1.9263	123
124	1.89233	3.85282	0.01622	182.98	182.99	91.99	1022.90	1114.89	0.1715	1.7525	1.9240	124
125	1.94470	3.95945	0.01623	178.34	178.36	92.99	1022.32	1115.31	0.1732	1.7485	1.9217	125
126	1.99831	4.06860	0.01623	173.85	173.86	93.99	1021.74	1115.73	0.1749	1.7445	1.9195	126
127	2.05318	4.18032	0.01623	169.47	169.49	94.99	1021.16	1116.15	0.1766	1.7406	1.9172	127
128	2.10934	4.29465	0.01624	165.23	165.25	95.99	1020.58	1116.57	0.1783	1.7366	1.9150	128
129	2.16680	4.41165	0.01624	161.11	161.12	96.99	1020.00	1116.99	0.1800	1.7327	1.9127	129
130	2.22560	4.53136	0.01625	157.11	157.12	97.99	1019.42	1117.41	0.1817	1.7288	1.9105	130
131	2.28576	4.65384	0.01625	153.22	153.23	98.99	1018.84	1117.83	0.1834	1.7249	1.9083	131
132	2.34730	4.77914	0.01626	149.44	149.46	99.99	1018.26	1118.25	0.1851	1.7210	1.9061	132
133	2.41025	4.90730	0.01626	145.77	145.78	100.99	1017.68	1118.67	0.1868	1.7171	1.9039	133
134	2.47463	5.03839	0.01627	142.21	142.23	101.99	1017.10	1119.08	0.1885	1.7132	1.9017	134
135	2.54048	5.17246	0.01627	138.74	138.76	102.99	1016.52	1119.50	0.1902	1.7093	1.8995	135
136	2.60782	5.30956	0.01627	135.37	135.39	103.98	1015.93	1119.92	0.1919	1.7055	1.8974	136
137	2.67667	5.44975	0.01628	132.10	132.12	104.98	1015.35	1120.34	0.1935	1.7017	1.8952	137
138	2.74707	5.59308	0.01628	128.92	128.94	105.98	1014.77	1120.75	0.1952	1.6978	1.8930	138
139	2.81903	5.73961	0.01629	125.83	125.85	106.98	1014.18	1121.17	0.1969	1.6940	1.8909	139
140	2.89260	5.88939	0.01629	122.82	122.84	107.98	1013.60	1121.58	0.1985	1.6902	1.8888	140

Table 163-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
141	2.96780	6.04250	0.01630	119.90	119.92	108.98	1013.01	1122.00	0.2002	1.6864	1.8867	141
142	3.04465	6.19897	0.01630	117.05	117.07	109.98	1012.43	1122.41	0.2019	1.6827	1.8845	142
143	3.12320	6.35888	0.01631	114.29	114.31	110.98	1011.84	1122.83	0.2035	1.6789	1.8824	143
144	3.20345	6.52229	0.01631	111.60	111.62	111.98	1011.26	1123.24	0.2052	1.6752	1.8803	144
145	3.28546	6.68926	0.01632	108.99	109.00	112.98	1010.67	1123.66	0.2068	1.6714	1.8783	145
146	3.36924	6.85984	0.01632	106.44	106.45	113.98	1010.09	1124.07	0.2085	1.6677	1.8762	146
147	3.45483	7.03410	0.01633	103.96	103.98	114.98	1009.50	1124.48	0.2101	1.6640	1.8741	147
148	3.54226	7.21211	0.01633	101.55	101.57	115.98	1008.91	1124.89	0.2118	1.6603	1.8721	148
149	3.63156	7.39393	0.01634	99.21	99.22	116.98	1008.32	1125.31	0.2134	1.6566	1.8700	149
150	3.72277	7.57962	0.01634	96.93	96.94	117.98	1007.73	1125.72	0.2151	1.6529	1.8680	150
151	3.81591	7.76925	0.01635	94.70	94.72	118.99	1007.14	1126.13	0.2167	1.6492	1.8659	151
152	3.91101	7.96289	0.01635	92.54	92.56	119.99	1006.55	1126.54	0.2184	1.6455	1.8639	152
153	4.00812	8.16061	0.01636	90.44	90.46	120.99	1005.96	1123.95	0.2200	1.6419	1.8619	153
154	4.10727	8.36247	0.01636	88.39	88.41	121.99	1005.37	1127.36	0.2216	1.6383	1.8599	154
155	4.20848	8.56854	0.01637	86.40	86.41	122.99	1004.78	1127.77	0.2233	1.6346	1.8579	155
156	4.31180	8.77890	0.01637	84.45	84.47	123.99	1004.19	1128.18	0.2249	1.6310	1.8559	156
157	4.41725	8.99360	0.01638	82.56	82.58	124.99	1003.60	1128.59	0.2265	1.6274	1.8539	157
158	4.52488	9.21274	0.01638	80.72	80.73	125.99	1003.00	1128.99	0.2281	1.6238	1.8519	158
159	4.63472	9.43637	0.01639	78.92	78.94	126.99	1002.41	1129.40	0.2297	1.6202	1.8500	159
160	4.7468	9.6646	0.01639	77.175	77.192	127.99	1001.82	1129.81	0.2314	1.6167	1.8480	160
161	4.8612	9.8974	0.01640	75.471	75.488	128.99	1001.22	1130.22	0.2330	1.6131	1.8461	161
162	4.9778	10.1350	0.01640	73.812	73.829	130.00	1000.63	1130.62	0.2346	1.6095	1.8441	162
163	5.0969	10.3774	0.01641	72.196	72.313	131.00	1000.03	1131.03	0.2362	1.6060	1.8422	163
164	5.2183	10.6246	0.01642	70.619	70.636	132.00	999.43	1131.43	0.2378	1.6025	1.8403	164
165	5.3422	10.8768	0.01642	69.084	69.101	133.00	998.84	1131.84	0.2394	1.5989	1.8383	165
166	5.4685	11.1340	0.01643	67.587	67.604	134.00	998.24	1132.24	0.2410	1.5954	1.8364	166
167	5.5974	11.3963	0.01643	66.130	66.146	135.00	997.64	1132.64	0.2426	1.5919	1.8345	167
168	5.7287	11.6638	0.01644	64.707	64.723	136.01	997.04	1133.05	0.2442	1.5884	1.8326	168
169	5.8627	11.9366	0.01644	63.320	63.336	137.01	996.44	1133.45	0.2458	1.5850	1.8308	169
170	5.9993	12.2148	0.01645	61.969	61.986	138.01	995.84	1133.85	0.2474	1.5815	1.8289	170
171	6.1386	12.4983	0.01646	60.649	60.666	139.01	995.24	1134.25	0.2490	1.5780	1.8270	171
172	6.2806	12.7874	0.01646	59.363	59.380	140.01	994.64	1134.66	0.2506	1.5746	1.8251	172
173	6.4253	13.0821	0.01647	58.112	58.128	141.02	994.04	1135.06	0.2521	1.5711	1.8233	173
174	6.5729	13.3825	0.01647	56.887	56.904	142.02	993.44	1135.46	0.2537	1.5677	1.8214	174
175	6.7232	13.6886	0.01648	55.694	55.711	143.02	992.83	1135.86	0.2553	1.5643	1.8196	175
176	6.8765	14.0006	0.01648	54.532	54.549	144.02	992.23	1136.26	0.2569	1.5609	1.8178	176
177	7.0327	14.3186	0.01649	53.397	53.414	145.03	991.63	1136.65	0.2585	1.5575	1.8159	177

Table 164-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
178	7.1918	14.6426	0.01650	52.290	52.307	146.03	991.02	1137.05	0.2600	1.5541	1.8141	178
179	7.3539	14.9727	0.01650	51.210	51.226	147.03	990.42	1137.45	0.2616	1.5507	1.8123	179
180	7.5191	15.3091	0.01651	50.155	50.171	148.04	989.81	1137.85	0.2632	1.5473	1.8105	180
181	7.6874	15.6518	0.01651	49.126	49.143	149.04	989.20	1138.24	0.2647	1.5440	1.8087	181
182	7.8589	16.0008	0.01652	48.122	48.138	150.04	988.60	1138.64	0.2663	1.5406	1.8069	182
183	8.0335	16.3564	0.01653	47.142	47.158	151.05	987.99	1139.03	0.2679	1.5373	1.8051	183
184	8.2114	16.7185	0.01653	46.185	46.202	152.05	987.38	1139.43	0.2694	1.5339	1.8034	184
185	8.3926	17.0874	0.01654	45.251	45.267	153.05	986.77	1139.82	0.2710	1.5306	1.8016	185
186	8.5770	17.4630	0.01654	44.339	44.356	154.06	986.16	1140.22	0.2725	1.5273	1.7998	186
187	8.7649	17.8455	0.01655	43.448	43.465	155.06	985.55	1140.61	0.2741	1.5240	1.7981	187
188	8.9562	18.2350	0.01656	42.579	42.595	156.07	984.94	1141.00	0.2756	1.5207	1.7963	188
189	9.1510	18.6316	0.01656	41.730	41.746	157.07	984.32	1141.39	0.2772	1.5174	1.7946	189
190	9.3493	19.0353	0.01657	40.901	40.918	158.07	983.71	1141.78	0.2787	1.5141	1.7929	190
191	9.5512	19.4464	0.01658	40.092	40.108	159.08	983.10	1142.18	0.2803	1.5109	1.7911	191
192	9.7567	19.8648	0.01658	39.301	39.317	160.08	982.48	1142.57	0.2818	1.5076	1.7894	192
193	9.9659	20.2907	0.01659	38.528	38.544	161.09	981.87	1142.95	0.2834	1.5043	1.7877	193
194	10.1788	20.7242	0.01659	37.774	37.790	162.09	981.25	1143.34	0.2849	1.5011	1.7860	194
195	10.3955	21.1653	0.01660	37.035	37.052	163.10	980.63	1143.73	0.2864	1.4979	1.7843	195
196	10.6160	21.6143	0.01661	36.314	36.331	164.10	980.02	1144.12	0.2880	1.4946	1.7826	196
197	10.8404	22.0712	0.01661	35.611	35.628	165.11	979.40	1144.51	0.2895	1.4914	1.7809	197
198	11.0687	22.5361	0.01662	34.923	34.940	166.11	978.78	1144.89	0.2910	1.4882	1.7792	198
199	11.3010	23.0091	0.01663	34.251	34.268	167.12	978.16	1145.28	0.2926	1.4850	1.7776	199
200	11.5374	23.4904	0.01663	33.594	33.610	168.13	977.54	1145.66	0.2941	1.4818	1.7759	200
201	11.7779	23.9800	0.01664	32.951	32.968	169.13	976.92	1146.05	0.2956	1.4786	1.7742	201
202	12.0225	24.4780	0.01665	32.324	32.340	170.14	976.29	1146.43	0.2971	1.4755	1.7726	202
203	12.2713	24.9847	0.01665	31.710	31.726	171.14	975.67	1146.81	0.2986	1.4723	1.7709	203
204	12.5244	25.5000	0.01666	31.110	31.127	172.15	975.05	1147.20	0.3002	1.4691	1.7693	204
205	12.7819	26.0241	0.01667	30.523	30.540	173.16	974.42	1147.58	0.3017	1.4660	1.7677	205
206	13.0436	26.5571	0.01667	29.949	29.965	174.16	973.80	1147.96	0.3032	1.4628	1.7660	206
207	13.3099	27.0991	0.01668	29.388	29.404	175.17	973.17	1148.34	0.3047	1.4597	1.7644	207
208	13.5806	27.6503	0.01669	28.839	28.856	176.18	972.54	1148.72	0.3062	1.4566	1.7628	208
209	13.8558	28.2108	0.01669	28.303	28.319	177.18	971.92	1149.10	0.3077	1.4535	1.7612	209
210	14.1357	28.7806	0.01670	27.778	27.795	178.19	971.29	1149.48	0.3092	1.4503	1.7596	210
212	14.7096	29.9489	0.01671	26.763	26.780	180.20	970.03	1150.23	0.3122	1.4442	1.7564	212
214	15.3025	31.1563	0.01673	25.790	25.807	182.22	968.76	1150.98	0.3152	1.4380	1.7532	214
216	15.9152	32.4036	0.01674	24.861	24.878	184.24	967.50	1151.73	0.3182	1.4319	1.7501	216
218	16.5479	33.6919	0.01676	23.970	23.987	186.25	966.23	1152.48	0.3212	1.4258	1.7469	218

Table 165-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
220	17.2013	35.0218	0.01677	23.118	23.134	188.27	964.95	1153.22	0.3241	1.4197	1.7438	220
222	17.8759	36.3956	0.01679	22.299	22.316	190.29	963.67	1153.96	0.3271	1.4136	1.7407	222
224	18.5721	37.8131	0.01680	21.516	21.533	192.31	962.39	1154.70	0.3301	1.4076	1.7377	224
226	19.2905	39.2758	0.01682	20.765	20.782	194.33	961.11	1155.43	0.3330	1.4016	1.7347	226
228	20.0316	40.7848	0.01683	20.045	20.062	196.35	959.82	1156.16	0.3359	1.3957	1.7316	228
230	20.7961	42.3412	0.01684	19.355	19.372	198.37	958.52	1156.89	0.3389	1.3898	1.7287	230
232	21.5843	43.9461	0.01686	18.682	18.709	200.39	957.22	1157.62	0.3418	1.3839	1.7257	232
234	22.3970	45.6006	0.01688	18.056	18.073	202.41	955.92	1158.34	0.3447	1.3780	1.7227	234
236	23.2345	47.3060	0.01689	17.446	17.463	204.44	954.62	1159.06	0.3476	1.3722	1.7198	236
238	24.0977	49.0633	0.01691	16.860	16.877	206.46	953.31	1159.77	0.3505	1.3664	1.7169	238
240	24.9869	50.8738	0.01692	16.298	16.314	208.49	952.00	1160.48	0.3534	1.3606	1.7140	240
242	25.9028	52.7386	0.01694	15.757	15.774	210.51	950.68	1161.19	0.3563	1.3548	1.7111	242
244	26.8461	54.6591	0.01695	15.238	15.255	212.54	949.35	1161.90	0.3592	1.3491	1.7083	244
246	27.8172	56.6364	0.01697	14.739	14.756	214.57	948.03	1162.60	0.3621	1.3434	1.7055	246
248	28.8169	58.6717	0.01698	14.259	14.276	216.60	946.70	1163.29	0.3649	1.3377	1.7026	248
250	29.8457	60.7664	0.01700	13.798	13.815	218.63	945.36	1163.99	0.3678	1.3321	1.6998	250
252	30.9043	62.9218	0.01702	13.355	13.372	220.66	944.02	1164.68	0.3706	1.3264	1.6971	252
254	31.9934	65.1391	0.01703	12.928	12.945	222.69	942.68	1165.37	0.3735	1.3208	1.6943	254
256	33.1135	67.4197	0.01705	12.526	12.147	226.73	939.99	1166.72	0.3764	1.3153	1.6691	256
258	34.2653	69.7649	0.01707	12.123	12.140	226.76	939.97	1166.73	0.3792	1.3097	1.6889	258
260	35.4496	72.1760	0.01708	11.742	11.759	228.79	938.61	1167.40	0.3820	1.3042	1.6862	260
262	36.6669	74.6545	0.01710	11.376	11.393	230.83	937.25	1168.08	0.3848	1.2987	1.6835	262
264	37.9180	77.2017	0.01712	11.024	11.041	232.87	935.88	1168.74	0.3876	1.2932	1.6808	264
266	39.2035	79.8190	0.01714	10.684	10.701	234.90	934.50	1169.41	0.3904	1.2877	1.6781	266
268	40.5241	82.5078	0.01715	10.357	10.374	236.94	933.12	1170.07	0.3932	1.2823	1.6755	268
270	41.8806	85.2697	0.01717	10.042	10.059	238.98	931.74	1170.72	0.3960	1.2769	1.6729	270
272	43.2736	88.1059	0.01719	9.737	9.755	241.03	930.35	1171.38	0.3988	1.2715	1.6703	272
274	44.7040	91.0181	0.01721	9.445	9.462	243.07	928.95	1172.02	0.4016	1.2661	1.6677	274
276	46.1723	94.0076	0.01722	9.162	9.179	245.11	927.55	1172.67	0.4044	1.2608	1.6651	276
278	47.6794	97.0761	0.01724	8.890	8.907	247.16	926.15	1173.31	0.4071	1.2554	1.6626	278
280	49.2260	100.2250	0.01726	8.627	8.644	249.20	924.74	1173.94	0.4099	1.2501	1.6600	280
282	50.8128	103.4558	0.01728	8.373	8.390	251.25	923.32	1174.57	0.4127	1.2448	1.6575	282
284	52.4406	106.7701	0.01730	8.128	8.146	253.30	921.90	1175.20	0.4154	1.2396	1.6550	284
286	54.1103	110.1695	0.01731	7.892	7.910	255.35	920.47	1175.82	0.4182	1.2343	1.6525	286
288	55.8225	113.6556	0.01733	7.664	7.681	257.40	919.03	1176.44	0.4209	1.2291	1.6500	288

Table 166-1:
Thermodynamic properties of water at saturation

Temp	Absolute	Pressure	Specific volume, ft3/lb			Enthalpy btu/lb			Entropy, btu/lb - °F			Temp.
			Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	Sat. solid	Evap.	Sat. vapor	
t, °F	psi	in. Hg	V_f	V_{fg}	V_f	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	°F
290	57.5780	117.2299	0.01735	7.444	7.461	259.45	917.59	1177.05	0.4236	1.2239	1.6476	290
292	59.3777	120.8941	0.01737	7.231	7.248	261.51	916.15	1177.66	0.4264	1.2187	1.6451	292
294	61.2224	124.6498	0.01739	7.026	7.043	263.56	914.69	1178.26	0.4291	1.2136	1.6427	294
296	63.1128	128.4987	0.01741	6.827	6.844	265.62	913.24	1178.86	0.4318	1.2084	1.6402	296
298	65.0498	132.4425	0.01743	6.635	6.652	267.68	911.77	1179.45	0.4345	1.2033	1.6378	298
300	67.0341	136.4827	0.01745	6.450	6.467	269.74	910.30	1180.04	0.4372	1.1982	1.6354	300

Conversion Data, Calculations, Factors & Tables

Table 167-1: Conversion Data - Power, Work and Heat

1 Btu	778.2 ft-lb
1 watt hour	2,655.4 ft-lb
	56.9 Btu per minute
1 kilowatt (1,000 watts)	1.3405 horsepower
	56.9 Btu per minute
1 horsepower	0.746 kilowatts
1 pound (force)	32.2 poundals (ft-lb per sec^2)
1 mechanical horsepower	42.4 Btu per minute
	2,544 Btu per hour
	33,000 ft-lb per minute
1 boiler horsepower	33,479 Btu per hour
	evaporation of 34.5 lb per hour of water at 212°F
1 ton of refrigeration	200 Btu per minute
	12,000 Btu per hour
Latent heat of ice	144 Btu per lb

Table 167-2: Conversion Data - Weight and Volume

1 gal (U.S.)	231.0 cubic inches
	0.13368 cubic feet
	3,785 cubic centimeters
1 gal (British or Imperial)	277.274 cubic inches
1 cubic foot	7.4805 gal
	1,728 cubic inches
1 cubic foot of water at 60°F	62.37 lbs
	7.4805 gal
1 cubic foot of water at 212°F	59.76 lbs
1 gallon of water at 60°F	8.34 lbs
1 gallon of water at 212°F	7.99 lbs
1 pound (avoirdupois)	16 oz.
	7,000 grains
1 bushel	1.244 cubic feet
1 short ton	2,000 lbs
1 long ton	2,240 lbs
1 cubic foot of free air at 65°F	0.07203 lbs
1 lb of free air at 65°F	13.88 cubic feet
1 grain	0.06480 grams

CONVERSION DATA - PRESSURE

1 PSI EQUALS

144 lbs per sq ft
2.0416 inches of mercury at 62°F
2.309 feet of water at 62°F
27.71 inches of water of 62°F
6.895 Kilo Pascal (kPa)

1 OZ PER SQUARE INCHES EQUALS

0.1276 inches of mercury at 62°F
1.732 inches of water at 62°F

1 INCH OF MERCURY AT 62°F EQUALS

0.491 lb per square inch
7.86 oz per square inch
1.132 feet of water at 62°F
13.58 inches of water at 62°F

1 ATMOSPHERE EQUALS

2,116.8 lb per square foot
33.542 feet of water at °F
30.0 inches of mercury at 62°F
29.92 inches of mercury at 32°F
760 mm of mercury at 32°F

1 INCH OF WATER AT 62°F EQUALS

0.03609 lbs per square inch
0.5774 oz per square inch
5.196 lbs per square foot

1 FOOT OF WATER AT 62°F

0.443 PSI
62.335 lbs per square foot

1 BAR

14.5 lbs per square inch
33.49 ft of water 62°F
29.530 inches of mercury 32°F
100 Kilo Pascal (kPa)

1 KILO PASCAL (KPA)

.1450 psi

Conversion Data - Metric Units

1 cm	0.3937 inches
1 in.	0.254 cm
1 meter	3.281 ft
1 ft	0.3048 meter
1 sq cm	0.155 sq in
1 sq in	0.645 sq cm
1 sq meter	10.765 sq ft
1 sq ft	0.0929 sq meter
1 cubic cm	0.061 cubic in 0.03381 oz (liquid)
1 cu in	16.38 cu ft
1 cu meter	35.32 cu ft
1 cu ft	0.0283 cu meter
1 kilometer per hr	0.6214 mph
1 kilogram	2.2046 lb
1 lb	0.4536 kilogram
1 metric ton	2,205 lb (avdp)
1 gram	980.59 cynes 15.4324 grains
1 oz (liquid)	29.574 cu cm
1 oz (avdp)	29.3495 grams
1 gram per sq cm	0.2896 in mercury at 0°C 0.394 in water at 15°F
1 kg per sq cm (metric atmosphere)	14.23 psi
1 gram per cu cm	0.03613 lb per cu in 62.43 lb per cu ft
1 liter	1,000 cu cm 0.264 gal

Calculations - Heating and Cooling - Steam

Condensing capacity of a steam coil:

$$\text{Steam in lb per hr} = \frac{\text{Coil heat load (Btu per hr)}}{\text{Latent heat of steam in Btu per lb}}$$

$$\text{Steam in lb per hr} = \frac{\text{cfm} \times 60 \times \text{weight per cu ft} \times \text{sp ht} \times (T - T_0)}{\text{Latent heat of steam in Btu per lb}}$$

$$\text{Steam in lb per hr} = \frac{\text{cfm} \times 60 \times 0.075 \times 0.24 \times (T - T_0)}{960}$$

$$\text{Steam in lb per hr} = \frac{\text{cfm} \times (T - T_0)}{890}$$

Equivalent Direct Radiation:

EDR = Btu/240

lb per hr = EDR/4

Glossary:

cfm	Cubic feet of air per minute passing through the coil
weight per cu ft	Weight of 1 pound (0.075)
sp ht	Btu required to raise the temperature of 1 lb of air 1°
T_0	Temperature of air entering coil in °F
T	Temperature of air leaving coil in °F
1 EDR	Emission 240 Btu per hour

Calculations - Heating and Cooling - Hot Water

Capacity of hot water coil:

$$\text{gpm} = \frac{\text{Coil heat load (Btu per hr)}}{(T_2 - T_1) \times 60 \times 8.34 \text{ lb per gal}}$$

$$\text{gpm} = \frac{\text{Coil heat load (Btu per hr)}}{(T_2 - T_1) \times 500}$$

Equivalent Direct Radiation:

EDR = Btu/150

Glossary:

T_1	Temperature of air leaving coil in °F
T_2	Temperature of air entering coil in °F
1 EDR	Emission 150 Btu per hour

Calculations - Heating and Cooling - Chilled Water

Capacity of chilled water coil:

$$\text{gpm} = \frac{\text{Coil heat load (Btu per hr)}}{(T_2 - T_1) \times 500}$$

Total cooling load = $(H_1 - H_2) \times$ cfm $\times$ 0.075 $\times$ 60

Equivalent Direct Radiation:

EDR = Btu/150

Glossary:

H1 .... Total heat content of entering air
H2 .... Total heat content of leaving air
0.075...................... Specific weight of standard air in lb per cu ft

Calculations - Miscellaneous

Work
W = FC
where, W = work (in. – lb)
F - force (lb)
D = distance (in.)

Torque
T - FL
where, T = torque (in. – lb)
F = force (lb)
L = length of lever arm (in.)

Valve Flow Rate (Fluid)
Q = Cv x Sq rt of (Dp/Sf)
Q = Liquid flow rate, gpm
Cv - Valve coefficient
Dp = Pressure drop across valve in psi
Sf = Specific gravity of fluid

Mix Air Temperature

$$T_m = \frac{(VrTr) + (VoTo)}{VT}$$

Tm = Mix air temperature
Tr= Return temperature
To = Outside temperature
Vr = Return air cfm
Vo = Outside air cfm
Vt = Vo + Vr x Total cfm

Air Distribution Systems
Pt = Pv + Ps
where, Pt = total pressure
Pv = velocity pressure
Ps - static pressure

V = 4005 x Sq rt of (Pv)
where, V = velocity (ft per min)
Pv = velocity pressure (in. water)

Standard CFM

$$\text{SCFM} = \text{CFM} * \frac{\text{Va } 59°\text{F}}{\text{Va at duct temp. }°\text{F}}$$

SCFM = Standard cubic feet of air per minute
CFM = Cubic feet of air per minute
Va59°F = Standard volume of air at 59°F
Va at duct temp °F = Volume of air at duct temp. °F

Valve Flow Rate (Steam)

$$Q = 3 \times Cv \times \text{Sq rt of } \frac{(DP \times P2)}{K}$$

Cv = Coefficient of flow
Q = Lbs. per hour of steam
DP = Differential pressure in PSI (pressure drop)
P2 = Outlet pressure in PSIA (absolute)
PSIG + 14.7 = PSIA (absolute)
K = 1 + (.0007 x °F super-heat)

Note: K normally is 1 (K = 1 for saturated steam)

CONVERSION FACTORS

Multiply	By	To Get
atmospheres	76.0	cms of mercury
atmospheres	29.92	inches of mercury
atmospheres	33.90	feet of water
atmospheres	14.70	pounds per sq inch
British thermal units	777.5	foot-pounds
British thermal units	3.927 x 10^4	horsepower-hours
British thermal units	2.928 x 10^4	kilowatt-hours
Btu per min	12.96	foot-pounds per sec
Btu per min	0.02356	horsepower
Btu per min	0.01757	kilowatts
Btu per min	17.57	watts
centigrams	0.01	grams
centimeters	0.3937	inches
centimeters	0.01	meters
centimeters	10	millimeters
centimeter-grams	7.233 x 105	pound-feet
centimeters of mercury	0.01316	atmospheres
centimeters of mercury	0.4461	feet of water
centimeters of mercury	27.85	pounds per sq foot
centimeters of mercury	0.1934	pounds per sq inch
centimeters per second	1.969	feet per minute
centimeters per second	0.03281	feet per second
centimeters per second	3.728 x 10^4	miles per minute
cubic centimeters	3.351 x 10^5	cubic feet
cubic centimeters	6.10 x 10^2	cubic inches
cubic centimeters	10^6	cubic meters
cubic centimeters	1.308 x 10^6	cubic yards
cubic centimeters	2.113 x 10^3	pints (liq)
cubic feet	2.832 x 10^4	cubic cms
cubic feet	1728	cubic inches
cubic feet	0.02832	cubic meters
cubic feet	0.03704	cubic yards
cubic inches	16.39	cubic centimeters
cubic inches	5.787 x 10^4	cubic feet
cubic inches	1.639 x 10^5	cubic meters
cubic inches	2.143 x 10^5	cubic yards
cubic inches	0.01732	quarts (liq)
cubic meters	10^6	cubic centimeter
cubic meters	35.31	cubic feet
cubic meters	61.023	cubic inches
cubic meters	1057	quarts(liq)
cubic yards	27	cubic feet
cubic yards	46,656	cubic inches
cubic yards	.07646	cubic meters
feet	1/3	yards
feet of water	0.02950	atmospheres
feet of water	0.8826	inches of mercury
feet of water	32.43	pounds per sq ft
feet of water	0.4335	pounds per sq inch
feet per minute	0.5080	centimeters per sec
feet per minute	0.01667	feet per sec
feet per minute	0.3048	meters per minute
feet per minute	0.01136	miles per hour
feet per second	30.48	centimeters per sec
feet per second	18.29	meters per minute
feet per second	0.6818	miles per hour

CONVERSION FACTORS (continued)

Multiply	By	To Get
feet per second	0.01136	miles per minute
foot-pounds	1.286 x 10^{3}	British thermal units
foot-pounds	5.050 x 10^{7}	horsepower hours
foot-pounds	1.286 x 10^{3}	kilowatt-hours
foot-pounds per minute	3.766 x 10^{7}	Btu per minute
foot-pounds per minute	0.01667	foot-pounds per sec
foot-pounds per minute	3.030 x 10^{5}	horsepower
foot-pounds per minute	2.260 x 10^{5}	kilowatts
foot-pounds per second	7.717 x 10^{2}	Btu per minute
foot-pounds per second	1.818 x 10^{3}	horsepower
foot-pounds per second	1.356 x 10^{3}	kilowatts
gallons	0.1337	cubic feet
gallons	231	cubic inches
gallons per minute	2.228 x 10^{3}	cubic feet per second
grams	0.03527	ounces
grams	2.205 x 10^{3}	pounds
horsepower	42.44	Btu per min
horsepower	33,000	foot-pounds per min
horsepower	55	foot-pounds per sec
horsepower	1.014	horsepower (metric)
horsepower	0.7457	kilowatts
horsepower	745.7	watts
horsepower (boiler)	33,520	Btu per hour
horsepower (boiler)	9.804	kilowatts
horsepower-hours	2547	British thermal units
horsepower-hours	1.98 x 10^{6}	foot-pounds
horsepower-hours	0.7457	kilowatt-hours
inches	2.540	centimeters
inches of mercury	0.03342	atmospheres
inches of mercury	1.133	feet of water
inches of mercury	70.73	pounds per square ft
inches of mercury	0.4912	pounds per square in
decigrams	0.1	grams
decimeters	0.1	meters
degrees (angle)	60	minutes
feet	30.48	centimeters
feet	12	inches
feet	0.3048	meters
kilometers	1093.6	yards
kilometers per hour	0.6214	miles per hour
kilowatts	56.92	Btu per min
kilowatts	4.425 x 10^{4}	foot-pounds per min
kilowatts	737.6	foot-pounds per sec
kilowatts	1.341	horsepower
kilowatts	10^{3}	watts
kilowatt-hours	3415	British thermal units
kilowatt-hours	2.655 x 10^{6}	foot-pounds
kilowatt-hours	1.341	horsepower-hours
megohms	10^{6}	ohms
meters	100	centimeters
meters	3.2808	feet
meters	39.37	inches
meters	10^{3}	kilometers
meters	1.0936	yards
meters per minute	3.281	feet per minute
meters per minute	0.05468	feet per second

CONVERSION FACTORS (continued)

Multiply	By	To Get
meters per minute	0.03728	miles per hour
miles	5280	feet
miles	1.6093	kilometers
miles	1760	yards
miles per hour	44.70	feet per minute
miles per hour	1.467	feet per second
miles per hour	1.6093	kilometers per hour
miles per hour	26.83	meters per minute
miles per minute	2683	centimeters per second
miles per minute	88	feet per second
miles per minute	1.6093	kilometers per min
miles per minute	60	miles per hour
millimeters	0.1	centimeters
ohms	10^6	megohms
ounces	28.35	grams
ounces	0.0625	pounds
ounces (fluid)	1.805	cubic inches
ounces per square inch	0.0625	pounds per sq inch
pounds	453.6	grams
inches of water	0.002458	atmospheres
inches of water	0.07355	inches of mercury
inches of water	5.204	pounds per square ft
inches of water	0.03613	pounds per square in
kilograms	2.2046	pounds
kilometers	0.6214	miles
pounds	16	ounces
pound-feet	13,825	centimeter-grams
pounds of water	0.1198	gallons
pounds per square foot	0.01602	feet of water
pounds per square foot	6.944×10^6	pounds per sq in
pounds per square inch	0.06804	atmospheres
pounds per square inch	2.307	feet of water
pounds per square inch	2.036	inches of mercury
pounds per square inch	144	pounds per sq ft
square centimeters	0.1550	square inches
square feet	144	square inches
square feet	0.09290	square meters
square feet	3.587×10^6	square miles

CONVERSION FACTORS (continued)

Multiply	By	To Get
square feet	1/9	square yards
square inches	6.452	square centimeters
square inches	6.944×10^3	square feet
square meters	10.764	square feet
square miles	27.88×10^6	square feet
square miles	3.098×10^6	square yards
square yards	9	square feet
square yards	3.228×10^7	square miles
temp (degs C) + 273	1	abs tem (degs C)
temp (degs C) + 17.8	1.8	temp (degs F)
temp (degsF) + 460	1	abs temp (degs F)
temp (degs F) - 32	5/9	temp (degs C)
watts	0.05692	Btu per min
watts	44.26	foot-pounds per min
watts	0.7376	foot-pounds per sec
watts	1.341×10^3	horsepower
watts	10^3	kilowatts
watt-hours	3.415	British thermal units
watt-hours	2655	foot-pounds
watt-hours	10^3	kilowatt-hours
yards	3	feet
yards	36	inches
yards	0.9144	meters

CONVERSION TABLES

Table 174-1:
Fahrenheit degrees to Celsius degrees

F	C	F	C
212	100.0	85	29.4
210	98.9	80	26.7
205	96.1	75	23.9
200	93.3	70	21.1
195	90.6	65	18.3
190	87.8	60	15.6
185	85.0	55	12.8
180	82.2	50	10.0
175	79.4	45	7.2
170	76.7	40	4.4
165	739.9	35	1.7
160	71.1	30	-1.1
155	68.3	25	-3.9
150	65.6	20	-6.7
145	62.8	15	-9.4
140	60.0	10	-12.2
135	57.0	5	-15.0
130	54.4	0	-17.8
125	51.7	-5	-20.6
120	58.9	-10	-23.3
115	46.1	-15	-26.1
110	43.3	-20	-28.9
105	40.6	-25	-31.7
100	37.8	-30	-34.4
95	35.0	-35	-37.2
90	32.2	-40	-40.0

Table 174-2:
Celsius degrees to Fahrenheit degrees

C	F	C	F
100	212.0	28	82.4
98	208.1	26	78.8
96	204.8	24	75.2
95	201.2	22	746
92	198.6	20	68.0
90	194.0	18	64.4
88	190.4	16	60.8
86	186.8	14	57.2
84	183.2	12	53.6
82	179.6	10	50.0
80	176.0	8	46.4
78	172.4	6	42.8
76	168.8	4	39.2
75	165.2	2	35.6
72	161.6	0	32.0
70	158.0	-2	28.4
68	154.4	-4	24.8
66	150.8	-6	21.2
64	147.2	-8	17.6
62	143.6	-10	14.0
60	140.0	-12	10.4
58	136.4	-14	6.8
56	132.8	-16	3.2
54	129.2	-18	-0.1
52	125.6	-20	-4.0
50	122.0	-22	-7.6
48	114.8	-28	-14.8
44	111.2	-28	-18.4
42	107.6	-30	-22.0
40	104.0	-32	-25.6
38	96.8	-38	-32.8
36	96.8	-38	-32.8
34	93.2	-38	-36.4
32	39.6	-40	-40.0
30	36.0	-42	-43.6

CONVERSION TABLES (continued)

Table 175-1:
Inches (in.) to centimeters (cm) to millimeters (mm)

in	cm	mm
1	2.5	25
2	5	50
3	7.5	75
4	10	100
5	12.5	125
6	15	150
7	17.5	175
8	20	200
9	22.5	225
10	25	250
20	50	500
30	75	750
36	90	900
40	100	1000
50	125	1250
60	150	1500
70	175	1750
80	200	2000
90	225	2250
100	250	2500

Table 175-2:
Square inches to square centimeters and square millimeters

in	cm^2	mm^2
1	6.5	645
2	13	1292
3	19.5	1935
4	26	2580
5	32.5	3225
6	39	3870
7	45.5	4515
8	52	5160
9	58.5	5805
10	65	6450
15	97.5	–
20	130	–
30	162	–
36	195	–
40	260	–
50	325	–
60	390	–
70	455	–
80	520	–
90	585	–
100	650	–

CONVERSION TABLES (continued)

Table 176-1:
Cubic inches (cu. in.) to cubic centimeters (cc) and liters (1)

cu. in.	cc	1
1	16.4	–
2	33	–
3	49	–
4	69	–
5	82	–
6	98	–
7	115	–
8	131	–
9	148	–
10	164	–
15	246	–
20	330	–
30	410	–
36	490	–
40	660	–
50	820	–
60	980	–
70	1150	1.2
80	1310	1.3
90	1480	1.5
100	1640	1.6

Table 176-2:
Feet (ft) to meters (m)

ft.	m
1	0.3
2	0.6
3	0.9
4	1.2
5	1.5
6	1.8
7	2.1
8	2.4
9	2.7
10	3
15	4.5
20	6
25	7.5
30	9
40	12
80	18
100	30
250	75
500	150
750	225
1000	300

Table 176-3:
Square feet (sq. ft) to square meters (m^2)

sq. ft.	m^2
1	0.09
2	0.18
3	0.27
4	0.36
5	0.45
6	0.54
7	0.63
8	0.72
9	0.82
10	0.90
20	1.8
25	2.3
30	2.7
40	3.6
50	4.5
60	5.4
70	6.3
80	7.2
90	8.1
100	9

Air Pressure Conversions

1 Atmosphere	= 29.92″ Hg (32°F)
1 Atmosphere	= 14.9#/sq. in.
1 Atmosphere	= 403″ water gage (62°F)
1 Atmosphere	= 33.5 ft. water gage (62°F)
1″ Hg (62°F)	= .49 lb./sq. in.
1″ water gage (62°F)	= .074″ Hg (62°F)
1″ water gage (62°F)	= .036 lb./sq. in.
1″ water gage (62°F)	= 25.4 mm water gage

CONVERSION TABLES (continued)

Table 177-1:
Pounds per square inch (psi) kilograms per square centimeter (kg/cm^2) (Use to convert either gauge or absolute readings)

psi	kg/cm^2
1	0.0703
2	0.14
3	0.21
4	0.28
5	0.35
6	0.42
7	049
8	0.58
9	0.63
10	0.7
11	0.77
12	0.84
13	0.91
14	0.98
15	1.1
20	1.4
25	1.9
30	2.1
35	2.5
40	2.8
45	3.2
50	3.5
60	4.2
70	4.9
80	5.6
90	6.3
100	7
150	10.5
200	14.1
250	17.6
300	21.1
350	24.6
400	28

Table 177-2:
Cubic feet (cu. ft.) to cubic meters (m^2)

cu. ft.	m^2
1	0.03 (0.028)
2	0.06
3	0.09
4	0.12
5	0.15
6	0.18
7	0.21
8	0.24
9	0.27
10	0.3
15	0.45
20	0.6
25	0.75
30	0.9
40	1.2
50	1.5
60	1.8
70	2.1
80	2.4
90	2.7
100	3

Table 177-3:
Cubic feet per minute (cfm) to liters per second (1/sec)

For conversions resulting in less than one liter per second, multiply cfm by 472 to get cubic centimeters per second (cc sec.)

cfm	1/sec.
1	0.5 (472 cc sec.)
2	1
3	1.5
4	2
5	2.5
6	3
7	3.5
8	4
9	4.5
10	5
20	10
30	15
40	20
50	25
60	30
70	35
80	40
90	45
100	50
250	125
400	200
500	250
750	375
1000	500

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